Environmental design

CIBSE Guide A

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Note from the publisher

This publication is primarily intended to provide guidance to those responsible for the design, installation, commissioning, operation and maintenance of building services. It is not intended to be exhaustive or definitive and it will be necessary for users of the guidance given to exercise their own professional judgement when deciding whether to abide by or depart from it.

Foreword

CIBSE Guide A: *Environmental design* is the premier reference source for designers of low energy sustainable buildings. This edition is the 7th revision and contains significant changes from its predecessor. The contents acknowledge and satisfy the Energy Performance of Buildings Directive and UK legislation, specifically the 2006 Building Regulations Approved Documents L and F. Additionally, the authors have incorporated the latest research and best practice in order to enable environmental design engineers to practise at the forefront of their profession.

The changes made for the 7th revision may briefly be summarised as follows:

Chapter 1: *Environmental criteria for design*: this chapter has been extensively revised to include the adaptive approach and thermal comfort criteria based on the outdoor running mean temperature for offices in both the free running mode (naturally ventilated and mixed mode buildings) and for sealed buildings served by heating and cooling systems. Guidance on overheating criteria has also been included. The health relevant issues associated with environmental design have been transferred to a completely new section (section 8) in order to provide more comprehensive guidance.

Chapter 2: *External design data*: UK dry bulb and wet bulb temperature data have been updated to 2002. New Test Reference Years and Design Summer Years for 14 sites have been identified for which hourly data are available separately. The text in this chapter has also been expanded to include a new section (2.9) giving the latest guidance on future UK climate trends based on UKCIP02 scenarios, and a new section (2.10) on the heat island effect.

Chapter 3: *Thermal properties of building structures*: all the data in this section have been reviewed and updated where necessary to reflect the changes in European and International standards, including test methods for thermal conductivity and thermal transmittance and those related to specification of thermal properties and the calculation of heat transmission. The data for glazed units and windows and for non-steady state properties (admittances etc.) have been reviewed and re-calculated.

Chapter 4: *Ventilation and air infiltration*: the previous edition of Guide A referred only to natural ventilation; this chapter now covers all modes of ventilation. The chapter also specifies minimum ventilation rates to conform to both the revised Approved Document F under the Building Regulations for England and Wales, and the ventilation requirements specified in new European Standards. A new section on empirical data for air infiltration gives design and peak annual average infiltration values for a range of building types and sizes.

Chapter 5: *Thermal response and plant sizing*: this chapter details the design information required to calculate heating and cooling loads and the installed plant capacity. Recognising the iterative nature of the design processes used by practicing engineers, both steady state (manual) calculations using the admittance procedures, and dynamic calculation techniques using computer programs are included in this chapter. The text and data have been comprehensively reviewed and updated. Recommended quality assurance procedures, including the need for a software assessment test for building services design programmes, are also included.

Chapter 6: *Internal heat gains*: this section provides the latest design information on heat emissions from a wide range of internal heat gains to enable designers to use either benchmark values typical of the building and its intended usage, or to make specific estimates where sufficient reliable data are available.

Chapter 7: *Moisture transfer and condensation*: this chapter addresses the widespread concerns amongst clients and building professionals with regard to surface condensation (or, more importantly, mould growth) and also the accumulation of moisture within the structure. The chapter has been expanded to present methods for the prediction of both surface and interstitial condensation and guidelines on how to minimise these problems. The latest British and European Standards methodology and national best practice and the appropriate boundary conditions are also covered.

Chapter 8: *Health issues*: this is a totally new chapter. Its purpose is to advise building service designers and building managers of the health implications of their decisions, and give recommendations for limiting, or preferably avoiding, any adverse health interactions. It has proved impractical to include the full text of this complete and very comprehensive document within this Guide and therefore an abridged version only has been included, with the complete text included on the CD-ROM that accompanies this Guide. Additionally, the complete text is published on the CIBSE website (www.cibse.org) as CIBSE TM 40: *Health issues in building services*.

I would like to express my personal thanks to the individual section authors and their contributors for the many hours of voluntary effort attending meetings, researching, drafting, reading proofs and commenting on not only their own sections but also associated sections in this guide. I would also like to thank the committee secretary, Alan Watson, the editor, Ken Butcher, and Peter Koch for checking and harmonising the symbols and notation used throughout the Guide. My personal thanks are also given to Jacqueline Balian, CIBSE Publishing Director, and to Brian Moss, Chairman of Publications and Research Outputs Delivery Committee (PROD), for their encouragement, support and forbearance during the lengthy gestation period of this 'light' revision.

Finally, I wish to thank all members of our Institution who have provided the section authors, contributors and myself with many useful and constructive suggestions including positive ideas for improving this Guide A revision.

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Chapter 8 consists of extracts from CIBSE TM40: *Health issues in building services*, the full text of which may be found on the CD-ROM that accompanies this Guide. TM40 was prepared by the CIBSE Health Issues Task Group.

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Contents

1 Environmental criteria for design

1.1 Introduction

1.1.1 Comfort

Comfort has been defined as ‘that condition of mind that expresses satisfaction with the ... environment’[(1)].

The indoor environment should be designed and controlled so that occupants’ comfort and health are assured. There are individual differences in perception and subjective evaluation, resulting in a base level of dissatisfaction within the building population. This dissatisfaction may be with a specific aspect of the environment or may be general and non-specific. The aim of design should be to minimise this dissatisfaction as far as is reasonably practicable.

The environmental factors considered here include the thermal, visual and acoustic conditions, indoor air quality, electromagnetic fields and static electricity. It is not practicable to formulate a single index that quantifies the individual’s response to all these factors, and there may be additive or synergistic effects resulting from interactions among a number of them. For example irritant contaminants, such as formaldehyde, become more noticeable at low air humidity[(2)].

Therefore, it is necessary to specify measurable limits or ranges for each of the environmental factors, making allowance, where possible, for any interactions that might occur.

1.1.2 Health aspects

See also chapter 8: *Health issues*.

The constitution of the World Health Organisation defines good health as ‘a state of complete physical, mental and social well-being, not merely the absence of disease and infirmity’. While for most people this may be an ideal rather than reality, it indicates that the indoor environment should be managed in such a way as to promote health, not merely to avoid illness.

In some cases occupants experience symptoms which may not be obviously related to a particular cause, but which become less severe or disappear when they leave a particular environment. These symptoms, such as nausea, mucosal dryness or irritation, runny nose, eye problems, headaches, skin problems, heavy head and flu-like symptoms, may be quite severe and lead to reduced productivity or absenteeism. If a significant proportion of occupants experience these symptoms then, by definition[(3)], the occupants are suffering from ‘sick building syndrome’.

It is likely that the cause of sick building syndrome is multi-factorial. Researchers have identified a statistically significant correlation between symptom prevalence and many different and unrelated factors. It would appear that if environmental conditions are within the ranges suggested in this Guide then the risk of occupant dissatisfaction and sick building syndrome is reduced, though not eliminated.

1.2 Notation

1.2.1 Symbols

The symbols used within this section are defined as follows.

A	Total area of internal surfaces (ceiling, floor, windows and walls) (m^2)
A_w	Net glazed area of window (m^2)
C_p	Concentration of pollutant (ppm)
C_p'	Concentration of pollutant by volume ($mg \cdot m^{-3}$)
C_{pi}	Limit of concentration of pollutant in indoor air (ppm)
C_{po}	Concentration of pollutant in outdoor air (ppm)
DF	Average daylight factor (%)
DR	Draught rating (%)
E_v	Ventilation effectiveness
f_{cl}	Ratio of the area of clothed human body to that of unclothed human body
H	Heat transfer ratio: $h_c / (h_c + h_r)$
h_c	Convective heat transfer coefficient at body surface ($W \cdot m^{-2} \cdot K^{-1}$)
h_r	Radiative heat transfer coefficient at body surface ($W \cdot m^{-2} \cdot K^{-1}$)
I_{cl}	Thermal resistance of clothing ($m^2 \cdot K \cdot W^{-1}$)
M_a	Activity level (met)
M_p	Molar mass of pollutant ($kg \cdot mole^{-1}$)
P	Pollutant emission rate ($L \cdot s^{-1}$)
p_s	Partial water vapour pressure in air surrounding the body (Pa)
Q	Outdoor air supply rate ($L \cdot s^{-1}$)
Q'	Reduced outdoor air supply rate to control intermittent pollution ($L \cdot s^{-1}$)
Q_c	Outdoor air supply rate to account for total contaminant load ($L \cdot s^{-1}$)
PMV	Predicted mean vote
PPD	Predicted percentage dissatisfied
R_a	Area-weighted average reflectance of interior surfaces (ceiling, floor, windows and walls)
T	Diffuse transmittance of glazing material including effects of dirt
T_u	Turbulence intensity (%)
t_p	Duration of release of pollutant (s)
V	Volume of space (m^3)
v	Air speed ($m \cdot s^{-1}$)

v_{SD} Standard deviation of air speed (m·s^{-1})
α Angle in degrees subtended, in the vertical plane normal to the window, by sky visible from centre of window (degree)
α_{rm} Constant related to running mean temperature
θ_{ai} Indoor air temperature (°C)
θ_{com} Comfort temperature (°C)
θ_{ed} Daily mean outdoor temperature (°C)
θ_c Operative temperature (°C)
θ_{on} Operative temperature at thermal neutrality (°C)
θ_r Mean radiant temperature (°C)
θ_{rm} Exponentially weighted running mean of the daily mean outdoor temperature (°C)
θ_{sc} Surface temperature of clothing (°C)
Φ_c Heat loss by convection from surface of clothed body (W)
Φ_e Heat loss by evaporation from surface of clothed body (W)
Φ_k Heat flow by conduction from surface of clothed body (W)
Φ_m Metabolic rate per m^2 of body surface (W)
Φ_{rad} Heat loss by radiation from surface of clothed body (W)
Φ_{re} Heat exchange by evaporation in respiratory tract (W)
Φ_{rc} Heat exchange by convection in respiratory tract (W)
Φ_s Body heat storage (W)
Φ_w Rate of performance of external work (W)

Note: in compound units, the abbreviation 'L' has been used to denote 'litre'.

1.2.2 Thermal comfort: annotated definitions of main thermal parameters

For the purposes of this Guide, the following terminology is adopted.

Indoor air temperature (θ_{ai})

The dry bulb temperature of the air in the space.

Mean radiant temperature (θ_r)

The uniform surface temperature of a radiantly black enclosure in which an occupant would exchange the same amount of radiant heat as in the actual non-uniform space. (see BS EN ISO 7726[4] for derivation). (*Note*: if the surface temperatures of the internal surfaces of the enclosure are unequal, mean radiant temperature varies throughout the enclosure and depends upon the posture and orientation of the occupant.)

Relative air speed (v_r)

The net mean air speed across the body. For sedentary occupancy, v_r is taken as the room air movement only (v). For people in motion it will take account of the speed of their movement in addition to the mean room air speed.

Humidity

The humidity of room air expressed in absolute terms, i.e. moisture content (mass of water vapour per unit mass of dry air (kg·kg^{-1})) or vapour pressure (partial pressure of water vapour (Pa)).

Relative humidity

The ratio of vapour pressure to saturation vapour pressure at same dry bulb temperature, expressed as a percentage (% RH).

Percentage saturation

The ratio of moisture content to moisture content of saturated air at same dry bulb temperature, expressed as a percentage (% sat). (*Note*: at ambient air temperatures and humidities the difference between relative humidity and percentage saturation is small and may be ignored.)

Clo

The unit for thermal insulation of clothing[5], where 1 clo = 0.155 m^2·K·W^{-1}. A clothing ensemble that approximates to 1 clo consists of underwear, blouse/shirt, slacks/trousers, jacket, socks and shoes.

Met

The unit used to express the physical activity of humans is the met[6], where 1 met = 58.2 W·m^{-2}. One met is approximately the metabolic rate of a person seated at rest. The average body surface area for adults is about 1.8 m^2, therefore 1 met is equivalent to approximately 100 W of total heat emission.

Operative and dry resultant temperatures

In previous editions of this Guide, dry resultant temperature was used as a temperature index for moderate thermal environments. In concept it is identical to the 'operative temperature' (θ_c) which is used in both International Standards[7] and ANSI/ASHRAE[1] standards. In the interests of international consistency of nomenclature, the CIBSE has decided to replace the term 'dry resultant temperature' with the term 'operative temperature'. This entails no change of substance.

The operative temperature (θ_c), like the dry resultant temperature, combines the air temperature and the mean radiant temperature into a single value to express their joint effect. It is a weighted average of the two, the weights depending on the heat transfer coefficients by convection (h_c) and by radiation (h_r) at the clothed surface of the occupant.

The operative temperature is defined as:

$$\theta_c = H\,\theta_{ai} + (1 - H)\,\theta_r \tag{1.1}$$

where θ_c is the operative temperature (°C), θ_{ai} is the indoor air temperature (°C), θ_r is the mean radiant temperature (°C), H is the ratio $h_c / (h_c + h_r)$ and $(1 - H)$ is the ratio $h_r / (h_c + h_r)$ where h_c and h_r are the surface heat transfer coefficients by convection and by radiation respectively (W·m^{-2}·K^{-1}).

Researchers have differed in their estimates of the values of these heat transfer coefficients, and hence of the value of H. In this Guide the value of $\sqrt{(10\ v)}$, where v is the air speed ($m \cdot s^{-1}$) is retained for the ratio of h_c to h_r (as for dry resultant temperature in previous editions), and so:

$$\theta_c = \frac{\theta_{ai} \sqrt{(10\ v)} + \theta_r}{1 + \sqrt{(10\ v)}} \tag{1.2}$$

At indoor air speeds below 0.1 $m \cdot s^{-1}$, natural convection is assumed to be equivalent to $v = 0.1$, and equation 1.2 becomes:

$$\theta_c = {}^1/_2\ \theta_{ai} + {}^1/_2\ \theta_r \tag{1.3}$$

Operative temperature approximates closely to the temperature at the centre of a painted globe of some 40 mm diameter, see Appendix 1.A2. A table-tennis ball is a suitable size, and may be used to construct a globe thermometer appropriate for indoor spaces[8].

In well insulated buildings that are predominantly heated by convective means, the difference between air and the mean radiant temperatures (and hence between the air and operative temperatures) is small.

Note: from the presence of the air speed in the equation 1.2, it has sometimes been assumed that operative temperature fully allows for the effect of air speed on the occupant. This is not so. Increased air movement has two related effects: (1) it alters the ratio $h_c / (h_c + h_r)$, thus potentially altering operative temperature, and (2) it alters the absolute value of the combined heat transfer coefficient ($h_c + h_r$) between the clothed surface and the enclosure. Thus the surface temperature of the occupant requires for its estimation both the operative temperature and the air speed.

1.3 Thermal environment

1.3.1 Factors affecting thermal comfort

A person's sensation of warmth is influenced by the following main physical parameters, which constitute the thermal environment:

— air temperature

— mean radiant temperature

— relative air speed

— humidity.

Besides these environmental factors there are personal factors that affect thermal comfort:

— metabolic heat production

— clothing.

It is also required that there be no local discomfort (either warm or cold) at any part of the human body due to, for example, asymmetric thermal radiation, draughts, warm or cold floors, or vertical air temperature differences.

Table 1.1 Thermal sensation scale[1]

Index value	Thermal sensation
+3	Hot
+2	Warm
+1	Slightly warm
0	Neutral
–1	Slightly cool
–2	Cool
–3	Cold

1.3.1.1 Temperature

The room air temperature and radiant temperature may be combined as the operative temperature. Temperature is usually the most important environmental variable affecting thermal comfort. A change of three degrees will change the response on the scale of subjective warmth (Table 1.1[1]) by about one scale unit for sedentary persons. More active persons are less sensitive to changes in room temperature. Guidance on temperatures suitable for various indoor spaces in heated or air conditioned buildings is given in Table 1.5 and, for unheated spaces in buildings in warm weather, in Table 1.7.

1.3.1.2 Air movement and draughts

The cooling effect of air movement is well known. If this cooling is not desired, it can give rise to complaints of draught. The temperature of the moving air is not necessarily that of the room air nor that of the incoming ventilation air but will generally lie between these values. It should also be noted that people are more tolerant of air movement if the direction of the air movement varies.

Where air speeds in a room are greater than 0.15 $m \cdot s^{-1}$ the operative temperature should be increased from its 'still air' value to compensate for the cooling effect of the air movement. Suitable corrections are given in Figure 1.1. The figure applies to sedentary or lightly active people. Alternatively, the influence of mean relative air speed can be calculated using the PMV index, as described in section 1.3.2. Note that air speeds greater than about 0.3 $m \cdot s^{-1}$ are probably unacceptable except in naturally ventilated buildings in summer when higher air speeds may be desirable for their cooling effect.

The relative air speed over the body surface increases with activity. A correction can be estimated where activity level (M_a) exceeds 1 met by adding $0.3 \times (M_a - 1)$ to the air speed relative to a stationary point. For example, for a person whose activity is equivalent to 1.8 met in a room in which the air speed is 0.1 $m \cdot s^{-1}$, the relative air speed over that person's body is: $0.1 + 0.3\ (1.8 - 1) = 0.34\ m \cdot s^{-1}$. This assumes that the direction of the airflow is at random. At higher air speeds, where airflow may by mono-directional, the relative air speed will depend on the direction of travel of the person.

However, studies[9] have shown that dissatisfaction due to draught is not only a function of mean air speed and local air temperature, but also of fluctuations of air speed. It has been suggested that people are particularly sensitive if air speeds fluctuate at a frequency in the range 0.3–0.6 Hz[10].

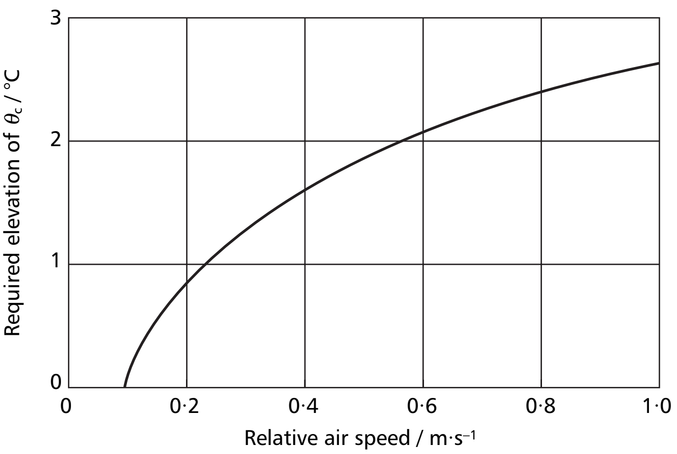


Figure 1.1 Correction to operative temperature (θ_c) to take account of air movement

Fluctuations in air speed may be described by the standard deviation of the air speed (v_{SD}), or the turbulence intensity (T_u) which is defined as the ratio of standard deviation of the air speed to the mean air speed, i.e.:

$$T_u = 100\ (v_{SD} / v) \tag{1.4}$$

where T_u is the turbulence intensity (%), v_{SD} is the standard deviation of the air speed (m·s^{-1}) and v is the mean air speed (m·s^{-1}). There is debate about the magnitude of the effect of turbulence on the sensation of draught[11–13]. One estimate of the effect is incorporated in the draught rating (DR), see below.

For air conditioned and mechanically ventilated buildings, the draught rating (DR), expressed as a percentage, is given by[7]:

$$\mathrm{DR} = (34 - \theta_{ai})\ (v - 0.05)^{0.62}\ (0.37\ v\ T_u + 3.14) \tag{1.5}$$

where DR is the draught rating (%) and θ_{ai} is the indoor air temperature (°C). (For air speeds less than 0.05 m·s^{-1}, take v = 0.05 m·s^{-1}; for calculated DR values greater than 100%, use DR = 100%.)

A draught rating of more than 15% is taken to be unacceptable[9]. Figure 1.2[7] shows solutions for equation 1.5 for DR = 15% based on light, mainly sedentary, activity (i.e. 1.2 met). Each line on the graph shows the limits of acceptable temperature and velocity for a given turbulence intensity. For example, if the temperature of the air passing over the body is 23 °C and the turbulence intensity is 60%, the draught rating criterion of 15% corresponds to an air speed of 0.14 m·s^{-1}. However if the turbulence intensity is only 10%, the limiting velocity for comfort is 0.23 m·s^{-1}.

In the main body of most rooms, away from supply air jets, the turbulence intensity is usually between 30% and 50%.

1.3.1.3 Humidity

Humidity has little effect on feelings of warmth unless the skin is damp with sweat. For sedentary, lightly clothed people moisture may become apparent as operative temperatures rise above 26–28 °C. Thus, for most practical purposes, the influence of humidity on warmth in moderate thermal environments may be ignored[14] and humidity in the range 40–70 % RH is generally acceptable[15]. However, humidity may be important in the context of microbiological growth, the preservation of artefacts and the reduction of static electricity[16], see section 8.3.3.

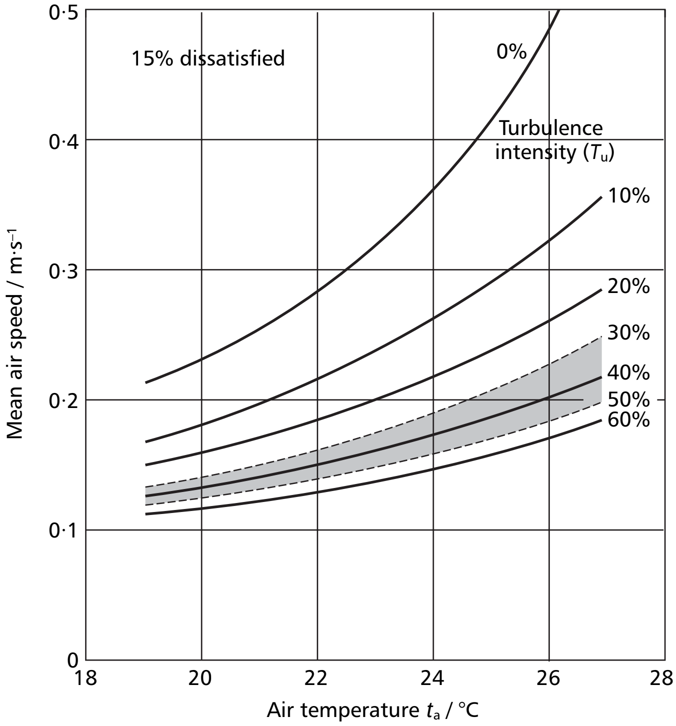


Figure 1.2 Combinations of mean air speed, air temperature and turbulence intensity for a draught rating of 15%[22]; data from climate chamber experiment at 20, 23 and 26 °C, 15 minute exposures, subjects adjusted clothing for comfort; the data apply to people comfortable had the air been still; people who are feeling cold may complain of draught even in still air (reproduced from BS EN ISO 7730[7] by permission of the British Standards Institution)

High room humidity may occur through a combination of evaporation from moisture sources and poor ventilation, and/or high outdoor humidity, see chapter 7: *Moisture transfer and condensation*. Bathrooms and kitchens are particularly prone.

For the purposes of designing air conditioning systems, a maximum room relative humidity of 60% within the recommended range of summer design operative temperatures would provide acceptable comfort conditions for human occupancy and minimise the risk of mould growth and house dust mites. Condensation should be avoided within buildings on surfaces that could support microbial growth or be stained or otherwise damaged by moisture. This may be achieved by ensuring that, where possible, surfaces are above the dew-point of the adjacent air.

If possible, at the design temperatures normally appropriate to sedentary occupancy, the room humidity should be above 40% RH. Lower humidity is often acceptable for short periods. Humidity of 30% RH or below may be acceptable but precautions should be taken to limit the generation of dust and airborne irritants and to prevent static discharge from occupants. Note that for heated-only buildings in the UK, the humidity can remain below 40% RH during periods of sustained cold weather.

Shocks due to static electricity (see section 8.3.3) are unlikely with humidities above 40% RH or at lower humidities if precautions are taken in the specification of materials and equipment to prevent the build-up of static electricity.

1.3.1.4 Clothing

Clothing worn by people indoors is modified by the season and outdoor weather, as well as by the indoor thermal environment. During the summer months typical clothing ensembles in commercial premises may consist of lightweight dresses or trousers, short or long-sleeved shirts or blouses, and occasionally a suit jacket or sweater. Without jacket or sweater, these ensembles have clothing insulation values ranging from 0.35 to 0.6 clo (see definition in section 1.2.2).

During winter, people wear thicker, heavier ensembles, usually with more layers. A typical indoor winter ensemble would have an insulation value of 0.8 to 1.0 clo, although recent studies of office workers found values generally at the lower end of this range[17].

Clothing insulation values for typical clothing ensembles are given in Table 1.2[1,5]. The insulation provided by other clothing ensembles may be estimated by summing the insulation values for individual garments, see Table 1.3[7].

The wearing or otherwise of an article of clothing is equivalent in its effect on subjective feelings of warmth to raising or lowering the operative temperature. Table 1.3 shows these equivalencies, which may be used to modify the design temperature ranges given in Table 1.5 (page 1-8).

The clothing insulation provided by an individual garment consists of the effective resistance of the material from which the garment is made plus the thermal resistance of the air layer trapped between the clothing and the skin. If the thickness of this layer is reduced, e.g. by air movement or change in posture, then the thermal resistance of the air layer is reduced leading to a reduction in the overall insulation provided by the clothing. In addition, body movement such as walking can lead to a pumping action in loose clothing that forces cool air between the skin and the surrounding clothing. Therefore, factors other than the thermal resistance of the clothing, e.g. looseness of fit, also affect the clo value. The value of clothing insulation of an ensemble, if estimated from Tables 1.2 or 1.3 is not precise, but will usually be within 20%.

For sedentary occupants, the insulating properties of the chair will affect thermal comfort, see footnotes to Tables 1.2 and 1.3.

1.3.1.5 Metabolic heat production

Metabolic heat production is largely dependent on activity. Table 1.4[1,6,7] gives metabolic rates for specific activities. For most people, daily activity consists of a mixture of specific activities and/or a combination of work and rest periods. A weighted-average metabolic rate may be used, provided that the activities frequently alternate, i.e. several times per hour.

Table 1.2 Thermal insulation values for typical clothing ensembles for work and daily wear; these values were determined by measurement on a standard thermal mannequin (adapted from ANSI/ASHRAE 55-2004[1] and BS ISO 9920[5])

Description	Insulation level / clo
Underpants plus:	
— shirt (short sleeves), lightweight trousers, light socks, shoes	0.5
— shirt, lightweight trousers, socks, shoes	0.6
— boiler suit, socks, shoes	0.7
— shirt, trousers, socks, shoes	0.75
— shirt, boiler suit, socks, shoes	0.8
— shirt, trousers, jacket, socks, shoes	0.85
— shirt, trousers, smock, socks, shoes	0.9
Underwear (short sleeves/legs) plus:	
— tracksuit (sweater and trousers), long socks, training shoes	0.75
— shirt, trousers, jacket or sweater, socks, shoes	1.0
— shirt, trousers, boiler suit, socks, shoes	1.1
— shirt, trousers, jacket, insulated jacket, socks, shoes	1.25
— boiler suit, insulated jacket and trousers, socks, shoes	1.4
— shirt, trousers, jacket, insulated jacket and trousers, socks, shoes	1.55
— shirt, trousers, jacket, quilted jacket and overalls, socks, shoes	1.85
— shirt, trousers, jacket, quilted jacket and overalls, socks, shoes, cap, gloves	2.0
Underwear (long sleeves/legs) plus:	
— shirt, trousers, pullover, jacket, socks, shoes	1.3
— insulated jacket and trousers, insulated jacket and trousers, socks, shoes	2.2
— insulated jacket and trousers, quilted parka, quilted overalls, socks, shoes, cap, gloves	2.55
Bra and pants plus:	
— T-shirt, shorts, light socks, sandals	0.3
— petticoat, stockings, lightweight dress (with sleeves), sandals	0.45
— stockings, blouse (short sleeves), skirt, sandals	0.55
— petticoat, stockings, dress, shoes	0.7
— petticoat, shirt, skirt, thick socks (long), shoes	0.8
— shirt, skirt, sweater, thick socks (long), shoes	0.9
— shirt, trousers, jacket, socks, shoes	1.0
— blouse (long sleeves), long skirt, jacket, stockings, shoes	1.1
Pyjamas (long sleeves/legs), bath robe, slippers (no socks)	0.95

Notes:

(1) For sedentary persons, an allowance should be made for the insulating effect of the chair, i.e. 0.15 clo for an office chair (corresponding to a temperature change of 0.9 K) and 0.3 clo for an upholstered armchair (corresponding to a temperature change of 1.8 K)

(2) Guidance on the clo-values of a wider range of ensembles, including some non-Western forms of dress, may be found in BS EN ISO 9920[5]

For example, the average metabolic rate for a person typing for 50% of the time, filing while seated for 25% of the time and walking on the level for 25% of the time will be: (0.5 × 1.1) + (0.25 × 1.2) + (0.25 × 1.6) = 1.25 met.

For people dressed in normal casual clothing (I_{cl} = 0.5 – 1.0 clo), a rise in activity of 0.1 met corresponds to a possible reduction of approximately 0.6 K in the design operative temperatures given in Table 1.5. A greater reduction is possible for heavily clad people.

For example, a seated person with an activity level equivalent to 1.0 met who experiences optimum comfort at 24 °C would find 22.8 °C better when carrying out filing for a period (1.2 met).

Table 1.3 Thermal insulation values for typical garments and corresponding reduction in acceptable operative temperature for sedentary occupants (adapted from BS EN ISO 7730[7])

Description	Insulation level / clo	Corresponding change in operative temperature / K
Underwear:		
— briefs/underpants	0.03	0.2
— underpants (long legs)	0.10	0.6
— singlet	0.04	0.2
— T-shirt	0.09	0.5
— vest (long sleeves)	0.12	0.7
— bra	0.01	0.06
Shirts/blouses:		
— short sleeve	0.15	0.9
— light blouse (long sleeves)	0.15	0.9
— lightweight (long sleeves)	0.20	1.2
— mediumweight (long sleeves)	0.25	1.5
— flannel shirt (long sleeves)	0.30	1.8
Trousers:		
— shorts	0.06	0.4
— lightweight	0.20	1.2
— mediumweight	0.25	1.5
— flannel	0.28	1.7
Skirts/dresses:		
— light skirt (summer)	0.15	0.2
— heavy skirt (winter)	0.25	1.5
— light dress (short sleeves)	0.20	1.2
— winter dress (long sleeves)	0.40	2.4
Boiler suit	0.55	3.3
Sweaters/pullovers:		
— sleeveless waistcoat	0.12	0.7
— thin	0.20	1.2
— medium	0.28	1.7
— thick	0.35	2.1
Jackets:		
— light (summer)	0.25	1.5
— medium	0.35	2.1
— smock	0.30	1.8
Highly insulative:		
— overall/ski suit	0.90	5.4
— trousers	0.35	2.1
— jacket	0.40	2.4
— sleeveless body-warmer	0.20	1.2
Outdoor clothing:		
— coat	0.60	3.6
— jacket	0.55	3.3
— parka	0.70	4.2
— heavyweight overalls	0.55	3.3
Miscellaneous:		
— ankle socks	0.02	0.1
— thick ankle socks	0.05	0.3
— thick long socks	0.10	0.6
— stockings	0.03	0.2
— shoes (thin soles)	0.02	0.1
— shoes (thick soles)	0.04	0.2
— boots	0.10	0.6
— gloves	0.05	0.3

Note: for sedentary persons, an allowance should be made for the insulating effect of the chair, i.e. 0.15 clo for an office chair (corresponding to a temperature change of 0.9 K), and 0.3 clo for an upholstered armchair (corresponding to a temperature change of 1.8 K)

Table 1.4 Typical metabolic rate and heat generation per unit area of body surface for various activities[1,5,7]

Activity	Metabolic rate / met	Heat generation / $W{\cdot}m^{-2}$
Resting:		
— sleeping	0.7	41
— reclining	0.8	46
— seated, quiet	1.0	58
— standing, relaxed	1.2	70
Walking (on level):		
— 0.9 $m{\cdot}s^{-1}$	2.0	116
— 1.3 $m{\cdot}s^{-1}$	2.6	151
— 1.8 $m{\cdot}s^{-1}$	3.8	221
Office work:		
— reading, seated	1.0	58
— writing	1.0	58
— typing	1.1	64
— filing, seated	1.2	70
— filing, standing	1.4	81
— lifting/packing	2.1	122
Occupational:		
— cooking	1.4–2.3	81–134
— house cleaning	1.7–3.4	99–198
— seated, heavy limb movement	2.2	128
— machine sawing	1.8	105
— light machine work	1.6–2.0	93–116
— heavy machine work	3.0	175
— handling 50 kg bags	4.0	233
Leisure:		
— dancing (social)	1.4–4.4	82–256
— callisthenics/exercise	3.0–4.0	175–233
— tennis (singles)	3.6–4.0	210–233
— basketball	5.0–7.6	291–442
— wrestling (competitive)	7.0–8.7	407–506

Note: average surface area of an adult human body is about 1.8 m^2

Care must be used when applying Table 1.4 due to uncertainties in measuring metabolic rates and in defining the tasks. It is reasonably accurate (i.e. ± 20%) for engineering purposes for well-defined activities with $M_a < 1.5$. However, for poorly defined activities with $M_a > 3.0$ the error may be as high as ±50%.

1.3.2 Heat balance model of thermal comfort: predicted mean vote (PMV) and predicted percentage dissatisfied (PPD)

The human thermo-regulatory system attempts to maintain a deep-body temperature of about 37 °C. When this temperature is exceeded, the body initiates heat control mechanisms, e.g. dilation of peripheral blood vessels and sweating. In response to cold, the body instigates constriction of peripheral blood vessels, changes in muscular tone, erection of body hair and shivering. The thermo-regulatory system can maintain the appropriate deep-body temperature in a wide range of combinations of activity level and environmental variables.

The heat balance of the human body may be written as:

$$\Phi_m - \Phi_w = \Phi_{rc} + \Phi_{re} + \Phi_k + \Phi_r + \Phi_c + \Phi_e + \Phi_s \tag{1.6}$$

where Φ_m is the metabolic rate (W), Φ_w is the rate of performance of external work (W), Φ_{rc} is the heat exchange by convection in the respiratory tract (W), Φ_{re} is the heat exchange by evaporation in the respiratory tract (W), Φ_k is the heat flow by conduction from the surface of the clothed body (W), Φ_r is the heat loss by radiation from the surface of the clothed body (W), Φ_c is the heat loss by convection from the surface of the clothed body (W), Φ_e is the heat loss by evaporation from the skin (W) and Φ_s is the body heat storage (W).

In steady state conditions Φ_s would be zero but this does not necessarily mean that a comfortable thermal state is achieved. It is also necessary for skin temperatures and sweat rates to be neither too high nor too low, their values for comfort depending on the metabolic rate[18,19]. Steady state conditions never truly exist, because the relation to the thermal environment is one of dynamic interaction, and comfort conditions based on the steady state are therefore approximate.

The 'predicted mean vote' (PMV) combines the influence of air temperature, mean radiant temperature, air movement and humidity with that of clothing and activity level into one value on a thermal sensation scale, see Table 1.1. The PMV is the predicted mean value of the 'votes' of a large group of persons, exposed to the same environment, and with identical clothing and activity.

Appendix 1.A1 gives an equation for PMV derived by Fanger[9] and the listing (in BASIC) of a computer program for solution of the equation, based on that given in BS EN ISO 7730[7]. Solutions to this equation in tabular form, based on 50% saturation, are given in BS EN ISO 7730.

People are thermally dissimilar. Where a group of people is subject to the same environment, it will normally not be possible to satisfy everyone at the same time. The aim, therefore, is to create optimum thermal comfort for the whole group, i.e. a condition in which the highest possible percentage of the group is thermally comfortable.

As the individual thermal sensation votes will be scattered around the mean predicted value (i.e. PMV), it is useful also to predict the percentage of people who would be dissatisfied, taken as those who would vote $> +1$ or < -1 on the sensation scale. The predicted percentage dissatisfied (PPD) attempts to do this and is obtained from the PMV using the following equation[7]:

$$\text{PPD} = 100 - 95 \exp\left[-(0.03353\ \text{PMV}^4 + 0.2179\ \text{PMV}^2)\right] \tag{1.7}$$

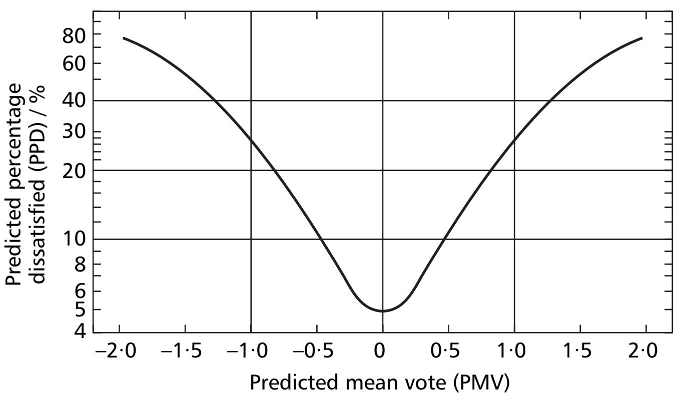


Figure 1.3 PPD as a function of PMV

The predicted percentage dissatisfied (PPD) as a function of predicted mean vote (PMV) is shown in Figure 1.3. It applies to a large group of people in the same thermal environment and with identical clothing and activity level. Alternatively it may be interpreted as the notional probability that a randomly chosen person, having that clothing and activity, would experience discomfort in that thermal environment. Since PPD assumes uniform clothing, and people are often free to choose their clothing for comfort, there will normally be less discomfort than that predicted by the PPD. PPD is a function of PMV and is seriously affected by any error in the estimation of PMV.

1.3.3 Discrepancies between field-study findings and the PMV/PPD index

The PMV/PPD index is a mathematical model of human thermal physiology, calibrated against the warmth sensations reported by people during experiments in climate-controlled spaces. The index has not always been found to agree with the sensations reported by people in the circumstances of daily life, as obtained during field studies of thermal comfort[20–22]. Reasons for these differences are diverse and not fully understood. In general, people are found to be more tolerant of diversity in ordinary circumstances than would have been predicted from the PMV/PPD model. The greatest systematic discrepancies occur when indoor temperature is warm and outdoor temperatures are high[23], when PMV predicts that people would be warmer than has been found in practice, and in this circumstance empirical results from field-studies should be preferred.

1.4 Design criteria

1.4.1 General

Table 1.5 gives general guidance and recommendations on suitable winter and summer temperature ranges (together with outdoor air supply rates, filtration grades, maintained illuminances and noise ratings) for a range of room and building types. The operative temperature* ranges correspond to a predicted mean vote (PMV) of ±0.25, see section 1.3.2, and assume the clothing insulation and metabolic rates indicated. From these values adjustments can be made for the circumstance that the clothing and activity differ from those assumed in Table 1.5 (see Tables 1.3 and 1.4). The temperature ranges may be widened by approximately 1 °C at each end if a PMV of ±0.5 (i.e. PPD of 10%) is acceptable.

Guidance on adapting these general recommendations to other situations is given in various sections and Table 1.6 indicates which section should be consulted for further guidance on any given design parameter.

Spaces occupied only briefly, such as bathrooms, toilets, halls and landings are outside the scope of PMV/PPD

* Operative temperature can be measured using a globe thermometer (see Appendix 1.A2), typically with a globe diameter of 40 mm, placed away from direct sun. Temperatures can vary within a space, so readings should be taken in several places.

because the thermal steady state is not normally reached. It is often convenient for their resultant temperatures to be similar to those of adjoining spaces.

The summer comfort temperatures given in Table 1.5 apply to air conditioned buildings. Higher temperatures may be acceptable if full air conditioning is not present, and guidance on this may be found in section 1.4.2, with a detailed discussion of the adaptive approach in section 1.6.

The Fuel and Electricity (Heating) (Control) Order 1974(24) and the Fuel and Electricity (Heating) (Control) (Amendment) Order 1980(25) prohibit the use of fuels or electricity to heat premises above 19 °C. This does not mean that the temperature in buildings must be kept below 19 °C but only that fuel or electricity must not be used to raise the temperature above this level. In Table 1.5, for some applications, the recommended winter design temperatures exceed 19 °C. In these cases, it is assumed that the recommended temperatures can be maintained by contributions from heat sources other than the heating system. These may include solar radiation, heat gains from lighting, equipment and machinery and heat gains from the occupants themselves.

Table 1.5 Recommended comfort criteria for specific applications

Building/room type	Winter operative temp. range for stated activity and clothing levels*			Summer operative temp. range (air conditioned buildings†) for stated activity and clothing levels*			Suggested air supply rate / ($L \cdot s^{-1}$ per person) unless stated otherwise	Filtration grade‡	Maintained illuminance¶ / lux	Noise rating§ (NR)
	Temp. / °C	Activity / met	Clothing / clo	Temp. / °C	Activity / met	Clothing / clo				
Airport terminals:										
— baggage reclaim	12–19[1]	1.8	1.15	21–25[1]	1.8	0.65	10[2]	F6–F7	200	45
— check–in areas[3]	18–20	1.4	1.15	21–23	1.4	0.65	10[2]	F6–F7	500[4]	45
— concourse (no seats)	19–24[1]	1.8	1.15	21–25[1]	1.8	0.65	10[2]	F6–F7	200	45
— customs area	18–20	1.4	1.15	21–23	1.4	0.65	10[2]	F6–F7	500	45
— departure lounge	19–21	1.3	1.15	22–24	1.3	0.65	10[2]	F6–F7	200	40
Art galleries — see *Museums and art galleries*										
Banks, building societies, post offices:										
— counters	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F6–F7	500	35–40
— public areas	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F5–F7	300	35–45
Bars/lounges	20–22	1.3	1.0	22–24	1.3	0.65	10[2]	F5–F7	100–200[5]	30–40
Bus/coach stations — see *Railway/coach stations*										
Churches	19–21	1.3	1.15	22–24	1.3	0.65	10[2]	G4–F6	100–200	25–30
Computer rooms[6]	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F7–F9	300	35–45
Conference/board rooms	22–23	1.1	1.0	23–25	1.1	0.65	10[2]	F6–F7	300/500[7]	25–30
Drawing offices	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F7	750	35–45
Dwellings:										
— bathrooms	20–22	1.2	0.25	23–25	1.2	0.25	15 $L \cdot s^{-1}$	G2–G4 (extract)[8]	150[4]	—
— bedrooms	17–19	0.9	2.5	23–25	0.9	1.2	0.4–1 ACH to control moisture[8]	G2–G4	100[4]	25
— hall/stairs/landings	19–24[1]	1.8	0.75	21–25[1]	1.8	0.65	—	—	100	—
— kitchen	17–19	1.6	1.0	21–23	1.6	0.65	60 $L \cdot s^{-1}$	G2–G4 (extract)[8]	150–300	40–45
— living rooms	22–23	1.1	1.0	23–25	1.1	0.65	0.4–1 ACH to control moisture[8]	G2–G4	50–300	30
— toilets	19–21	1.4	1.0	21–23	1.4	0.65	> 5 ACH	G2–G4	100[4]	—
Educational buildings:										
— lecture halls[9]	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	G4–G5	500[10]	25–35
— seminar rooms	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	G4–G5	300[10]	25–35
— teaching spaces[9]	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	G4–G5	300[10]	25–35
Exhibition halls	19-21	1.4	1.0	21–23	1.4	0.65	10[2]	G3–G4	300	40
Factories:										
— heavy work	11–14[11]	2.5	0.85	—[12]	—	—	—[13]	Depends on use	—[14,15]	50–65
— light work	16–19	1.8	0.85	—[12]	—	—	—[13]	Depends on use	—[14,15]	45–55
— sedentary work	19–21	1.4	1.0	21–23	1.4	0.65	—[13]	Depends on use	—[14,15]	45
Fire/ambulance stations:										
— recreation rooms	20–22	1.3	1.0	22–24	1.3	0.65	10[2]	F5	300	35–40
— watchroom	22–23	1.1	1.0	24–26	1.1	0.65	10[2]	F5	200	35–40

Table continues

Table 1.5 Recommended comfort criteria for specific applications — *continued*

Building/room type	Winter operative temp. range for stated activity and clothing levels*			Summer operative temp. range (air conditioned buildings†) for stated activity and clothing levels*			Suggested air supply rate / (L.s^{-1} per person) unless stated otherwise	Filtration grade‡	Maintained illuminance¶ / lux	Noise rating§ (NR)
	Temp. / °C	Activity / met	Clothing / clo	Temp. / °C	Activity / met	Clothing / clo				
Garages:										
— parking	—	—	—	—	—	—	6 ACH (extract)	—	75/300	55
— servicing	16–19	1.8	0.85	—	—	—	—	G2–G3	300/500	45–50
General building areas:										
— corridors	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	—[16]	100	40
— entrance halls/lobbies	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	—[16]	100/200[4]	35–40
— kitchens (commercial)	15–18	1.8	1.0	18–21	1.8	0.65	—[17]	G2–G4	500	40–45
— toilets	19–21	1.4	1.0	21–23	1.4	0.65	> 5 ACH	G4–G5	200	35–45
— waiting areas/rooms	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	—[16]	200	30–35
Hospitals and health care buildings:										
— bedheads/wards	22–24	0.9	1.4	23–25	0.9	1.2	10[2]	F7–F9	—[18]	30
— circulation spaces (wards)[19]	19–24[1]	1.8	0.75	21–25[1]	1.8	0.65	10[2]	F7–F9	—[18]	35
— consulting/treatment rooms	22–24	1.4	0.55	23–25	1.4	0.45	10[2]	F7–F9	300/500[18]	30
— nurses' station[19]	19–22	1.4	0.9	21–23	1.4	0.65	10[2]	F7–F9	—[18]	35
— operating theatres	17–19	1.8	0.8	17–19	1.8	0.8	0.65–1.0 $m^3 \cdot s^{-1}$	F9	—[18]	30–35
Hotels:										
— bathrooms	20–22	1.2	0.25	23–25	1.2	0.25	12[2]	F5–F7	150	40
— bedrooms	19–21	1.0	1.0	21–23	1.0	1.2	10[2]	F5–F7	50/100	20–30
Ice rinks	12	—	—	—	—	—	3 ACH	G3	—[20]	40–50
Laundries:										
— commercial	16–19	1.8	0.85	—[12]	—	—	—[21]	G3–G4	300/500	45
— launderettes	16–18	1.6	1.15	20–22	1.6	0.65	—[21]	G2–G3	300	45–50
Law courts	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F5–F7	300	25–30
Libraries:										
— lending/reference areas[22]	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F5–F7	200	30–35
— reading rooms	22–23	1.1	1.0	24–25	1.1	0.65	10[2]	F5–F7	500[23]	30–35
— store rooms	15	—	—	—	—	—	—	F6–F8	200	—
Museums and art galleries:										
— display[24]	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F7–F8	200[25]	30–35
— storage[24]	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F7–F8	50[25]	30–35
Offices:										
— executive	21–23	1.2	0.85	22–24	1.2	0.7	10[2]	F7	300–500[7]	30
— general	21–23	1.2	0.85	22–24	1.2	0.7	10[2]	F6–F7	300–500[7]	35
— open–plan	21–23	1.2	0.85	22–24	1.2	0.7	10[2]	F6–F7	300–500[7]	35
Places of public assembly:										
— auditoria[26]	22–23[1]	1.0	1.0	24–25	1.1	0.65	10[2]	F5–F7	100–150[5]	20–30
— changing/dressing rooms	23–24	1.4	0.5	23–25	1.4	0.4	10[2]	F5–F7	300	35
— circulation spaces	13–20[1]	1.8	1.0	21–25[1]	1.8	0.65	10[2]	G4–G5	200	40
— foyers[27]	13–20[1]	1.8	1.0	21–25[1]	1.8	0.65	10[2]	F5–F7	200	40
— multi-purpose halls[28]	—	—	—	—	—	—	10[2]	G4–G5	300	—
Prison cells	19–21	1.0	1.7	21–23	1.0	1.2	10[2]	F5	100[4]	25–30
Railway/coach stations:										
— concourse (no seats)	12–19[1]	1.8	1.15	21–25[1]	1.8	0.65	10[2]	G4–G5	200	45
— ticket office	18–20	1.4	1.15	21–23	1.4	0.65	10[2]	G4–G5	300	40
— waiting room	21–22	1.1	1.15	24–25	1.1	0.65	10[2]	G4–G5	200	40
Restaurants/ dining rooms	21–23	1.1	1.0	24–25	1.1	0.65	10[2]	F5–F7	50–200[5]	35–40
Retailing:										
— shopping malls	12–19[1]	1.8	1.15	21–25[1]	1.8	0.65	10[2]	G4–G5	50–300	40–50
— small shops, department stores[22]	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F5–F7	500	35–40
— supermarkets[29]	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F5–F7	750/1000	40–45

Table continues

Table 1.5 Recommended comfort criteria for specific applications — *continued*

Building/room type	Winter operative temp. range for stated activity and clothing levels*			Summer operative temp. range (air conditioned buildings†) for stated activity and clothing levels*			Suggested air supply rate / (L.s^{-1} per person) unless stated otherwise	Filtration grade‡	Maintained illuminance¶ / lux	Noise rating§ (NR)
	Temp. / °C	Activity / met	Clothing / clo	Temp. / °C	Activity / met	Clothing / clo				
Sports halls[30]:										
— changing rooms	22–24	1.4	0.55	24–25	1.4	0.35	6–10 ACH	G3	100[20]	35–45
— hall	13–16	3.0	0.4	14–16	3.0	0.35	10[2]	G3–F5	300[20]	40–50
Squash courts[30]	10–12	4.0	0.25	—	—	—	4 ACH	G3	—[20]	50
Swimming pools:										
— changing rooms	23–24	1.4	0.5	24–25	1.4	0.35	10 ACH	G3	100[20]	35–45
— pool halls	23–26[31]	1.6	< 0.1	23–26[31]	1.6	< 0.1	0–15 L·s^{-1}·m^{-2} (of wet area)	G3	—[20]	40–50
Television studios[26]	19–21	1.4	1.0	21–23	1.4	0.65	10[2]	F5–F7	—[32]	25

Notes: Except where indicated[1], temperature ranges based on stated values of met and clo and a PMV of ±0.25. Upper temperature of stated range may be increased and lower temperature decreased by approximately 1°C if PMV of ±0.5 (i.e. 90 PPD) is acceptable (see section 1.3.2). Calculation assumes RH = 50% and v_r = 0.15 m·s^{-1}. Insulation value of chair assumed to be 0.15 clo for all applications except dwellings, for which 0.3 has been assumed.

* See section 1.4.3. for additional data and variations due to different activities and levels of clothing.

† Higher temperatures may be acceptable if air conditioning is not present, see section 1.3.1.

‡ See also chapter 8, Table 8.2, which gives requirements for specific pollutants.

§ Illumination levels given thus: 200–500 indicate that the required level varies through the space depending on function and/or task. Illumination levels given thus: 300/500, indicate that one or the other level is appropriate depending on exact function. Illumination levels in this table give only a general indication of requirements. Reference must be made to the table of recommended illuminances in the SLL *Code for lighting*(33) and CIBSE/SLL Lighting Guides for design guidance on specific applications (see notes to individual entries).

¶ See also Table 1.11.

[1] Based on PMV of ±0.5

[2] Assumes no smoking. For spaces where smoking is permitted, see section 1.7.2.

[3] Based on comfort requirements for check-in staff

[4] Local illumination may be required for specific tasks

[5] Dimming normally required

[6] Follow computer manufacturers' recommendations if necessary, otherwise design for occupant comfort

[7] Refer to Lighting Guide LG7: *Office lighting*(34)

[8] Refer to The Building Regulations: Part F1: Means of ventilation(35)

[9] Podium may require special consideration to cater for higher activity level

[10] Refer to Lighting Guide LG5: *The visual environment in lecture, conference and teaching spaces*(36)

[11] The Workplace (Health, Safety and Welfare) Regulations 1992(37) require 13 °C where there is severe physical effort

[12] In the UK, air conditioning is not normally appropriate for this application. Cooling may be provided by local air jets. Some applications (e.g. steel mills, foundries) require special attention to reduce risk of heat stress

[13] As required for industrial process, if any, otherwise based on occupants' requirements

[14] Depends on difficulty of task

[15] Refer to Lighting Guide LG1: *The industrial environment*(38)

[16] Filtration should be suitable for the areas to which these spaces are connected

[17] See CIBSE Guide B(39), section 2.3.6.

[18] Refer to SLL *Code for lighting*(33)

[19] Design for clothing and activity levels appropriate to nurses

[20] Refer to Lighting Guide LG4: *Sports*(40)

[21] As required for removal of heat and moisture

[22] Based on comfort requirements of staff

[23] Study tables and carrels require 500 lux

[24] Conditions required for preservation/conservation of exhibits may override criteria for human comfort; abrupt changes in temperature and humidity should be avoided.

[25] Critical conservation levels may apply, refer to Lighting Guide LG8: *Lighting in museums and art galleries*(41)

[26] Performers may have wider range of met and clo values than audience, along with higher radiant component, necessitating special provision

[27] Dependent on use

[28] Design for most critical requirement for each parameter

[29] Special provision required for check-out staff to provide conditions as for small shops

[30] Audience may require special consideration depending on likely clothing levels

[31] 2 °C above pool water temperature, to a maximum of 30 °C

[32] Depends on production requirements

Table 1.6 Location of detailed guidance to environmental criteria

Parameter	Application and conditions	Section or table reference
Temperature	Known application, normal conditions Other levels of clothing and/or activity	Table 1.5 Section 1.3.1
Humidity	Relating to temperature Relating to comfort Relating to static electricity	Section 1.3.1.3 Section 1.3.1.3; ch. 8, section 8.3 Section 1.11.3; ch. 8, section 8.3
Outdoor air supply	Known application, odour sources unknown Specific pollutants, exposure limits Specific pollutants, known emission rates, design exposure limits	Table 1.5 and section 1.7.2 Chapter 8, section 8.4.2 Chapter 8, section 8.4.4
Filter selection	—	Table 1.5; ch. 8, section 8.4.3.2
Visual criteria	—	Table 1.5 and section 1.8
Noise	—	Table 1.5 and section 1.9
Vibration	—	Section 1.10
Electromagnetic fields	—	Section 1.11.1
Ionisation	—	Section 1.11.2
Static electricity	—	Section 1.11.3

1.4.2 Summer design temperatures and overheating criteria for free-running buildings in the UK

1.4.2.1 Introduction

Table 1.5 provides guidance for indoor temperatures for buildings with full year round temperature control. However the guidance is not always applicable to buildings without cooling or air conditioning systems under summertime operation. For free-running* modes, such as non-air conditioned buildings operating in summer, higher internal temperatures may be generally acceptable. Experience indicates, and research confirms, that people adapt over time and a temperature that may feel uncomfortably warm in a sudden short hot spell in April may be quite acceptable during warm weather in July.

The adaptive approach to comfort, see section 1.6, considers the way in which acceptable indoor conditions are related to those found outside. In the UK, in free-running (i.e. non-air conditioned) office buildings in the summer, research has shown that during warm summer weather 25 °C is an acceptable indoor temperature. At this temperature few people will be uncomfortable. As the indoor temperature rises from this design value an increasing number of people may become uncomfortable and there may be a decline in the productivity of office work and of learning in schools[26,27]. The peak temperature during the day should preferably not be more than 3 K above the design temperature, giving a benchmark maximum of 28 °C.

* Free-running can be defined as a mode of operation of a building rather than a specific building type. A building can be said to be free-running when it is not, at the time in question, consuming energy for the purpose either of heating or of cooling. Thus, typically, non-air conditioned UK buildings are in the free-running mode in summer, but not in winter. (Incidental gains from occupancy, equipment, insolation and use of energy by desk or ceiling fans are not considered in the classification.) The converse of the free-running mode is therefore the heated or cooled mode. This includes typical HVAC all the time and normal UK buildings during the heating season. The status of mixed mode buildings will vary.

1.4.2.2 Summer design conditions

For the free-running mode Table 1.7 indicates acceptable values for general summer indoor temperatures for a range of buildings.

However, in normal operation, it may not be possible to meet these summer internal design criteria under all conditions without the provision of mechanical cooling, and it is necessary to analyse the risk of overheating and aim to minimise the length and severity of any discomfort.

1.4.2.3 Overheating risk

In the past the summertime peak indoor temperature has been calculated for proposed building designs to give a measure of overheating or to decide whether cooling is required. However it is now recognised that more detailed analysis is needed, that considers both the frequency and the length of time that high temperatures may occur.

Summertime thermal performance of buildings is usually measured against a benchmark temperature that should not be exceeded for a designated numbers of hours or a percentage of the annual occupied period. The benchmark

Table 1.7 General summer indoor comfort temperatures for non-air conditioned buildings

Building type	Operative temp. for indoor comfort in summer / °C	Notes
Offices	25	Assuming warm summer conditions in UK
Schools	25	Assuming warm summer conditions in UK
Dwellings:		
— living areas	25	Assuming warm summer conditions in UK
— bedrooms	23	Sleep may be impaired above 24 °C
Retail	25	Assuming warm summer conditions in UK

temperature is usually related to the likelihood of discomfort, although it may be related to other factors, such as productivity or health. When the benchmark temperature is exceeded the building is said to have 'overheated' and if this occurs for more than the designated amount of time the building is said to suffer from 'overheating'. Accordingly, a design target for the assessment of overheating risk is set and this is called the overheating criterion.

1.4.2.4 Benchmark summer temperatures and overheating criteria

There has been little generally accepted UK guidance on benchmark summer peak temperatures or overheating criteria for use in the design of non-air conditioned buildings or spaces, with the exception of schools, where guidance is given in DfES Building Bulletin BB87(30), to which standard Building Regulations Approved Document L2(28) now refers.

CIBSE has undertaken considerable consultation and research on the impact of climate change on the indoor environment and on weather data. CIBSE TM36(29) provides an extensive discussion on overheating criteria, from which the guidelines given in Table 1.8 have been developed. This table gives guideline benchmark summer peak temperatures and overheating criteria for use in design for three non-air conditioned building types: offices, schools and dwellings. However, given the probable impact of climate change, this guidance will be kept under review.

It is recommended that the CIBSE Design Summer Years (DSYs) are used to assess the overheating risk (see section 5.10.4.1 for guidance on the assessment of overheating risk) as these provide a more stringent test of overheating risk than do the CIBSE Test Reference Years (TRYs) as the peak summer temperatures in the former are significantly warmer than those in the latter. CIBSE Design Summer Years are available for 14 UK locations, see chapter 2, section 2.1.2.

Table 1.8 Benchmark summer peak temperatures and overheating criteria

Building	Benchmark summer peak temp. / °C	Overheating criterion
Offices	28	1% annual occupied hours over operative temp. of 28 °C
Schools	28	1% annual occupied hours over operative temp. of 28 °C
Dwellings:		
— living areas	28	1% annual occupied hours over operative temp. of 28 °C
— bedrooms	26	1% annual occupied hours over operative temp. of 26 °C

* DfES Building Bulletin BB87(30) recommends an allowable overheating criterion of 80 occupied hours in a year over an air temperature of 28 °C.

Notes:

(1) It is reasonable to calculate the percentage of occupied hours over a year to reflect true hours of occupation, e.g. 08:00–18:00, and to allow for 5-, 6- or 7-day working as appropriate.

(2) It is recommended that the overheating criteria be assessed against the CIBSE Design Summer Years (DSYs) using the calculation methods recommended in chapter 5, section 5.10.4.1. It is incumbent upon the designer to ensure that any software used for the purpose of predicting overheating risk is validated for that purpose and operated in accordance with quality assurance procedures described in chapter 5.

1.4.2.5 Peak indoor temperatures in operation

Although design can predict and can seek to limit the length and degree of overheating, in operation temperatures will exceed design values during hot spells. During hot summers, internal temperatures may rise above the design temperature and could also rise above the benchmark summer peak temperatures for periods of the day. It then becomes the responsibility of the building owner/operator to recognise this situation and to act to minimise the length and severity of any discomfort.

To date there has been little guidance on peak indoor temperatures in operation. CIBSE has reviewed this area and the following summarises CIBSE guidance on peak temperatures for some types of non-air conditioned UK building in summer.

Offices (non-air conditioned)

Table 1.7 recommends 25 °C as an acceptable summer indoor design operative temperature for non-air conditioned office buildings, and Table 1.8 recommends limiting the expected occurrence of operative temperatures above 28 ºC to 1% of the annual occupied period (e.g. around 25–30 hours).

Between 25 ºC and 28 ºC increasing numbers of people may feel hot, uncomfortable and show lower productivity. Indoor operative temperatures that stay at or over 28 ºC for long periods of the day will, except during prolonged hot spells, result in dissatisfaction for many occupants.

Good practice ways to reduce discomfort for occupants of office buildings in hot summer conditions when indoor operative temperatures rise above 25 ºC include:

— relaxation of formal office dress to encourage individual adaptation to conditions

— individual control over the thermal environment, where practicable, such as by opening windows, the use of blinds, or moving out of sunny areas

— flexible working so people can work at more comfortable times

— availability of hot* or cold drinks

— increased air movement; e.g. the cooling effect of local fans can be equivalent to reducing the operative temperature by around 2 K.

Indoor operative temperatures of 30 ºC or more are rarely acceptable to occupants of office buildings in the UK.

Schools

The guidance given above for offices is also applicable to non-air conditioned schools, with the current probable exception of flexible working.

* Hot drinks can be very beneficial in hot weather because they trigger the sweating response

Retail premises

Many retail premises are air conditioned, as the additional heat gains due to display glazing and lighting necessitate this. Non-air conditioned retail outlets are of two main kinds: small premises and large 'retail sheds'.

The good practice guidelines outlined for offices above are applicable and, in particular, the use of additional air movement is likely to be beneficial. Both increased natural ventilation and the use of local fans are recommended and should be considered as part of the design process. Further guidance on natural ventilation strategies is given in CIBSE AM10[31].

Retail sheds can be hot in hot spells and could exceed 30 °C in the occupied zone for short periods of time. The majority of customers visit for short periods and will tend to have adapted to external conditions by dressing accordingly. For staff, some further relaxation in dress code may be acceptable to management, which could enable these temperatures to be tolerated for short periods. Conditions for staff in fixed locations such as the checkout and enquiry areas should be particularly considered, as these locations limit the opportunity for individual adaptation.

Industrial buildings

The guidance given above for offices and retail premises is generally applicable to non-air conditioned industrial buildings. However, depending on the processes carried out within the space, other criteria, such as heat stress, may need to be considered.

Dwellings

The individual has more freedom to adapt to conditions at home than at work. Bedroom temperatures are likely to be more critical than living area temperatures as most people find sleeping difficult in the heat. Research[32] has shown that high bedroom temperatures can result in poor sleep quality and poor performance on the following day at work. This is further discussed in section 1.6. The use of shading to reduce solar gain during the day and of night-time ventilation when feasible can reduce internal night-time temperatures. Additional air movement from quiet fans can also help improve comfort. CIBSE TM36[29] provides further discussion and relevant case studies.

1.5 Other factors potentially affecting comfort

1.5.1 Temperature variations

Relatively slow variations in temperature produce results which are directly predictable at any time from the steady state relationships between temperature and subjective sensations[42]. As long as changes in operative temperature are within the ranges shown in Table 1.5, no significant discomfort should result. For guidance on day-to-day changes in response to outdoor temperature see section 1.6.

1.5.2 Age

Studies[17,43–46] have found that at a given activity and clothing level the thermal environments preferred by older people did not differ significantly from those preferred by younger ones. The lower metabolism in older people is compensated by a lower evaporative loss[47].

1.5.3 Gender

Experiments[8,17,44] have shown that at the same activity and clothing levels men and women preferred almost the same thermal environments. Women's skin temperature and evaporative loss are slightly lower than those for men, and this balances the slightly lower metabolic rate of women. The reason that women sometimes prefer higher ambient temperatures to those preferred by men may be explained by the lower thermal insulation provided by some clothing ensembles worn by women.

1.5.4 Colour of surfaces and lighting

Studies[12,48] have found no significant relationship between the colour of interior surfaces or lighting and perceptions of thermal comfort.

1.5.5 Occupants' state of health

There is limited knowledge of the comfort requirements for people who are ill, disabled, undergoing treatments involving drugs etc. The comfort of immobilised people will depend on the insulation of their clothes or bed-clothes, along with clinical factors related to the nature of the illness or disability and the treatment regime. Recent studies have indicated that disabled people are more varied in their thermal responses than is the general population. It is therefore important that they should be given individual control over their thermal environment if it is practical to provide it[49,50].

1.5.6 Vertical air temperature differences[51–53]

A relationship between vertical air temperature differences and the percentage of occupants who are likely to be dissatisfied is given in Figure 1.4. Studies have focused on a rise in temperature with distance from the floor. In general, it is recommended that the gradient should be not more than 3 K between ankles and head[7]. If air velocities are higher at floor level than across the upper part of the body then a maximum gradient of 2 $K \cdot m^{-1}$ is recommended.

1.5.7 Horizontal air temperature differences

The temperature at any given position within the occupied zone of a room should normally be within the ranges specified in Table 1.5. However, in rooms where people are free to choose their location, a variety of thermal environment may be advantageous, and increase the likelihood of comfort.

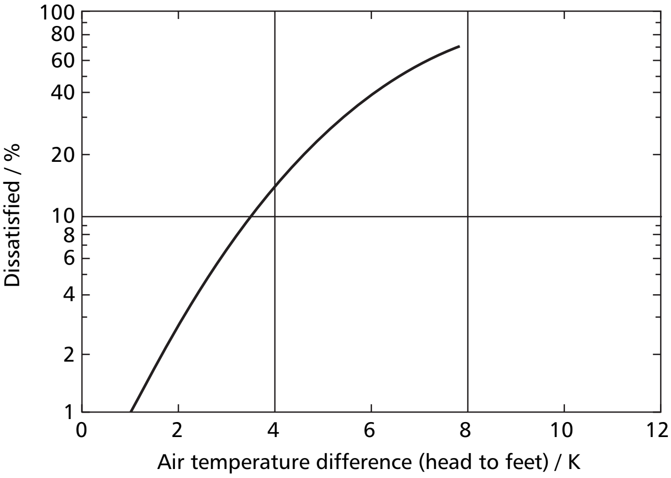


Figure 1.4 Percentage dissatisfied as a function only of vertical air temperature difference between head and ankles[51]

1.5.8 Warm or cold floors

Local discomfort of the feet can be caused by the floor temperature being too high or too low.

For rooms in which occupants spend much of their time with bare feet (e.g. swimming pools, bathrooms, dressing rooms etc.) or with their bodies in contact with the floor (e.g. gymnasia, kindergartens etc.), studies have found that the flooring material is important[54,55]. Comfort ranges for surface temperature of some typical flooring materials are given in Table 1.9.

Table 1.9 Comfortable temperature ranges for typical flooring materials

Material	Surface temperature range / °C
Textiles	21–28
Pine wood	21.5–28
Oak wood	24.5–28
Hard thermoplastic floor covering	24–28
Concrete	26–28

Note: data from laboratory experiments; 0.6 clo, 10 min standing exposures, subjects otherwise in thermal comfort (operative temperature ≈24 °C). The range is that in which fewer than 15% would be expected to report foot discomfort

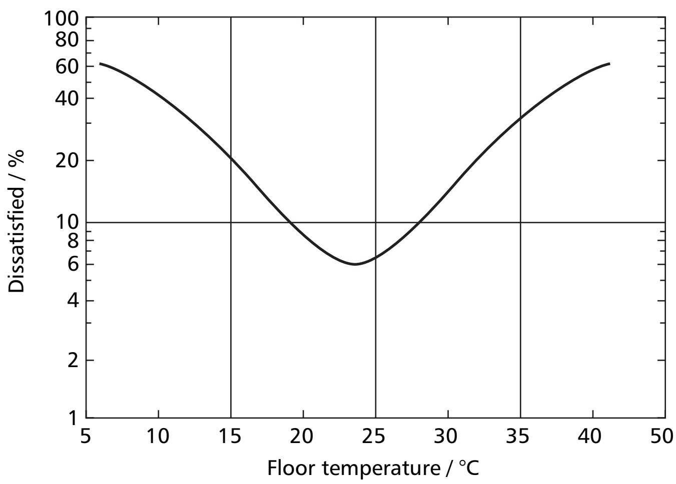


Figure 1.5 Percentage dissatisfied as a function of floor temperature only[51]; derived from climate laboratory experiments; sedentary and walking subjects combined, college age and elderly, light footwear, 3 h exposures

For floors occupied by people wearing normal footwear, flooring material is unimportant. Studies have found an optimal surface temperature of 25 °C for sedentary and 23 °C for standing or walking persons[56–58]. The floor temperature is not critical. Figure 1.5 shows percentage dissatisfied as a function of floor temperature for seated and standing people combined. In general, it is recommended that floor temperature should be in the range 19–29 °C. (For the design of floor heating systems, BS EN 1264[59] suggests that a surface temperature of 29 °C is appropriate.)

1.5.9 Asymmetric thermal radiation

There are three cases of asymmetric radiation that may lead to discomfort:

— *local cooling*: radiation exchange with adjacent cool surfaces, such as cold windows

— *local heating*: radiation from adjacent hot surfaces, such as overhead lighting or overhead radiant heaters

— *intrusion of short-wavelength radiation*: such as solar radiation through glazing.

Radiant temperature asymmetry is defined as the difference between the plane radiant temperatures on opposite sides of the human body. The plane radiant temperature is the radiant temperature resulting from surfaces on one side of a notional plane passing through the point or body under consideration. The measurement and calculation of radiant temperature asymmetry are dealt with in BS EN ISO 7726[4].

The radiant temperature asymmetry in the vertical direction is calculated from the difference in plane radiant temperature between the upper and lower parts of the space with respect to a small horizontal plane, taken as 0.6 m above the floor for a seated person and 1.1 m above the floor for a standing person.

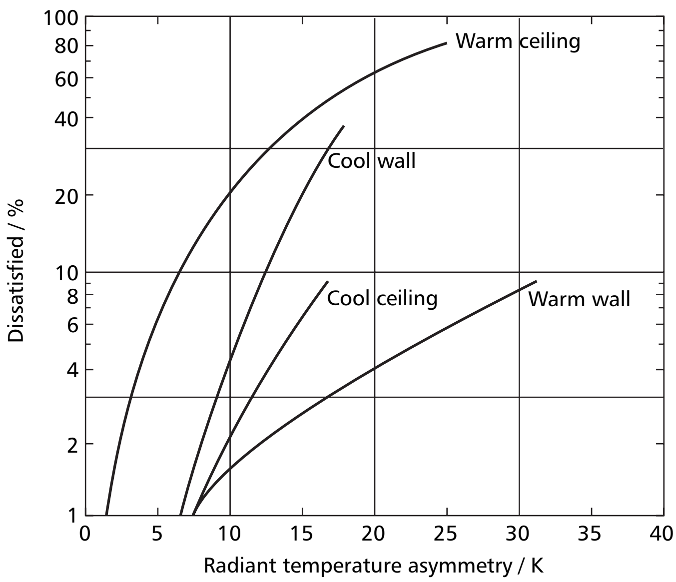


Figure 1.6 Percentage dissatisfied due to asymmetric radiation only[60,61]; data from climate chamber experiments, air temperature adjusted to compensate for cool or warm wall or ceiling; subjects seated, thermally neutral, 0.6 clo, resultant temperature 25 °C, 30 min. exposures

In the horizontal direction it is the difference between plane radiant temperatures in opposite directions from a small vertical plane with its centre located 0.6 m (seated) or 1.1 m (standing) above the floor.

Figure 1.6[60,61] can be used to predict dissatisfaction where surface temperatures are known and radiant temperature asymmetry can be calculated.

It is recommended that radiant temperature asymmetry should contribute no more than 5% dissatisfied. Hence, in the vertical direction radiant temperature asymmetry (warm ceiling) should be less than 5 K, and in the horizontal direction (cool wall) less than 10 K. Similarly, for a cool ceiling the maximum recommended radiant temperature asymmetry is 14 K and for a warm wall 23 K.

It appears that comfort conditions in rooms with chilled ceiling and displacement ventilation conform to this advice, and with that for temperature gradients[62].

1.5.10 Low temperature radiant heating systems

It may not be possible to provide an economic ceiling-mounted radiant heating system while keeping the radiant temperature asymmetry within 5 K. For such systems it is permissible to design for a maximum radiant temperature asymmetry of 10 K, although this could lead to 20% dissatisfaction. Based on this criterion, Figure 1.7 suggests design limits of downward emission from horizontal panels for various head to panel distances.

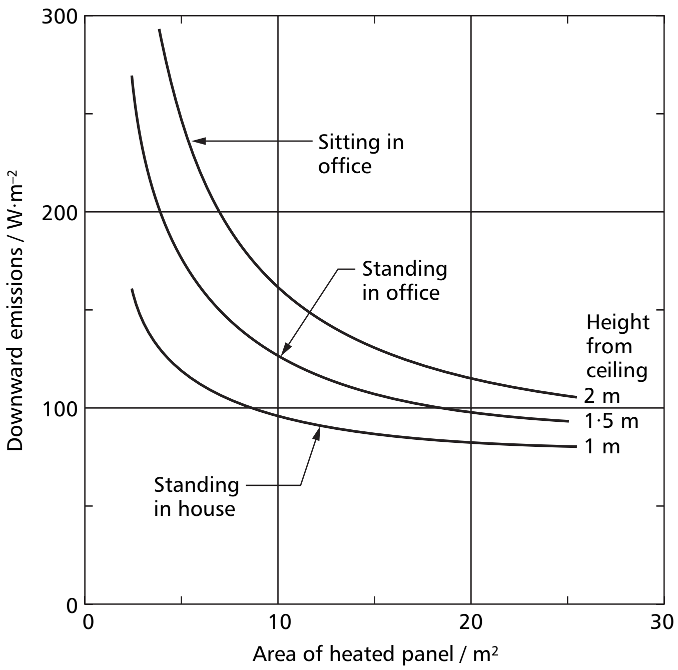


Figure 1.7 Downward heat emission from centre of a square low-temperature radiant panel

1.5.11 Lighting

Another possible source of radiant heat is lighting. Fluorescent lamps are relatively cool. For example, for an illuminance of 1000 lux, fluorescent lighting would increase the mean radiant temperature such that it would be necessary to reduce the air temperature by 0.25–0.5 K, compared to the same room without lighting, in order to maintain the same operative temperature. For the same illuminance provided by tungsten filament lamps, a reduction in the air temperature of about 1.5 K would be required. Radiant temperature asymmetry could be a problem in the latter case.

1.5.12 Short-wave radiation

When solar radiation falls on a window the transmitted short-wave radiation is almost all absorbed by the internal surfaces. This raises the temperature of these surfaces which, as well as contributing to the convective gain, augments the mean radiant temperature. In comfort terms, the most significant component is the direct radiation falling on occupants near the window[63].

Figure 1.8 shows the elevation of mean radiant temperature and operative temperature due to incident short-wave radiation. Clothing absorptance of short-wave radiation will depend on the colour and the texture but can be taken as 0.7 for typical summer wear and 0.8 for the darker clothing typically worn in cold climates.

People exposed to direct solar gain may also experience discomfort due to glare and veiling reflections. Where extensive glazed areas face other than north, it will be necessary to consider the provision of solar control devices.

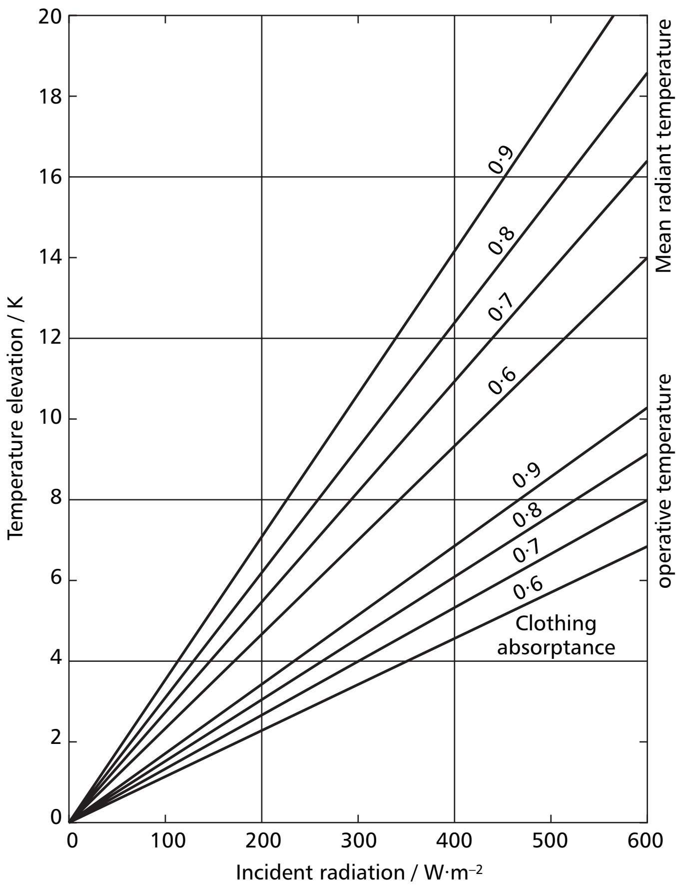


Figure 1.8 Effect of short-wave radiation on the mean radiant and operative temperatures

1.6 The adaptive approach and field-studies of thermal comfort

1.6.1 Introduction

The adaptive approach[64] to thermal comfort has been developed from field studies of people in daily life. While lacking the rigour of laboratory experiments, field studies have a more immediate relevance to ordinary living conditions; for examples see references 20, 65 and 66.

The adaptive method, unlike the heat-exchange method, does not require knowledge of the clothing insulation and the metabolic rate in order to establish the temperature required for thermal comfort. Rather it is a behavioural approach, and rests on the observation that people in daily life are not passive in relation to their environment, but tend to make themselves comfortable, given time and opportunity. They do this by making adjustments (adaptations) to their clothing, activity and posture, as well as to their thermal environment.

1.6.2 Occupant control

Adaptation is assisted by the provision of control over the thermal environment. So where practicable, convenient and effective means of control should be provided, sufficient for the occupants to adjust the thermal environment to their own requirements. This 'adaptive opportunity'[67] may be provided, for instance, by ceiling fans and openable windows in summertime, or by local temperature controls in winter. A control band of ±2 K (or an equivalent band of air speed)[68] should be sufficient to accommodate the great majority of people. Individual control is more effective in promoting comfort than is group-control.

1.6.3 Customary thermal environments and comfort

People tend to become well-adapted to thermal environments they are used to, and find them comfortable. The building services engineer should therefore aim to provide a thermal environment that is within the range customary for the particular type of accommodation, according to climate, season and cultural context. The values of the operative temperature given in Table 1.5 can be regarded as estimates of such customary temperatures, established from professional experience, and appropriate in buildings that are heated or air conditioned, and set in temperate climates.

1.6.3.1 Drift of comfort conditions

These customary temperatures, each intended to be a group-optimum for comfort, are not fixed, but are subject to gradual drift in response to changes in both outdoor and indoor temperature, and are modified by climate and social custom. A departure from the current group-optimum temperature (referred to hereafter as the 'comfort temperature'), if suddenly imposed upon the occupants, is likely to provoke discomfort and complaint, while a similar change, occurring gradually over several days or longer, would be compensated by a corresponding change in clothing, and would not provoke complaint.

1.6.3.2 Dress codes

The extent of seasonal variation in indoor temperature that is consistent with comfort depends on the extent to which the occupants wear cool clothing in summertime and warm clothing in wintertime. Some dress codes restrict this freedom, and therefore have consequences for thermal design, for services provision, and consequently for energy consumption. Organisations that have dress codes should be made aware of this, and be encouraged to incorporate adequate seasonal flexibility.

1.6.3.3 Temperature drift during a day

Field research can indicate the extent and rapidity of clothing adaptation, and hence of the temperature drifts that are acceptable. During any working day, field-studies have found rather little systematic clothing-adjustment in response to variations in room temperature[69], so it is desirable that the temperature during occupied hours in any day should not vary much from the comfort temperature. Temperature drifts within ±1 K of the comfort temperature would attract little notice, while ±2 K would be likely to attract attention and could result in mild discomfort among a small proportion of the occupants.

1.6.3.4 Temperature drift over several days

Clothing and other adjustments in response to day-on-day changes in temperature, such as occur when a building is responding to weather and seasonal changes, occur quite gradually[69–71], and take a week or so to complete. So it is desirable that the day-to-day change in mean indoor operative temperature during occupied hours should not normally exceed about 1 K, nor should the cumulative change over a week exceed about 3 K. These figures apply to sedentary or lightly active people.

1.6.4 Probable comfort temperatures in relation to climate

During the summer months many buildings in temperate climates are free-running (i.e. not heated or cooled). The temperatures in such buildings will change according to the weather outdoors, as will the clothing of the occupants. Even in air conditioned buildings the clothing has been found to change according to the weather[22,70]. As a result the temperature that people find comfortable indoors also changes[22,65,72]. Guidance for limits on indoor temperature may therefore be related to the outdoor temperature[1,73]. The relationship between indoor comfort and outdoor temperature has usually been expressed in terms of the monthly mean of the outdoor temperature[1,72]. Important variations of outdoor temperature do however occur at much shorter than monthly intervals. Adaptive theory suggests that people respond on the basis of their thermal experience, with more recent experience being more important. A running mean of outdoor temperatures, weighted according to their distance in the past, is therefore more appropriate than a monthly mean.

1.6.4.1 Exponentially weighted running mean outdoor temperatures

An exponentially weighted running mean of the daily mean outdoor air temperature, θ_{rm}, is an appropriate expression of the outdoor temperature, and is calculated from the formula:

$$\theta_{rm} = (1 - \alpha_{rm})\,[\theta_{e(d-1)} + \alpha_{rm}\,\theta_{e(d-2)} + \alpha_{rm}^{2}\,\theta_{e(d-3)}\,...] \tag{1.8}$$

where α_{rm} is a constant between 0 and 1 which defines the speed at which the running mean responds to outdoor temperature, $\theta_{e(d)}$ is the daily mean outdoor temperature (°C) for the previous day, $\theta_{e(d-1)}$ is the daily mean outdoor temperature (°C) for the day before that, and so on.

The use of an infinite series would be impracticable were not equation 1.8 reducible to the form:

$$\theta_{rm(n)} = (1 - \alpha_{rm})\,\theta_{e(d-1)} + \alpha_{rm}\,\theta_{rm(n-1)} \tag{1.9}$$

where $\theta_{rm(n)}$ is the running mean temperature (°C) for day n, $\theta_{rm(n-1)}$ is the running mean temperature (°C) for day ($n-1$), and so on.

So if the running mean has been calculated (or assumed) for one day, then it can be readily calculated for the next day, and so on.

1.6.4.2 Application to offices

Data applicable to Europe are available from extensive surveys of office workers(68,74). A value in the region of 0.8 was found to be suitable for α_{rm} in the running mean temperature, a value previously found suitable for data from the UK(71). This value suggests that the characteristic time subjects take to adjust fully to a change in the outdoor temperature is about a week.

Bands within which comfortable conditions have been found to lie are shown in relation to the running mean outdoor temperature in Figure 1.9(74), both for the free-running mode of operation and for the heated or cooled mode. Comfortable conditions for mixed mode operation lie within and between these bands. The bands indicate the indoor temperatures within which people readily adapt, in relation to the outdoor temperature. A thermally successful building is one whose indoor temperatures change only gradually in response to changes in the outdoor temperature (see 1.6.2.3 and 1.6.2.4 above), and rarely stray beyond these bands. The limits of the bands are given by the following equations.

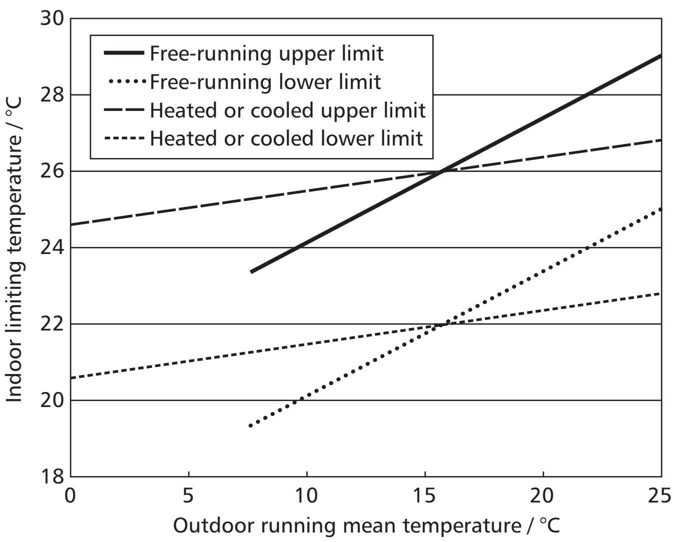


Figure 1.9 Bands of comfort temperatures in offices related to the running mean of the outdoor temperature; separate bands are shown for buildings in the free-running and the heated and cooled modes (from field surveys in Europe(51)

For free-running operation:

(*a*) upper margin:

$$\theta_{com} = 0.33\,\theta_{rm} + 20.8 \tag{1.10}$$

(*b*) lower margin:

$$\theta_{com} = 0.33\,\theta_{rm} + 16.8 \tag{1.11}$$

For heated or cooled operation:

(*a*) upper margin:

$$\theta_{com} = 0.09\,\theta_{rm} + 24.6 \tag{1.12}$$

(*b*) lower margin:

$$\theta_{com} = 0.09\,\theta_{rm} + 20.6 \tag{1.13}$$

where θ_{com} is the comfort temperature (°C)

Example 1.1: Naturally ventilated office in summer

For the assessment of the adequacy of the building in summer, the upper margin of the free-running zone is examined. This line may be used to indicate probable upper limit of the comfort temperature. In the UK the running mean outdoor temperature rarely exceeds 20 °C. At this temperature the upper limit of the band is 27.4 °C. So the temperature during occupied hours should preferably not exceed this value. Operative temperatures drifting a little above this value might attract little notice, but temperatures 2 K or more above it would be likely to attract increasing complaint. On a more normal summer day the running mean outdoor temperature would be about 15 °C. This would give a value of 25.8 °C, and the indoor temperature should preferably not be higher than this. Again, temperatures a little above this value would attract little notice, while temperatures more than about 2 K above the line would be likely to attract increasing complaint.

Expected percentages of occupants experiencing discomfort have sometimes been estimated, e.g. reference 1, but the percentage varies from building to building, depending on where its comfort temperature lies within the band, and on the adaptive opportunity it affords(75). Temperatures below these values would be found satisfactory provided the advice on within-day and day-on-day temperature changes (section 1.6.3 above) is observed.

1.6.4.3 Houses in summer

There are insufficient data to provide similar advice for houses. Oseland(76) among others has suggested that people are less sensitive to temperature changes in their own home than at work, and in general people have more adaptive opportunity at home. However, attention should

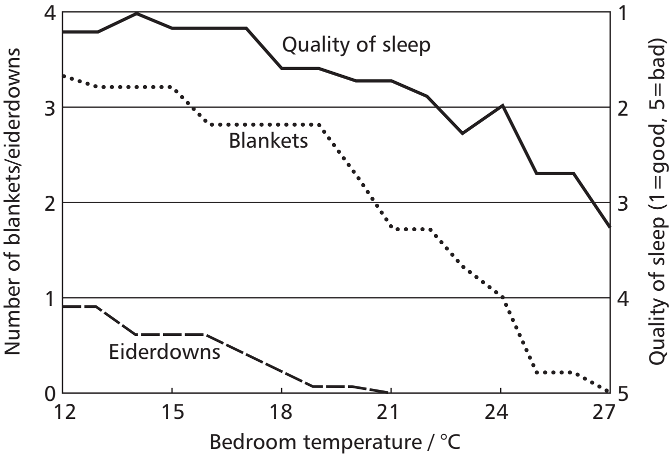


Figure 1.10 Bedclothing and sleep quality; data collected from UK subjects showing the drop in the number of bedclothes used as the bedroom temperature increases and the drop in quality of sleep above 24 °C when all bedclothes except the sheet are shed and little further adaptation is possible[49]

be given to the bedroom temperature at night. Available field study data for the UK[69] show that thermal discomfort and quality of sleep begin to decrease if the bedroom temperature rises much above 24 °C, see Figure 1.10. At this temperature just a sheet is used for cover. It is desirable that bedroom temperatures at night should not exceed 26 °C unless ceiling fans are available.

1.7 Determination of required outdoor air supply rate

1.7.1 General

For consideration of indoor air quality, see chapter 8, section 8.4

Ventilation requirements for a wide range of building types are summarised in Table 1.5. Detailed information on specific applications is given in chapter 2 of CIBSE Guide B[39]. For some industrial applications outdoor air may be required both to dilute specific pollutants and to make up the air exhausted through local extract ventilation systems, see CIBSE Guide B[39], chapter 3. Specialist advice should be sought in dealing with toxic and/or high emission pollutants.

In the following sections three methods are described for determining the outdoor air supply rate required for particular applications.

The first method (see section 1.7.2) is prescriptive, providing either an outdoor air supply rate per person or an air change rate, depending on the application. These values are based primarily on chamber studies, in which all sources of odour other than body odour and/or cigarette smoke were excluded. Therefore these prescribed rates may underestimate the outdoor air supply requirements if odour sources other than body odour or smoking dominate. Examples of such situations are spaces having large areas of new floor covering, upholstery, curtains etc. or spaces in which the standards of cleaning and maintenance are less than excellent.

Method 2 (see section 1.7.3) should be used in situations where there are known pollutants being released into the space at a known rate and local extract ventilation (LEV) is not practicable. To apply method 2, it is necessary to know the appropriate concentration limits for the pollutants. Local extract should be used wherever source location permits and for all applications where risks to the health of the occupants are not acceptable. The ventilation strategy should be based on a risk assessment under the Control of Substances Hazardous to Health Regulations 1994[77]. Design guidance is given in CIBSE Guide B[39].

A further method has been suggested[78] which is intended for use where the pollution sources are known but (*a*) the emission rates of specific malodorous pollutants cannot be predicted, (*b*) their limiting concentrations are not known, or (*c*) odours are likely to result from complex mixtures of contaminants. Further research will be required to establish benchmark criteria. Ventilation rates calculated by this method will usually be higher than the prescribed rates determined using method 1. Details of this method are given in WHO publication CR 1752[79]. At the time of writing (November 2005) this method has not gained international acceptance.

1.7.2 Method 1: Prescribed outdoor air supply rates

For applications in which the main odorous pollutants arise due to human activities, e.g. body odour, it is possible to supply a quantity of outdoor air based on the number of occupants in a given space. If smoking is prohibited, as is increasingly the case, then the recommended outdoor air supply rates given in Table 1.5 apply.

Spaces in which smoking is permitted should be regarded as 'smoking rooms' and an outdoor air supply rate of 45 $L \cdot s^{-1}$ per person is suggested for such rooms. However, it should be noted that this recommendation aims only to reduce discomfort and does not ensure health protection.

1.7.3 Method 2: Specific pollutant control

1.7.3.1 Steady state conditions

For pollutants emitted at a constant rate, the ventilation rate required to prevent the mean equilibrium concentration rising above a prescribed level may be calculated from the following equation:

$$Q = \frac{P\,(10^6 - C_{pi})}{E_v\,(C_{pi} - C_{po})} \tag{1.14}$$

where Q is the outdoor air supply rate ($L \cdot s^{-1}$), P is the pollutant emission rate ($L \cdot s^{-1}$), C_{po} is the concentration of pollutant in the outdoor air (ppm), E_v is the ventilation effectiveness and C_{pi} is the limit of concentration of pollutant in the indoor air (ppm). Values for E_v are given in section 1.7.4, Table 1.10.

If the pollutant threshold is quoted in $mg \cdot m^{-3}$, the concentration in parts per million may be obtained from:

$$C_p = (C_p' \times 24.05526) / M_p \quad (1.15)$$

where C_p is the concentration of pollutant (ppm), C_p' is the concentration of pollutant by volume (mg·m^{-3}), M_p is the molar mass of the pollutant (kg·mole^{-1}). The numerical factor is the molar volume of an ideal gas (m^3·mole^{-1}) at 20 °C and pressure of 1 atmosphere. Molecular masses for pollutants are given in EH40: *Occupational exposure limits*[80] or from manufacturers' data.

If C_{pi} is small (i.e. $C_{pi} \ll 10^6$), equation 1.14 becomes:

$$Q = \frac{P \times 10^6}{E_v (C_{pi} - C_o)} \quad (1.16)$$

If the incoming air is not contaminated by the pollutant in question, this equation simplifies to:

$$Q = (P \times 10^6) / E_v \, C_{pi} \quad (1.17)$$

Where there is more than one known pollutant, the calculation should be performed for each pollutant separately. The outdoor air supply rate for ventilation is then the highest of these calculated rates.

If E_v is equal to one in equations 1.14, 1.16 and 1.17, this indicates that a substantially uniform concentration exists throughout the space. If the ventilation results in a non-uniform concentration so that higher than average concentrations occur in the inhaled air, the outdoor air supply rate would need to be increased above the value calculated by these equations.

Example 1.2

Toluene is being released at a rate of 20 mL·h^{-1}; determine the rate of ventilation required to meet the WHO *Air Quality Guidelines for Europe*[81], assuming a ventilation effectiveness (E_v) of 1.0.

The release rate of the pollutant, P = 20 mL·h^{-1}, which is equivalent to 5.56×10^{-6} L·s^{-1}.

From chapter 8, Table 8.2, the WHO AQG threshold for toluene is 68 ppb (i.e. 0.068 ppm).

Assuming there is no toluene in the outdoor air, $C_o = 0$.

Therefore, using equation 1.17:

$$Q = (5.56 \times 10^{-6} \times 10^6) / (1 \times 0.068) = 81.7 \text{ L·s}^{-1}$$

1.7.3.2 Non-steady state conditions

The ventilation rate given by equation 1.17 is independent of the room or building volume. However the volume of the space affects the time taken for the equilibrium condition to be reached. This becomes important when the emission of a pollutant occurs for a limited duration only. In such cases the ventilation rate derived from equation 1.17 will exceed that required to maintain the concentration below the specified limit.

The ratio by which the steady state ventilation rate may be reduced in these circumstances is given by:

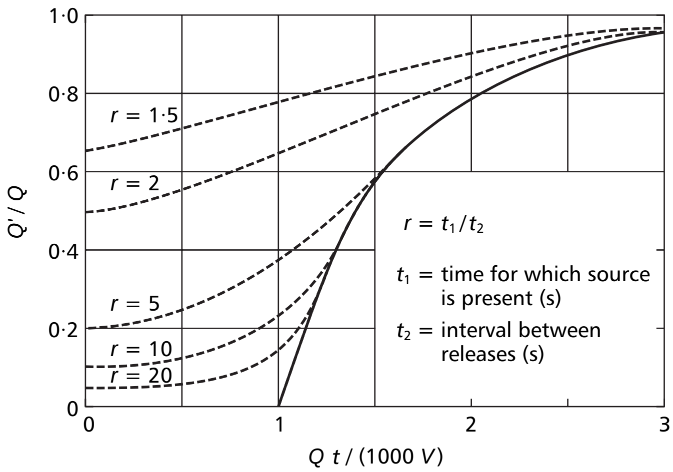


Figure 1.11 Reduction in fresh air rate for intermittent pollutant source

$$Q' / Q = f(Q \, t_p / 1000 \, V) \quad (1.18)$$

where Q' is the reduced ventilation rate (L·s^{-1}), Q is the steady state ventilation rate (L·s^{-1}), t_p is the duration of release of the pollutant (s) and V is the volume of the space (m^3).

The form of the function $f(Q \, t_p / 1000 \, V)$ is given by the solid curve in Figure 1.11. Although theoretically no ventilation is required when $(Q \, t_p / 1000 \, V) < 1.0$, some ventilation should be provided because subsequent releases of pollutant are likely to occur.

Recurrent emissions can be taken into account by considering a regular intermittent emission where the releases occur for periods of t_1 seconds at intervals of t_2 seconds. The ventilation rate ratio then becomes a function of $(Q \, t_1 / V)$ and the ratio of t_1 to t_2. The broken lines in Figure 1.11 may be used to determine (Q' / Q) where these parameters are known.

Example 1.3

If the toluene in Example 1.2 were released into a ventilated space of 160 m^3 volume over a 40 minute period each day, determine the continuous outdoor air supply rate required to maintain the concentration of toluene at or below the WHO AQG value.

Initial data: Q = 81.7 L·s^{-1}, t_p = 2400 s, V = 160 m^3.

Therefore:

$$(Q \, t_p / 1000 \, V) = (81.7 \times 2400) / (1000 \times 160) = 1.23$$

From Figure 1.11:

$$(Q' / Q) = 0.35$$

Therefore:

$$Q' = 0.35 \times 81.7 = 28.6 \text{ L·s}^{-1}$$

Example 1.4

If the toluene in Example 1.2 were periodically released for periods of 40 minutes with an interval of 20 minutes between releases, determine the appropriate outdoor air supply rate.

Initial data: $Q = 81.7\ \mathrm{L{\cdot}s^{-1}}$, $t_p = 2400$ s, $V = 160\ \mathrm{m^3}$.

As for Example 1.3:

$$(Q\ \theta_p / 1000\ V) = 1.23$$

Ratio of source duration to return interval is given by:

$$r = t_1 / t_2 = 2400 / 1200 = 2$$

From Figure 1.11, for $r = 2$:

$$(Q' / Q) = 0.75$$

Therefore:

$$Q' = 0.75 \times 81.7 = 61.3\ \mathrm{L{\cdot}s^{-1}}$$

1.7.3.3 Indoor air pollutants

See chapter 8, section 8.4.2.

1.7.4 Ventilation effectiveness

Guidance on the ventilation effectiveness for the ventilation arrangements shown in Figure 1.12 is given in Table 1.10. In each case, the space is considered as divided into two zones:

— the zone into which air is supplied/exhausted

— the remainder of the space, i.e. the 'breathing zone'.

In mixing ventilation (cases (a) and (b) in Figure 1.12), the outside air supply rates given in Table 1.10 assume that the supply zone is usually above the breathing zone. The best conditions are achieved when mixing is sufficiently effective that the two zones merge to form a single zone. In displacement ventilation (Figure 1.12(c)), the supply zone is usually at low level and occupied with people, and the exhaust zone is at a higher level. The best conditions are achieved when there is minimal mixing between the two zones. The values given in Table 1.10 consider the effects of air distribution and supply temperature but not the location of the pollutants, which are assumed to be evenly distributed throughout the ventilated space. For other types of displacement system, the ventilation effectiveness (E_v) may be assumed to be 1.0.

Table 1.10 Ventilation effectiveness for ventilation arrangements shown in Figure 1.12

Ventilation arrangement	Temp. difference (/ K) between supply air and room air, $(\theta_s - \theta_{ai})$	Ventilation effectiveness, E_v
Mixing; high-level supply and exhaust (Figure 1.12(a))	< 0	0.9 – 1.0
	0 – 2	0.9
	2 – 5	0.8
	> 5	0.4 – 0.7
Mixing; high-level supply, low- level exhaust (Figure 1.12(b))	< –5	0.9
	(–5) – 0	0.9 – 1.0
	> 0	1.0
Displacement (Figure 1.12(c))	< 0	1.2 – 1.4
	0 – 2	0.7 – 0.9
	> 2	0.2 – 0.7

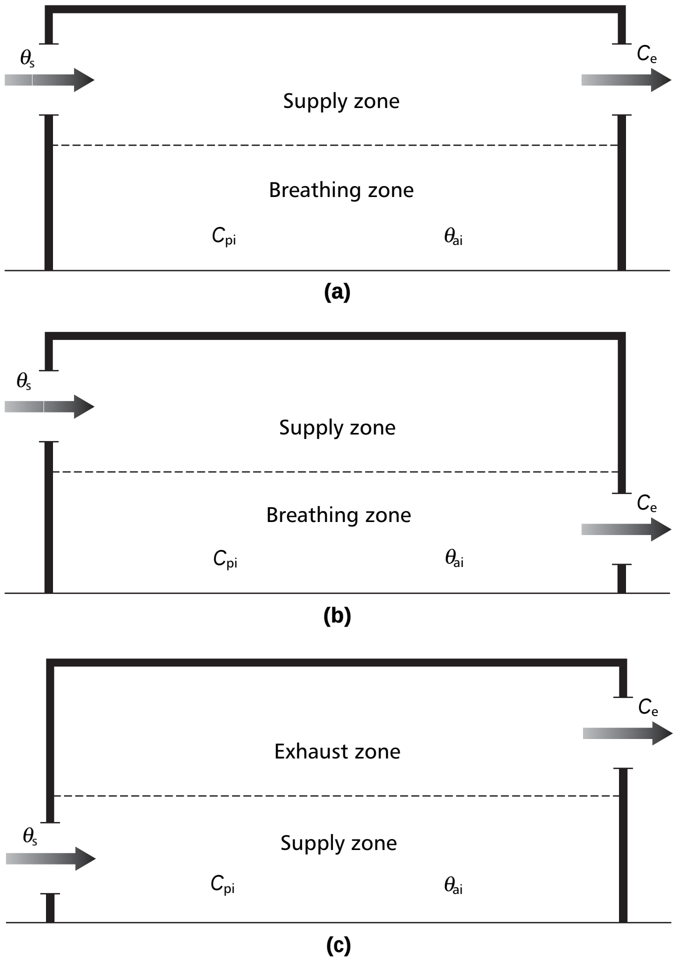


Figure 1.12 Supply/extract arrangements for ventilation; (*a*) mixing, supply and exhaust at high level, (*b*) mixing, supply at high level, exhaust at low level, (*c*) displacement

1.8 Visual environment

1.8.1 General

Lighting in a building has three purposes:

— to enable the occupant to work and move about in safety

— to enable tasks to be performed correctly and at an appropriate pace

— to create a pleasing appearance.

A satisfactory visual environment can be achieved by electric lighting alone, but most people have a strong preference for some daylight. This is supported by the Workplace (Health, Safety and Welfare) Regulations 1992[82], which require access to daylight for all workers where reasonably practicable. Where daylight is available a good design will make use of it to save energy and enhance internal appearance without glare, distracting reflections, overheating or excessive heat loss.

Good lighting can aid the avoidance of hazards during normal use of a building and in emergencies by revealing obstacles and clearly indicating exits. It makes tasks easier to perform and it can contribute to an interior that is considered satisfactory and, even, inspiring by providing emphasis, colour and variety.

The balance between lighting for performance and for pleasantness is usually dependent upon the primary purpose for which the interior is intended. For example, in an engineering workshop the primary requirement of the lighting is to enable some product to be made quickly, easily and accurately. However, a cheerful but non-distracting atmosphere produced by careful lighting can be an aid to productivity. At the other extreme, the prime purpose of the lighting in restaurants is often to produce a particular atmosphere while ensuring that the food is not difficult to see. The lighting must be matched to the context and the operational requirements in order to be successful.

1.8.2 Lighting for safety

There are two aspects of lighting for safety. The first refers to the conditions prevailing in an interior when the normal lighting system is in operation, see chapter 8, section 8.5.

The second aspect becomes apparent when the normal lighting system fails, in which circumstances the alternative/standby lighting constitutes emergency lighting.

1.8.2.1 Emergency lighting

Emergency lighting can be divided into two classes:

(*a*) standby lighting, which is intended to enable essential work to be carried out, e.g. hospital operating theatres

(*b*) escape lighting which enables people to evacuate a building quickly and safely.

For general use an absolute minimum illuminance of 0.2 lux is required along the centre line of all escape routes and 1 lux average over open areas with no defined escape routes. However, such escape routes must be permanently clear of obstructions and this can rarely be guaranteed. The Workplace Regulations 1992[82] require that emergency lighting be 'suitable and sufficient'. Detailed guidance on the design of emergency lighting systems is given in BS 5266[83] and SLL Lighting Guide LG12: *Emergency lighting design guide*[84].

1.8.2.2 Layout of lighting

Escape routes

Emergency lighting must be located so as not to create glare and confusion. For example a 'headlight' style emergency light must not be positioned such that it throws light along an escape route towards the escapees. For emergency lighting, both BS 5266[83] and LG12[84] recommend that the ratio of maximum to minimum illuminance on escape routes should not exceed 40:1. It is particularly important that luminaires providing emergency lighting are arranged to draw attention to intersections and changes of direction or level. In addition to providing light for evacuation, an organised installation of luminaires serves to give a sense of orientation and direction. This is also true for normal lighting. Emergency lighting can be integrated into the normal lighting installation if required.

Staircases

The lighting of staircases must be given careful consideration, particularly if there are changes of direction at short intervals. Lighting schemes designed for appearance only may combine visual confusion with poor visibility. For example, uplighters that throw light into the faces of people descending a staircase must be avoided. Staircases constitute zones of potential hazard and the guiding principle should be to reveal clearly the stair treads (by locating sources so that each riser is shadowed) and to make evident the location and direction of the stairway.

Workplaces

In workplaces, reflections from shiny surfaces can make objects, machinery, controls or indicators difficult to see. This may be avoided by the use of matt surfaces or arranging the lighting to avoid specular reflections towards the subject. Another hazard with rotating machinery is the stroboscopic effect produced by discharge lamps operating from an AC supply. This can be reduced by wiring adjacent luminaires to different phases of the supply or by the use of high frequency control gear. Such hazards are considered in CIBSE Lighting Guide LG1[38].

1.8.3 Lighting for performance

1.8.3.1 General

The exact relationship between visual performance and illuminance or luminance has been the subject of many investigations[85–88]. All of these studies indicate that this relationship depends upon many factors, which vary with task, individual and environment.

Where tasks involve the observation of fine detail (i.e. requiring high acuity), if the contrast (i.e. the difference in appearance of two parts of a visual field seen simultaneously or successively) is low then no amount of increase in illuminance will raise the visual performance to the level which can be attained by providing high contrast. However, performance can be improved by higher contrast, even with very low values of illuminance.

For larger task detail (i.e. requiring low acuity), the visual performance does not decline with low contrast or luminance to the same extent.

Task performance also depends on other factors such as the visual complexity of the task, task movement, the age and eyesight of the workers and the significance to the worker of the visual component of the work.

The SLL *Code for lighting*[33] contains illuminance recommendations for many different working situations. Table 1.11 summarises these in relation to different categories of visual task. It should be noted that these illuminances are intended to be measured on the appropriate working plane (i.e. horizontal, vertical or intermediate). Also, it is important to note that it is often more economic to improve performance by making the task easier through increase in apparent size of detail (e.g. by using optical aids) and improved contrast (e.g. by selecting a suitable task background) rather than by increasing illuminance.

Note that Table 1.11 is for information purposes only. Reference should be made to the comprehensive tables of specific task illuminance values in the *Code*, modified to

Table 1.11 Examples of activities/interiors appropriate for each maintained illuminance*

Standard maintained illuminance / lux	Characteristics of activity/interior	Representative activities/interiors
50	Interiors used rarely, with visual tasks confined to movement and casual seeing without perception of detail	Cable tunnels, indoor storage tanks, walkways
100	Interiors used occasionally, with visual tasks confined to movement, and casual seeing calling for only limited perception of detail	Corridors, changing rooms, bulk stores, auditoria
150	Interiors used occasionally, with visual tasks requiring some perception of detail or involving some risk to people, plant or product	Loading bays, medical stores, switchrooms plant rooms
200	Continuously occupied interiors, visual tasks not requiring perception of detail	Foyers and entrances, monitoring automatic processes, casting concrete, turbine halls, dining rooms
300	Continuously occupied interiors, visual tasks moderately easy, i.e. large details > 10 min. arc and/or high contrast	Libraries, sports and assembly halls, teaching spaces, lecture theatres, packing
500	Visual tasks moderately difficult, i.e. details to be seen are of moderate size (5–10 min. arc) and may be of low contrast; also colour judgement may be required	General offices, engine assembly, painting and spraying, kitchens, laboratories, retail shops
750	Visual tasks difficult, i.e. details to be seen are small (3–5 min. arc) and of low contrast; also good colour judgements may be required	Drawing offices, ceramic decoration, meat inspection, chain stores
1000	Visual tasks very difficult, i.e. details to be seen are very small (2–3 min. arc) and can be of very low contrast; also accurate colour judgements may be required	General inspection, electronic assembly, gauge and tool rooms, retouching paintwork, cabinet making, supermarkets
1500	Visual tasks extremely difficult, i.e. details to be seen extremely small (1–2 min. arc) and of low contrast; visual aids and local lighting may be of advantage	Fine work and inspection, hand tailoring, precision assembly
2000	Visual tasks exceptionally difficult, i.e. details to be seen exceptionally small (< 1 min. arc) with very low contrasts; visual aids and local lighting will be of advantage	Assembly of minute mechanisms, finished fabric inspection

* Maintained illuminance is defined as the average illuminance over the reference surface at the time maintenance has to be carried out by replacing lamps and/or cleaning the equipment and room surfaces

take account of task contrast, age of operatives, consequences of error etc. as described in the *Code*.

These recommendations do not identify the source that is required to provide these illuminances and the recommended levels may be met using either daylight or electric light. However, when using daylight it is only possible to give the criteria in terms of daylight factor since the daylight illuminance varies continuously. Detailed guidance on daylighting is given in CIBSE Lighting Guide LG10*: Daylighting and window design*(89) and the SLL *Code for lighting*(33).

1.8.3.2 Distribution of light

When considering the distribution of light in an interior, alongside the visual performance of the occupants it is important to consider both the attractiveness of the space and the energy implications of the design.

For good visual performance, both sharp changes in surface luminance and the blandness caused by washing every surface with light should be avoided. The light needed for the visual task should be provided only over the immediate task area, with the task surround at a lower level and the circulation spaces at a lower level still. The maximum illuminance ratio between adjacent areas should be not more than 3:1.

Only in exceptional circumstances therefore should the whole area of a room be lit to the illuminance recommended for the tasks undertaken. This is wasteful of energy and can lead to bland, uninteresting spaces.

It is important if spaces are to appear pleasant that the walls are well lit, along with other parts of the visual scene in a 40 degree band above and below the horizontal at the observer's position.

If the lighting is controlled by presence/absence detectors, care is needed in open spaces that egress routes remain lit even when only a small proportion of the workers are present. This is both to provide a safe means of exit and to alleviate gloomy and oppressive conditions caused by odd pools of light in a dark area.

1.8.3.3 Directional effects

Some directional effects of light make it easier to recognise the details of a task, others make recognition more difficult.

The contrasts perceived in a task depend on the reflection characteristics of its surface and on how the task is lit. Contrast is reduced if the images of bright sources, such as luminaires or the sky, are seen in shiny surfaces. This veiling effect, see section 1.8.3.4, is often most apparent when a bright source is reflected from glossy paper.

Legibility of print is sometimes seriously impaired by veiling reflections.

A similar problem occurs when using display screen equipment where the screen tends to reflect back to the viewer images of bright objects in front of the screen. For this reason screens need to be positioned to avoid the images of windows and brightly lit areas of the room from being reflected in the screen, see chapter 8, section 8.5. A full analysis of the problems involved and the solutions available is given SLL Lighting Guide LG7: *Office lighting*[34].

In general terms loss of contrast due to veiling reflections can be minimised by careful positioning of the viewer, the task and the source. If a light source lies within a certain solid angle behind the viewer then veiling reflections will occur and bright sources should not be positioned in this region. For screens which are near flat, the size of the offending zone can be determined by extending the solid angle created between the eyes and the screen as shown in Figure 1.13.

'Modelling' is the term used to describe ability of light to reveal solid form. Modelling may be harsh or flat depending on the strength of the light flow. Fairly strong and coherent modelling helps to reveal three-dimensional shapes.

Each task has special requirements and the extent to which modelling can assist perception should be determined from a combination of experience and practical trials. The details of some tasks may be revealed more clearly by careful adjustment of the direction of the light rather than by an increase in illuminance.

Surface texture and relief are normally emphasised if light is directed across the surface and subdued, or flat, if the surface is lit mainly from the front. Particular tasks should be lit to provide the maximum relevant visual information

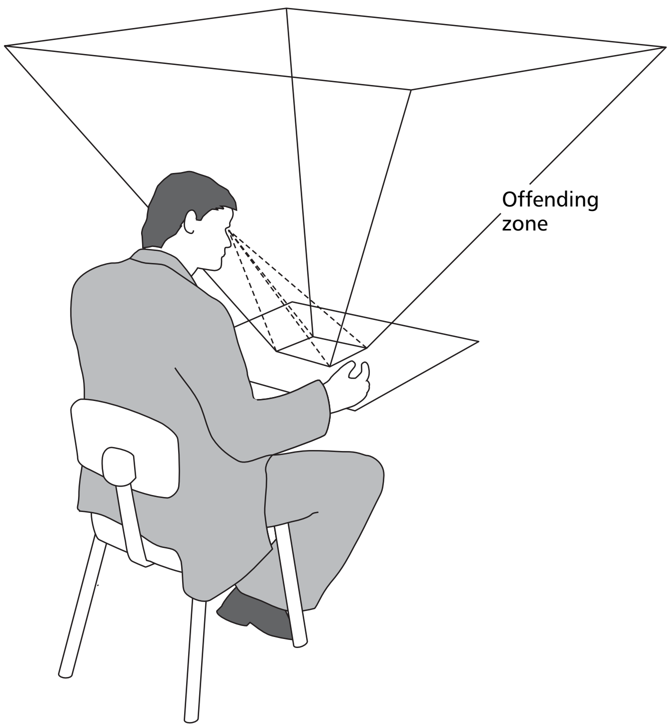


Figure 1.13 Avoidance of veiling reflections

and the best arrangement is usually achieved through adjustable luminaires or by experiment.

1.8.3.4 Disability glare

Veiling reflections directly affect the visibility of the task by reflection from the task area. However, disability glare can influence the task visibility without reflection from the task. This effect is due to light entering the eye and then being scattered in such a way that it forms a veil over the retinal image of the task.

The effect is most noticeable when the source is close to the line of sight between the observer and the task. Therefore, disability glare caused by the reflection of light sources in areas adjacent to the work is particularly troublesome.

The only way of eliminating disability glare is to separate all areas of high luminance from areas immediately surrounding the task. This is usually a matter of avoiding the use of glossy surfaces close to the task or moving the task to another location. In practice, disability glare direct from luminaires is rare in interior lighting.

1.8.3.5 Health effects

See chapter 8, section 8.5.

1.8.4 Criteria for design using daylight

The average daylight factor may be used as an initial design parameter. It is calculated as follows:

$$\text{DF} = (T\,A_w\,\alpha\;\;M)\,/\,A\,(1-R_a^{\,2}) \tag{1.19}$$

where DF is the average daylight factor (%), T is the diffuse transmittance of the glazing material including effects of dirt (see Table 1.12[89,90]), A_w is the net glazed area of the window (m^2), α is the vertical angle subtended by sky that is visible from the centre of the window (degree) (see Figure 1.14), M is the maintenance factor (see Table 1.13), A is the total area of the internal surfaces (ceiling, floor, windows and walls) (m^2) and R_a is the area-weighted average reflectance of the interior surfaces (ceiling, floor windows and walls) (Table 1.14[33]).

Table 1.12 Approximate diffuse transmittances for various glazing types (clean)[90]

Glazing type	Diffuse transmittance
Clear glazing:	
— single	0.8
— double	0.7
Double glazing, low emissivity	0.69
Double glazing with light shelf:	
— internal light shelf only	0.55
— internal and external light shelves	0.4
Double glazing with coated prismatic glazing	0.3
Double glazing with prismatic film	0.55
Double glazing with solar control mirrored louvres	0.3

Table 1.13 Calculation of maintenance factor for daylight factor

Room use	Percentage loss of daylight compared with clean glazing / %	
	Rural/suburban	Urban
Residential (private and communal); rooms with few occupants, good maintenance	4	8
Commercial, educational; rooms used by groups of people, office equipment	4	8–12
Polluted atmosphere; gymnasia, swimming pools, heavy smoking	12–24	12–24

Note: values in table must be adjusted for special conditions and exposure by applying multipliers as follows:

(1) Multiplier for special conditions: vertical glazing sheltered from rain (×3); weathered or corroded glazing (no correction for rain) (×3); leaded glass (×3)

(2) Multiplier for exposure:
- (*a*) normal exposure for location: vertical glazing (×1); inclined glazing (×2); horizontal glazing (×3)
- (*b*) exposed to heavy rain: vertical glazing (×0.5); inclined glazing (×1.5); horizontal glazing (×3)
- (*c*) exposed to snow: vertical glazing (×1); inclined glazing (×3); horizontal glazing (×4)

Maintenance factor is then given by (100 – adjusted daylight loss) / 100

Table 1.14 Reflectances for early design calculations[89]

Surface	Reflectance
Light walls and floor cavity	0.6
Medium walls and floor cavity	0.5
Dark walls and floor cavity	0.4

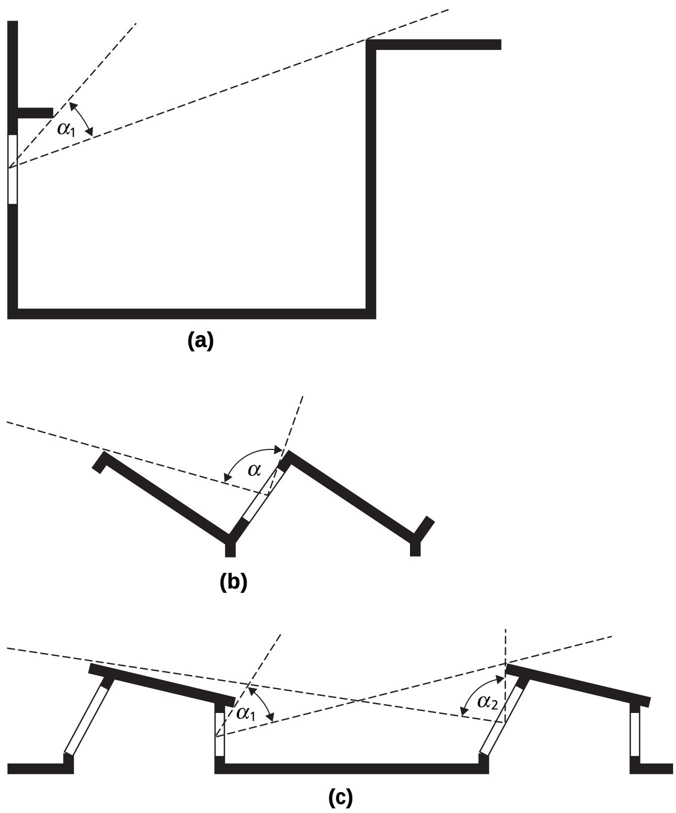


Figure 1.14 Angle of sky (α) seen from centre of window; (a) window obstructed by overhang and nearby wall, (b) rooflight obstructed by roof construction, (c) different windows facing different obstructions

If the average daylight factor exceeds 5% on the horizontal plane, an interior will look cheerfully daylit, even in the absence of sunlight. If the average daylight factor is less than 2% the interior will not be perceived as well daylit and electric lighting may need to be in constant use. BS 8206[91] recommends average daylight factors of at least 1% in bedrooms, 1.5% in living rooms and 2% in kitchens, even if a predominantly daylit appearance is not required.

Where different windows face different obstructions, see Figure 1.14(c), the average daylight factor for each window should be calculated separately and the results added together.

Equation 1.19 can also be used to provide a preliminary estimate of window size for design purposes.

The determination of daylight factor at specific points in a room is more complex, see CIBSE LG10: *Daylighting and window design*[89].

1.8.5 Lighting for pleasantness

1.8.5.1 Quantity of light

It has been established that working environments lit uniformly to less than 200 lux tend to be rated as unsatisfactory for continuous occupation[92]. Following the recommendations of BS EN 12464-1[92], the SLL *Code for lighting*[33] recommends this value as a minimum amenity level in continuously occupied spaces, even though it may not be justified on performance grounds for occupations where the visual tasks are not demanding. Studies[93] have also shown that the preference for high lighting levels declines above 2000 lux.

Considerations of energy efficiency also affect the specification of a general preferred illuminance and energy savings may be achieved by separating the task lighting from the general building lighting. Daylight is often inappropriate as task lighting but even modest daylight admission can provide satisfactory building lighting for much of the working day. Furthermore, the provision of daylight is in itself a desirable amenity.

If a combination of general and task lighting is to be employed, utilising daylight where available, it is important to note that human reaction to sunlight is less predictable than to electric lighting[94,95] and effective sunlight control may be required for working environments.

For some interiors (e.g. circulation areas) there is no obvious visual task and reference to an illuminance on a working plane is not appropriate.

1.8.5.2 Distribution of light

Selective lighting may be used to make particular objects or spaces more conspicuous. In general, the brightest elements in the field of view should be those that it is wished to emphasise. If this is not so discomfort is experienced due to the visual conflict.

For working situations, the ratio of wall illuminance to working plane illuminance should be in the range 0.5–0.8

and ceiling/working plane illuminance ratio within the range 0.3–0.9[33]. The upper limits are directed by the expectation that the working plane should appear to be more strongly illuminated than either the walls or the ceiling. The exception to this is uplighting, which creates a visual environment more akin to daylight outdoors. The provision of lighting that achieves these illuminance ratios will usually ensure an acceptable distribution of light onto the main room surfaces but will not necessarily ensure a satisfactory balance of lighting on objects within the room.

1.8.5.3 Directional effects

The directional characteristics of light may be defined by an 'illumination vector'[33]. The magnitude of the illumination vector is the difference in illuminance on opposite sides of a flat surface that is so orientated to maximise this difference. Its direction is normal to this surface, the positive direction being from the higher illuminance to the lower.

Studies of the appearance of the human face show that a flow of light from above and to one side of the face gives the most natural appearance. This flow should not be too dominant or hard shadows below the brow and nose make faces appear harsh. A dominant flow of light across the space with general light to soften shadows is recommended, especially in areas where eye to eye contact is normal, e.g. reception areas and meeting rooms.

1.8.5.4 Discomfort glare

When the brightness of a surface or luminaire is higher than recommended then people may experience visual discomfort. Discomfort glare does not directly affect the visual difficulty of tasks.

For a detailed discussion of glare and its avoidance, reference should be made to the SLL *Code for lighting*[33].

1.8.5.5 Colour of light

All sources of light, both natural and electric, differ in their spectral composition. Surface colour is produced by a combination of the wavelengths of the incident light and the spectral reflectance of the surface. Therefore, different sources will produce changes of colour of the surface. For most purposes these changes are modified by chromatic adaption, whereby the observer adapts to the particular composition of the light. However, for some tasks, the perceived colour of the surface is important and a suitable light source must be chosen.

Natural light sources such as sunlight and daylight have visible spectra which approximate to that of a black-body radiator at the appropriate temperature. Non-incandescent electric lamps generally have discontinuous spectra, an extreme example being the low-pressure sodium lamp, which is a monochromatic source. Lamps of this type are widely used for street lighting. They have very poor colour properties, particularly colour rendering. Light sources having much better colour rendering properties are required for interior lighting.

The general colour rendering index (CRI) adopted by the Commission Internationale de l'Eclairage (CIE), R_a, specifies the accuracy with which lamps reproduce colours relative to a standard source. The CRI takes values up to a maximum of 100. The SLL *Code for lighting*[33] recommends ranges for the CIE colour rendering index for specific applications.

In general, the higher the CRI the more accurately colours are reproduced with respect to the standard, and the greater is the enhancement of differences between colours. The advice of the lamp manufacturer should be sought when selecting lamps for applications where colour judgement is particularly important.

The colour rendering index is not the only parameter that needs to be considered. The colour appearance of the source is also important. For almost all interiors, the recommended light sources are nominally white in appearance with varying degrees of warmth. This degree of warmth is quantified by the correlated colour temperature (CCT) of the lamp. This is the temperature of the black-body radiator which most closely approximates to the colour appearance of the lamp.

The commonly used light sources are classified as follows:

— warm (CCT < 3300 K)

— intermediate (3300 K < CCT < 5300 K)

— cold (CCT > 5300 K).

Having selected a source with the required CCT, a source having a high colour rendering index should be chosen to ensure that surface colours are rendered to an accuracy appropriate to the application. Detailed guidance is given in the SLL *Code for lighting*[33].

For daylight, this variation is catered for naturally when bright, blue-sky conditions give way to warm tints of sunset. Where electric lighting is being used in a daylit space, lamps with a cool colour appearance give a good blend with daylight.

Refer to the *Code*[33] for advice on the selection of warm or cool source colour appearance. Satisfaction with the appearance of surface colours is likely to be increased by selecting a source having a high colour rendering index. Reference should also be made to the *Code*[33] for guidance on allowances for maintenance.

1.9 Acoustic environment

Noise affects people in different ways depending on its level and may cause annoyance, interference to speech intelligibility or hearing damage. The acoustic environment must be designed, as far as possible, to avoid such detrimental effects.

1.9.1 Sound level in a room

The sound energy emitted by a source can be quantified by its sound power level. This is a property of the source and is not affected by the characteristics of the room in which it is located. The effects of a noise source are assessed in terms of sound pressure level. The sound power level of a source may be used to calculate the resulting sound pressure level in a room, which depends on the volume of the room and the amount of absorbing material it contains. These acoustic characteristics of a room contribute to its reverberation time, i.e. the time taken for a sound to decay by 60 dB.

For a constant sound power input, the sound pressure level within a room will vary from place to place. The highest level is experienced near the noise source(s), it then decreases roughly with the square of the distance from the source until it reaches an approximately constant level. This constant level depends on the reverberation time of the room.

The calculation of noise levels within a space due to individual noise sources has been investigated by Beranek[97]. Noise from heating, ventilating and air conditioning plant is considered in CIBSE Guide B[39], chapter 5.

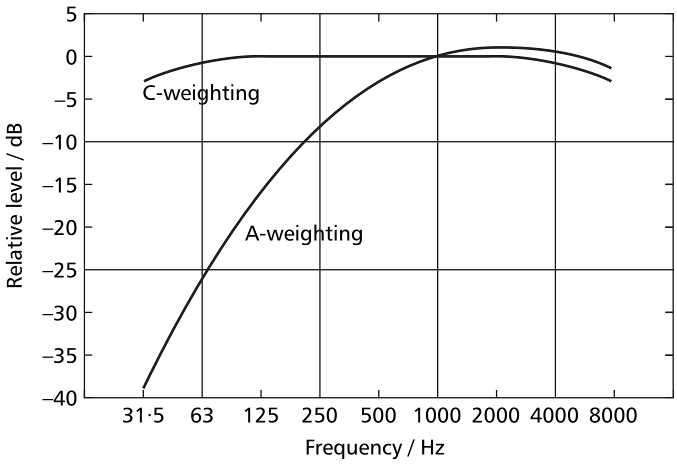


Figure 1.16 A- and C-weighting curves

1.9.2 Human hearing response

The human hearing system responds to frequencies in the range 20 Hz to 20 000 Hz. The precise range differs from person to person and hearing acuity at high frequencies tends to diminish with age due to deterioration in the receptor cells in the ear.

The response of the hearing system is non-linear and it is less sensitive to low and high frequencies than to mid-range frequencies. The sensitivity of the ear is represented by the curves of equal loudness shown in Figure 1.15. These curves have been derived by subjective experiments and show that the sensitivity of the ear varies with both sound pressure level and frequency.

The unit of loudness level is the phon. For example, the curve representing a loudness of 60 phon illustrates that a 1000 Hz note at a sound pressure level of 60 dB is perceived as being of equal loudness to a 100 Hz note at 66 dB. However, this method of assessing loudness is too complicated for everyday use.

When sound levels are measured, the variation in the sensitivity of the ear can be taken into account by incorporating frequency-weighting networks in the measuring instrument. The most widely used of these is the A-weighting network. Other networks are known as B- and C-weightings. The B-weighting has fallen into disuse so the main measurement curves are the A- and C-weighting curves, see Figure 1.16. The C-weighting gives more prominence to lower frequencies than does the A-weighting, having an approximately level response above 31.5 Hz. In contrast, the A-weighting rises gradually to 1000 Hz, thus discriminating against lower frequencies. The reason for these differences arises from the different equal loudness responses of the human ear over a range of sound pressure levels, see Figure 1.15.

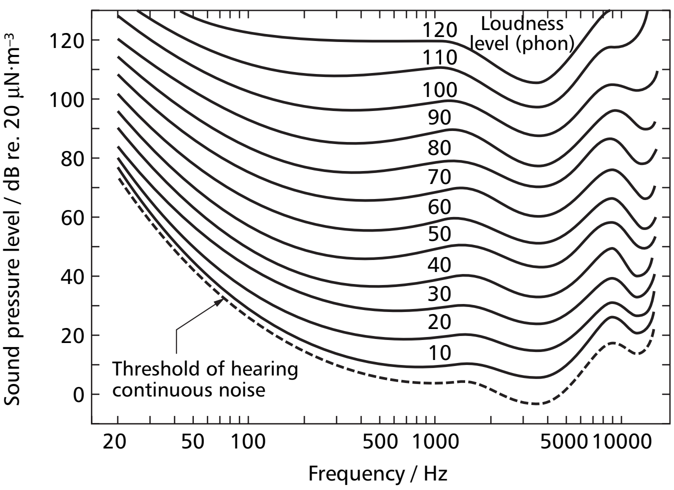


Figure 1.15 Equal loudness level contours

A-weighting was proposed in the 1930s for low sound pressure levels and C-weighting for high sound pressure levels, see Figure 1.15. This distinction has since been lost with the result that the A-weighting is now employed at sound levels for which it was not originally intended. Problems may arise if there is an excess of low frequency noise, since this will not register its full subjective impact when measured using the A-weighting.

At the time that the weighting networks were devised, the complexities of the human hearing system were not fully understood. Methods of loudness evaluation were developed in the 1970s which take account of frequency and sound level in far more detail than do the simple A- and C-weighting networks.

1.9.3 Noise assessment

The dBA measure is often used as an indicator of human subjective reactions to noise across the full range of frequencies audible to humans. This index is simple to measure using a sound level meter incorporating an A-weighting network. In addition, the measured noise spectrum can be compared with reference curves such as the NR or NC curves which aid identification of any tonal frequency components. This is the usual method of assessment for mechanical services installations[98].

Noise rating (NR) curves, see Figure 1.17, are commonly used in Europe for specifying noise levels from mechanical services in order to control the character of the noise. However, it should be noted that NR is not recognised by the International Standards Organisation or similar standardisation bodies. Noise criteria (NC) curves, see Figure 1.18, are similar to NR but less stringent at high frequencies and more stringent at low frequencies. The curves are very close at middle frequencies and, as long as there are no spectrum irregularities at low and high frequencies, they may be regarded as reasonably interchangeable. More recent developments in North America have lead to the introduction of room criterion (RC) curves[99], see see Figure 1.19.

The relationship between NR and dBA is not constant because it depends upon the spectral characteristics of the noise. However, for ordinary intrusive noise found in buildings, dBA is usually between 4 and 8 dB greater than the corresponding NR. If in doubt, both should be determined for the specific noise spectrum under consideration.

Noise from many sources, such as road traffic and aircraft, varies with time and the human response to the noise depends on its amplitude and temporal characteristics. Single number indices, such as $L_{A10,T}$, $L_{A90,T}$ and $L_{Aeq,T}$ may be used to describe these types of noise.

$L_{A10,T}$ is the A-weighted sound pressure level exceeded for 10% of the measurement period, T, which must be stated. Similarly, $L_{A90,T}$ denotes the level exceeded for 90% of the measurement period. It is often used to measure background noise levels. $L_{Aeq,T}$ is the A-weighted sound pressure level of a continuous steady sound having the same energy as the variable noise over the same time period. It is found to correlate well with subjective response to noises having different characteristics, and is used with BS 8233[100], which recommends appropriate design limits for common situations.

Noise from plant may be audible outside the building and control measures may be necessary to avoid complaints. Noise limits may also be set by the local authority. Noise emanating from industrial premises in mixed industrial and residential areas is usually assessed according to BS 4142[101].

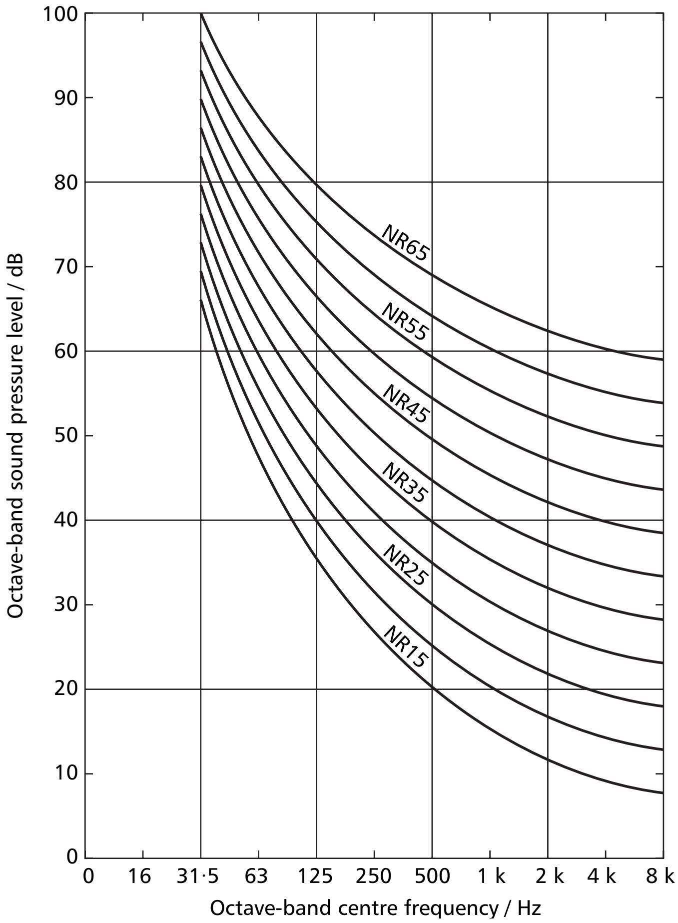


Figure 1.17 Noise rating (NR) curves

The past 15–20 years has seen significant developments in North American usage of noise criteria but these have not yet made an impact in Europe. NR was never adopted in the USA and ASHRAE no longer recommends NC as a design criterion. For some time the preferred assessment method has been room criterion (RC) curves, see Figure 1.19, as proposed by Blazier[102] in the early 1980s.

RC curves are based on actual measurements in air conditioned buildings in which the occupants are judged to have good acoustical environments. The result is a set of parallel lines falling from low to high frequencies at

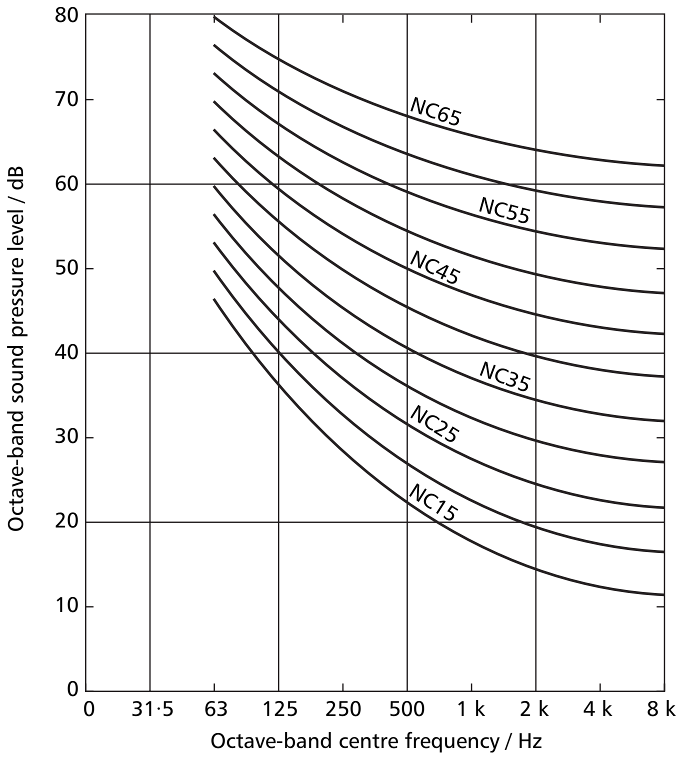


Figure 1.18 Noise criterion (NC) curves

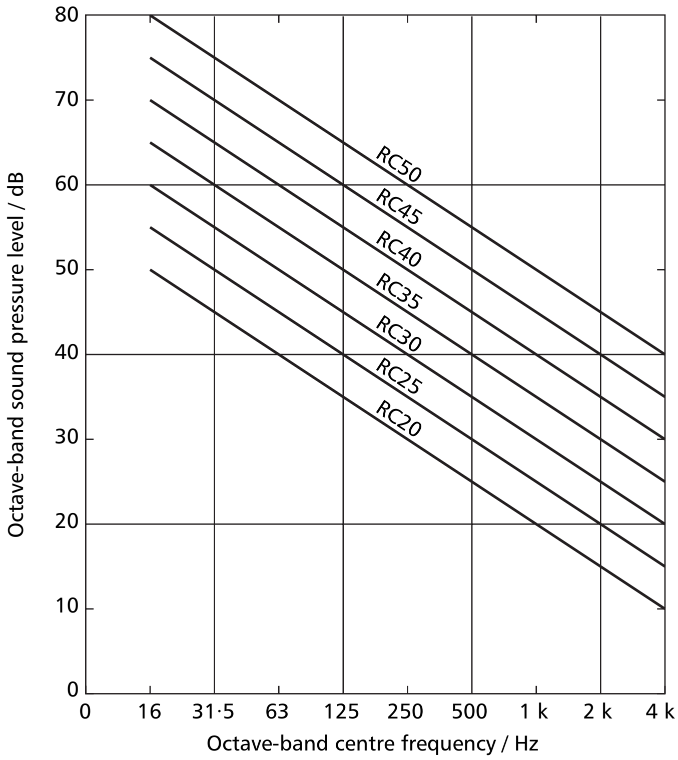


Figure 1.19 Room criterion (RC) curves

–5 dB/octave. Noise spectra following these slopes are acoustically 'neutral'. The use of the curves is explained in ASHRAE Handbook: *Applications*[99], where it is shown how to determine whether a noise has 'rumble', 'neutral' or 'hiss' characteristics, and the frequency regions where improvement is required. More recent work by Blazier[103] has aimed at using the RC curves to determine the acoustical 'quality' of a noise by separating its low frequency (rumble), middle frequency (roar) and high frequency (hiss) characteristics and comparing these with subjective acceptability ratings.

RC curves provide a more detailed description of the noise than is available from NR. In addition, room criteria are more prescriptive at low frequencies than NR. At 31.5 Hz, permitted NR35 levels are 19 dB higher than RC35 levels. A noise which follows the NR curve will be unpleasantly 'rumbly' compared with the neutral sound of a noise that follows the RC curve. Therefore NR continues to be adequate only where low frequency noise levels are well below the NR limit. The potential drawback of NRs is that they permit unacceptable noises that would have been rejected if RC had been applied.

1.9.4 Noise due to building services and other sources

In specifying noise design goals for a building, a balance must be sought between noise from the building services and noise from the activities taking place within the building. The acceptability of noise from building services does not depend only upon its absolute level and frequency content, but also on its relationship with noise from other sources. It is important that the designer considers the likely activity-related and extraneous noise level and frequency content at an early stage of design. However, while noise from the building services is controlled by the building services engineer, activity noise is a function of the office or other equipment in the space and is therefore under the control of the office management. Noise from outside the building, e.g. traffic noise, is controlled mainly by the fabric of the building, which is the responsibility of the architect. The building services engineer must work to an agreed specification for the noise level from the building services and may be able to influence the level set down in the specification.

Reasonable design limits to minimise annoyance from broadband continuous noise from building services installations are given in Table 1.15. If the noise contains recognisable tones or is intermittent or impulsive it will be more annoying and the appropriate NR value from Table 1.15 should be corrected using the factors given in Table 1.16.

Noise levels for building services are often specified for the unoccupied space. Noise from the building services becomes more noticeable when other noise is at its minimum level, as is usually the case outside the occupied period. The building services designer cannot rely on external or activity noise in order to permit noise levels from services higher than those given in Table 1.15. Indeed, it may be necessary to control noise from these other sources to achieve an acceptable acoustical environment.

Table 1.15 Suggested maximum permissible background noise levels generated by building services installations[100]

Situation	Noise rating (NR)
Studios and auditoria:	
— sound broadcasting (drama)	15
— sound broadcasting (general), television (general), sound recording	20
— television (audience studio)	25
— concert hall, theatre	20–25
— lecture theatre, cinema	25–30
Hospitals:	
— audiometric room	20–25
— operating theatre, single bed ward	30–35
— multi–bed ward, waiting room	35
— corridor, laboratory	35–40
— wash room, toilet, kitchen	35–40
— staff room, recreation room	30–40
Hotels:	
— individual room, suite	20–30
— ballroom, banquet room	30–35
— corridor, lobby	35–40
— kitchen, laundry	40–45
Restaurants, shops and stores:	
— restaurant, department store (upper floors)	35–40
— night club, public house, cafeteria, canteen, department store (main floors)	40–45
Offices:	
— boardroom, large conference room	25–30
— small conference room, executive office, reception room	30–35
— open plan office	35
— drawing office, computer suite	35–45
Public buildings:	
— law court	25–30
— assembly hall	25–35
— library, bank, museum	30–35
— washroom, toilet	35–45
— swimming pool, sports arena	40–50
— garage, car park	55
Ecclesiastical and academic buildings:	
— church	25–30
— classroom, lecture theatre	25–35
— laboratory, workshop	35–40
— corridor, gymnasium	35–45
Industrial:	
— warehouse, garage	45–50
— light engineering workshop	45–55
— heavy engineering workshop	50–65
Dwellings (urban):	
— bedroom	25
— living room	30

Note: dBA ≈ NR + 6

Table 1.16 Corrections to noise rating for certain types of noise

Type of noise	NR correction
Pure tone easily perceptible	+5
Impulsive and/or intermittent noise	+3

In the absence of noise from other sources, noise from building services may become noticeable. If so, the aim should be to reduce the services noise rather than to attempt to mask it by noise from other sources. However, provided that it is within the limits required by the specification, a steady level of services noise can sometimes help to improve acoustical privacy in open plan offices.

In modern offices, a prominent source of noise at workstations is the cooling fans in office equipment, such as personal computers. The characteristic of this noise is different from that of services noise, tending to have a tonal spectrum with peaks at about 250 Hz and associated harmonics, depending on the design of the fan. Clearly, this source of noise is not under the control of the building services engineer.

1.9.5 Speech intelligibility

Speech intelligibility is dependent upon the ambient noise and the distance between listener and speaker. Table 1.17 gives an indication of the distance at which normal speech will be intelligible for various ambient noise levels[100].

Ambient noise may also interfere with the intelligibility of telephone conversations. However, conversation can be carried out in reasonable comfort if the ambient level is below 60 dBA, which should be the case in well-designed offices where the maximum levels are not likely to exceed 45 dBA.

Table 1.17 Maximum steady noise levels for reliable speech communication[100] (reproduced from BS 8233 by permission of the British Standards Institution)

Distance between talker and listener (m)	Noise level, L_{Aeq} (dB)	
	Normal voice	Raised voice
1	57	62
2	51	56
4	45	50
8	39	44

1.9.6 Hearing damage

Exposure to high noise levels, such as may occur in a plant room, can cause temporary or permanent hearing damage. Where workers are exposed to high levels of noise, the noise levels must be assessed by a qualified person. The Noise at Work Regulations 2005[104] identify two levels of 'daily personal noise exposure' (measured in a manner similar to $L_{Aeq,T}$) at which actions become necessary. These levels are 80 dBA for the lower level and 85 dBA for the higher level, corresponding to advisory and compulsory requirements. In addition, for impulse noise, there is a lower peak level limit of 112 Pa and a higher peak level limit of 140 Pa. These peak action levels control exposure to impulse noise. Suppliers of machinery must provide noise data for machines likely to cause exposure to noise above the action levels.

1.10 Vibration

1.10.1 General

In the context of building services installations, vibrations arise from reciprocating machines or from unbalanced forces in rotating machines. The vibration is often most noticeable during machine start-up (i.e. low-frequency movement), during which some machines pass through a critical (resonant) speed before reaching their normal operating condition. Vibration associated with start-up may not be important if the machine operates for long periods, since that condition occurs only infrequently. However, machines which switch on and off under thermostatic control, for example, may require special precautions.

Vibrations transmitted from machines through their bases to the building structure may be felt and heard at considerable distances from the plant and, in extreme cases, even in neighbouring buildings. Therefore, adequate isolation is important in those cases where vibration is expected. Vibration isolators must be chosen to withstand the static load of the machine as well as isolate it from the structure. Efficient vibration isolation is the preferred way of controlling structure-borne noise, which occurs when vibration transmitted to building surfaces is re-radiated as noise. Structure-borne noise is enhanced when the excitation frequency corresponds with a structural resonance frequency, which may cause unexpected noise problems.

1.10.2 Response of the human body to vibration

Vibrating motion of the human body can produce both physical and biological effects. The physical effect is the excitation of parts of the body and under extreme conditions physical damage may result. Building vibration may affect the occupants by reducing both quality of life and working efficiency. Complaints about continuous vibration in residential situations are likely to arise from occupants when the vibration levels are only slightly greater than the threshold of perception.

The levels of complaint resulting from vibration and acceptable limits for building vibration depend upon the characteristics of the vibration and the building environment, as well as individual response. These factors are incorporated in guidance given in BS 6472[105] which gives magnitudes of vibrations below which the probability of complaints is low. This guidance takes the form of base curves of RMS acceleration against frequency over the range 1–80 Hz, for vibration along *x*-, *y*-, and *z*-axes. The axes of vibration with respect to the human body are shown in Figure 1.20. The values for the *x*- and *y*-axis curves are more severe than that for the *z*-axis curve, reflecting the greater sensitivity of the human body to *x*- and *y*-axis motion at low frequencies.

The base curves (not shown here) are modified by factors appropriate to the building environment, time of day and type of vibration (see Table 1.18) to produce curves of design maximum vibration magnitude, Figure 1.21. The curves are labelled to correspond to the multiplying factors appropriate to the situations listed in Table 1.18.

1.10.3 Effect on structure

Vibration can damage building structures. The degree of damage depends largely on the magnitude and frequency of vibration. In general, the level of vibration likely to cause cosmetic damage, such as plaster cracking, is significantly greater than that which would be easily perceptible to the occupants. Therefore, the occupants provide early warning of vibration levels likely to cause damage to the fabric.

Table 1.18 Multiplying factors used to specify satisfactory magnitudes of building vibration with respect to human response(105) (reproduced from BS 6472 by permission of the British Standards Institution)

Situation	Multiplying factor for stated vibration[1,5]	
	Exposure to continuous vibration (16 hour day, 8 hour night)[2]	Intermittent vibration with up to 3 occurrences
Critical working areas, e.g. hospital operating theatre, precision laboratory[3,9]		
— daytime	1	1
— night	1	1
Residential buildings:		
— daytime	2–4 [4]	60–90 [4,8]
— night	1.4	20
Offices:		
— daytime	4	128 [6]
— night	4	128
Workshops:		
— daytime	8[7]	128 [6,7]
— night	8	128

Notes:

[1] Magnitude of vibration below which the probability of adverse comments is low (any acoustical noise caused by structural vibration is not considered)

[2] Doubling of suggested vibration magnitudes may result in adverse comment and this may increase significantly if magnitudes are quadrupled (where available, dose/response curves may be consulted)

[3] Magnitudes of vibration in hospital operating theatres and critical working places pertain to periods of time when operations are in progress or critical work is being performed; at other times magnitudes as high as those for residences are satisfactory provided there is due agreement and warning

[4] In residential areas people exhibit wide variations of vibration tolerance; specific values dependent upon social and cultural factors, psychological attitudes and expected degree of intrusion

[5] Vibration to be measured at point of entry of the vibration to the subject; where this is not possible it is essential that transfer functions be evaluated

[6] Magnitudes for vibration in offices and workshop areas should not be increased without considering the possibility of significant disruption of work activity

[7] Vibration acting on operators in certain processes such as drop forges or crushers, which cause the working place to vibrate, may be in a separate category from the workshop areas considered in the table. Vibration magnitudes specified in relevant standards apply to the operators of these processes

[8] When short term works such as piling, demolition and construction give rise to impulsive vibrations, undue restriction on vibration levels can significantly prolong these operations resulting in greater annoyance. In certain circumstances higher magnitudes can be used

[9] Where sensitive equipment or delicate tasks impose more stringent criteria than human comfort, the corresponding more stringent values should be applied

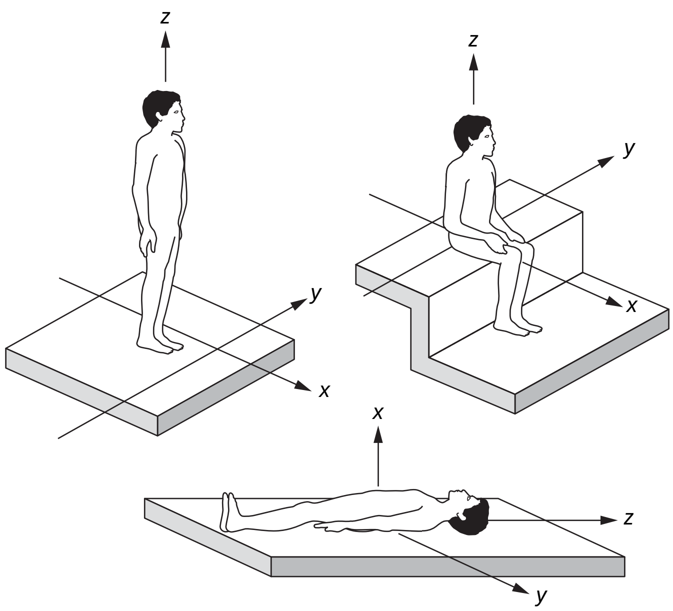


Figure 1.20 Definition of axes of vibration with respect to the human body(105); *x*-axis = back to chest, *y*-axis = right side to left side, *z*-axis = foot to head (reproduced from BS 6472 by permission of the British Standards Institution)

Although vibrations in buildings are often noticeable, there is little documented evidence to show that they produced even cosmetic damage(105–107).

1.11 Electromagnetic and electrostatic environment

1.11.1 Electromagnetic fields

Since the 1960s concern has been expressed regarding the possible effects on health of extremely low frequency electromagnetic fields (i.e. below 300 Hz). Some reports have suggested that exposure to these fields, such as might be experienced by those living near high voltage overhead power lines, increases the risk of cancer, particularly leukaemia, especially amongst children. Other studies have raised the possibility that 'electrical' occupations, such as those that entail prolonged proximity to visual display terminals, result in an increased risk of illness.

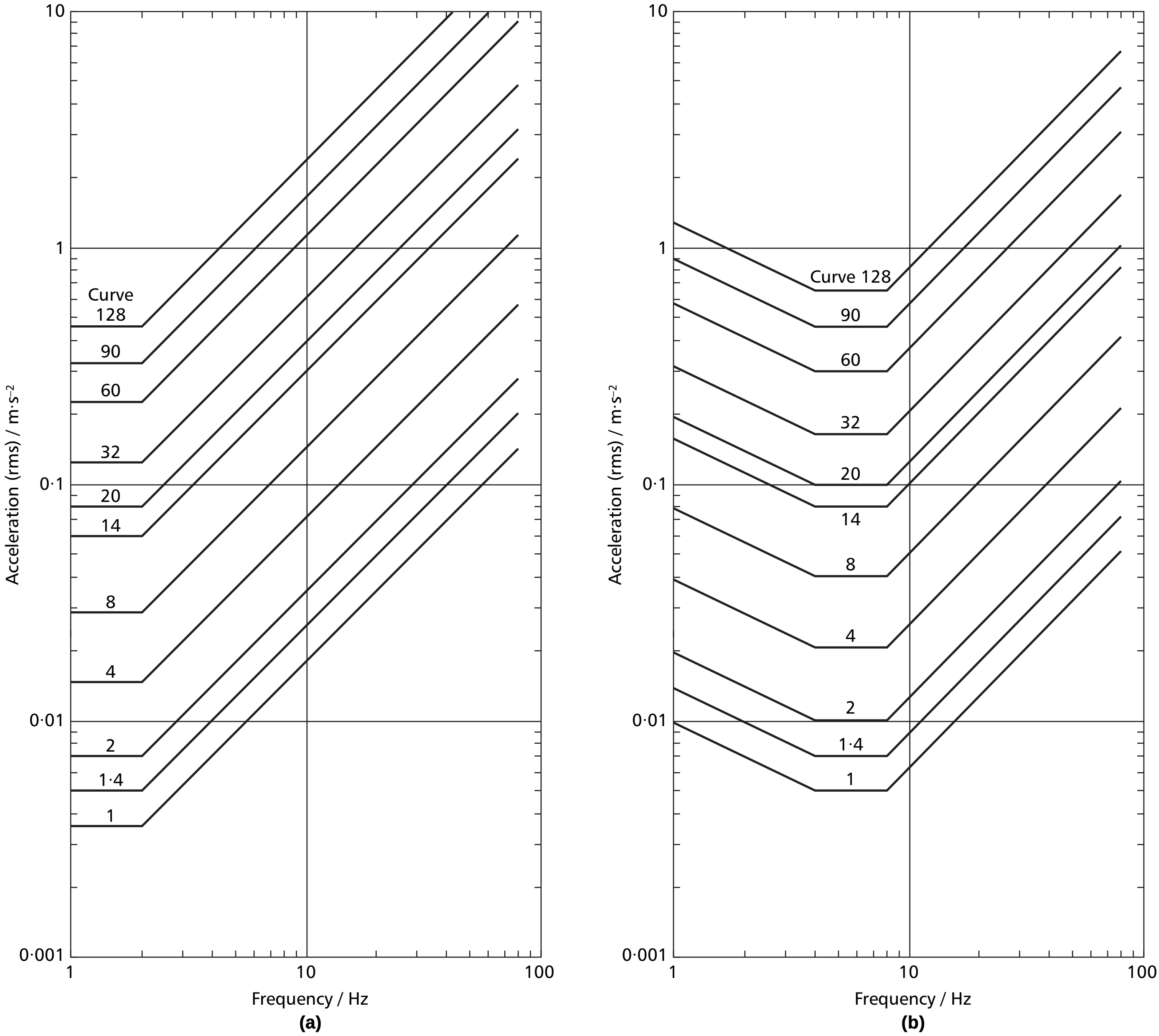


Figure 1.21 Design maximum vibration curves[105] corresponding to multiplying factors for the situations given in Table 1.17; (*a*) for *x*- and *y*-axes, (*b*) for *z*-axis (reproduced from BS 6472 by permission of the British Standards Institution)

A review of these studies[108] reveals that all suffer from methodological or other shortcomings but it is not clear whether these are sufficient to explain the results. Experiments with animals have produced conflicting and confusing results, and their relevance to the effect on humans is difficult to assess. No plausible mechanism for carcinogenesis due to exposure to electrical or magnetic fields has yet been deduced. It has been established that such fields affect the function of cardiac pacemakers but this is unlikely to be a hazard at the field strengths normally encountered and most modern pacemakers are designed to cope with high field strengths.

Therefore, current evidence does not permit firm conclusions to be drawn on the relationship between electromagnetic fields and physiological or psychological effects on humans. Until the situation is clarified by further research and provided hat no significant cost penalties result, it is suggested that potential fields be minimised. Often this can be achieved by ensuring that line and return cables are in close proximity, as is usual practice for mains wiring.

1.11.2 Air ionisation

It has been suggested that the ion balance of the air is an important factor in human comfort in that negative ions tend to produce sensations of freshness and well-being and positive ions cause headache, nausea and general malaise. Present evidence on the effects of air ions and, in particular, the effectiveness of air ionisers is inconclusive and hence no design criteria can be established[109,110].

1.11.3 Static electricity

Static electricity can lead to shocks when occupants are not adequately earthed via the floor covering. The incidence of electrostatic shocks depends on the electrical resistance of the floor covering. The resistance is a function of the material itself and its moisture content. The highest electrical resistance is produced by fibrous carpets with an insulating backing when low in moisture content.

At low room humidity, some types of carpet can become highly charged and electrostatic shocks may be experienced, see chapter 8, section 8.3.3.

References

1 ASHRAE Standard 55-04: *Thermal environmental conditions for human occupancy* (Atlanta, GA: American Society of Heating Refrigeration and Air-Conditioning Engineers) (2004)

2 Cain W S, See L C and Tosun T Irritation and odour from formaldehyde: chamber studies *Proc. ASHRAE Conf. IAQ '86: Managing the Indoor Air for Health and Energy Conservation, Atlanta GA, USA* (1986)

3 *Indoor air pollutants: exposure and health effects — Report of WHO meeting, Norlinger, 8–11 June 1982* (Copenhagen: World Health Organisation Regional Office for Europe) (1982)

4 BS EN ISO 7726: 2001: *Ergonomics of the thermal environment. Instruments for measuring physical quantities* (London: British Standards Institution) (2001)

5 BS EN ISO 9920: 2003: *Ergonomics of the thermal environment. Estimation of the thermal insulation and evaporative resistance of a clothing ensemble* (London: British Standards Institution) (2003)

6 BS EN 28996: 1994 (ISO 8996: 1990): *Ergonomics. Determination of metabolic heat production* (London: British Standards Institution) (1994)

7 BS EN ISO 7730: *Moderate thermal environments. Determination of the PMV and PPD indices and specification of the conditions for thermal comfort* (London: British Standards Institution) (1995)

8 Humphreys M A The optimum diameter for a globe thermometer for use indoors. *Ann. Occup. Hyg.* **20** 135–140 (1977)

9 Fanger P O, Melikov A, Hanzawa H and Ring J Air turbulence and sensation of draught *Energy and Buildings* **12** 21–39 (1988)

10 Fanger P O and Pedersen C J Discomfort due to air velocities in spaces *Proc. IIR Commissions B1/B2/E1, Belgrade, 1977* (1977)

11 Fountain M E Laboratory studies of the effect of air movement on thermal comfort: a comparison *ASHRAE Trans.* **97** (1), 1991]

12 Fountain M E *An empirical model for predicting air movement preferred in warm environments* in Nicol F, Humphreys M and Sykes O (eds.) *Standards for thermal comfort* (London: Spon) (1995)

13 Griefahn B, Kunemund C and Gehring U *Annoyance caused by draught, the extension of the draught-rating model (ISO 7730)* in McCartney K (ed.) *Moving thermal comfort standards into the 21st century* (Oxford: Oxford Brookes University) (2001)

14 McIntyre D A Response to atmospheric humidity: a comparison of three experiments *Ann. Occupational Hygiene* **21** 177–190 (1978)

15 Nevins R G, Rohles F H, Springer W E and Feyerherm A M Temperature–humidity chart for thermal comfort of seated persons *ASHRAE Trans.* **72** (1) 283 (1966)

16 Morey P R, Hodgson M J, Sorenson W G, Kullman G J, Rhodes W W and Visvesvara G S Environmental studies in mouldy office buildings: biological agents, sources and preventive measures *Ann. American Conference Governmental Industrial Hygienists — Evaluating Office Environmental Problems* **10** (1984)

17 de Dear R, Brager G and Cooper D *Developing an adaptive model of thermal comfort and preference* ASHRAE Final Report RP-884 (Sydney: Macquarie University) (1997)

18 Fanger P O *Thermal comfort* (Malabar, FL, USA: Krieger) (1982)

19 Hensel H *Thermoreception and temperature regulation* Physiological Society Monograph No. 38 (London: Academic Press) (1981)

20 Humphreys M A Field studies of thermal comfort compared and applied *J. Inst. Heating and Ventilating Engineers* **44** 5–27 (1975)

21 Humphreys M A *Thermal Comfort requirements, climate and energy* in Sayigh A A M (ed.) *Renewable Energy, Technology and the Environment* (Oxford: Pergamon) (1992)

22 de Dear R and Brager G Thermal comfort in naturally ventilated buildings: revisions to ASHRAE Standard 55 *Energy and Buildings* **34** (6) (2002)

23 Humphreys M A and Nicol J F The validity of ISO-PMV for predicting comfort votes in every-day thermal environments *Energy and Buildings* **34** (6) 667–684 (2002)

24 Control of Fuel and Electricity. The Fuel and Electricity (Heating) (Control) Order 1974 Statutory Instrument 1974 No. 2160 (London: The Stationery Office) (1974)

25 Control of Fuel and Electricity. The Fuel and Electricity (Heating) (Control) (Amendment) Order 1980 Statutory Instrument 1980 No. 1013 (London: The Stationery Office) (1980)

26 Witterseh T, Wyton D P and Clausen G The effects of moderate heat stress and open plan office noise distraction on SBS symptoms and on the performance of office work *Indoor Air* **14** (supplement 8) 30–40 (2004)

27 Wyon D P Studies of children under imposed noise and heat stress *Ergonomics* **13** (5) 598–609 (1970)

28 *Conservation of fuel and power* The Building Regulations 2000 Approved Document L1/L2 (London: The Stationery Office) (2002)

29 *Climate change and the internal environment* CIBSE TM36 (London: Chartered Institution of Building Services Engineers) (2005)

30 *Guidelines for environmental design in schools* Building Bulletin BB87 (London: Department for Education and Skills) (2003)

31 *Natural ventilation in non-domestic buildings* CIBSE AM10 (London: Chartered Institution of Building Services Engineers) (2005)

32 Thomas M et al. Brain and behaviour changes during sleep deprivation *Proc. Human Factors and Ergonomics Society* **2** (1998)

33 *Code for lighting* (London: Society of Light and Lighting) (2004)

34 *Office lighting* SLL Lighting Guide LG7 (London: Society of Light and Lighting) (2005)

35 *Ventilation* The Building Regulations 1991 Approved Document F (London: Her Majesty's Stationery Office) (1991)

36 *The visual environment in lecture, teaching and conference rooms* CIBSE Lighting Guide LG5 (London: Chartered Institution of Building Services Engineers) (1991)

37 The Workplace (Health, Safety and Welfare) Regulations 1992 (London: The Stationery Office) (1992)

38 *The industrial environment* CIBSE Lighting Guide LG1 (London: Chartered Institution of Building Services Engineers) (1989)

39 *Heating, ventilating, air conditioning and refrigeration* CIBSE Guide B (London: Chartered Institution of Building Services Engineers) (2001–2)

40 *Sports* CIBSE Lighting Guide LG4 (London: Chartered Institution of Building Services Engineers) (1990 plus addendum 1999)

41 *Lighting for museums and art galleries* CIBSE Lighting Guide LG8 (London: Chartered Institution of Building Services Engineers) (1994)

42 Griffiths I D and McIntyre D A Sensitivity to temporal changes in thermal conditions *Ergonomics* **17** 499–507 (1974)

43 Griffiths I D and McIntyre D A The balance of radiant and air temperature for warmth in older women *Environmental Res.* **6** (4) 382–388 (1973)

44 Fanger P O and Langkilde G Inter-individual differences in ambient temperature preferred by seated persons *ASHRAE Trans.* **81** (2) 140–147 (1975)

45 Langkilde G Thermal comfort for people of high age Comfort thermique: *Aspects Physiologiques et Psychologiques (INSERM)* **75** 187–193 (1979)

46 Rohles F H and Johnson M A Thermal comfort in the elderly *ASHRAE Trans.* **78** (1) 131 (1972)

47 Collins K J and Hoinville E Temperature requirements in old age *Building Serv. Eng. Res. Technol.* **1** (4) 165–172 (1980)

48 Berry P C The effect of coloured illumination on perceived temperature *J. Applied Psychol.* **45** 248–250 (1961)

49 Parsons K C The effects of gender, acclimation state, and the opportunity to adjust clothing and physical disability on requirements for thermal comfort *Energy and Buildings* **34** (6) 593–599 (2002)

50 Haghighat F, Megri A C, Donnini G and Giorgi G Responses of disabled, temporarily ill, and elderly persons to thermal environments *ASHRAE Trans.* **5** (2) (2000)

51 Olesen B W, Schfler M and Fanger P O *Vertical air temperature differences and comfort* in Fanger and Valbjfrn (eds.) *Indoor Climate* (Copenhagen: Danish Building Research Institute) (1979)

52 McNair H P *A preliminary study of the subjective effects of vertical air temperature gradients* British Gas Corporation Reports WH/T/R and D/73/94 (London: British Gas Corporation) (1973)

53 McNair H P and Fishman D S *A further study of the subjective effects of vertical air temperature gradients* British Gas Corporation Reports WH/T/R and D/74/2 (London: British Gas Corporation) (1974)

54 Olesen B W Thermal comfort requirements for floors *Proc. IIR Commissions B1/B2/E1 Belgrade* 337–343 (1977)

55 Olesen B W Thermal comfort requirements for floors occupied by people with bare feet *ASHRAE Trans.* **83** (2) (1977)

56 Nevins R G, Michaels K B, Feyerherm A M The effect of floor surface temperatures on comfort, Part I: College-age males *ASHRAE Trans.* **70** (1964)

57 Nevins R G, Michaels K B, Feyerherm A M The effect of floor surface temperatures on comfort, Part II: College-age females *ASHRAE Trans.* **70** 1964

58 Springer W E, Nevins R G, Feyerherm A M and Michaels K B The effect of floor surface temperatures on comfort, Part III: the elderly *ASHRAE Trans.* **72** (1) (1966)

59 BS EN 1264: *Floor heating. Systems and components* (London: British Standards Institution) (1998)

60 Fanger P O, Banhidi L, Olesen B W and Langkilde G Comfort limits for heated ceilings *ASHRAE Trans.* **86** (1980)

61 Fanger P O, Ipsen G, Langkilde G, Olesen B W, Christiansen N K and Tanabe S Comfort limits for asymmetric thermal radiation *Energy and Buildings* **8** 225– 236 (1985)

62 Loveday D L, Parsons K C, Taki A H and Hodder S G Displacement ventilation environments with chilled ceilings, thermal comfort design within the context of the BS EN ISO 7730 *Energy and Buildings* **34** (6) 573–579 (2002)

63 Owens P G T Air conditioned comfort and sunshine *J. Inst. Heating Ventilating Engineers* **37** 92–96 (July 1969)

64 Humphreys M A and Nicol J F Understanding the adaptive approach to thermal comfort *ASHRAE Trans.* **104** (1) 991–1004 (1998)

65 Auliciems A Towards a psycho-physiological model of thermal perception *Int. J. Biometeorology* **25** 109–122 (1981)

66 de Dear R A Global database of thermal comfort field experiments *ASHRAE Trans.* **104** (1) 1141–1152 (1998)

67 Baker N V and Standeven M A Thermal comfort in free-running buildings *Energy and Buildings* **23** (1996)

68 Nicol J F and Humphreys M A Maximum temperatures in European offices to avoid heat discomfort *Solar Energy J.* (to be published)

69 Humphreys M A *The influence of season and ambient temperature on human clothing behaviour* in Fanger P O and Valbjorn O (eds.) *Indoor Climate* (Copenhagen: Danish Building Research) (1979)

70 Morgan C A, de Dear R and Brager G Climate clothing and adaptation in the built environment *Proc. Conf. Indoor Air 2002: 9th Int. Conf. on Indoor Air Quality and Climate, Santa Cruz CA, USA* **5** 98–103 (2002)

71 Nicol F and Raja I *Thermal comfort, time and posture: exploratory studies in the nature of adaptive thermal comfort* (Oxford: School of Architecture, Oxford Brookes University) (1996)

72 Humphreys M A *The dependence of comfortable temperature upon indoor and outdoor climate* in Cena K and Clark J A (eds.) *Bioengineering, Thermal Physiology and Comfort* (Oxford: Elsevier) (1981)

73 Boestra A et al. A New Dutch adaptive thermal comfort guideline *Proc. Conf. Indoor Air 2002: 9th Int. Conf. on Indoor Air Quality and Climate, Santa Cruz CA, USA* **2** (2002)

74 McCartney K J and Nicol J F Developing an adaptive control algorithm for Europe: results of the SCATs project *Energy and Buildings* **34** (6) 623–635 (2002)

75 Baker N V and Standeven M A A behavioural approach to thermal comfort assessment in naturally ventilated buildings. *Proc. CIBSE Nat. Conf., Eastbourne, 1995* 76–84 (1995)

76 Oseland N A Predicted and reported thermal sensation in climate chambers, offices and homes *Energy and Buildings* **23** (2) 105–116 (1995)

77 The Control of Substances Hazardous to Health Regulations 1988 (London: The Stationery Office) (1988)

78 Fanger P O The new comfort equation for indoor air quality *Proc. ASHRAE Conf. IAQ '89: The Human Equation — Health and Comfort, San Diego CA, USA* (April 1989)

79 PD CR 1752: 1999: *Ventilation for buildings. Design criteria for the indoor environment* (London: British Standards Institution) (1999)

80 *Occupational Exposure Limits* HSE EH40 (London: The Stationery Office) (published annually)

81 *Air quality guidelines for Europe* (Copenhagen: World Health Organisation) (1987)

82 The Workplace (Health, Safety and Welfare) Regulations 1992 (London: The Stationery Office) (1992)

83 BS 5266-1: 1999: *Emergency lighting. Code of practice for the emergency lighting of premises other than cinemas and certain other specified premises used for entertainment* (London: British Standards Institution) (1999)

84 *Emergency lighting design guide* SLL Lighting Guide LG12 (London: Society of Light and Lighting) (2004)

85 Weston H C *The relation between illumination and visual performance* Industrial Health Research Board Report 87 (London: The Stationery Office) (1945)

86 Stenzel A G and Sommer J The effect of illumination on tasks which are independent of vision *Lichttechnik* **21** 143–146 (1969)

87 Boyce P R Age, illuminance, visual performance and preference *Lighting Res. Technol.* **5** (3) 125–144 (1973)

88 Rea M S and Ouellette M J Visual performance using reaction times *Lighting Res. Technol.* **20** (4) 139–153 (1988)

89 *Daylighting and window design* CIBSE Lighting Guide LG10 (London: Chartered Institution of Building Services Engineers) (1999)

90 Littlefair P J *Designing with innovative daylighting* BRE Report BR305 (London: Construction Research Communications) (1996)

91 BS 8206-2: 1992: *Lighting for buildings. Code of practice for daylighting* (London: British Standards Institution) (1999)

92 BS EN 12464-1: 2002: *Light and lighting. Lighting of work places. Indoor work places* (London: British Standards Institution) (2002)

93 Saunders J E The role of the level and diversity of horizontal illumination in an appraisal of a simple office task *Lighting Res. Technol.* **1** (1) 37–46 (1969)

94 Fischer D The European approach to the integration of lighting and air-conditioning *Lighting Res. Technol.* **2** (3) 174–185 (1970)

95 Bitter C and Van Ireland J F A A Appreciation of sunlight in the home *Proc. CIE Int. Conf. Newcastle-on-Tyne* (April 1965)

96 Ne'eman E Visual aspects of sunlight in buildings *Lighting Res. Technol.* **6** (3) 159–164 (1974)

97 Beranek L L Noise and vibration control (Washington DC: Institute of Noise Control Engineering) (1988)

98 *Noise Control in Building Services* (London: Pergamon Press) (1988)

99 *Sound and vibration control* in ASHRAE Handbook: *HVAC Applications* (Atlanta GA: American Society of Heating, Ventilating and Air-conditioning Engineers) (1995)

100 BS 8233: 1999: *Sound insulation and noise reduction for buildings. Code of practice* (London: British Standards Institution) (1999)

101 BS 4142: 1997: *Method for rating industrial noise affecting mixed residential and industrial areas* (London: British Standards Institution) (1997)

102 Blazier W E Revised noise quality criteria for application in the acoustic design and control of HVAC systems *Noise Control Eng.* **16** (2) 64–73 (1981)

103 Blazier W E Sound quality criteria in rating noise from heating, ventilating and air-conditioning (HVAC) systems in buildings *Noise Control Eng.* **43** (3) 53–63 (1995)

104 The Noise at Work Regulations 2005 Statutory Instrument 2005 No. 1643 (London: The Stationery Office) (2005)

105 BS 6472: 1992: *Guide to evaluation of human exposure to vibration in buildings (1 Hz to 80 Hz)* (London: British Standards Institution) (1992)

106 *Damage to structures from ground-borne vibration* BRE Digest 353 (Garston: Building Research Establishment) (1990)

107 BS 7385: *Evaluation and measurements for vibration in buildings*: Part 1: 1990: *Guide for measurement of vibration and evaluation of their effects on buildings* (London: British Standards Institution) (1990)

108 Naismith O F *Electromagnetic fields: A Review of the evidence for effects on health* BR 206 (Garston: Building Research Establishment) (1991)

109 Hamilton G and Kew J *Negative air ionisation in buildings* Technical Note TN 4/85 (Bracknell: Building Services Research and Information Association) (1985)

110 Raw G *Sick Building Syndrome: a review of the evidence* HSE CRR 42/1992 (London: Health and Safety Executive) (1992)

Appendix 1.A1: Determination of predicted mean vote (PMV)

1.A1.1 Equation for determination of PMV

The predicted mean vote (PMV) is given by the equation:

$$\begin{aligned}\text{PMV} = {} & (0.303\ \mathrm{e}^{-0.036M} + 0.028)\ \{(M - W) \\ & - 0.00305\ [5733 - 6.99\ (M - W) - p_s] \\ & - 0.42\ [M - W - 58.15] \\ & - (1.7 \times 10^{-5})\ M\ (5867 - p_s) \\ & - 0.0014\ M\ (34 - \theta_{ai}) \\ & - (3.96 \times 10^{-8})\ f_{cl}\ [(\theta_{cl} + 273)^4 \\ & - (\theta_c + 273)^4] - [f_{cl}\ h_c\ (\theta_{cl} - \theta_{ai})]\}\end{aligned} \tag{1.20}$$

where PMV is the predicted mean vote, M is metabolic rate ($\mathrm{W{\cdot}m^{-2}}$ of body surface), W is external work ($\mathrm{W{\cdot}m^{-2}}$ of body surface) (0 for most activities), f_{cl} is the ratio of the area of the clothed human body to that of the unclothed human body, θ_{ai} is the average air temperature surrounding the body (°C), θ_c is the operative temperature (°C), p_s is the partial water vapour pressure in the air surrounding the body (Pa), h_c is the convective heat transfer coefficient at the body surface ($\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$) and θ_{cl} is the surface temperature of clothing (°C).

The surface temperature of clothing (θ_{cl}) is given by:

$$\begin{aligned}\theta_{cl} = {} & 35.7 - 0.028\ (M - W) - I_{cl}\ \{(3.96 \times 10^{-8}) \\ & \times f_{cl}\ [(\theta_{cl} + 273)^4 - (\theta_c + 273)^4] \\ & + f_{cl}\ h_c\ (\theta_{cl} - \theta_{ai})\}\end{aligned} \tag{1.21}$$

where I_{cl} is the thermal resistance of clothing ($\mathrm{m^2{\cdot}K{\cdot}W^{-1}}$).

For $\{2.38\ (\theta_{cl} - \theta_{ai})^{0.25}\} > 12.1 \sqrt{v_r}$:

$$h_c = 2.38\ (\theta_{cl} - \theta_{ai})^{0.25} \tag{1.22}$$

For $\{2.38\ (\theta_{cl} - \theta_{ai})^{0.25}\} < 12.1 \sqrt{v_r}$:

$$h_c = 12.1 \sqrt{v_r} \tag{1.23}$$

For $I_{cl} \leq 0.078\ \mathrm{m^2{\cdot}K{\cdot}W^{-1}}$:

$$f_{cl} = 1 + 1.29\ I_{cl} \tag{1.24}$$

For $I_{cl} > 0.078\ \mathrm{m^2{\cdot}K{\cdot}W^{-1}}$:

$$f_{cl} = 1.05 + 0.645\ I_{cl} \tag{1.25}$$

1.A1.2 Computer program for determination of PMV

The following BASIC program computes the PMV for a given set of input variables, as follows:

CLO	Clothing (clo)
MET	Metabolic rate (met)
WME	External work (met)
TAI	Air temperature (°C)
TR	Mean radiant temperature (°C)
VEL	Relative air velocity (m/s)
SAT	Percentage saturation (%)
PS	Partial vapour pressure (Pa)

```
10   'Computer program (BASIC) for calculation of
20   'Predicted Mean Vote (PMV) in accordance with
30   'CIBSE Guide A1: Environmental criteria for design (2006)
40   CLS:PRINT"DATA ENTRY":'data entry
50   INPUT " Clothing                              (clo)"; CLO
60   INPUT " Metabolic rate                        (met)"; MET
70   INPUT " External work, normally around 0      (met)"; WME
80   INPUT " Air temperature                       (C)"; TA
90   INPUT " Mean radiant temperature              (C)"; TR
100  INPUT " Relative air velocity                 (m/s)"; VEL
110  PRINT " ENTER EITHER %SAT OR VAPOUR PRESSURE BUT NOT BOTH"
120  INPUT " Percentage saturation                 (%)"; SAT
130  INPUT " Vapour pressure                       (Pa)"; PS
140  DEF FNPS(T)=EXP(16.6536-4030.183/(T+235):'saturated vapour pressure,KPa
150  IF PS=0 THEN PS=SAT*10*FNPS(TAI)   :'vapour pressure,Pa
```

```
160   ICL = .155 * CLO           :'thermal insulation of the clothing in m2K/W
170   M = MET * 58.15            :'metabolic rate in W/m2
180   W = WME * 58.15            :'external work in W/m2
190   MW = M - W      :'internal heat production in W/m2
200   IF ICL < .078 THEN FCL = 1 + 1.29 * ICL ELSE FCL=1.05+.645*ICL:'clothing area factor
210   HCF=12.1*SQR(VEL)
220   TAA = TA + 273             :'air temperature in kelvins
230   TRA = TR + 273             :'mean radiant temperature in kelvins
240   '———-CALCULATE SURFACE TEMPERATURE OF CLOTHING BY ITERATION———-
250   TCLA = TAA + (35.5-TA) / (3.5*(6.45*ICL+.1)):'first guess for surface temp of clothing
260   P1 = ICL * FCL             :'calculation term
270   P2 = P1 * 3.96             :'calculation term
280   P3 = P1 + 100              :'calculation term
290   P4 = P1 * TAA              :'calculation term
300   P5 = 308.7 - .028 * MW + P2 * (TRA/100) ^ 4 :'calculation term
310   XN = TCLA / 100
320   XF = XN
330   N = 0           :'N: number of iterations
340   EPS = .00015               :'stop criterion in iteration
350   XF = (XF+XN)/2
360   HCN=2.38*ABS(100*XF-TAA) ^ .25 :'heat trans coeff by nat convection
370   IF HCF>HCN THEN HC=HCF ELSE HC=HCN
380   XN=(P5+P4*HC-P2*XF ^ 4) / (100+P3*HC)
390   N = N+1
400   IF N > 150 THEN GOTO 550
410   IF ABS(XN-XF) > EPS GOTO 350
420   TCL=100*XN-273    : 'surface temperature of the clothing
430   '————————————HEAT LOSS COMPONENTS————————————
440   HL1 = 3.05*.001*(5733-6.99*MW-PS)     :'heat loss diff. through skin
450   IF MW > 58.15 THEN HL2 = .42*(MW-58.15) ELSE HL2 = 0!:'heat loss by sweating (comfort)
460   HL3 = 1.7*.00001*M*(5867-PA)          :'latent respiration heat loss
470   HL4 = .0014*M*(34-TA)                 :'dry respiration heat loss
480   HL5 = 3.96*FCL*(XN ^ 4-(TRA/100) ^ 4):'heat loss by radiation
490   HL6 = FCL*HC*(TCL-TA)                 :'heat loss by convection
500   '————————————CALCULATE PMV————————————
510   TS = .303*EXP(-.036*M)+.028           :'thermal sensation trans coeff
520   PMV = TS*(MW-HL1-HL2-HL3-HL4-HL5-HL6) :'predicted mean vote
530   GOTO 550
540   PMV = 999999!
550   PRINT:PRINT"OUTPUT"                   :'output
560   PRINT " Predicted mean vote        (PMV): ";:PRINT USING "##.#"; PMV
570   PRINT: INPUT "NEXT RUN (Y/N)" ; R$
580   IF (R$="Y" OR R$="y") THEN RUN
590   END
```

1.A1.3 Example

DATA ENTRY:

Clothing (clo)? 1.0
Metabolic rate (met)? 1.2
External work, normally around 0 (met)? 0
Air temperature (C)? 19.0
Mean radiant temperature (C)? 18.0
Relative air velocity (m/s)? 0.1

ENTER EITHER %SAT OR WATER VAPOUR PRESSURE BUT NOT BOTH

Percentage saturation (%)? 40
Water vapour pressure (Pa)?

OUTPUT:

Predicted mean vote (PMV): -0.7

Appendix 1.A2: Measuring operative temperature

An ordinary liquid-in-glass or digital thermometer is not suitable for measuring operative temperature if the radiant temperature differs greatly from the air temperature.

The globe thermometer is an instrument that combines the effects of air and radiant temperature in a way related to the response of a human subject. It is essentially an integrating sphere (made of metal or plastic) whose temperature will closely approximate the operative temperature (see page 1-2).

Spheres of various diameters have been used for globe thermometers, but it has been estimated[(A2.1)] that the optimum diameter for a globe thermometer to sense operative temperature to be about 40 mm (similar to that of a table tennis ball). The surface of the sphere should be painted grey or black to approximate the reflectivity of the clothed human body to any diffuse solar radiation reflected from the room surfaces.

A globe thermometer can be made by inserting a temperature sensor (electronic or liquid-in-glass) into a suitable 40 mm sphere, with a grey or black surface. The sensor should be at the centre of the globe. The thermometer should fit closely through the globe, to prevent the exchange of air between its interior and the room. The temperature measured at the centre will approximate the mean temperature of the enclosing sphere. Depending on the thermal capacity of the sphere and of the sensor itself, the instrument will take some time to settle. This means that from 5 to 20 minutes may need to elapse before taking the final reading.

To assess the operative temperature of a space several readings of the globe thermometer should be taken, in places representative of the occupied area, such as on the working plane, but out of direct sunlight. The thermometer should be suspended or clamped, and not held in the hand. Each time the thermometer is moved it needs time to stabilise, so it may be useful to have two or more identical globe thermometers, allowing multiple readings to be taken in different locations in the space over a period of 30 minutes or so. This will be particularly important if the temperature is changing. The operative temperature for the space is taken to be the average of the readings.

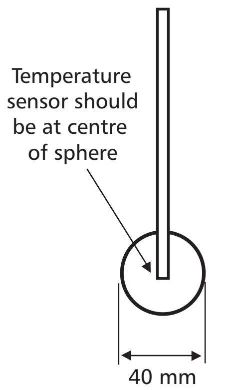


Figure 1.22 Schematic of a globe thermometer

Reference

A2.1 Humphreys M A The optimum diameter for a globe thermometer for use indoors *Ann. Occupational Hygiene* **20**(2) 135–140 (1977)

2 External design data

2.1 Introduction

2.1.1 General

The intention of this chapter is to provide the basic weather and solar data required for manual calculation of heating and cooling loads in the UK and Europe. More extensive data are contained in CIBSE Guide J: *Weather, solar and illuminance data*[1], which deals in detail with meteorological and solar data and provides the basis for the data presented in this chapter.

The main changes from the 1999 edition of CIBSE Guide A involve moving the 20-year basis for the averages from 1976–1995 to 1983–2002 in order to bring the tables and figures up-to-date. Some changes to the contents of the chapter were also felt necessary to reflect the need for useful and meaningful data.

2.1.2 Scope of data

Cold and warm weather data for eight locations in the UK are given in sections 2.3 and 2.4. Since the 1999 edition of this Guide some Met. Office stations have closed. In addition, the formats for data storage have changed with the result that, for some sites, almost a whole year's worth of data are missing. Therefore, their 20-year windows span 1982–2002 with the missing data year omitted. Furthermore, data for some sites had to be compiled from two neighbouring stations. In doing so, efforts were made to ensure that the data are geographically consistent. Table 2.1 shows the 20-year windows for all eight UK sites.

Heating and cooling design temperatures for a range of world-wide locations are given in section 2.6. These data have been selected from the extensive tables of design temperatures for world-wide locations contained in ASHRAE Handbook: *Fundamentals*[2] and are reproduced here by kind permission of ASHRAE.

Measured solar radiation data and sol-air temperatures derived from surface observations are given in this Guide for three UK locations (but not for the same periods), as follows:

— London area (Bracknell)

— Manchester (Aughton) (on CD-ROM)

— Edinburgh (Mylnefield/Strathallan) (on CD-ROM)

Tables of data for Bracknell are included within this section. Tables for all three sites are included on the CD-ROM that accompanies this Guide. The CD-ROM also contains tables of theoretical solar irradiances, based on computer algorithms, for all latitudes. The theoretical basis for these tables is fully explained in CIBSE Guide J: *Weather, solar and illuminance data*[1].

Incomplete radiation data sets are available for a number of UK locations[3].

In addition to the tabulated data, for building simulation purposes, hourly data for CIBSE-defined Test Reference Years (TRYs) and Design Summer Years (DSYs) have been derived for 14 locations in the UK, see Table 2.2. These are available separately from CIBSE. The basis for selection of the DSYs is given in CIBSE Guide J[1].

Table 2.1 Locations and averaging periods for tabulated data

Location	Station	WMO station number	Lat.	Long.	Altitude / m	Start year	End year	Missing year
Belfast	Aldergrove	39170	54.66 °N	6.22 °W	63	1983	2002	—
Birmingham*	Elmdon	35340	52.45 °N	1.74 °W	96	1982	1996	1997
	Coleshill	35350	52.48 °N	1.69 °W	96	1998	2002	—
Cardiff*	Rhoose	37150	51.40 °N	3.34 °W	65	1982	1996	1997
	St. Athan	37160	51.40 °N	3.44 °W	49	1998	2002	—
Edinburgh	Turnhouse	31600	55.95 °N	3.35 °W	35	1983	2002	—
Glasgow	Abbotsinch	31400	55.87 °N	4.43 °W	5	1983	2002	—
London	Heathrow	37720	51.48 °N	0.45 °W	25	1982	2002	1997
Manchester	Ringway	33340	53.36 °N	2.28 °W	69	1983	2002	—
Plymouth	Mount Batten	38270	50.35 °N	4.12 °W	50	1982	2002	1999

* Data for this site compiled from neighbouring stations

Table 2.2 Locations for which hourly data for CIBSE Design Summer Year and Test Reference Year are available

Location	Station	WMO station number	Lat.	Long.	Alt. / m
Belfast	Aldergrove	39170	54.66 °N	6.22 °W	63
Birmingham*	Elmdon	35340	52.45 °N	1.74 °W	96
	Coleshill	35350	52.48 °N	1.69 °W	96
Cardiff*	Rhoose	37150	51.40 °N	3.34 °W	65
	St. Athan	37160	51.40 °N	3.44 °W	49
Edinburgh	Turnhouse	31600	55.95 °N	3.35 °W	35
Glasgow	Abbotsinch	31400	55.87 °N	4.43 °W	5
Leeds	Leeds W.C.	33470	53.80 °N	1.56 °W	64
London	Heathrow	37720	51.48 °N	0.45 °W	25
Manchester	Ringway	33340	53.36 °N	2.28 °W	69
Newcastle	Newcastle W.C.	32450	54.98 °N	1.59 °W	52
Norwich	Coltishall	34950	52.76 °N	1.36 °E	17
Nottingham	Nottingham W.C.	33540	53.01 °N	1.25 °W	117
Plymouth	Mount Batten	38270	50.35 °N	4.12 °W	50
Southampton	S'hampton W.C.	38650	50.90 °N	1.41 °W	3
Swindon	Boscombe Down	37460	51.16 °N	1.75 °W	126

* Data for this site compiled from neighbouring stations

2.1.3 Time differences

With the exception of solar radiation data, climatological data for the UK are referred to mean time, i.e. Greenwich Mean Time (GMT). Solar radiation data are referred to sun time, i.e. local apparent time (LAT), on which scale the sun crosses the north–south meridian at noon. The difference between the two scales leads to GMT being slow relative to sun time by a maximum of 16 minutes in early November and fast by a maximum of 14 minutes in mid-February. There is a further difference at longitudes other than 0° (the reference longitude for GMT), GMT being slow/fast on sun time by 4 minutes for every degree east/west of 0° longitude, respectively.

In other countries, local mean time (LAT) is that given by the corresponding time zone, which usually differs by an integer number of hours from GMT. LAT in these other countries differs from LAT in the same manner that LAT differs from GMT in the UK. Care must be taken in design to account for daylight saving time where applicable.

2.2 Notation

A Weibull coefficient
C Proportion of the year (%)
$C_R(z)$ Roughness coefficient at height z (for wind speed calculations)
C_T Topography coefficient (for wind speed calculations)
DF Daylight factor (%)
E_v Diffuse illuminance received on the horizontal plane (klux)
$E_{vd}(c)$ Diffuse horizontal irradiance available for proportion c (%) of the year (klux)
f_o Orientation factor
$F(v)$ Probability that wind speed v will be exceeded
$f(v)$ Frequency of occurrence of wind speed v
H Effective height of topographical feature (for wind speed calculations) (m)
k Weibull coefficient
K_R Terrain factor (for wind speed calculations)
L_d Length of the downwind slope in the wind direction (m)
L_e Effective length of upwind slope (for wind speed calculations) (m)
L_u Length of the upwind slope in the wind direction (m)
s Slope factor (for wind speed calculations)
v Wind speed to nearest metre per second ($m \cdot s^{-1}$)
v_{ref} Reference regional wind speed ($m \cdot s^{-1}$)
v_z Wind speed at height z ($m \cdot s^{-1}$)
x Horizontal distance of site from the top of crest (for wind speed calculations) (m)
z Vertical distance from ground level of site (for wind speed calculations) (m)
z_{min} Minimum height (for wind speed calculations) (m)
z_0 Roughness length (for wind speed calculations) (m)
Φ Upwind slope (H / L_u) in wind direction

2.3 UK cold weather data

2.3.1 Winter design temperatures

2.3.1.1 General

Outside design temperatures are near-extreme values of dry bulb temperature used to determine the sizes of central plant, distribution systems and room terminals for heating in winter. Design temperature has a large influence on the capital cost of building services systems, and some influence on running costs. No single design temperature is given for a particular location; rather, a range from which the designer can select an appropriate design temperature in consultation with the client (bearing in mind the previous sentence).

The selection method described in this Guide is founded in that first presented by Jamieson(4). This method reflects the thermal response of the building by defining two different averaging times. For most buildings, a 24-hour mean temperature is appropriate. However, a 48-hour mean temperature is more suitable for buildings with high thermal inertia (i.e. high thermal mass, low heat losses), with a response factor $f_r \geq 6$. Response factor is defined in chapter 5.

It is important to choose the correct temperature for the type of building and the level of performance needed. This Guide recommends selection based on variable risk, so that different design temperatures may be chosen for different risks of exceedence. This choice needs to be agreed between designer and client, taking account of the consequences for the building, its users and contents, and of the severity and duration of periods when external design temperatures are exceeded. For buildings with low thermal inertia, earlier editions of Guide A recommended design temperatures such that, on average, only one day in each heating season had a lower mean temperature. Similarly, for buildings with high inertia, a design temperature was recommended such that only one two-day period in each heating season had a lower mean temperature.

Initial selection of an external winter design temperature should be based upon meteorological data from the nearest recording station to the site of the subject building. Then corrections for specific sites should be applied to account for altitude and heat island effects. Suitable corrections are suggested in sections 2.3.1.3 and 2.10, respectively. Correction of tabulated wind data to the actual site conditions is complicated and methods are given in section 2.8

2.3.1.2 Average frequency of occurrence of low temperatures

Data for eight sites are presented to provide an indicative guide to the selection of an external winter design temperature, based on 24-hour and 48-hour means, suitable for determining building fabric heat loss, see Table 2.3. The data are presented in graphical form in Figures 2.1 to 2.8.

The graphs show the average number of times per year that 24-hour and 48-hour mean temperatures fall below a given value. Overlapping 48-hour periods have been eliminated to avoid giving undue weight to several consecutive days of cold weather. Thus, if it were found that, for example, the average daily temperature for January 1 and January 2 of a year fell between –1 °C and 2 °C, the next 48-hour average would be taken between January 3 and January 4, and not between January 2 and January 3. However, if it were found that the average did not fall below –1 °C, then the next 48-hour average would be taken between January 2 and January 3 in order to check if the temperature fell below –1 °C.

Note that two different vertical scales are used for frequency: 0 to 12 for the colder sites (Glasgow, Birmingham and Edinburgh), and 0 to 6 for the remainder. The numerical data on which these graphs are based are given in Table 2.3.

In Table 2.3 the data are cumulative, i.e. the –1 limit represents all values that were less than –1, the –2 value represents all values less than –2, and so on.

Table 2.3 Binned frequencies of occurrence of low 24-hour and 48-hour average temperatures

Upper bin temp.* / °C	Binned frequency of occurrence of low 24-hour and 48-hour average temperatures / °C							
	Belfast (Aldergrove) (altitude: 68 m)		Birmingham (Elmdon) (altitude: 96 m)		Cardiff (Rhoose) (altitude: 67 m)		Edinburgh (Turnhouse) (altitude: 35 m)	
	24-hour	48-hour	24-hour	48-hour	24-hour	48-hour	24-hour	48-hour
–13	0	0	0	0	0	0	0	0
–12	0	0	0	0	0	0	0	0
–11	0	0	0	0	0	0	0.05	0
–10	0	0	0	0.05	0	0	0.05	0
–9	0	0	0.05	0.05	0	0	0.1	0.05
–8	0	0	0.1	0.05	0	0	0.15	0.05
–7	0	0	0.2	0.2	0.05	0	0.15	0.05
–6	0	0	0.45	0.25	0.1	0.05	0.2	0.05
–5	0.15	0	0.9	0.35	0.3	0.1	0.7	0.1
–4	0.3	0.05	1.55	0.5	0.5	0.2	1	0.2
–3	0.6	0.2	2.5	1.15	0.85	0.3	2.25	0.75
–2	0.85	0.4	5.05	2.5	1.8	0.6	3.95	1.5
–1	1.8	0.8	8.05	4.15	3.45	1.75	6.3	3.2

* Temperature equal to or less than stated upper bin temperature; 1 degree bins

Note: single zero indicates no occurrence

Table 2.3 Binned frequencies of occurrence of low 24-hour and 48-hour average temperatures — *continued*

Upper bin temp.* / °C	Binned frequency of occurrence of low 24-hour and 48-hour average temperatures / °C							
	Glasgow (Abbotsinch) (altitude: 68 m)		London (Heathrow) (altitude: 96 m)		Manchester (Ringway) (altitude: 67 m)		Plymouth (Mount Batten) (altitude: 35 m)	
	24-hour	48-hour	24-hour	48-hour	24-hour	48-hour	24-hour	48-hour
–13	0	0.05	0	0	0	0	0	0
–12	0	0.05	0	0	0	0	0	0
–11	0.05	0.05	0	0	0	0	0	0
–10	0.05	0.05	0	0	0	0	0	0
–9	0.05	0.05	0	0	0	0	0	0
–8	0.1	0.1	0	0	0	0	0	0
–7	0.15	0.15	0.05	0	0.1	0	0	0
–6	0.25	0.2	0.1	0.05	0.1	0.1	0	0
–5	0.6	0.25	0.15	0.05	0.25	0.1	0.1	0
–4	1.25	0.4	0.55	0.1	0.55	0.1	0.15	0.05
–3	2.55	0.95	0.8	0.25	0.95	0.1	0.25	0.1
–2	4.1	1.55	1.95	0.75	1.85	0.7	0.6	0.25
–1	6.9	3.2	3.55	1.85	4	2.25	1.15	0.55

* Temperature equal to or less than stated upper bin temperature; 1 degree bins

Note: single zero indicates no occurrence

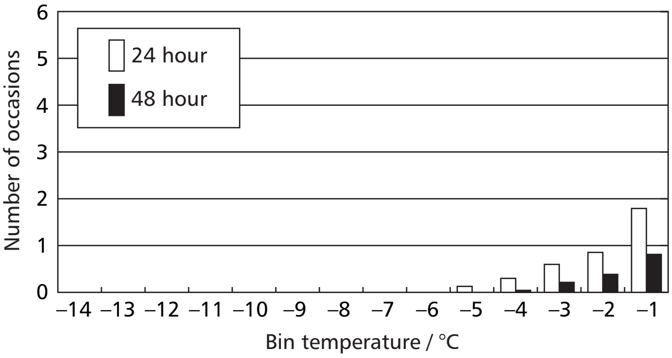


Figure 2.1 Winter temperature distribution: Belfast (Aldergrove) (1983–2002)

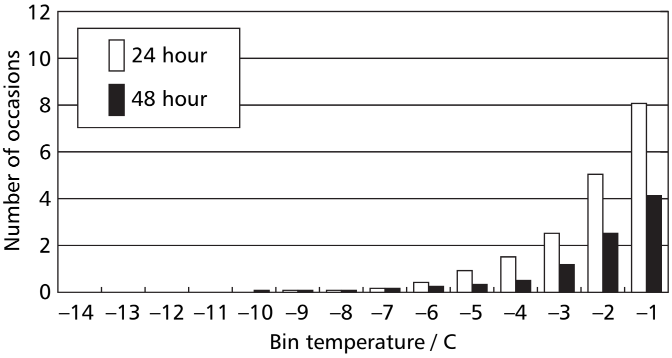


Figure 2.2 Winter temperature distribution: Birmingham (Elmdon) (1982–1996), (Coleshill) (1998–2002)

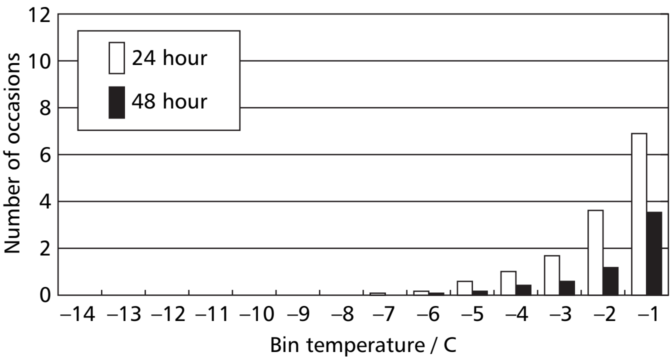


Figure 2.3 Winter temperature distribution: Cardiff (Rhoose) (1982–1996), (St. Athan) (1998–2002)

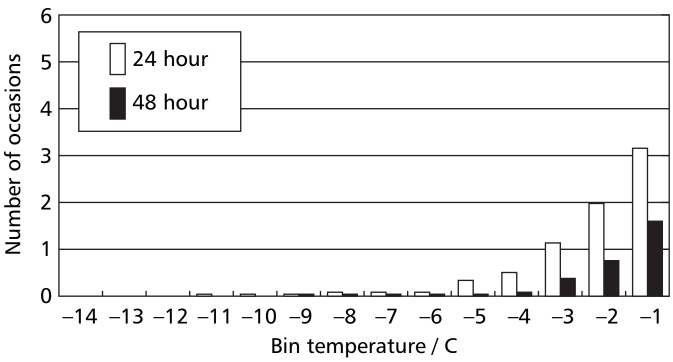


Figure 2.4 Winter temperature distribution: Edinburgh (Turnhouse) (1983–2002)

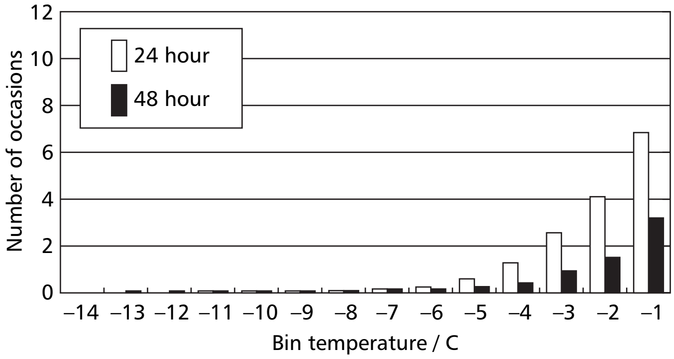


Figure 2.5 Winter temperature distribution: Glasgow (Abbotsinch) (1983–2002)

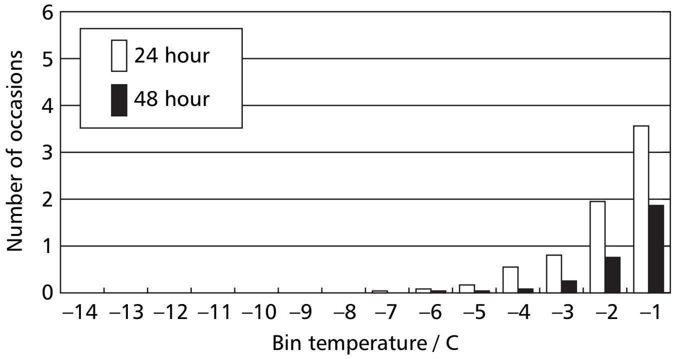


Figure 2.6 Winter temperature distribution: London (Heathrow) (1982–2002)

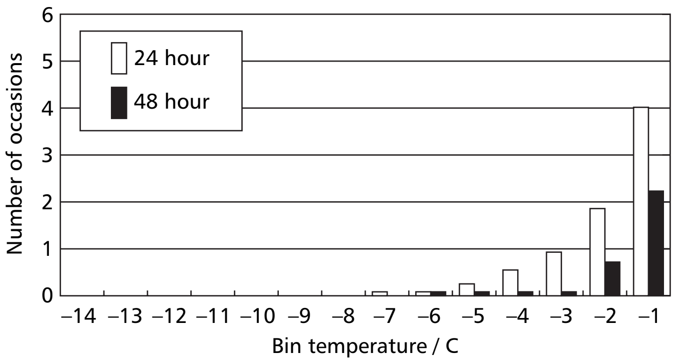


Figure 2.7 Winter temperature distribution: Manchester (Ringway) (1982–2002)

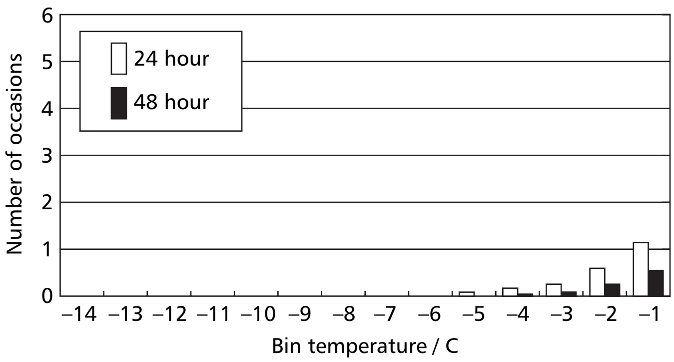


Figure 2.8 Winter temperature distribution: Plymouth (Mount Batten) (1983–2002)

Table 2.4 Wintertime dry bulb temperatures and coincident wet bulb temperatures equal to or exceeded for given percentages of hours in the year (approx. 1982–2002; see Table 2.1)

Location	Hourly temperature (/ °C) equal to or exceeded for stated percentage of hours in the year							
	99.6%		99%		98%		95%	
	Dry-bulb	Wet-bulb	Dry-bulb	Wet-bulb	Dry-bulb	Wet-bulb	Dry-bulb	Wet-bulb
Belfast	–2.6	–3.1	–1.2	–1.8	–0.2	–0.8	1.3	0.5
Birmingham	–5.4	–5.6	–3.4	–3.8	–2.0	–2.4	0.3	–0.4
Cardiff	–3.2	–4.0	–1.6	–2.4	–0.4	–1.2	1.5	0.6
Edinburgh	–5.4	–5.6	–3.4	–3.7	–1.9	–2.3	0.3	–0.5
Glasgow	–5.9	–6.0	–3.9	–4.1	–2.1	–2.6	0.2	–0.5
London	–3.3	–4.0	–1.8	–2.5	–0.6	–1.3	1.4	0.5
Manchester	–3.6	–4.0	–2.2	–2.7	–0.9	–1.7	0.9	0.0
Plymouth	–1.6	–2.6	–0.2	–1.2	0.9	–0.1	2.9	1.9

2.3.1.3 Design temperatures corresponding to given frequencies of occurrence

An alternative approach, useful for an air heating system, is to determine the temperature that is not exceeded for a given frequency of occurrence. Table 2.4 provides for eight sites, the winter dry bulb temperatures equal to or exceeded by specified percentage of hours in the year with coincident wet bulb temperatures.

These design temperatures should be reduced by 0.6 °C for every complete 100 m by which the altitude of the design site exceeds that of the recording station indicated.

2.3.2 Warm front condensation

In addition to the condensation that may occur due to moisture generated within buildings, see chapter 7, condensation may also occur when the weather changes at the end of a cold spell, if a cold air-mass is replaced within a few hours by a warm, moist air-mass. For heavyweight structures, the atmospheric dew-point may rise more quickly than the surface temperature, producing temporary condensation on such surfaces exposed to outside air. This is most likely to happen internally on surfaces in poorly heated or unheated buildings such as warehouses and storage buildings. Condensation may also occur on the contents of the building. In such conditions, unheated buildings should, as far as possible, be closed to exclude humid external air.

Table 2.5 shows the frequency of such conditions for eight UK locations. These values are based on positive differences between the dew-point temperature in the middle of the day (mean of values at 09, 12 and 15 GMT) and the mean dry bulb temperature of the previous day (mean of hourly values from 01 to 24 GMT). Almost all days with a risk of condensation from external air occur during the cold half of the year. The values given in Table 2.5 indicate the likely frequency of condensation following a rapid rise in atmospheric dew-point assuming that the surface temperature of an unheated, heavyweight building at the end of a cold spell is equal to the mean outside dry bulb temperature. Note that Table 2.5 is not a cumulative distribution. Therefore, the 0.0 to 0.9 bin represents only those values that lay between 0.0 and 0.9.

For a typical heavyweight structure that has approached thermal equilibrium during a cold spell and is ventilated at 1 air change per hour, the surface temperature of the inside walls will increase by only one-quarter of the rise in outside air temperature during the first 12 hours. Condensation will occur on the walls but at only about 20% of the possible rate because the low ventilation rate limits the amount of water vapour available. Increasing the rate of ventilation causes a corresponding increase in condensation.

Table 2.5 Average number of occasions per year when the mean of the dew-point temperature at 09:00, 12:00 and 15:00 exceeds the preceding day's dry bulb temperature by the amount indicated (approx. 1982–2002; see Table 2.1)

Amount by which dry bulb temperature is exceeded / K	Average number of occasions per year for stated location*							
	Belfast	Birmingham	Cardiff	Edinburgh	Glasgow	London	Manchester	Plymouth
0.0–0.9	21.7	18.8	21.9	17.4	18.8	14.7	16.1	21.5
1.0–1.9	10.7	10.3	10.9	11.2	11.3	8.1	8.9	10.6
2.0–2.9	6	5.9	6	5.4	5.6	4.4	4.2	5.4
3.0–3.9	3.4	2.4	2.7	3.1	3.2	3	2.7	2.3
4.0–4.9	2.1	1.9	1.5	1.5	2.3	1.2	1.2	0.9
5.0–5.9	0.4	0.9	0.4	0.75	1.3	0.9	0.4	0.7
6.0–6.9	0.3	0.3	0.3	0.2	0.4	0.3	0.2	0.2
7.0–7.9	0.05	0.3	0.05	0.2	0.3	0.1	0.05	0.05
8.0–8.9	0	0.1	0.05	0.1	0.3	0.2	0.05	0
9.0–9.9	0	0.05	0	0.05	0.05	0	0	0
10.0–10.9	0	0.05	0	0	0	0	0	0
11.0–11.9	0	0	0	0	0.05	0	0	0
> 12	0	0	0	0	0	0	0	0

Notes: single zero indicates no occurrences; calculated using 24-hour average daily dry and wet bulb temperatures and standard pressure.

Sensitivity to condensation varies with the thermal characteristics and ventilation strategy of the building. Most unheated 'heavyweight' buildings are liable to suffer condensation from this cause at least once per year. Condensation may also occur in buildings held at a steady temperature other than that of the outside air. The values in Table 2.5 may be adapted for this purpose by adding to, or subtracting from, the temperature difference values in the left-hand column. For example, for a building in Glasgow, Table 2.5 shows there are on average 0.05 occasions per year (i.e. 1 occasion in 20 years) when the temperature difference is in the range 9.0 to 9.9 °C. If the initial temperature of the building were 3 °C above mean outside dry bulb temperature, there would be 0.05 occasions per year when the temperature difference is 6.0 to 6.9 °C.

2.4 UK warm weather data

2.4.1 Coincidence of wet and dry bulb temperatures

For use in air conditioning plant design, Table 2.6 gives hourly dry bulb temperatures equal to or exceeded by specified percentages of hours in the year with coincident hourly wet bulb temperatures.

The frequency of coincidence of wet and dry bulb temperatures is important for air conditioning and natural ventilation design for warm weather. Tables 2.7 to 2.14 give these data for eight sites, over 24 hours, for the four months from June to September. Tables 2.15 and 2.16 show the percentage frequency with which the hourly dry bulb and wet bulb temperatures respectively exceed given values for eight UK locations. These are simply the cumulative frequencies for wet and dry bulb temperature derived from the totals given in Tables 2.7 to 2.14.

The data in Tables 2.7 to 2.14 may be used to plot the percentage frequencies of combinations of hourly dry bulb and wet bulb temperatures on a psychrometric chart. This enables the frequency with which the specific enthalpy exceeds given values to be determined, from which summer design conditions may be established. Figure 2.9 shows the data for London (Heathrow) plotted on a psychrometric chart.

Table 2.6 Summertime dry bulb temperatures and coincident wet bulb temperatures equal to or exceeded for given percentages of hours in the year (approx. 1982–2002; see Table 2.1)

Location	Hourly temperature (/ °C) equal to or exceeded for stated percentage of hours in the year							
	0.4%		1%		2%		5%	
	Dry-bulb	Wet-bulb	Dry-bulb	Wet-bulb	Dry-bulb	Wet-bulb	Dry-bulb	Wet-bulb
Belfast	22.6	18.0	20.8	17.1	19.3	16.3	17.3	14.9
Birmingham	26.1	19.2	24.1	18.2	22.4	17.3	19.6	15.9
Cardiff	24.6	19.0	22.6	18.0	21.0	17.2	18.6	16.0
Edinburgh	22.2	17.8	20.6	16.8	19.2	15.9	17.2	14.6
Glasgow	23.5	18.2	21.3	17.1	19.7	16.2	17.4	14.7
London	28.0	20.0	26.0	19.1	24.3	18.2	21.5	16.9
Manchester	25.5	18.8	23.4	17.9	21.7	17.0	19.0	15.6
Plymouth	23.5	18.7	21.8	17.9	20.4	17.2	18.5	16.1

Table 2.7 Percentage frequency of combinations of hourly dry bulb and wet bulb temperatures for June to September: Belfast (Aldergrove) (1983–2002)

Dry bulb temp. / °C	Wet bulb temperature / °C													Total
	–2 to 0	0 to 2	2 to 4	4 to 6	6 to 8	8 to 10	10 to 12	12 to 14	14 to 16	16 to 18	18 to 20	20 to 22	22 to 24	
–2 to 0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0 to 2	0	0.01	0	0	0	0	0	0	0	0	0	0	0	0.01
2 to 4	0	0.02	0.18	0	0	0	0	0	0	0	0	0	0	0.20
4 to 6	0	0	0.15	0.63	0	0	0	0	0	0	0	0	0	0.78
6 to 8	0	0	0	0.67	1.77	0	0	0	0	0	0	0	0	2.44
8 to 10	0	0	0	0.03	2.93	5.05	0	0	0	0	0	0	0	8.01
10 to 12	0	0	0	0	0.55	9.11	9.29	0	0	0	0	0	0	18.95
12 to 14	0	0	0	0	0.04	2.30	14.27	8.58	0	0	0	0	0	25.19
14 to 16	0	0	0	0	0	0.25	4.26	12.15	5.02	0	0	0	0	21.67
16 to 18	0	0	0	0	0	0.01	0.50	4.61	6.29	1.62	0	0	0	13.03
18 to 20	0	0	0	0	0	0	0.04	0.72	2.78	2.09	0.16	0	0	5.79
20 to 22	0	0	0	0	0	0	0.01	0.06	0.65	1.33	0.37	0	0	2.41
22 to 24	0	0	0	0	0	0	0	0.01	0.17	0.55	0.34	0.02	0	1.09
24 to 26	0	0	0	0	0	0	0	0	0.02	0.11	0.19	0.03	0	0.34
26 to 28	0	0	0	0	0	0	0	0	0	0.01	0.05	0.02	0	0.08
28 to 30	0	0	0	0	0	0	0	0	0	0	0	0	0	0.01
30 to 32	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Total	0	0.03	0.33	1.33	5.29	16.71	28.36	26.13	14.92	5.71	1.11	0.07	0	

Table 2.8 Percentage frequency of combinations of hourly dry bulb and wet bulb temperatures for June to September: Birmingham (Elmdon) (1983–1996), (Coleshill) (1998–2002)

Dry bulb temp. / °C	Wet bulb temperature / °C													Total
	–2 to 0	0 to 2	2 to 4	4 to 6	6 to 8	8 to 10	10 to 12	12 to 14	14 to 16	16 to 18	18 to 20	20 to 22	22 to 24	
–2 to 0	0.01	0	0	0	0	0	0	0	0	0	0	0	0	0.01
0 to 2	0	0.07	0	0	0	0	0	0	0	0	0	0	0	0.07
2 to 4	0	0	0.19	0	0	0	0	0	0	0	0	0	0	0.19
4 to 6	0	0	0.07	0.55	0	0	0	0	0	0	0	0	0	0.62
6 to 8	0	0	0	0.52	1.65	0	0	0	0	0	0	0	0	2.16
8 to 10	0	0	0	0.05	2.28	3.77	0	0	0	0	0	0	0	6.10
10 to 12	0	0	0	0	0.49	6.19	6.48	0	0	0	0	0	0	13.16
12 to 14	0	0	0	0	0.12	2.42	10.52	7.23	0	0	0	0	0	20.29
14 to 16	0	0	0	0	0.01	0.68	4.60	10.25	4.37	0	0	0	0	19.91
16 to 18	0	0	0	0	0	0.04	1.57	5.76	6.43	1.73	0	0	0	15.55
18 to 20	0	0	0	0	0	0	0.25	2.53	4.17	2.65	0.20	0	0	9.79
20 to 22	0	0	0	0	0	0	0.01	0.51	2.26	2.36	0.63	0.01	0	5.78
22 to 24	0	0	0	0	0	0	0	0.06	0.77	1.62	0.88	0.05	0	3.39
24 to 26	0	0	0	0	0	0	0	0.02	0.22	0.67	0.71	0.15	0	1.77
26 to 28	0	0	0	0	0	0	0	0	0.05	0.21	0.34	0.15	0	0.76
28 to 30	0	0	0	0	0	0	0	0	0.01	0.08	0.17	0.08	0.01	0.34
30 to 32	0	0	0	0	0	0	0	0	0	0.02	0.05	0.01	0	0.09
Total	0.01	0.07	0.26	1.12	4.55	13.11	23.43	26.37	18.28	9.35	2.98	0.44	0.01	

Table 2.9 Percentage frequency of combinations of hourly dry bulb and wet bulb temperatures for June to September: Cardiff (Rhoose) (1982–1996), (St. Athan) (1998–2002)

Dry bulb temp. / °C	Wet bulb temperature / °C													Total
	–2 to 0	0 to 2	2 to 4	4 to 6	6 to 8	8 to 10	10 to 12	12 to 14	14 to 16	16 to 18	18 to 20	20 to 22	22 to 24	
–2 to 0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0 to 2	0	0	0	0	0	0	0	0	0	0	0	0	0	0
2 to 4	0	0.01	0.03	0	0	0	0	0	0	0	0	0	0	0.03
4 to 6	0	0	0.05	0.26	0	0	0	0	0	0	0	0	0	0.31
6 to 8	0	0	0	0.34	1.05	0	0	0	0	0	0	0	0	1.38
8 to 10	0	0	0	0.01	1.61	2.49	0	0	0	0	0	0	0	4.11
10 to 12	0	0	0	0	0.20	4.42	5.14	0	0	0	0	0	0	9.77
12 to 14	0	0	0	0	0.03	1.42	10.24	8.73	0	0	0	0	0	20.42
14 to 16	0	0	0	0	0	0.21	3.72	13.91	8.69	0	0	0	0	26.53
16 to 18	0	0	0	0	0	0.01	0.69	5.18	10.31	3.84	0	0	0	20.02
18 to 20	0	0	0	0	0	0	0.06	1.08	4.10	3.75	0.30	0	0	9.28
20 to 22	0	0	0	0	0	0	0.01	0.15	1.15	2.28	0.62	0.04	0	4.24
22 to 24	0	0	0	0	0	0	0	0.02	0.32	0.92	0.71	0.05	0.03	2.05
24 to 26	0	0	0	0	0	0	0	0	0.07	0.35	0.39	0.11	0.02	0.94
26 to 28	0	0	0	0	0	0	0	0	0.01	0.10	0.22	0.06	0.01	0.40
28 to 30	0	0	0	0	0	0	0	0	0	0.02	0.07	0.04	0	0.14
30 to 32	0	0	0	0	0	0	0	0	0	0	0.01	0.01	0	0.02
Total	0	0.01	0.08	0.61	2.89	8.55	19.86	29.07	24.64	11.26	2.32	0.31	0.06	

Table 2.10 Percentage frequency of combinations of hourly dry bulb and wet bulb temperatures for June to September: Edinburgh (Turnhouse) (1983–2002)

Dry bulb temp. / °C	Wet bulb temperature / °C													Total
	–2 to 0	0 to 2	2 to 4	4 to 6	6 to 8	8 to 10	10 to 12	12 to 14	14 to 16	16 to 18	18 to 20	20 to 22	22 to 24	
–2 to 0	0.03	0	0	0	0	0	0	0	0	0	0	0	0	0.03
0 to 2	0.01	0.13	0	0	0	0	0	0	0	0	0	0	0	0.14
2 to 4	0	0.04	0.33	0	0	0	0	0	0	0	0	0	0	0.38
4 to 6	0	0	0.21	1.09	0	0	0	0	0	0	0	0	0	1.31
6 to 8	0	0	0.01	0.91	2.75	0	0	0	0	0	0	0	0	3.66
8 to 10	0	0	0	0.08	3.44	5.86	0	0	0	0	0	0	0	9.37
10 to 12	0	0	0	0	0.71	8.96	9.09	0	0	0	0	0	0	18.76
12 to 14	0	0	0	0	0.07	2.67	12.91	8.44	0	0	0	0	0	24.10
14 to 16	0	0	0	0	0	0.50	5.07	10.31	3.91	0	0	0	0	19.79
16 to 18	0	0	0	0	0	0.01	1.26	4.94	5.61	0.98	0	0	0	12.79
18 to 20	0	0	0	0	0	0.01	0.10	1.24	2.76	1.82	0.07	0	0	6.00
20 to 22	0	0	0	0	0	0	0.01	0.13	0.80	1.19	0.24	0	0	2.37
22 to 24	0	0	0	0	0	0	0	0.02	0.17	0.41	0.29	0.01	0	0.90
24 to 26	0	0	0	0	0	0	0	0.01	0.02	0.13	0.11	0.04	0	0.30
26 to 28	0	0	0	0	0	0	0	0	0	0.02	0.03	0.01	0	0.05
28 to 30	0	0	0	0	0	0	0	0	0	0.01	0.03	0	0	0.03
30 to 32	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Total	0.04	0.17	0.55	2.07	6.97	18.00	28.43	25.09	13.27	4.55	0.77	0.06	0	

Table 2.11 Percentage frequency of combinations of hourly dry bulb and wet bulb temperatures for June to September: Glasgow (Abbotsinch) (1983–2002)

Dry bulb temp. / °C	Wet bulb temperature / °C													Total
	–2 to 0	0 to 2	2 to 4	4 to 6	6 to 8	8 to 10	10 to 12	12 to 14	14 to 16	16 to 18	18 to 20	20 to 22	22 to 24	
–2 to 0	0.03	0	0	0	0	0	0	0	0	0	0	0	0	0.03
0 to 2	0.01	0.24	0	0	0	0	0	0	0	0	0	0	0	0.25
2 to 4	0	0.05	0.48	0	0	0	0	0	0	0	0	0	0	0.53
4 to 6	0	0	0.21	1.19	0	0	0	0	0	0	0	0	0	1.40
6 to 8	0	0	0.01	0.79	2.67	0	0	0	0	0	0	0	0	3.47
8 to 10	0	0	0	0.07	3.11	5.56	0	0	0	0	0	0	0	8.75
10 to 12	0	0	0	0	0.50	8.45	8.83	0	0	0	0	0	0	17.77
12 to 14	0	0	0	0	0.10	2.37	13.26	8.50	0	0	0	0	0	24.23
14 to 16	0	0	0	0	0	0.46	4.63	11.33	4.64	0	0	0	0	21.07
16 to 18	0	0	0	0	0	0.01	0.97	4.72	5.52	1.25	0	0	0	12.48
18 to 20	0	0	0	0	0	0	0.04	1.02	2.69	1.66	0.13	0	0	5.56
20 to 22	0	0	0	0	0	0	0.01	0.14	0.82	1.20	0.24	0	0	2.41
22 to 24	0	0	0	0	0	0	0	0.04	0.17	0.62	0.38	0.01	0	1.23
24 to 26	0	0	0	0	0	0	0	0	0.04	0.19	0.31	0.05	0	0.59
26 to 28	0	0	0	0	0	0	0	0	0	0.04	0.10	0.06	0	0.21
28 to 30	0	0	0	0	0	0	0	0	0	0.01	0.01	0	0	0.03
30 to 32	0	0	0	0	0	0	0	0	0	0	0	0	0	0
Total	0.04	0.29	0.70	2.05	6.38	16.86	27.74	25.76	13.89	4.98	1.17	0.14	0.01	

Table 2.12 Percentage frequency of combinations of hourly dry bulb and wet bulb temperatures for June to September: London (Heathrow) (1982–2002)

Dry bulb temp. / °C	Wet bulb temperature / °C													Total
	–2 to 0	0 to 2	2 to 4	4 to 6	6 to 8	8 to 10	10 to 12	12 to 14	14 to 16	16 to 18	18 to 20	20 to 22	22 to 24	
–2 to 0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0 to 2	0	0	0	0	0	0	0	0	0	0	0	0	0	0
2 to 4	0	0	0.01	0	0	0	0	0	0	0	0	0	0	0.01
4 to 6	0	0	0.01	0.14	0	0	0	0	0	0	0	0	0	0.15
6 to 8	0	0	0	0.20	0.57	0	0	0	0	0	0	0	0	0.77
8 to 10	0	0	0	0.01	1.08	1.67	0	0	0	0	0	0	0	2.75
10 to 12	0	0	0	0	0.21	3.87	3.65	0	0	0	0	0	0	7.73
12 to 14	0	0	0	0	0.03	1.33	8.32	5.33	0	0	0	0	0	15.02
14 to 16	0	0	0	0	0	0.39	3.87	10.68	4.88	0	0	0	0	19.82
16 to 18	0	0	0	0	0	0.03	1.77	6.16	9.26	2.06	0	0	0	19.28
18 to 20	0	0	0	0	0	0	0.23	3.35	5.60	4.54	0.30	0	0	14.01
20 to 22	0	0	0	0	0	0	0	0.91	3.48	3.54	0.95	0.01	0	8.89
22 to 24	0	0	0	0	0	0	0	0.05	1.20	2.79	1.38	0.08	0	5.50
24 to 26	0	0	0	0	0	0	0	0	0.24	1.43	1.37	0.24	0	3.27
26 to 28	0	0	0	0	0	0	0	0	0.04	0.42	0.87	0.35	0	1.69
28 to 30	0	0	0	0	0	0	0	0	0.01	0.09	0.38	0.26	0.01	0.74
30 to 32	0	0	0	0	0	0	0	0	0	0.01	0.11	0.15	0.03	0.29
Total	0	0	0.01	0.35	1.89	7.29	17.84	26.49	24.72	14.88	5.34	1.08	0.04	

Table 2.13 Percentage frequency of combinations of hourly dry bulb and wet bulb temperatures for June to September: Manchester (Ringway) (1983–2002)

Dry bulb temp. / °C	Wet bulb temperature / °C													Total
	–2 to 0	0 to 2	2 to 4	4 to 6	6 to 8	8 to 10	10 to 12	12 to 14	14 to 16	16 to 18	18 to 20	20 to 22	22 to 24	
–2 to 0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0 to 2	0	0.03	0	0	0	0	0	0	0	0	0	0	0	0.03
2 to 4	0	0.01	0.11	0	0	0	0	0	0	0	0	0	0	0.11
4 to 6	0	0	0.06	0.30	0	0	0	0	0	0	0	0	0	0.36
6 to 8	0	0	0	0.45	1.31	0	0	0	0	0	0	0	0	1.76
8 to 10	0	0	0	0.04	2.12	3.50	0	0	0	0	0	0	0	5.66
10 to 12	0	0	0	0.01	0.58	6.65	6.63	0	0	0	0	0	0	13.87
12 to 14	0	0	0	0	0.11	2.62	12.13	7.54	0	0	0	0	0	22.40
14 to 16	0	0	0	0	0	0.59	5.26	11.59	4.45	0	0	0	0	21.91
16 to 18	0	0	0	0	0	0.02	1.47	5.95	6.76	1.55	0	0	0	15.74
18 to 20	0	0	0	0	0	0	0.07	1.84	3.94	2.65	0.16	0	0	8.65
20 to 22	0	0	0	0	0	0	0.02	0.27	1.76	2.25	0.44	0	0	4.74
22 to 24	0	0	0	0	0	0	0	0.03	0.53	1.30	0.64	0.03	0	2.54
24 to 26	0	0	0	0	0	0	0	0.01	0.11	0.57	0.53	0.09	0	1.32
26 to 28	0	0	0	0	0	0	0	0	0.01	0.20	0.33	0.09	0	0.64
28 to 30	0	0	0	0	0	0	0	0	0	0.02	0.16	0.03	0	0.22
30 to 32	0	0	0	0	0	0	0	0	0	0	0.03	0.01	0	0.04
Total	0	0.04	0.17	0.80	4.13	13.40	25.58	27.23	17.56	8.54	2.29	0.26	0	

Table 2.14 Percentage frequency of combinations of hourly dry bulb and wet bulb temperatures for June to September: Plymouth (Mount Batten) (1982–2002)

Dry bulb temp. / °C	Wet bulb temperature / °C													Total
	–2 to 0	0 to 2	2 to 4	4 to 6	6 to 8	8 to 10	10 to 12	12 to 14	14 to 16	16 to 18	18 to 20	20 to 22	22 to 24	
–2 to 0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0 to 2	0	0.02	0	0	0	0	0	0	0	0	0	0	0	0.02
2 to 4	0	0	0.01	0	0	0	0	0	0	0	0	0	0	0.01
4 to 6	0	0	0	0.11	0	0	0	0	0	0	0	0	0	0.11
6 to 8	0	0	0	0.10	0.53	0	0	0	0	0	0	0	0	0.64
8 to 10	0	0	0	0	0.70	2.06	0	0	0	0	0	0	0	2.75
10 to 12	0	0	0	0	0.09	3.06	4.92	0	0	0	0	0	0	8.07
12 to 14	0	0	0	0	0.01	0.80	9.37	10.64	0	0	0	0	0	20.81
14 to 16	0	0	0	0	0.02	0.18	3.19	14.70	12.24	0	0	0	0	30.33
16 to 18	0	0	0	0	0	0.01	0.59	4.51	10.93	4.80	0	0	0	20.85
18 to 20	0	0	0	0	0	0	0.05	1.00	3.88	4.71	0.40	0	0	10.05
20 to 22	0	0	0	0	0	0	0	0.11	1.08	1.96	0.73	0.01	0	3.90
22 to 24	0	0	0	0	0	0	0	0.02	0.19	0.81	0.58	0.04	0	1.64
24 to 26	0	0	0	0	0	0	0	0	0.03	0.17	0.31	0.07	0	0.58
26 to 28	0	0	0	0	0	0	0	0	0	0.02	0.14	0.03	0	0.19
28 to 30	0	0	0	0	0	0	0	0	0	0	0.02	0.02	0	0.03
30 to 32	0	0	0	0	0	0	0	0	0	0	0	0.01	0	0.02
Total	0	0.02	0.01	0.22	1.35	6.10	18.12	30.99	28.36	12.48	2.18	0.18	0	

Table 2.15 Percentage frequency for which the hourly dry bulb temperatures exceeds the stated values in the period June to September (1983–2002)

Dry bulb temp. / °C	Frequency / %							
	Belfast	Birmingham	Cardiff	Edinburgh	Glasgow	London	Manchester	Plymouth
12	47.95	57.44	67.67	43.74	45.94	72.55	55.88	74.19
13	46.02	55.67	65.88	41.14	43.78	71.66	54.22	71.85
14	39.36	50.21	58.94	35.30	37.44	67.22	48.34	63.55
15	30.75	43.17	48.43	28.17	29.21	60.14	40.60	50.79
16	22.20	35.59	36.34	21.08	21.47	51.65	32.30	36.61
17	14.91	28.16	25.75	14.60	14.93	42.52	24.67	24.89
18	9.67	21.66	17.02	9.55	9.98	34.17	18.04	16.36
19	6.15	16.35	11.45	5.95	6.68	26.69	13.03	10.14
20	3.93	12.12	7.79	3.65	4.47	20.38	9.47	6.36
21	2.49	8.87	5.29	2.20	3.07	15.54	6.74	3.90
22	1.52	6.35	3.56	1.29	2.06	11.50	4.75	2.46
23	0.89	4.43	2.33	0.74	1.39	8.39	3.23	1.50
24	0.43	2.96	1.51	0.39	0.83	6.00	2.21	0.82
25	0.20	1.89	0.91	0.17	0.46	4.13	1.46	0.47
26	0.09	1.19	0.56	0.09	0.24	2.72	0.89	0.24
27	0.03	0.75	0.28	0.05	0.10	1.71	0.53	0.10
28	0.01	0.43	0.16	0.03	0.03	1.03	0.25	0.05
29	0.00	0.22	0.06	0.01	0.00	0.62	0.10	0.03
30	0.00	0.09	0.02	0.00	0.00	0.29	0.04	0.02
31	0.00	0.02	0.01	0.00	0.00	0.11	0.01	0.00

Note: 0.00 + indicates a value greater than zero but less than 0.005.

Table 2.16 Percentage frequency for which the hourly wet bulb temperatures exceed the stated values in the period June to September (1982–2002)

Dry bulb temp. / °C	Frequency / %							
	Belfast	Birmingham	Cardiff	Edinburgh	Glasgow	London	Manchester	Plymouth
12	47.95	57.44	67.67	43.74	45.94	72.55	55.88	74.19
13	33.52	43.76	53.32	29.57	31.64	59.85	41.55	59.55
14	21.82	31.07	38.60	18.65	20.18	46.06	28.65	43.20
15	13.04	20.88	24.70	10.67	11.62	32.61	18.60	27.20
16	6.89	12.79	13.95	5.38	6.29	21.35	11.09	14.84
17	3.23	6.98	6.60	2.34	3.13	12.63	5.69	6.55
18	1.18	3.44	2.69	0.82	1.31	6.46	2.55	2.36
19	0.38	1.39	1.08	0.24	0.49	3.02	0.88	0.76
20	0.08	0.46	0.37	0.06	0.14	1.12	0.26	0.18
21	0.02	0.11	0.13	0.00	0.02	0.24	0.03	0.03
22	0.00	0.01	0.06	0.00	0.01	0.04	0.00	0.00
23	0.00	0.00	0.02	0.00	0.00	0.00	0.00	0.00

Note: 0.00 + indicates a value greater than zero but less than 0.005.

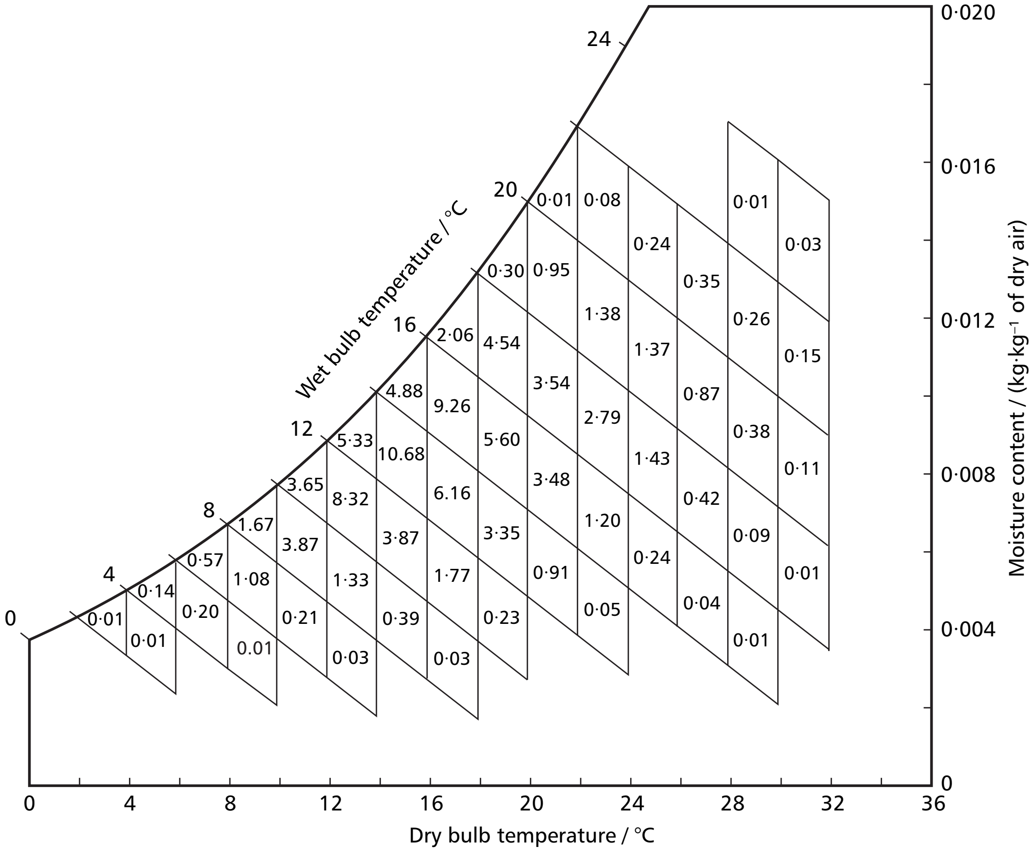


Figure 2.9 Percentage frequencies of wet and dry bulb temperatures plotted on a psychrometric chart: London (Heathrow) (1982–2002)

2.4.2 Design temperatures — approximate method

Where wet and dry bulb temperature data for the required locality are not sufficiently comprehensive to enable the analysis illustrated in Figure 2.9, the following method may be employed. This uses only general information available from the UK Met Office or other sources. Note that this is *not* the basis of the world-wide cooling design temperatures given in section 2.6.

The method requires the following data:

— average monthly maximum dry bulb temperature for the hottest month (e.g. for July, the average over a period of years of the highest temperature in each July within that period)

— average daily maximum dry bulb temperature (e.g. for July, the average over a period of years of the highest temperatures for each July day within that period; i.e. for a 30-year period, the average of the maximum temperatures on all 930 July days within that period)

— average daily minimum relative humidity (e.g. for July, the average over a period of years of the lowest relative humidity for each July day in that period).

The design temperature is obtained as follows:

(1) The month having the highest average *monthly* maximum dry bulb temperature is selected; this highest average *monthly* maximum dry bulb temperature is taken as the design dry bulb temperature.

(2) For the month selected in step 1, using psychrometric charts or tables[5], a moisture content, dew-point temperature or vapour pressure is determined for the average *daily* maximum dry bulb temperature and average *daily* minimum relative humidity.

(3) The moisture content, dew-point temperature or vapour pressure determined in step 2 is combined with the highest average *monthly* maximum dry bulb temperature determined in step 1 to give a screen wet bulb temperature, which is taken as the design wet bulb temperature.

Note that if the CIBSE psychrometric chart is used, step 3 yields the sling wet bulb temperature rather than the screen wet-bulb. The difference between screen and sling wet bulb temperatures varies with position on the chart.

Example 2.1

For Berlin, meteorological data indicate July as the month having the highest average monthly maximum dry bulb temperature (31.8 °C). For July, the average daily maximum dry bulb temperature is 23.5 °C, the average daily minimum relative humidity (at 14:00 h) is 60%.

Step 1: the design dry bulb temperature is 31.8 °C (rounded to 32 °C).

Step 2: from psychrometric tables[5], for relative humidity of 61% and dry bulb temperature of 23.5 °C:

— moisture content = 0.01102 $kg{\cdot}kg^{-1}$

— dew-point temperature = 15.5 °C

— vapour pressure = 1.757 kPa

Step 3: from psychrometric tables, for dry bulb temperature of 32 °C and moisture content of 0.01102 $kg{\cdot}kg^{-1}$ (or equivalent dew-point temperature or vapour pressure), the screen wet bulb temperature is 21.7 °C. Therefore, the design wet bulb temperature is 21.7 °C (rounded to 22 °C).

2.5 Accumulated temperature difference (degree-days and degree-hours)

Accumulated temperature differences are relatively simple forms of climatic data, useful as an index of climatic severity as it affects energy use for space heating or cooling. Accumulated temperature differences are calculated as the difference between the prevailing external, dry bulb temperature and a 'base temperature'. This is the external temperature at which, in theory, no artificial heating (or cooling) is required to maintain an acceptable internal temperature.

Two types of degree-day are used in building services engineering. Heating degree-days (K·day) indicate the severity of the heating season and therefore heating energy requirements. Cooling degree-days (K·day), or cooling degree-hours (K·h), indicate the warmth of the summer and hence cooling requirements. The most widely used form of accumulated temperature difference is heating degree-days, which have proved particularly useful in monitoring heating energy consumption in buildings from year to year[(6)].

Table 2.17 gives 20-year averages of monthly and annual heating degree-day totals for all 18 degree-day regions, referred to the traditional standard base temperature of 15.5 °C. These data are standard degree-day totals, calculated from daily maximum and minimum temperature.

Tables 2.18 to 2.20 give heating degree-day and cooling degree-hour totals for a range of base temperatures for London (Heathrow), Manchester (Ringway) and Edinburgh (Turnhouse), respectively. Cooling degree-hours are calculated when the external dry bulb temperature exceeds the stated base temperature. Degree-day data may be obtained from (http://vesma.com).

Table 2.17 Mean monthly and annual heating degree-day totals (base temperature 15.5°C) for 18 UK degree-day regions (1976–1995)

Degree-day region		Mean total degree-days (K·day)												
		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Year
1	Thames Valley (Heathrow)	340	309	261	197	111	49	20	23	53	128	234	308	2033
2	South-eastern (Gatwick)	351	327	283	218	135	68	32	38	75	158	254	324	2255
3	Southern (Hurn)	338	312	279	222	135	70	37	42	77	157	246	311	2224
4	South-western (Plymouth)	286	270	249	198	120	58	23	26	52	123	200	253	1858
5	Severn Valley (Filton)	312	286	253	189	110	46	17	20	48	129	217	285	1835
6	Midland (Elmdon)	365	338	291	232	153	77	39	45	85	186	271	344	2425
7	W Pennines (Ringway)	360	328	292	220	136	73	34	42	81	170	259	331	2228
8	North-western (Carlisle)	370	329	309	237	159	89	45	54	101	182	271	342	2388
9	Borders (Boulmer)	364	328	312	259	197	112	58	60	102	186	270	335	2483
10	North-eastern (Leeming)	379	339	304	235	159	83	40	46	87	182	272	345	2370
11	E Pennines (Finningley)	371	339	294	228	150	79	39	45	82	174	266	342	2307
12	E Anglia (Honington)	371	338	294	228	143	74	35	37	70	158	264	342	2254
13	W Scotland (Abbotsinch)	380	336	317	240	159	93	54	64	107	206	286	358	2494
14	E Scotland (Leuchars)	390	339	320	253	185	104	57	65	113	204	290	362	2577
15	NE Scotland (Dyce)	394	345	331	264	194	116	62	72	122	216	295	365	2668
16	Wales (Aberporth)	328	310	289	231	156	89	44	44	77	156	234	294	2161
17	N Ireland (Aldergrove)	362	321	304	234	158	88	47	56	102	189	269	330	2360
18	NW Scotland (Stornoway)	336	296	332	260	207	124	85	88	135	214	254	330	2671

Table 2.18 Monthly heating degree-day and cooling degree-hour totals to various base temperatures: Belfast (Aldergrove) (1983–2002)

Base temp. / °C	Monthly heating degree-days (/ K·day) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
10	172	153	126	87	35	9	1	2	11	42	102	150
12	233	209	184	138	70	26	6	9	28	81	156	209
14	295	265	246	194	117	58	22	28	61	133	214	270
15.5	341	307	293	240	159	91	47	53	98	177	258	317
16	357	322	309	254	172	102	55	62	109	192	273	332
18	419	378	371	314	231	157	102	110	166	253	333	394
18.5	434	393	386	328	245	172	117	124	180	268	349	409
20	481	435	433	373	292	215	158	167	225	315	393	455
Base temp. / °C	**Monthly cooling degree-hours (/ K·h) for stated base temperature**											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
5	784	768	1387	2175	4215	5705	7439	7178	5395	3683	1795	1092
12	2	3	9	122	618	1272	2377	2191	1024	324	30	7
18	0	0	0	4	35	108	271	198	29	1	0	0

Table 2.19 Monthly heating degree-day and cooling degree-hour totals to various base temperatures: Birmingham (Elmdon) (1983–1996), (Coleshill) (1998–2002)

Base temp. / °C	Monthly heating degree-days (/ K·day) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
10	183	166	126	87	35	8	2	3	10	41	109	164
12	241	219	181	134	66	22	6	8	27	76	161	222
14	303	276	241	187	109	47	18	23	52	124	218	282
15.5	349	319	286	231	147	75	36	40	82	166	263	328
16	365	332	302	245	159	84	43	47	92	180	277	344
18	427	389	364	304	215	131	76	85	142	240	337	406
18.5	444	402	380	319	229	143	88	98	156	255	351	421
20	489	445	426	364	274	182	122	133	199	302	397	467
Base temp. / °C	Monthly cooling degree-hours (/ K·h) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
5	936	902	1617	2469	4736	6629	8696	8296	6095	4072	1857	1166
12	3	9	55	254	993	2049	3607	3251	1603	524	63	22
18	0	0	1	10	120	344	838	651	147	10	0	0

Table 2.20 Monthly heating degree-day and cooling degree-hour totals to various base temperatures: Cardiff (Rhoose) (1982–1996), (St. Athan) (1998–2002)

Base temp. / °C	Monthly heating degree-days (/ K·day) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
10	147	142	109	74	23	6	1	1	5	24	78	128
12	208	196	167	121	51	16	3	4	14	51	124	185
14	270	253	229	177	93	39	11	13	36	92	180	246
15.5	316	296	276	222	132	66	26	29	63	134	225	292
16	332	310	291	236	144	76	32	35	72	148	240	308
18	394	366	353	296	201	125	71	74	123	208	300	370
18.5	410	379	369	310	216	139	82	86	137	223	315	385
20	456	422	415	355	261	180	118	124	180	270	360	431
Base temp. / °C	Monthly cooling degree-hours (/ K·h) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
5	1214	1030	1640	2497	4952	6575	8529	8350	6488	4726	2473	1550
12	1	3	16	147	902	1868	3388	3228	1781	641	70	10
18	0	0	0	5	90	222	541	410	81	3	0	0

Table 2.21 Monthly heating degree-day and cooling degree-hour totals to various base temperatures: Edinburgh (Turnhouse) (1983–2002)

Base temp. / °C	Monthly heating degree-days (/ K·day) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
10	193	167	142	101	42	13	3	5	17	55	122	176
12	253	221	199	151	83	34	11	13	38	95	175	235
14	315	277	260	208	132	67	31	33	71	146	233	296
15.5	362	319	306	252	173	100	55	58	107	189	279	340
16	377	334	322	266	187	112	65	67	119	204	293	357
18	438	391	384	326	247	165	111	113	174	264	353	419
18.5	453	405	401	341	264	179	126	126	190	279	367	436
20	500	447	446	386	308	222	168	169	233	326	413	480
Base temp. / °C	Monthly cooling degree-hours (/ K·h) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
5	688	737	1311	1965	3825	5526	7231	7157	5245	3535	1643	962
12	3	6	13	112	478	1209	2250	2242	1007	355	41	8
18	0	0	0	4	24	100	226	218	43	2	0	0

Table 2.22 Monthly heating degree-day and cooling degree-hour totals to various base temperatures: Glasgow (Abbotsinch) (1983–2002)

Base temp. / °C	Monthly heating degree-days (/ K·day) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
10	187	163	137	95	39	13	4	6	19	57	119	177
12	248	217	195	142	73	32	11	15	39	96	172	236
14	309	274	256	198	118	64	30	35	75	148	230	297
15.5	357	317	303	242	159	97	54	61	108	192	276	342
16	371	330	319	256	172	107	63	71	120	206	290	359
18	433	387	381	316	230	159	109	116	176	267	350	421
18.5	448	400	396	330	245	171	122	130	190	282	365	438
20	495	443	443	375	290	214	163	172	234	329	410	483
Base temp. / °C	Monthly cooling degree-hours (/ K·h) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
5	764	755	1326	2184	4303	5758	7412	7160	5212	3513	1712	1040
12	1	0	6	140	684	1374	2422	2261	993	314	31	5
18	0	0	0	9	54	144	317	248	30	1	0	0

Table 2.23 Monthly heating degree-day and cooling degree-hour totals to various base temperatures: London (Heathrow) (1982–2002)

Base temp. / °C	Monthly heating degree-days (/ K·day) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
10	150	140	99	61	16	2	0	0	4	22	84	132
12	207	192	151	101	37	8	1	2	11	46	130	187
14	267	247	208	150	72	24	6	8	28	86	184	246
15.5	314	290	255	192	105	45	16	18	51	124	228	293
16	329	304	269	206	117	52	20	23	59	135	243	307
18	391	360	331	264	168	91	45	50	100	192	302	369
18.5	406	373	345	277	182	102	55	58	113	207	317	384
20	453	417	393	323	224	138	82	87	152	253	362	431
Base temp. / °C	Monthly cooling degree-hours (/ K·h) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
5	1347	1216	2166	3236	5935	7820	9965	9630	7232	5101	2507	1622
12	8	20	109	443	1626	2972	4787	4467	2454	962	158	43
18	0	0	2	32	274	635	1388	1158	308	33	0	0

Table 2.24 Monthly heating degree-day and cooling degree-hour totals to various base temperatures: Manchester (Ringway) (1983–2002)

Base temp. / °C	Monthly heating degree-days (/ K·day) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
10	174	154	116	79	26	6	0	1	7	34	98	155
12	233	208	174	125	56	19	3	6	20	69	150	213
14	295	264	235	179	98	47	15	19	48	116	207	274
15.5	341	306	281	223	137	76	33	38	78	158	251	320
16	357	320	296	237	148	86	40	45	89	172	266	336
18	419	377	358	296	204	135	79	86	140	232	326	398
18.5	435	391	374	310	219	148	89	99	154	247	342	414
20	481	433	420	355	264	189	127	136	199	294	386	460
Base temp. / °C	Monthly cooling degree-hours (/ K·h) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
5	907	919	1627	2539	4923	6420	8432	8136	6052	4206	2001	1192
12	3	7	39	237	1006	1813	3306	3059	1491	530	68	18
18	0	0	0	14	120	270	643	525	95	9	0	0

Table 2.25 Monthly heating degree-day and cooling degree-hour totals to various base temperatures: Plymouth (Mount Batten) (1982–2002)

Base temp. / °C	Monthly heating degree-days (/ K·day) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
10	110	104	78	58	18	2	0	0	3	15	52	88
12	165	156	133	104	43	9	1	2	9	35	92	136
14	227	212	195	159	84	29	8	8	26	71	143	196
15.5	274	254	241	204	122	57	22	23	50	109	186	241
16	289	268	257	218	135	67	29	29	58	122	203	256
18	351	323	318	278	193	117	67	66	109	183	262	317
18.5	367	337	334	293	209	132	80	80	123	199	278	333
20	413	379	380	338	254	173	119	118	167	245	321	378
Base temp. / °C	Monthly cooling degree-hours (/ K·h) for stated base temperature											
	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
5	1864	1527	2217	2821	5144	6699	8444	8428	6779	5290	3135	2279
12	1	1	22	151	908	1857	3274	3269	1962	873	140	19
18	0	0	0	5	82	167	421	356	65	4	0	0

2.6 World-wide weather data

This section and the data on the CD-ROM that accompanies this Guide provide climatic design information for some 250 locations world-wide. These have been selected from the 4422 locations for which similar data are given on the CD-ROM that accompanies the 2005 ASHRAE Handbook: *Fundamentals*(2). The CIBSE gratefully acknowledges the American Society of Heating, Refrigerating and Air-Conditioning Engineers for permission to reproduce these data and the accompanying text.

Figure 2.10 shows an example data sheet and Table 2.26 lists the locations for which similar data sheets are included on the CD-ROM that accompanies this Guide.

This climatic design information is commonly used for design, sizing, distribution, installation, and marketing of heating, ventilating, air conditioning, and dehumidification equipment, as well as for other energy-related processes in residential, agricultural, commercial, and industrial applications. These summaries include values of dry bulb, wet bulb, and dew-point temperatures, and wind speed with direction at various frequencies of occurrence.

The information includes design values of dry-bulb with mean coincident wet bulb temperature, design wet-bulb with mean coincident dry bulb temperature, and design dew-point with mean coincident dry bulb temperature and corresponding humidity ratio. These data allow the designer to consider various operational peak conditions.

Warm-season temperature and humidity conditions are based on annual percentiles of 0.4, 1.0, and 2.0. Cold-season conditions are based on annual percentiles of 99.6 and 99.0. The use of annual percentiles to define design conditions ensures that they represent the same probability of occurrence in any climate, regardless of the seasonal distribution of extreme temperature and humidity.

Other elements included in the data sheets are cold-season dew point temperature with coincident humidity ratio and dry bulb temperature; warm-season enthalpy with coincident dry bulb temperature; extreme maximum wet bulb temperature; 5-, 10-, 20- and 50-year return period for maximum and minimum extreme dry bulb temperature; hottest/coldest month; UTC (Universal Time Coordinated) offset; time zone code; and monthly mean daily dry bulb temperature range. Also, monthly tables of 0.4, 1.0, and 2.0% dry bulb temperature with mean coincident wet bulb temperature, and wet bulb temperature with mean coincident dry bulb temperature are included for all stations.

2.6.1 Climatic design conditions

2.6.1.1 Annual design conditions

The annual climatic design conditions for Atlanta GA are shown in Figure 2.10 to illustrate the format of the data. Columns are numbered and described as follows:

— *Station information*:
  - (*a*) name of observing station
  - (*b*) World Meteorological Organization (WMO) station identifier
  - (*c*) station latitude of (°N/S)
  - (*d*) station longitude (°E/W)
  - (*e*) station elevation (m)
  - (*f*) standard pressure at elevation (kPa)
  - (*g*) time zone (h ± UTC)
  - (*h*) time zone code (e.g. NAE = Eastern Time, USA and Canada)
  - (*i*) period analysed (e.g. 7201 = 1972–2001)

— *Annual heating and humidification design conditions*:
  - (2) coldest month (i.e., month with lowest average dry bulb temperature; 1 = January, 12 = December)
  - (3) dry bulb temperature corresponding to 99.6 and 99.0% annual cumulative frequency of occurrence (cold conditions) (°C)
  - (4) dew-point temperature corresponding to 99.6 and 99.0% annual cumulative frequency of occurrence (°C); corresponding humidity ratio, calculated at standard atmospheric pressure at elevation of station

(g·kg^{-1} of dry air); mean coincident dry bulb temperature (°C)

(5) wind speed corresponding to 0.4 and 1.0% cumulative frequency of occurrence for coldest month (column 2) (m·s^{-1}); mean coincident dry bulb temperature (°C)

(6) mean wind speed coincident with 99.6% dry bulb temperature (column 3a) (m·s^{-1}); corresponding most frequent wind direction, degrees from north (east = 90°)

— *Annual cooling, dehumidification and enthalpy design conditions*:

(7) hottest month (i.e. month with highest average dry bulb temperature; 1 = January, 12 = December)

(8) daily temperature range for hottest month, (°C) (defined as mean of the difference between daily max. and daily min. dry bulb temperatures for hottest month (column 7))

(9) dry bulb temperature corresponding to 0.4, 1.0, and 2.0% annual cumulative frequency of occurrence (warm conditions) (°C); mean coincident wet bulb temperature (°C)

(10) wet bulb temperature corresponding to 0.4, 1.0, and 2.0% annual cumulative frequency of occurrence (°C); mean coincident dry bulb temperature (°C)

(11) mean wind speed coincident with 0.4% dry bulb temperature (column 9a) (m·s^{-1}); corresponding most frequent wind direction, degrees from north.

(12) dew-point temperature corresponding to 0.4, 1.0, and 2.0% annual cumulative frequency of occurrence (°C); corresponding humidity ratio, calculated at the standard atmospheric pressure at elevation of station (g·kg^{-1} of dry air); mean coincident dry bulb temperature (°C)

(13) enthalpy corresponding to 0.4, 1.0, and 2.0% annual cumulative frequency of occurrence (kJ·kg^{-1}); mean coincident dry bulb temperature (°C).

— *Extreme annual design conditions*:

(14) wind speed corresponding to 1.0, 2.5, and 5.0% annual cumulative frequency of occurrence (m·s^{-1})

(15) extreme maximum wet bulb temperature (°C)

(16) mean and standard deviation of extreme annual maximum and minimum dry bulb temperature (°C)

(17) 5-, 10-, 20-, and 50-year return period values for maximum and minimum extreme dry bulb temperature (°C).

2.6.1.2 Monthly design conditions

Monthly percentiles of dry bulb and wet bulb temperature, and monthly mean daily dry bulb temperature range are provided in columns 18–20 of the data sheets (see Figure 2.10). These values are derived from the same analysis that results in the annual design conditions.

The monthly summaries are useful when seasonal variations in solar geometry and intensity, building or facility occupancy, or building use patterns require consideration. In particular, these values can be used when determining air-conditioning loads during periods of maximum solar radiation.

The monthly information is identified by the following column numbers:

(18) dry bulb temperature corresponding to 0.4, 1.0, and 2.0% cumulative frequency of occurrence for indicated month (°C); mean coincident wet bulb temperature (°C)

(19) wet bulb temperature corresponding to 0.4, 1.0, and 2.0% cumulative frequency of occurrence for indicated month (°C); mean coincident dry bulb temperature (°C)

(20) mean daily temperature range for month indicated (°C) (defined as mean of difference between daily maximum/minimum dry bulb temperatures).

For a 30-day month, the 0.4, 1.0, and 2.0% values of occurrence represent the value that occurs or is exceeded for a total of 3, 7, or 14 h, respectively, per month on average over the period of record.

Monthly percentile values of dry or wet bulb temperature may be higher or lower than the design conditions corresponding to the same nominal percentile, depending on the month and the seasonal distribution of the parameter at that location. Generally, for the hottest or most humid months of the year, the monthly percentile value exceeds the design condition for the same element corresponding to the same nominal percentile. For example, in Figure 2.10, column 9a shows that the annual 0.4% design dry bulb temperature at Atlanta is 34.4 °C. Column 18 shows that the 0.4% monthly dry bulb temperature exceeds 34.4 °C for June, July, and August, with values of 34.8, 36.8, and 35.8 °C, respectively.

2.6.2 Data sources

The following three primary sources of observational data sets were used for the calculation of the design values given in the data sheets included on the CD-ROM that accompanies this Guide:

— *USA*: hourly weather observations from Surface Airways Meteorological and Solar Observing Network (SAMSON) data from the National Climatic Data Center (NCDC) for US observing locations from 1961 to 1990[7]

— *Canada*: hourly weather records for the period 1972 to 2001 for Canadian locations from the Canadian Weather Energy and Engineering Data Sets (CWEEDS) produced by Environment Canada[8]

— *Rest of world*: Integrated Surface Hourly (ISH) data for stations from around the world provided by NCDC for the period 1982 to 2001[9,10].

2005 ASHRAE Handbook - Fundamentals (SI)

Design conditions for ATLANTA, GA, USA

Station Information

Station name	WMO#	Lat	Long	Elev	StdP	Hours +/- UTC	Time zone code	Period
1a	*1b*	*1c*	*1d*	*1e*	*1f*	*1g*	*1h*	*1i*
ATLANTA	**722190**	**33.65N**	**84.42W**	**315**	**97.60**	**-5.00**	**NAE**	**7201**

Annual Heating and Humidification Design Conditions

Coldest month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB	
	99.6%	99%	99.6%			99%			0.4%		1%			
			DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD
2	*3a*	*3b*	*4a*	*4b*	*4c*	*4d*	*4e*	*4f*	*5a*	*5b*	*5c*	*5d*	*6a*	*6b*
1	**-7.3**	**-4.5**	**-16.7**	**0.9**	**-4.0**	**-14.1**	**1.2**	**-1.7**	**11.7**	**2.5**	**10.8**	**3.0**	**5.3**	**320**

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

Hottest month	Hottest month DB range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB	
		0.4%		1%		2%		0.4%		1%		2%			
		DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD
7	*8*	*9a*	*9b*	*9c*	*9d*	*9e*	*9f*	*10a*	*10b*	*10c*	*10d*	*10e*	*10f*	*11a*	*11b*
7	**9.7**	**34.4**	**23.8**	**33.1**	**23.5**	**31.8**	**23.1**	**25.2**	**31.4**	**24.6**	**30.4**	**24.0**	**29.5**	**4.0**	**300**

Dehumidification DP/MCDB and HR									Enthalpy/MCDB					
0.4%			1%			2%			0.4%		1%		2%	
DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB
12a	*12b*	*12c*	*12d*	*12e*	*12f*	*12g*	*12h*	*12i*	*13a*	*13b*	*13c*	*13d*	*13e*	*13f*
23.5	**19.0**	**27.8**	**22.9**	**18.4**	**26.9**	**22.4**	**17.8**	**26.5**	**78.4**	**31.5**	**75.9**	**30.6**	**73.7**	**29.8**

Extreme Annual Design Conditions

Extreme Annual WS			Extreme Max WB	Extreme Annual DB				n-Year Return Period Values of Extreme DB							
				Mean		Standard deviation		n=5 years		n=10 years		n=20 years		n=50 years	
1%	2.5%	5%	WB	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min
14a	*14b*	*14c*	*15*	*16a*	*16b*	*16c*	*16d*	*17a*	*17b*	*17c*	*17d*	*17e*	*17f*	*17g*	*17h*
9.8	**8.6**	**7.8**	**28.9**	**35.8**	**-11.8**	**2.0**	**3.9**	**37.2**	**-14.6**	**38.4**	**-16.9**	**39.5**	**-19.1**	**41.0**	**-21.9**

Monthly Design Dry Bulb and Mean Coincident Wet Bulb Temperatures

%	Jan		Feb		Mar		Apr		May		Jun	
	DB	MCWB	DB	MCWB	DB	MCWB	DB	MCWB	DB	MCWB	DB	MCWB
	18a	*18b*	*18c*	*18d*	*18e*	*18f*	*18g*	*18h*	*18i*	*18j*	*18k*	*18l*
0.4%	**21.2**	**15.2**	**23.5**	**14.9**	**27.1**	**16.9**	**29.8**	**18.7**	**32.0**	**22.2**	**34.8**	**23.3**
1%	**19.8**	**14.8**	**22.2**	**14.6**	**25.9**	**16.4**	**28.9**	**18.0**	**31.0**	**21.4**	**33.9**	**23.0**
2%	**18.5**	**14.5**	**20.9**	**14.7**	**24.8**	**15.6**	**27.9**	**17.5**	**30.2**	**20.9**	**33.2**	**22.8**

%	Jul		Aug		Sep		Oct		Nov		Dec	
	DB	MCWB	DB	MCWB	DB	MCWB	DB	MCWB	DB	MCWB	DB	MCWB
	18m	*18n*	*18o*	*18p*	*18q*	*18r*	*18s*	*18t*	*18u*	*18v*	*18w*	*18x*
0.4%	**36.8**	**24.3**	**35.8**	**24.0**	**33.5**	**23.2**	**28.6**	**20.5**	**25.3**	**17.8**	**22.2**	**17.2**
1%	**35.8**	**24.1**	**34.7**	**24.0**	**32.4**	**23.1**	**27.7**	**19.9**	**24.1**	**17.2**	**21.2**	**16.8**
2%	**34.9**	**24.0**	**33.8**	**23.8**	**31.5**	**22.7**	**26.8**	**19.1**	**23.2**	**16.6**	**19.9**	**16.0**

Monthly Design Wet Bulb and Mean Coincident Dry Bulb Temperatures

%	Jan		Feb		Mar		Apr		May		Jun	
	WB	MCDB	WB	MCDB	WB	MCDB	WB	MCDB	WB	MCDB	WB	MCDB
	19a	*19b*	*19c*	*19d*	*19e*	*19f*	*19g*	*19h*	*19i*	*19j*	*19k*	*19l*
0.4%	**17.6**	**19.1**	**18.4**	**19.7**	**19.1**	**22.9**	**21.4**	**25.8**	**23.8**	**28.8**	**25.1**	**31.4**
1%	**16.8**	**18.5**	**17.6**	**19.8**	**18.4**	**22.2**	**20.5**	**24.9**	**23.2**	**28.6**	**24.6**	**30.8**
2%	**15.9**	**17.7**	**16.8**	**19.2**	**17.9**	**21.9**	**19.9**	**24.2**	**22.6**	**28.0**	**24.1**	**30.1**

%	Jul		Aug		Sep		Oct		Nov		Dec	
	WB	MCDB	WB	MCDB	WB	MCDB	WB	MCDB	WB	MCDB	WB	MCDB
	19m	*19n*	*19o*	*19p*	*19q*	*19r*	*19s*	*19t*	*19u*	*19v*	*19w*	*19x*
0.4%	**26.7**	**33.8**	**25.8**	**32.0**	**24.9**	**30.4**	**22.4**	**26.1**	**20.1**	**22.5**	**19.0**	**20.6**
1%	**25.9**	**32.5**	**25.3**	**31.6**	**24.2**	**29.3**	**21.6**	**25.1**	**19.6**	**21.9**	**18.0**	**19.7**
2%	**25.5**	**31.8**	**25.0**	**31.1**	**23.9**	**28.8**	**21.0**	**24.3**	**19.0**	**21.2**	**17.2**	**18.9**

Monthly Mean Daily Temperature Range

Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
20a	*20b*	*20c*	*20d*	*20e*	*20f*	*20g*	*20h*	*20i*	*20j*	*20k*	*20l*
9.6	**10.6**	**11.2**	**11.6**	**10.6**	**10.0**	**9.7**	**9.3**	**9.3**	**10.5**	**10.3**	**9.6**

WMO#	World Meteorological Organization number	Lat	Latitude, °	Long	Longitude, °
Elev	Elevation, m	StdP	Standard pressure at station elevation, kPa		
DB	Dry bulb temperature, °C	DP	Dew point temperature, °C	WB	Wet bulb temperature, °C
WS	Wind speed, m/s	Enth	Enthalpy, kJ/kg	HR	Humidity ratio, grams of moisture per kilogram of dry air
MCDB	Mean coincident dry bulb temperature, °C	MCDP	Mean coincident dew point temperature, °C	MCWB	Mean coincident wet bulb temperature, °C
MCWS	Mean coincident wind speed, m/s	PCWD	Prevailing coincident wind direction, °, 0 = North, 90 = East		

Figure 2.10 Worldwide design information; example data sheet (copyright 2005, ©American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (www.ashrae.org). Reprinted by permission from 2005 ASHRAE Handbook— *Fundamentals*. This text may not be copied nor distributed in either paper or digital form without ASHRAE's permission)

Table 2.26 Locations for which external design information is provided on accompanying CD-ROM

Location	WMO number	Lat.	Long.	Elev. / m
ALGERIA				
Constantine/El Bey	604190	36.28 °N	6.62 °E	694
Dar-el-Beida	603900	36.72 °N	3.25 °E	25
ARGENTINA				
Buenos Aires	875760	34.82 °S	58.53 °W	20
San Juan	873110	31.57 °S	68.42 °W	598
ARMENIA				
Yerevan	377890	40.13 °N	44.47 °E	890
ASCENSION ISLAND				
Georgetown	619020	7.97 °S	14.40 °W	79
AUSTRALIA				
Adelaide	946720	34.93 °S	138.52 °E	4
Brisbane	945780	27.38 °S	153.10 °E	5
Canberra	949260	35.30 °S	149.18 °E	577
Darwin	941200	12.40 °S	130.87 °E	30
Perth	946100	31.93 °S	115.95 °E	29
Sydney	947670	33.95 °S	151.18 °E	3
AUSTRIA				
Innsbruck	111200	47.27 °N	11.35 °E	593
Salzburg	111500	47.80 °N	13.00 °E	450
Vienna	110350	48.25 °N	16.37 °E	200
AZORES				
Lajes	85090	38.77 °N	27.10 °W	55
BAHAMAS				
Nassau	780730	25.05 °N	77.47 °W	7
BAHRAIN				
Al–Manamah	411500	26.27 °N	50.65 °E	2
BELARUS				
Babruysk (Bobruysk)	269610	53.12 °N	29.25 °E	165
Minsk	268500	53.87 °N	27.53 °E	234
BELGIUM				
Antwerp	64500	51.20 °N	4.47 °E	14
Brussels	64510	50.90 °N	4.53 °E	58
Oostende	64070	51.20 °N	2.87 °E	5
BENIN				
Parakou	653300	9.35 °N	2.62 °E	393
BERMUDA				
Hamilton	780160	32.37 °N	64.68 °W	3
BOLIVIA				
La Paz	852010	16.52 °S	68.18 °W	4014
BOSNIA–HERZEGOVINA				
Banja Luka	132420	44.78 °N	17.22 °E	156
BOTSWANA				
Gaborone Int. A.	682400	24.22 °S	25.92 °E	1005
BRAZIL				
Brasilia	833780	15.87 °S	47.93 °W	1061
Rio De Janeiro	837460	22.82 °S	43.25 °W	6
Sao Paulo	837800	23.62 °S	46.65 °W	803
BULGARIA				
Sofia	156140	42.65 °N	23.38 °E	595
BRUNEI				
Brunei	963150	4.93 °N	114.93 °E	15
BURKINA FASO				
Ouagadougou	655030	12.35 °N	1.52 °W	306
CANADA				
Calgary	718770	51.12	114.02	1084
Edmonton	711230	53.30	113.58	723
Halifax	713950	44.88	63.50	145
CANADA (*continued*)				
Montreal	716270	45.47 °N	73.75 °W	36
Ottawa	716280	45.32 °N	75.67 °W	114
Quebec	717080	46.80 °N	71.38 °W	73
Toronto	716240	43.67 °N	79.63 °W	173
Vancouver	718920	49.18 °N	123.17 °W	2
Winnipeg	718520	49.90 °N	97.23 °W	239
CANARY ISLANDS				
Las Palmas	600300	27.93 °N	15.38 °W	25
Santa Cruz De Tenerife	600250	28.05 °N	16.57 °W	72
CAPE VERDE				
Sal Island	85940	16.73 °N	22.95 °W	55
CHAD				
Ndjamena	647000	12.13 °N	15.03 °E	295
CHILE				
Concepcion	856820	36.77 °S	73.05 °W	16
Santiago	855740	33.38 °S	70.78 °W	476
CHINA				
Beijing	545110	39.93 °N	116.28 °E	55
Dinghai	584770	30.03 °N	122.12 °E	37
Guangzhou	592870	23.13 °N	113.32 °E	8
Hangzhou	584570	30.23 °N	120.17 °E	43
Jinan	548230	36.68 °N	116.98 °E	58
Kowloon	450070	22.33 °N	114.18 °E	24
Macau	450110	22.20 °N	113.53 °E	59
Nanjing	582380	32.00 °N	118.80 °E	12
Shanghai	583670	31.17 °N	121.43 °E	7
Shenyang	543420	41.77 °N	123.43 °E	43
Tianjin	545270	39.10 °N	117.17 °E	5
Wuhan	574940	30.62 °N	114.13 °E	23
Zhanjiang	596580	21.22 °N	110.40 °E	28
COLOMBIA				
Bogota	802220	4.70 °N	74.13 °W	2548
CONGO				
Brazzaville/Maya-Maya	644500	4.25 °S	15.25 °E	316
CROATIA				
Split	144440	43.53 °N	16.30 °E	19
Zagreb	142410	45.73 °N	16.07 °E	107
CUBA				
Guantanamo	783670	19.90 °N	75.15 °W	17
Havana	782240	22.98 °N	82.40 °W	59
CYPRUS				
Akrotiri	176010	34.58 °N	32.98 °E	23
Larnaca	176090	34.88 °N	33.63 °E	2
CZECH REPUBLIC				
Brno	117230	49.15 °N	16.70 °E	246
Prague	115180	50.10 °N	14.28 °E	366
DENMARK				
Aalborg	60300	57.10 °N	9.87 °E	3
Copenhagen	61800	55.63 °N	12.67 °E	5
ECUADOR				
Quito	840710	0.15 °S	78.48 °W	2812
EGYPT				
Alexandria	623180	31.20 °N	29.95 °E	7
Cairo	623660	30.13 °N	31.40 °E	74
ESTONIA				
Tallinn	260380	59.35 °N	24.80 °E	44
FAEROE ISLANDS				
Torshavn	60110	62.02 °N	6.77 °W	39

Table continues

Table 2.26 Locations for which external design information is provided on accompanying CD-ROM — *continued*

Location	WMO number	Lat.	Long.	Elev. / m
FIJI				
Nadi	916800	17.75 °S	177.45 °E	18
FINLAND				
Helsinki	29740	60.32 °N	24.97 °E	56
Pello	28440	66.80 °N	24.00 °E	84
FRANCE				
Bordeaux	75100	44.83 °N	0.70 °W	61
Brest	71100	48.45 °N	4.42 °W	103
Lyon	74810	45.73 °N	5.08 °E	240
Marseille	76500	43.45 °N	5.23 °E	36
Nice	76900	43.65 °N	7.20 °E	10
Paris, Orly	71490	48.73 °N	2.40 °E	96
Strasbourg	71900	48.55 °N	7.63 °E	154
Toulouse	76300	43.63 °N	1.37 °E	153
GAMBIA				
Banjul	617010	13.35 °N	16.80 °W	36
GERMANY				
Berlin	103840	52.47 °N	13.40 °E	49
Dresden	094880	51.13 °N	13.77 °E	230
Dusseldorf	104000	51.28 °N	6.78 °E	44
Frankfurt	106370	50.05 °N	8.60 °E	113
Hamburg	101470	53.63 °N	10.00 °E	16
Hannover	103380	52.47 °N	9.70 °E	54
Koln	105130	50.87 °N	7.17 °E	99
Leipzig	94690	51.42 °N	12.23 °E	142
Munich	108660	48.13 °N	11.70 °E	529
Stuttgart	107380	48.68 °N	9.22 °E	419
GEORGIA				
Batumi	374840	41.65 °N	41.63 °E	6
Tbilisi	375490	41.68 °N	44.95 °E	467
GIBRALTAR				
North Front	84950	36.15 °N	5.35 °W	5
GREECE				
Athens	167160	37.90 °N	23.73 °E	15
Iraklion (Crete)	167540	35.33 °N	25.18 °E	39
Rodhos (Rhodes)	167490	36.40 °N	28.08 °E	11
Thessaloniki	166220	40.52 °N	22.97 °E	4
GREENLAND				
Godthab	42500	64.17 °N	51.75 °W	70
GUINEA				
Cayenne/Rochambeau	814050	4.83 °N	52.37 °W	9
HUNGARY				
Budapest	128390	47.43 °N	19.27 °E	185
ICELAND				
Reykjavik	40300	64.13 °N	21.90 °W	61
INDIA				
Bangalore	432950	12.97 °N	77.58 °E	921
Bombay	430030	19.12 °N	72.85 °E	14
Calcutta	428090	22.65 °N	88.45 °E	6
Goa/Panaji	431920	15.48 °N	73.82 °E	60
Hyderabad	431280	17.45 °N	78.47 °E	545
Jaipur	423480	26.82 °N	75.80 °E	390
Madras	432790	13.00 °N	80.18 °E	16
New Delhi	421820	28.58 °N	77.20 °E	216
IRELAND				
Dublin	39690	53.43 °N	6.25 °W	85
Kilkenny	39600	52.67 °N	7.27 °W	64
Rosslare	39570	52.25 °N	6.33 °W	25
Shannon	39620	52.70 °N	8.92 °W	20
ISRAEL				
Jerusalem	401840	31.78 °N	35.22 °E	754
ISRAEL (*continued*)				
Tel Aviv	401760	32.10 °N	34.78 °E	4
ITALY				
Bologna	161400	44.53 °N	11.30 °E	42
Milan	160800	45.43 °N	9.28 °E	103
Naples	162890	40.85 °N	14.30 °E	72
Palermo	164050	38.18 °N	13.10 °E	34
Rome	162420	41.80 °N	12.23 °E	3
Venice	161050	45.50 °N	12.33 °E	6
JAMAICA				
Kingston	783970	17.93 °N	76.78 °W	9
JAPAN				
Asahikawa	474070	43.77 °N	142.37 °E	116
Fukuoka	478080	33.58 °N	130.45 °E	12
Hiroshima	477650	34.40 °N	132.47 °E	53
Nagoya	476350	35.25 °N	136.93 °E	17
Sapporo	474120	43.05 °N	141.33 °E	19
Tokyo	476710	35.55 °N	139.78 °E	8
JORDAN				
Amman	402700	31.98 °N	35.98 °E	773
Pavlodar	360030	52.28 °N	76.95 °E	123
Zhambyl (Dzhambul)	383410	42.85 °N	71.38 °E	653
KENYA				
Kisumu	637080	0.10 °S	34.75 °E	1146
Nairobi	637400	1.32 °S	36.92 °E	1624
KOREA, NORTH				
Ch'ongjin	470080	41.78 °N	129.82 °E	43
Changjin	470310	40.37 °N	127.25 °E	1081
P'yongyang	470580	39.03 °N	125.78 °E	38
KOREA, SOUTH				
Cheju	471820	33.50 °N	126.55 °E	27
Inch'on	471120	37.48 °N	126.63 °E	70
Seoul	471100	37.55 °N	126.80 °E	19
KUWAIT				
Kuwait	405820	29.22 °N	47.98 °E	55
KYRGYZSTAN				
Bishkek (Frunze)	383530	42.85 °N	74.53 °E	635
LATVIA				
Riga	264220	56.97 °N	24.07 °E	3
LIBYA				
Banghazi	620530	32.08 °N	20.27 °E	132
Tripoli	620100	32.67 °N	13.15 °E	81
LIECHTENSTEIN				
Vaduz	69900	47.13 °N	9.53 °E	463
LITHUANIA				
Vilnius	267300	54.63 °N	25.28 °E	156
LUXEMBOURG				
Luxembourg	65900	49.62 °N	6.22 °E	379
MACEDONIA				
Skopje	135860	41.97 °N	21.65 °E	239
MADAGASCAR				
Antananarivo/Ivato	670830	18.80 °S	47.48 °E	1276
MADEIRA ISLANDS				
Funchal	85210	32.68 °N	16.77 °W	55
MALAYSIA				
Kuala Lumpur/Subang	486470	3.12 °N	101.55 °E	22

Table continues

Table 2.26 Locations for which external design information is provided on accompanying CD-ROM — *continued*

Location	WMO number	Lat.	Long.	Elev. / m	Location	WMO number	Lat.	Long.	Elev. / m
MALI					QATAR				
Bamako	612910	12.53 °N	7.95 °W	381	Ad Dawhah	411700	25.25 °N	51.57 °E	10
MALTA					ROMANIA				
Luqa	165970	35.85 °N	14.48 °E	91	Bucharest	154200	44.50 °N	26.13 °E	91
MAURITANIA					RUSSIA				
Nouadhibou	614150	20.93 °N	17.03 °W	3	Aldan	310040	58.62 °N	125.37 °E	682
MEXICO					Grozny	372350	43.35 °N	45.68 °E	162
Acapulco	768056	16.77 °N	99.75 °W	5	Moscow	276120	55.75 °N	37.63 °E	156
Cancun	765906	21.03 °N	86.87 °W	5	Omsk	286980	54.93 °N	73.40 °E	123
Mexico City	766790	19.43 °N	99.08 °W	2234	Rostov–Na–Donu	347310	47.25 °N	39.82 °E	77
MOLDOVA					Smolensk	267810	54.75 °N	32.07 °E	241
Chisinau (Kishinev)	338150	47.02 °N	28.87 °E	180	St Petersburg	260630	59.97 °N	30.30 °E	4
MONGOLIA					Vladivostok	319600	43.12 °N	131.90 °E	184
Ulaangom	442120	49.97 °N	92.08 °E	936	Volgograd	345600	48.68 °N	44.35 °E	145
MOROCCO					SAMOA				
Casablanca	601550	33.57 °N	7.67 °W	62	Pago Pago	917650	14.33 °S	170.72 °W	3
Tanger	601010	35.73 °N	5.90 °W	21	SAUDI ARABIA				
MOZAMBIQUE					Jiddah	410240	21.67 °N	39.15 °E	12
Maputo/Mavalane	673410	25.9 °S	32.57 °E	44	Khamis Mushayt	411140	18.30 °N	42.80 °E	2054
NAMIBIA					Riyadh	404380	24.72 °N	46.72 °E	612
Windhoek/Eros	681100	22.57 °S	17.10 °E	1725	SENEGAL				
NETHERLANDS					Dakar	616410	14.73 °N	17.50 °W	24
Amsterdam	62400	52.30 °N	4.77 °E	–2	SINGAPORE				
Eindhoven	63700	51.45 °N	5.42 °E	22	Singapore	486980	1.37 °N	103.98 °E	16
Groningen	62800	53.13 °N	6.58 °E	4	SLOVAKIA				
Rotterdam	63440	51.95 °N	4.45 °E	–4	Bratislava	118160	48.20 °N	17.20 °E	130
NEW ZEALAND					SOLVENIA				
Auckland	931190	37.02 °S	174.80 °E	6	Ljubljana	130140	46.22 °N	14.48 °E	385
Christchurch	937800	43.48 °S	172.55 °E	34	SOUTH AFRICA				
Wellington	934360	41.33 °S	174.80 °E	7	Cape Town	688160	33.98 °S	18.60 °E	42
NIGER					Durban	685880	29.97 °S	30.95 °E	8
Niamey	610520	13.48 °N	2.17 °E	227	Johannesburg	683680	26.13 °S	28.23 °E	1700
NORWAY					Pretoria	682620	25.73 °S	28.18 °E	1322
Bergen	13110	60.30 °N	5.22 °E	50	SPAIN				
Oslo	13840	60.20 °N	11.08 °E	204	Barcelona	81810	41.28 °N	2.07 °E	6
Stavanger	14150	58.88 °N	5.63 °E	9	Madrid	82210	40.45 °N	3.55 °W	582
Trondheim	12710	63.47 °N	10.93 °E	17	Malaga	84820	36.67 °N	4.48 °W	7
OMAN					Palma	83060	39.55 °N	2.73 °E	8
Masqat	412560	23.58 °N	58.28 °E	15	Santander	80230	43.47 °N	3.82 °W	65
PANAMA					Sevilla	83910	37.42 °N	5.90 °W	31
Panama	788060	8.92 °N	79.60 °W	16	Valencia	82840	39.50 °N	0.47 °W	62
PARAGUAY					SWEDEN				
Asuncion	862180	25.27 °S	57.63 °W	101	Goteborg	25260	57.67 °N	12.30 °E	169
PERU					Ostersund	22260	63.18 °N	14.50 °E	370
Lima	846280	12.00 °S	77.12 °W	13	Stockholm	24600	59.65 °N	17.95 °E	61
PHILLIPPINES					SWITZERLAND				
Manila	984290	14.52 °N	121.00 °E	21	Geneva	67000	46.25 °N	6.13 °E	416
POLAND					Lugano	67700	46.00 °N	8.97 °E	276
Gdansk	121500	54.38 °N	18.47 °E	138	Zurich	66600	47.38 °N	8.57 °E	569
Krakow	125660	50.08 °N	19.80 °E	237	SYRIA				
Warsaw	123750	52.17 °N	20.97 °E	107	Damascus	400800	33.42 °N	36.52 °E	605
PORTUGAL					TAIWAN				
Lisbon	85360	38.78 °N	9.13 °W	123	Taipei	466960	25.07 °N	121.55 °E	6
Porto	85450	41.23 °N	8.68 °W	73	TAJIKISTAN				
PUERTO RICO					Dushanbe	388360	38.55 °N	68.78 °E	803
San Juan	785260	18.43 °N	66.00 °W	19	TANZANIA				
					Dar es Salaam	638940	6.87 °S	39.20 °E	55
					THAILAND				
					Bangkok	484560	13.92 °N	100.60 °E	12

Table continues

Table 2.26 Locations for which external design information is provided on accompanying CD-ROM — *continued*

Location	WMO number	Lat.	Long.	Elev. / m
THAILAND (*continued*)				
Tak	483760	16.88 °N	99.15 °E	124
TRINIDAD				
Port of Spain	789700	10.62 °N	61.35 °W	15
TUNISIA				
Tunis	607150	36.83 °N	10.23 °E	4
TURKEY				
Ankara	171280	40.12 °N	32.98 °E	949
Istanbul	170600	40.97 °N	28.82 °E	37
TURKMENISTAN				
Dashhowuz (Tashauz)	383920	41.83 °N	59.98 °E	88
UKRAINE				
Kyyiv (Kiev)	333450	50.40 °N	30.45 °E	168
Odesa	338370	46.48 °N	30.63 °E	35
Poltava	335060	49.60 °N	34.55 °E	159
UNITED ARAB EMIRATES				
Abu Dhabi	412170	24.43 °N	54.65 °E	27
Dubai	411940	25.25 °N	55.33 °E	5
Sharjah	411960	25.33 °N	55.52 °E	33
UNITED KINGDOM				
Aberdeen/Dyce	30910	57.20 °N	2.22 °W	65
Aberporth	35020	52.13 °N	4.57 °W	134
Aughton	33220	53.55 °N	2.92 °W	56
Belfast	39170	54.65 °N	6.22 °W	81
Birmingham	35340	52.45 °N	1.73 °W	99
Bournemouth	38620	50.78 °N	1.83 °W	11
Bristol	37260	51.47 °N	2.60 °W	11
Cardiff	37150	51.40 °N	3.35 °W	67
Edinburgh	31600	55.95 °N	3.35 °W	41
Exeter	38390	50.73 °N	3.42 °W	30
Glasgow	31400	55.87 °N	4.43 °W	8
Jersey/Channel Islands	38950	49.22 °N	2.20 °W	84
Lerwick	30050	60.13 °N	1.18 °W	84
London, Heathrow	37720	51.48 °N	0.45 °W	24
Manchester	33340	53.35 °N	2.27 °W	78
Nottingham	33540	53.00 °N	1.25 °W	117
Plymouth	38270	50.35 °N	4.12 °W	27
Stornoway	30260	58.22 °N	6.32 °W	13
UNITED STATES OF AMERICA				
Atlanta (GA)	722190	33.65 °N	84.42 °W	315
Boston (MA)	725090	42.37 °N	71.03 °W	9
Chicago (IL)	725300	41.98 °N	87.90 °W	205
Cleveland (OH)	725240	41.42 °N	81.87 °W	245
Dallas/Fort Worth (TX)	722590	32.90 °N	97.03 °W	182
Denver (CO)	724690	39.77 °N	104.87 °W	1611
Fairbanks (AK)	702610	64.82 °N	147.87 °W	138
Honolulu (HI)	911820	21.35 °N	157.93 °W	5
Las Vegas (NV)	723860	36.08 °N	115.17 °W	664
Los Angeles (CA)	722950	33.93 °N	118.40 °W	32
Memphis (TN)	723340	35.05 °N	90.00 °W	87
Miami (FL)	722020	25.82 °N	80.28 °W	4
Minneapolis–St. Paul (MN)	726580	44.88 °N	93.22 °W	255
Montgomery (AL)	722260	32.30 °N	86.40 °W	62
New Orleans (LA)	722310	29.98 °N	90.25 °W	9
New York (NY)	744860	40.65 °N	73.78 °W	7
Oklahoma City (OK)	723530	35.40 °N	97.60 °W	397
Philadelphia (PA)	724080	39.88 °N	75.25 °W	9
Phoenix (AZ)	722780	33.43 °N	112.02 °W	337
Richmond (VA)	724010	37.50 °N	77.33 °W	54
Sacramento (CA)	724839	38.70 °N	121.58 °W	7
Salt Lake City (UT)	725720	40.78 °N	111.97 °W	1288
San Francisco (CA)	724940	37.62 °N	122.38 °W	5
Seattle (WA)	727930	47.45 °N	122.30 °W	137
St. Louis (MO)	724340	38.75 °N	90.37 °W	172
Washington DC	724050	38.85 °N	77.03 °W	20
URUGUAY				
Montevideo	865800	34.83 °S	56.00 °W	32
UZBEKISTAN				
Tashkent	384570	41.27 °N	69.27 °E	489
VANUATU				
Luganville	915540	15.52 °S	167.22 °E	44
VENEZUELA				
Caracas	804150	10.60 °N	66.98 °W	48
VIETNAM				
Ho Chi Minh City (Saigon)	489000	10.82 °N	106.67 °E	19
YUGOSLAVIA				
Belgrade	132720	44.82 °N	20.28 °E	99
ZIMBABWE				
Harare	677750	17.92 °S	31.13 °E	1503

2.6.3 Thermal design conditions

Values of ambient dry bulb, dew-point, and wet bulb temperature and wind speed corresponding to the various annual percentiles represent the value that is exceeded on average by the indicated percentage of the total number of hours in a year (8760). The 0.4, 1.0, 2.0, and 5.0% values are exceeded on average 35, 88, 175, and 438 h per year, respectively, for the period of record. The design values occur more frequently than the corresponding nominal percentile in some years and less frequently in others. The 99.0% and 99.6% (cold-season) values are defined in the same way but are usually viewed as the values for which the corresponding weather element is less than the design condition for 88 and 35 h, respectively.

Simple design conditions were obtained by binning hourly data into frequency vectors, then deriving from the binned data the design condition having the probability of being exceeded a certain percentage of the time. Mean coincident values were obtained by double-binning the hourly data into joint frequency matrices, then calculating the mean coincident value corresponding to the simple design condition.

The weather data sets used for the calculations often contain missing values—either isolated records, or because some stations report data only every third hour. Gaps up to 6 h were filled by linear interpolation to provide as complete a time series as possible. Dry bulb temperature, dew-point temperature, station pressure and humidity ratio were interpolated. However, wind speed and direction were not interpolated because of their more stochastic and unpredictable nature.

Some stations in the ISH data set also provide data that were not recorded at the beginning of the hour. When data at the exact hour were missing, they were replaced by data up to 0.5 h before or after, when available.

Finally, psychrometric quantities such as wet bulb temperature or enthalpy are not contained in the weather data sets. They were calculated from dry bulb temperature, dew-point temperature and station pressure.

Further details of the analysis procedures are available from ASHRAE[11].

2.7 Solar and illuminance data

2.7.1 Solar geometry

Two angles are used to define the angular position of the sun as seen from a given point on the surface of the earth, see Figure 2.11. These are:

— Solar altitude, γ_s: the angular elevation of the centre of the solar disk above the horizontal plane.

— Solar azimuth, α_s: the horizontal angle between the vertical plane containing the centre of the solar disk and the vertical plane running in a true N–S direction. Solar azimuth is measured clockwise from due south in the northern hemisphere and anti-clockwise measured from due north in the southern hemisphere. Values are negative before solar noon and positive after solar noon.

Other important angles for solar geometry are:

— Wall azimuth, α: the orientation of the wall, measured clockwise from due south in the northern hemisphere and anti-clockwise measured from due north in the southern hemisphere.

— Wall–solar azimuth angle, sometimes called the horizontal shadow angle, α_f : the angle between the vertical plane containing the normal to the surface and the vertical plane passing through the centre of the solar disk, i.e. the resolved angle on the horizontal plane between the direction of the sun and the direction of the normal to the surface.

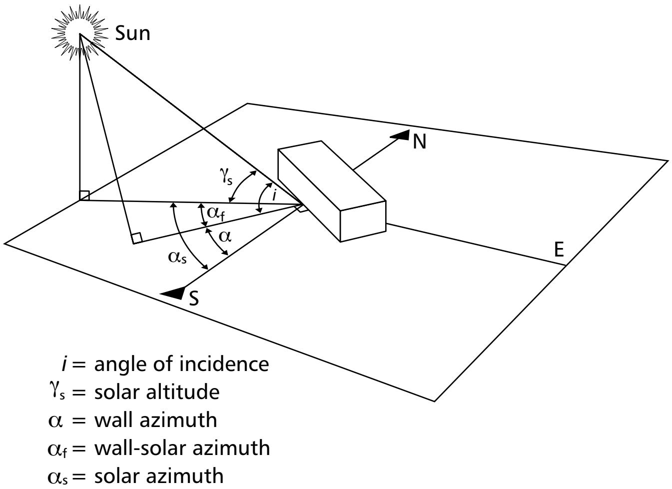


Figure 2.11 Solar geometry

Numerical values of altitude angle and bearing to the nearest degree are given in CIBSE Guide J[1], Appendix A6.

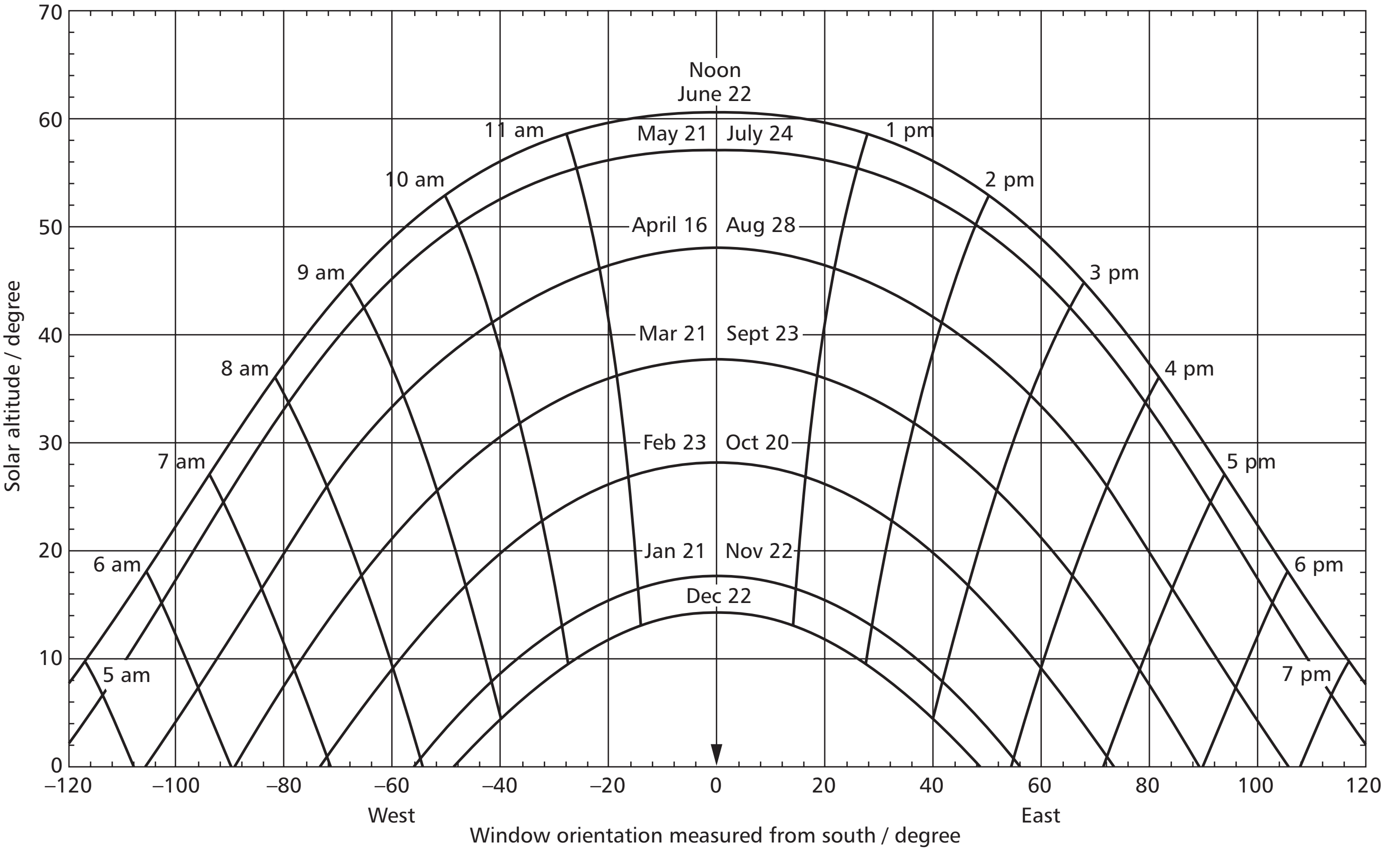


Figure 2.12 Sun-path diagram for latitude 52 °N; the times of day are given in local apparent (i.e. solar) time

2.7.2 Sun position

Data on sun position may be presented in graphical or tabular forms. Sun-path diagrams are useful for visualising the course of the sun across the year and can be used in various graphical methods for analysing the effects of solar shading, whether due to external obstructions, building features or window position. Various projections can be used but the simplest presentation for any given latitude is to plot the hourly solar altitude as a function of the hourly solar bearing for a range of declinations covering the year. Figure 2.12 shows an example sun-path diagram for latitude 52°N.

2.7.3 Solar irradiation

2.7.3.1 Monthly mean daily irradiation on inclined planes (UK)

Tables 2.27 to 2.29 provide values of monthly mean daily irradiation on a range of inclined planes are provided for three sites in the UK: London, Manchester and Edinburgh. These are based on measured data for the period 1983–2002. A ground albedo of 0.2 was assumed.

For Tables 2.28 and 2.29, refer to the CD-ROM that accompanies this Guide. Note that, for Manchester, the data were obtained from Aughton and Hulme due to the relocation of the measuring station. Likewise, for Edinburgh, the data are based on measurements made at Mylnefield and Strathallan.

2.7.3.2 Near-extreme global irradiation with associated diffuse irradiation (UK)

Tables 2.30 to 2.32 provide values of near-extreme hourly irradiation (formerly termed 'design maxima') for three UK sites: London area (Bracknell), Manchester (Aughton) and Edinburgh (Mylnefield), respectively. For Tables 2.31 and 2.32, refer to the CD-ROM that accompanies this Guide.

The tables provide mean hourly irradiation values associated with a known exceedence of daily global irradiation and are intended for use in risk-based design applications. They provide the basis for the tables of solar cooling load provided in chapter 5.

In the cases of Bracknell and Aughton, the input data sets include the observed hourly horizontal diffuse irradiation measured during the same hour as the corresponding hourly global irradiation. The diffuse data for Mylnefield have been estimated using the observed hourly global irradiation. The clear sky model used for this purpose is described in CIBSE Guide J: *Weather, solar and illuminance data*[1]. In all cases, the horizontal beam irradiation is the difference between the horizontal global and horizontal diffuse irradiation.

Table 2.27 Monthly mean daily irradiation on inclined planes: London area (Bracknell) (1983–2002)

Month	Mean irradiation ($W \cdot h \cdot m^{-2}$) for stated inclination from horizontal (°)																		
	Beam					Diffuse					Total					Ground reflected			
	0	30	45	60	90	0	30	45	60	90	0	30	45	60	90	30	45	60	90
(*a*) West																			
Jan	223	224	224	211	169	460	440	409	361	235	683	673	652	606	473	9	20	35	69
Feb	485	453	431	402	304	846	809	749	660	428	1332	1280	1219	1129	866	18	39	67	134
Mar	825	775	730	668	485	1410	1347	1248	1099	709	2235	2152	2044	1879	1417	30	66	112	224
Apr	1550	1503	1412	1283	917	2062	1996	1867	1662	1088	3611	3548	3385	3125	2367	48	106	181	362
May	2081	2025	1889	1701	1189	2565	2494	2337	2086	1370	4647	4581	4362	4019	3025	62	136	233	465
Jun	2190	2034	1851	1628	1083	2874	2744	2557	2260	1451	5064	4847	4557	4142	3042	68	149	254	508
Jul	2226	2080	1909	1690	1140	2702	2590	2412	2137	1378	4928	4737	4465	4073	3011	66	145	247	493
Aug	2013	1919	1784	1608	1133	2297	2224	2079	1856	1219	4310	4200	3990	3680	2783	58	126	216	432
Sep	1187	1193	1148	1067	804	1657	1619	1521	1354	897	2843	2850	2752	2564	1986	38	83	143	285
Oct	744	814	808	771	607	1001	997	944	850	576	1746	1834	1804	1708	1359	24	51	88	176
Nov	330	380	385	378	306	578	573	538	483	324	908	965	950	907	721	12	27	46	92
Dec	173	191	196	191	158	361	350	327	291	192	534	548	539	509	404	7	16	27	54
Mean	1169	1132	1064	966	691	1568	1515	1416	1258	822	2737	2684	2560	2362	1788	37	80	137	274
(*b*) South-west:																			
Jan	223	483	567	613	579	460	495	483	454	346	683	987	1071	1102	994	9	20	35	69
Feb	485	802	884	912	784	846	877	844	778	569	1332	1697	1768	1757	1486	18	39	67	134
Mar	825	1118	1166	1140	886	1410	1417	1342	1216	841	2235	2565	2574	2468	1951	30	66	112	224
Apr	1550	1846	1837	1715	1214	2062	2056	1944	1754	1189	3611	3950	3887	3650	2764	49	106	181	362
May	2081	2253	2151	1931	1238	2565	2523	2366	2115	1402	4647	4838	4652	4279	3106	62	136	233	465
Jun	2190	2217	2054	1788	1056	2874	2775	2578	2283	1480	5064	5059	4781	4325	3043	68	149	254	508
Jul	2226	2306	2166	1913	1171	2702	2624	2448	2173	1420	4928	4997	4758	4332	3084	66	145	247	493
Aug	2013	2277	2221	2037	1384	2297	2281	2153	1936	1303	4310	4616	4501	4189	3119	58	126	216	432
Sep	1187	1566	1620	1568	1204	1657	1687	1611	1471	1031	2843	3291	3315	3181	2520	38	84	143	285
Oct	744	1242	1374	1414	1220	1001	1078	1054	988	740	1746	2343	2480	2489	2136	24	51	88	175
Nov	330	694	811	872	817	578	636	630	597	462	908	1342	1467	1515	1371	12	27	46	92
Dec	173	424	511	562	549	361	401	399	380	298	534	833	925	969	901	7	16	27	54
Mean	1169	1436	1447	1372	1008	1568	1571	1488	1345	923	2737	3043	3015	2855	2206	37	80	137	274

Table continues

Table 2.27 Monthly mean daily irradiation on inclined planes: London area (Bracknell) (1983–2002) — *continued*

Month	Mean irradiation ($W{\cdot}h{\cdot}m^{-2}$) for stated inclination from horizontal (°)																		
	Beam					Diffuse					Total					Ground reflected			
	0	30	45	60	90	0	30	45	60	90	0	30	45	60	90	30	45	60	90
(*c*) South:																			
Jan	223	586	713	792	785	460	515	514	491	389	683	1111	1247	1317	1243	9	20	35	69
Feb	485	969	1119	1193	1097	846	912	893	839	639	1332	1899	2051	2099	1869	18	39	67	134
Mar	825	1266	1363	1367	1102	1410	1443	1380	1265	908	2235	2738	2808	2743	2233	30	66	112	224
Apr	1550	1925	1922	1788	1179	2062	2054	1935	1744	1185	3611	4027	3963	3714	2725	49	106	181	362
May	2081	2229	2097	1824	975	2565	2486	2314	2053	1334	4647	4777	4547	4109	2774	62	136	233	465
Jun	2190	2213	2018	1696	795	2874	2745	2541	2239	1429	5064	5026	4708	4189	2731	68	149	254	508
Jul	2226	2312	2139	1829	916	2702	2602	2417	2137	1374	4928	4980	4700	4213	2784	66	145	247	493
Aug	2013	2350	2291	2077	1265	2297	2274	2136	1917	1278	4310	4682	4554	4209	2975	58	127	216	432
Sep	1187	1660	1733	1688	1264	1657	1688	1609	1469	1039	2843	3386	3425	3300	2588	38	83	143	285
Oct	744	1346	1518	1587	1403	1001	1081	1058	994	752	1746	2450	2628	2668	2331	23	52	88	176
Nov	330	784	937	1027	995	578	647	645	616	487	908	1443	1609	1689	1574	12	27	46	92
Dec	173	508	629	707	716	361	417	421	407	331	534	932	1066	1141	1101	7	16	27	54
Mean	1169	1512	1540	1465	1041	1568	1572	1489	1348	929	2737	3121	3109	2949	2244	37	80	137	274
(*d*) South-east:																			
Jan	223	459	534	572	532	460	486	472	440	326	683	954	1026	1046	927	9	20	35	69
Feb	485	816	905	933	811	846	876	841	775	560	1332	1709	1785	1775	1504	18	39	67	134
Mar	825	1106	1154	1125	875	1410	1403	1326	1198	825	2235	2539	2546	2435	1924	30	66	112	224
Apr	1550	1720	1677	1537	1047	2062	2004	1877	1672	1099	3611	3773	3661	3390	2508	48	106	181	362
May	2081	2064	1920	1682	1021	2565	2453	2277	2008	1290	4647	4580	4333	3923	2775	62	136	233	465
Jun	2190	2114	1936	1668	960	2874	2737	2527	2219	1412	5064	4919	4612	4141	2879	68	149	254	508
Jul	2226	2195	2029	1771	1053	2702	2590	2402	2117	1357	4928	4852	4576	4135	2903	66	145	247	493
Aug	2013	2158	2071	1872	1229	2297	2234	2091	1860	1219	4310	4449	4288	3948	2880	58	126	216	432
Sep	1187	1421	1432	1353	986	1657	1628	1529	1371	916	2843	3088	3045	2866	2187	38	84	143	285
Oct	744	1049	1104	1099	877	1001	1004	951	864	602	1746	2076	2106	2050	1654	24	51	88	176
Nov	330	582	654	682	602	578	595	571	525	381	908	1189	1251	1253	1074	12	27	46	92
Dec	173	382	450	488	464	361	386	376	352	266	534	775	842	867	784	7	16	27	54
Mean	1169	1339	1322	1232	871	1568	1533	1437	1283	854	2737	2908	2839	2652	2000	37	80	137	274
(*e*) East:																			
Jan	223	200	190	179	135	460	433	397	346	216	683	642	608	560	420	9	20	35	69
Feb	485	467	453	425	321	846	805	744	653	415	1332	1289	1236	1144	870	18	39	67	134
Mar	825	761	714	647	467	1410	1336	1231	1075	680	2235	2128	2011	1834	1370	30	66	112	224
Apr	1550	1337	1207	1060	712	2062	1930	1782	1562	980	3611	3316	3095	2804	2053	48	106	181	362
May	2081	1752	1555	1337	848	2565	2396	2206	1929	1203	4647	4211	3898	3499	2517	62	136	233	465
Jun	2190	1886	1673	1438	899	2874	2681	2478	2163	1342	5064	4635	4300	3854	2749	68	149	254	508
Jul	2226	1927	1723	1493	958	2702	2532	2347	2056	1285	4928	4525	4215	3796	2736	66	145	247	493
Aug	2013	1759	1590	1395	928	2297	2164	1998	1759	1112	4310	3980	3715	3370	2471	58	126	216	432
Sep	1187	1019	933	830	577	1657	1554	1433	1251	786	2843	2610	2449	2223	1648	38	83	143	285
Oct	744	595	537	477	325	1001	930	852	741	462	1746	1549	1440	1306	963	24	52	88	176
Nov	330	254	233	205	147	578	533	489	421	262	908	799	749	671	501	12	27	46	92
Dec	173	143	138	126	98	361	337	309	268	168	534	487	463	420	320	7	16	27	54
Mean	1169	1008	912	801	535	1568	1469	1356	1185	743	2737	2514	2348	2123	1551	37	80	137	274

Table 2.28 Monthly mean daily irradiation on inclined planes: Manchester (Aughton) (1983–2002) — refer to CD-ROM

Table 2.29 Monthly mean daily irradiation on inclined planes: Edinburgh (Mylnefield) (1983–1995) (Strathallan) (1996–2002) — refer to CD-ROM

Monthly mean hourly near-extreme values of global and diffuse horizontal irradiation have been established using the hourly data for all days when the daily horizontal global daily irradiation data on that day fell into the category of the highest 5% of the days in each long-term monthly daily series. As the hourly mean horizontal values tabulated are based on taking hourly averages for all the days in a month with a daily global irradiation above 95% level, the tables may be considered to be representative of the 97.5 percentile daily global horizontal irradiation conditions month-by-month. Details of the quality control procedures used in the generation of these data have been published by Page(12).

The corresponding hourly slope irradiation values were calculated, hour-by-hour, using the monthly mean hourly values of the observed hourly global and diffuse horizontal observations for each of the selected monthly extreme-day data series. The declination values needed to generate the underlying solar geometry were set at the standard clear day design values, see CIBSE Guide J: *Weather, solar and illuminance data*. A ground albedo of 0.2 was assumed.

Table 2.30 Design 97.5 percentile of beam and diffuse irradiance on vertical and horizontal surfaces: London area (Bracknell) (1981–1992)

Date and times of sunrise/sunset	Orientation	Type	Daily mean irradiance (/ W·m^{-2}) and mean hourly irradiance (/ W·m^{-2}) for stated solar time*																		
			Mean	0330	0430	0530	0630	0730	0830	0930	1030	1130	1230	1330	1430	1530	1630	1730	1830	1930	2030
Jan 29	Normal to beam		156		—	—	—	*104*	288	466	593	617	612	507	363	224	*81*	—	—	—	—
	N	Beam	0	—	—	—	—	*0*	0	0	0	0	**0**	0	0	0	*0*	—	—	—	—
Sunrise: 07:37		Diffuse	15	—	—	—	—	*9*	23	40	53	61	**62**	57	42	24	*10*	—	—	—	—
Sunset: 16:23	NE	Beam	1	—	—	—	—	*23*	22	0	0	0	**0**	0	0	0	*0*	—	—	—	
		Diffuse	16	—	—	—	—	*25*	29	45	53	61	**62**	57	42	24	*10*	—	—	—	—
	E	Beam	34	—	—	—	—	*88*	217	**269**	215	77	0	0	0	0	*0*	—	—	—	—
		Diffuse	23	—	—	—	—	*34*	80	**99**	94	72	71	57	42	24	*10*	—	—	—	—
	SE	Beam	98	—	—	—	—	*101*	285	449	**521**	460	349	186	53	0	*0*	—	—	—	—
		Diffuse	36	—	—	—	—	*37*	97	135	**149**	146	128	101	57	27	*10*	—	—	—	—
	S	Beam	130	—	—	—	—	*55*	186	365	522	**574**	569	447	285	145	*43*	—	—	—	—
		Diffuse	43	—	—	—	—	*28*	72	118	150	**166**	169	157	114	70	*32*	—	—	—	—
	SW	Beam	89	—	—	—	—	*0*	0	68	217	352	**456**	446	350	222	*79*	—	—	—	—
		Diffuse	35	—	—	—	—	*9*	26	54	95	126	**148**	157	129	92	*42*	—	—	—	—
	W	Beam	28	—	—	—	—	*0*	0	0	0	0	76	184	**210**	169	*69*	—	—	—	—
		Diffuse	23	—	—	—	—	*9*	23	40	53	70	73	101	**97**	77	*39*	—	—	—	—
	NW	Beam	1	—	—	—	—	*0*	0	0	0	0	**0**	0	0	17	*18*	—	—	—	—
		Diffuse	16	—	—	—	—	*9*	23	40	53	61	**62**	57	47	30	*29*	—	—	—	—
	Horiz.	Beam	42	—	—	—	—	*3*	34	106	180	213	211	154	82	26	*2*	—	—	—	—
	Horiz.	Diffuse	24	—	—	—	—	*19*	41	65	77	87	89	91	74	47	*22*	—	—	—	—
	Horiz.	Global	66	—	—	—	—	*22*	75	171	257	300	300	245	156	73	*24*	—	—	—	—
Feb 26	Normal to beam		201	—	—	—	*67*	210	375	503	605	647	636	603	552	426	238	*77*	—	—	—
	N	Beam	0	—	—	—	*0*	0	0	0	0	**0**	0	0	0	0	0	*0*	—	—	—
Sunrise: 06:46		Diffuse	26	—	—	—	*9*	23	44	68	82	**92**	90	82	66	44	22	*9*	—	—	—
Sunset: 17:14	NE	Beam	6	—	—	—	*32*	**77**	57	0	0	0	0	0	0	0	0	*0*	—	—	—
		Diffuse	28	—	—	—	*26*	**49**	59	79	82	92	90	82	66	44	22	*8*	—	—	—
	E	Beam	46	—	—	—	*64*	192	294	**302**	229	83	0	0	0	0	0	*0*	—	—	—
		Diffuse	37	—	—	—	*34*	81	115	**138**	130	103	102	82	66	44	22	*8*	—	—	—
	SE	Beam	106	—	—	—	*59*	194	358	466	**509**	454	330	185	42	0	0	*0*	—	—	—
		Diffuse	49	—	—	—	*32*	81	129	173	**184**	181	155	121	71	49	22	*8*	—	—	—
	S	Beam	145	—	—	—	*18*	83	213	358	491	**558**	549	489	392	242	94	*21*	—	—	—
		Diffuse	57	—	—	—	*24*	50	97	150	181	**201**	196	179	146	99	51	*24*	—	—	—
	SW	Beam	111	—	—	—	*0*	0	0	39	185	336	446	**507**	512	407	220	*67*	—	—	—
		Diffuse	49	—	—	—	*8*	23	49	74	121	158	177	**183**	170	133	85	*32*	—	—	—
	W	Beam	50	—	—	—	*0*	0	0	0	0	0	82	228	**332**	334	217	*74*	—	—	—
		Diffuse	37	—	—	—	*8*	23	44	68	82	104	101	129	**135**	118	84	*34*	—	—	—
	NW	Beam	7	—	—	—	*0*	0	0	0	0	0	0	0	0	65	**87**	*37*	—	—	—
		Diffuse	30	—	—	—	*8*	23	44	68	82	92	90	129	77	59	**49**	*26*	—	—	—
	Horiz.	Beam	75	—	—	—	*1*	24	95	184	270	316	311	269	202	108	27	*1*	—	—	—
	Horiz.	Diffuse	40	—	—	—	*18*	44	77	109	121	131	129	120	100	73	42	*18*	—	—	—
	Horiz.	Global	115	—	—	—	*19*	68	172	293	391	447	440	389	302	181	69	*19*	—	—	—
Mar 29	Normal to beam		263	—	—	*70*	221	385	524	610	649	696	683	663	631	552	415	238	*76*	—	—
	N	Beam	0	—	—	*5*	0	0	0	0	0	0	**0**	0	0	0	0	0	*5*	—	—
Sunrise: 05:43		Diffuse	41	—	—	*27*	30	53	77	96	112	119	**120**	112	96	75	51	28	*24*	—	—
Sunset: 18:17	NE	Beam	20	—	—	*53*	145	**180**	131	12	0	0	0	0	0	0	0	0	*0*	—	—
		Diffuse	45	—	—	*35*	72	**97**	107	90	118	119	120	112	96	75	49	24	*9*	—	—
	E	Beam	72	—	—	*70*	219	355	**415**	371	248	90	0	0	0	0	0	0	*0*	—	—
		Diffuse	57	—	—	*41*	93	136	**167**	169	160	128	135	112	96	75	49	24	*9*	—	—
	SE	Beam	117	—	—	*46*	165	322	455	**513**	489	426	293	141	0	0	0	0	*0*	—	—
		Diffuse	65	—	—	*33*	78	129	176	**196**	205	194	173	136	117	79	49	24	*9*	—	—
	S	Beam	139	—	—	*0*	14	100	229	354	443	**512**	503	453	367	241	108	15	*0*	—	—
		Diffuse	67	—	—	*11*	32	78	129	166	196	**210**	210	196	165	126	75	30	*10*	—	—
	SW	Beam	121	—	—	*0*	0	0	0	0	138	298	418	499	**531**	480	347	177	*50*	—	—
		Diffuse	65	—	—	*9*	26	51	80	118	137	172	195	204	**195**	172	126	75	*30*	—	—
	W	Beam	75	—	—	*0*	0	0	0	0	0	0	89	253	384	**437**	383	236	*76*	—	—
		Diffuse	56	—	—	*9*	26	51	77	96	112	134	129	159	168	**164**	134	91	*37*	—	—
	NW	Beam	22	—	—	*0*	0	0	0	0	0	0	0	0	12	138	**194**	156	*57*	—	—
		Diffuse	45	—	—	*9*	26	51	77	96	112	119	120	118	89	104	**94**	70	*32*	—	—
	Horiz.	Beam	131	—	—	*2*	28	110	223	330	404	462	454	413	341	235	118	30	*2*	—	—
	Horiz.	Diffuse	59	—	—	*21*	50	87	120	139	158	161	163	155	135	113	81	46	*19*	—	—
	Horiz.	Global	190	—	—	*23*	78	197	343	469	562	623	617	568	476	348	199	76	*21*	—	—

* Mean over hour centred at stated solar time

Note: italicised values are calculated for time halfway between sunrise and the end of the sunrise hour or halfway between the beginning of the sunset hour and sunset; the figures shown in bold type, when added together, give the peak total irradiance (i.e. beam plus diffuse) for the stated orientation.

Table 2.30 Design 97.5 percentile of beam and diffuse irradiance on vertical and horizontal surfaces: London area (Bracknell) (1981–1992) — *continued*

Date and times of sunrise/sunset	Orient-ation	Type	Daily mean irradiance (/ W·m^{-2}) and mean hourly irradiance (/ W·m^{-2}) for stated solar time*																		
			Mean	0330	0430	0530	0630	0730	0830	0930	1030	1130	1230	1330	1430	1530	1630	1730	1830	1930	2030
Apr 28	Normal to beam		343	—	*74*	235	440	555	674	707	744	765	761	740	724	655	568	408	218	*69*	—
	N	Beam	7	—	*27*	59	24	0	0	0	0	0	**0**	0	0	0	0	22	55	*26*	—
Sunrise: 04:46		Diffuse	56	—	*25*	43	53	87	101	117	130	139	**141**	131	120	100	88	52	41	*25*	—
Sunset: 19:14	NE	Beam	47	—	*68*	202	316	**298**	220	71	0	0	0	0	0	0	0	0	0	*0*	—
		Diffuse	64	—	*33*	86	123	**142**	142	119	145	139	141	131	120	100	77	47	22	*8*	—
	E	Beam	103	—	*69*	226	423	497	**519**	417	276	96	0	0	0	0	0	0	0	*0*	—
		Diffuse	74	—	*34*	93	146	183	**196**	185	171	143	155	131	120	100	77	47	22	*8*	—
	SE	Beam	128	—	*29*	118	282	405	513	**519**	483	392	253	92	0	0	0	0	0	*0*	—
		Diffuse	77	—	*25*	61	115	164	195	**202**	205	196	179	136	138	100	77	47	22	*8*	—
	S	Beam	122	—	*0*	0	0	76	207	317	407	458	**455**	404	324	202	78	0	0	*0*	—
		Diffuse	74	—	*8*	24	57	90	139	168	192	207	**212**	194	174	138	91	56	23	*8*	—
	SW	Beam	127	—	*0*	0	0	0	0	0	92	255	390	480	**531**	499	415	262	109	*27*	—
		Diffuse	77	—	*8*	23	48	77	101	134	135	175	201	207	**210**	192	165	110	58	*25*	—
	W	Beam	101	—	*0*	0	0	0	0	0	0	0	97	275	427	**504**	509	392	210	*64*	—
		Diffuse	74	—	*8*	23	48	77	101	117	130	152	146	173	192	**193**	184	139	86	*34*	—
	NW	Beam	45	—	*0*	0	0	0	0	0	0	0	0	0	73	214	**305**	293	187	*63*	—
		Diffuse	64	—	*8*	23	48	77	101	117	130	139	141	146	122	140	**143**	117	80	*33*	—
	Horiz.	Beam	199	—	*1*	26	119	235	377	475	558	605	602	555	486	367	240	110	25	*1*	—
	Horiz.	Diffuse	73	—	*18*	44	80	117	138	151	163	170	177	166	157	138	116	80	43	*18*	—
	Horiz.	Global	272	—	*19*	70	199	352	515	626	721	775	779	721	643	505	356	190	68	*19*	—
May 29	Normal to beam		386	—	*145*	350	482	580	685	753	787	794	793	787	758	693	623	511	371	*154*	—
	N	Beam	22	—	*73*	**114**	66	0	0	0	0	0	0	0	0	0	0	70	120	*78*	—
Sunrise: 04:01		Diffuse	68	—	*49*	**72**	77	108	116	126	137	143	142	134	125	113	104	73	68	*43*	—
Sunset: 19:59	NE	Beam	63	—	*140*	308	**360**	333	255	117	0	0	0	0	0	0	0	0	0	*0*	—
		Diffuse	74	—	*64*	121	**144**	159	154	136	159	143	142	134	125	108	87	60	36	*15*	—
	E	Beam	112	—	*125*	323	444	498	**505**	426	280	96	0	0	0	0	0	0	0	*0*	—
		Diffuse	81	—	*59*	124	162	191	**197**	188	170	145	153	134	125	108	87	60	36	*14*	—
	SE	Beam	117	—	*36*	148	268	372	460	**486**	443	343	207	47	0	0	0	0	0	*0*	—
		Diffuse	80	—	*44*	81	125	167	189	**197**	194	184	165	126	134	108	87	60	36	*14*	—
	S	Beam	98	—	*0*	0	0	28	145	261	347	**389**	388	347	262	147	30	0	0	*0*	—
		Diffuse	72	—	*16*	39	72	89	133	163	180	**191**	190	177	159	128	85	68	37	*14*	—
	SW	Beam	120	—	*0*	0	0	0	0	0	47	207	343	443	**489**	465	399	284	157	*38*	—
		Diffuse	79	—	*16*	39	64	90	111	137	128	165	184	190	**193**	183	160	118	76	*39*	—
	W	Beam	116	—	*0*	0	0	0	0	0	0	0	96	280	429	511	**535**	471	342	*132*	—
		Diffuse	80	—	*16*	39	64	90	111	126	137	153	145	168	184	191	**184**	153	117	*53*	—
	NW	Beam	66	—	*0*	0	0	0	0	0	0	0	0	0	118	258	357	**382**	327	*149*	—
		Diffuse	74	—	*17*	39	64	90	111	126	137	143	142	157	134	149	153	**136**	114	*56*	—
	Horiz.	Beam	243	—	*10*	74	175	296	439	564	649	685	685	649	567	444	318	186	79	*10*	—
	Horiz.	Diffuse	77	—	*33*	70	101	133	146	153	156	162	161	153	148	140	121	91	63	*29*	—
	Horiz.	Global	320	—	*43*	144	276	429	585	717	805	847	846	802	715	584	439	277	142	*39*	—
Jun 21	Normal to beam		414	*64*	191	387	554	683	747	797	826	837	840	809	771	705	639	531	390	192	*65*
	N	Beam	27	*40*	100	**132**	85	0	0	0	0	0	0	0	0	0	0	82	133	100	*40*
Sunrise: 03:49		Diffuse	71	*25*	55	**76**	85	111	117	125	134	139	140	135	127	121	114	85	77	58	*25*
Sunset: 20:11	NE	Beam	74	*64*	185	342	**417**	397	285	134	0	0	0	0	0	0	0	0	0	0	*0*
		Diffuse	77	*32*	72	124	**157**	158	150	134	158	139	140	135	127	113	94	69	42	21	*9*
	E	Beam	124	*50*	162	352	504	**579**	544	445	290	100	0	0	0	0	0	0	0	0	*0*
		Diffuse	83	*27*	66	126	175	**189**	189	180	163	140	149	135	127	113	94	69	42	20	*8*
	SE	Beam	122	*7*	44	155	296	422	484	**496**	447	345	204	36	0	0	0	0	0	0	*0*
		Diffuse	79	*21*	50	82	133	163	180	**187**	183	173	157	123	134	113	94	69	42	20	*8*
	S	Beam	94	*0*	0	0	0	18	141	256	342	387	**389**	335	247	133	17	0	0	0	*0*
		Diffuse	72	*8*	18	41	77	83	125	154	170	179	**180**	172	158	130	87	77	42	20	*8*
	SW	Beam	118	*0*	0	0	0	0	0	0	37	203	346	438	**479**	457	395	284	157	44	*7*
		Diffuse	80	*8*	18	41	69	91	110	132	122	156	175	185	**193**	188	169	133	83	52	*21*
	W	Beam	120	*0*	0	0	0	0	0	0	0	0	101	284	431	513	**542**	483	355	163	*51*
		Diffuse	83	*8*	18	41	69	91	110	125	134	148	140	165	185	197	**196**	175	128	69	*27*
	NW	Beam	71	*0*	0	0	0	0	0	0	0	0	0	0	130	269	371	**399**	345	186	*65*
		Diffuse	77	*9*	19	41	69	91	110	125	134	139	140	159	138	156	164	**157**	126	75	*32*
	Horiz.	Beam	264	*1*	17	91	214	362	493	610	694	735	738	679	590	465	339	205	92	18	*1*
	Horiz.	Diffuse	76	*16*	37	71	104	120	132	140	141	144	146	146	149	146	131	107	72	38	*16*
	Horiz.	Global	341	*17*	54	162	318	482	625	750	835	879	884	825	739	611	470	312	164	56	*17*

* Mean over hour centred at stated solar time

Table continues

Note: italicised values are calculated for time halfway between sunrise and the end of the sunrise hour or halfway between the beginning of the sunset hour and sunset; the figures shown in bold type, when added together, give the peak total irradiance (i.e. beam plus diffuse) for the stated orientation.

Table 2.30 Design 97.5 percentile of beam and diffuse irradiance on vertical and horizontal surfaces: London area (Bracknell) (1981–1992) — *continued*

Date and times of sunrise/sunset	Orient- ation	Type	Daily mean irradiance (/ W·m^{-2}) and mean hourly irradiance (/ W·m^{-2}) for stated solar time*																		
			Mean	0330	0430	0530	0630	0730	0830	0930	1030	1130	1230	1330	1430	1530	1630	1730	1830	1930	2030
Jul 4	Normal to beam		381	*57*	170	362	505	610	687	729	751	753	765	750	717	672	616	502	360	169	*57*
	N	Beam	24	*35*	88	**122**	75	0	0	0	0	0	0	0	0	0	0	74	121	87	*35*
Sunrise: 03:53		Diffuse	66	*20*	47	**64**	73	103	110	121	133	140	141	132	123	113	102	78	62	43	*18*
Sunset: 20:08	NE	Beam	67	*57*	165	320	**379**	353	260	119	0	0	0	0	0	0	0	0	0	0	*0*
		Diffuse	71	*25*	61	103	**132**	146	143	132	156	140	141	132	123	107	85	64	35	15	*6*
	E	Beam	114	*45*	145	331	461	**519**	502	409	265	90	0	0	0	0	0	0	0	0	*0*
		Diffuse	77	*22*	57	106	147	**174**	179	176	166	144	150	132	123	107	85	64	35	14	*5*
	SE	Beam	113	*7*	40	148	273	381	450	**459**	412	315	190	37	0	0	0	0	0	0	*0*
		Diffuse	76	*16*	43	69	113	151	171	**183**	186	179	162	124	131	107	85	64	35	14	*5*
	S	Beam	89	*0*	0	0	0	20	135	240	318	355	**361**	317	236	132	20	0	0	0	*0*
		Diffuse	70	*6*	16	35	67	81	121	152	173	185	**186**	172	156	124	80	71	35	14	*5*
	SW	Beam	112	*0*	0	0	0	0	0	0	37	187	320	411	**451**	440	385	272	147	40	*7*
		Diffuse	76	*6*	16	35	60	85	104	129	124	161	180	185	**188**	176	150	121	67	39	*15*
	W	Beam	113	*0*	0	0	0	0	0	0	0	0	92	264	402	491	**524**	459	329	144	*45*
		Diffuse	78	*6*	16	35	60	85	104	121	133	149	144	165	181	184	**173**	157	102	52	*20*
	NW	Beam	66	*0*	0	0	0	0	0	0	0	0	0	0	117	254	357	**377**	318	164	*57*
		Diffuse	72	*6*	17	35	60	85	104	121	133	140	141	155	135	147	145	**141**	100	56	*23*
	Horiz.	Beam	241	*0*	14	83	191	319	449	554	627	658	668	626	545	439	323	190	82	14	*0*
	Horiz.	Diffuse	75	*13*	31	60	90	116	130	143	154	162	162	153	150	137	114	97	58	29	*12*
	Horiz.	Global	315	*13*	45	143	281	435	579	697	781	820	830	779	695	576	437	287	140	43	*12*
Aug 4	Normal to beam		333	—	*93*	259	411	551	644	698	755	744	727	706	705	609	496	361	227	*82*	—
	N	Beam	10	—	*40*	73	36	0	0	0	0	0	**0**	0	0	0	0	32	64	*35*	—
Sunrise: 04:29		Diffuse	57	—	*33*	50	58	89	101	116	127	134	**135**	131	116	100	87	56	47	*32*	—
Sunset: 19:31	NE	Beam	49	—	*87*	225	301	**305**	222	86	0	0	0	0	0	0	0	0	0	*0*	—
		Diffuse	64	—	*44*	88	118	**137**	140	122	143	134	135	131	116	100	75	48	27	*11*	—
	E	Beam	101	—	*84*	245	389	486	**488**	406	276	93	0	0	0	0	0	0	0	*0*	—
		Diffuse	72	—	*43*	93	136	171	**186**	179	162	138	147	131	116	100	75	48	27	*11*	—
	SE	Beam	118	—	*31*	122	250	383	468	**488**	465	358	221	70	0	0	0	0	0	*0*	—
		Diffuse	74	—	*32*	62	107	152	182	**192**	189	182	166	133	130	100	75	48	27	*11*	—
	S	Beam	107	—	*0*	0	0	55	174	284	381	**413**	404	356	287	164	50	0	0	*0*	—
		Diffuse	70	—	*11*	28	59	84	131	160	177	**190**	193	187	159	131	83	55	28	*11*	—
	SW	Beam	113	—	*0*	0	0	0	0	0	74	226	350	434	**493**	442	345	219	107	*28*	—
		Diffuse	73	—	*11*	27	51	77	100	130	126	163	185	200	**191**	181	146	98	58	*31*	—
	W	Beam	94	—	*0*	0	0	0	0	0	0	0	91	258	410	**461**	438	342	215	*74*	—
		Diffuse	72	—	*11*	27	51	77	100	116	127	145	141	171	178	**184**	164	123	85	*41*	—
	NW	Beam	45	—	*0*	0	0	0	0	0	0	0	0	0	87	210	**274**	264	197	*77*	—
		Diffuse	64	—	*11*	27	51	77	100	116	127	134	135	149	121	140	**133**	107	80	*42*	—
	Horiz.	Beam	204	—	*4*	40	127	253	383	492	590	612	598	552	497	362	228	112	35	*3*	—
	Horiz.	Diffuse	72	—	*23*	51	84	113	136	147	148	159	166	167	146	140	116	82	50	*22*	—
	Horiz.	Global	276	—	*27*	91	211	366	519	639	738	771	764	719	643	502	344	194	85	*25*	—
Sep 4	Normal to beam		302	—	—	*139*	371	551	654	683	678	688	720	716	683	604	452	304	*114*	—	—
	N	Beam	1	—	—	*19*	0	0	0	0	0	**0**	**0**	0	0	0	0	0	*16*	—	—
Sunrise: 05:24		Diffuse	43	—	—	*30*	36	58	76	99	115	**118**	**118**	109	95	77	59	37	*33*	—	—
Sunset: 18:36	NE	Beam	34	—	—	*111*	252	**272**	182	32	0	0	0	0	0	0	0	0	*0*	—	—
		Diffuse	48	—	—	*40*	88	**105**	104	94	123	118	118	109	95	77	55	31	*12*	—	—
	E	Beam	93	—	—	*137*	365	505	**515**	412	257	89	0	0	0	0	0	0	*0*	—	—
		Diffuse	58	—	—	*45*	112	146	**157**	165	158	126	131	109	95	77	55	31	*12*	—	—
	SE	Beam	132	—	—	*84*	264	442	546	**551**	488	399	286	131	0	0	0	0	*0*	—	—
		Diffuse	64	—	—	*36*	90	135	162	**188**	197	182	159	123	113	79	55	31	*12*	—	—
	S	Beam	139	—	—	*0*	9	120	258	367	433	475	**498**	457	367	238	99	7	*0*	—	—
		Diffuse	64	—	—	*12*	32	76	117	158	187	194	**191**	175	150	119	77	34	*13*	—	—
	SW	Beam	125	—	—	*0*	0	0	0	0	124	273	418	515	**551**	505	363	217	*69*	—	—
		Diffuse	64	—	—	*11*	30	54	78	118	132	162	179	184	**178**	164	131	88	*39*	—	—
	W	Beam	85	—	—	*0*	0	0	0	0	0	0	93	272	412	**475**	414	299	*113*	—	—
		Diffuse	57	—	—	*11*	30	54	76	99	115	132	124	148	157	**159**	141	107	*49*	—	—
	NW	Beam	28	—	—	*0*	0	0	0	0	0	0	0	0	32	168	**223**	206	*91*	—	—
		Diffuse	48	—	—	*11*	30	54	76	99	115	118	118	116	90	107	**104**	85	*43*	—	—
	Horiz.	Beam	158	—	—	*7*	66	184	310	402	454	489	512	479	402	286	151	54	*6*	—	—
	Horiz.	Diffuse	56	—	—	*23*	51	79	99	128	151	153	145	134	120	107	88	57	*25*	—	—
	Horiz.	Global	214	—	—	*30*	117	263	409	530	605	642	657	613	522	393	239	111	*31*	—	—

* Mean over hour centred at stated solar time

Table continues

Note: italicised values are calculated for time halfway between sunrise and the end of the sunrise hour or halfway between the beginning of the sunset hour and sunset; the figures shown in bold type, when added together, give the peak total irradiance (i.e. beam plus diffuse) for the stated orientation.

Table 2.30 Design 97.5 percentile of beam and diffuse irradiance on vertical and horizontal surfaces: London area (Bracknell) (1981–1992) — *continued*

Date and times of sunrise/sunset	Orient-ation	Type	Daily mean irradiance (/ W·m^{-2}) and mean hourly irradiance (/ W·m^{-2}) for stated solar time*																		
			Mean	0330	0430	0530	0630	0730	0830	0930	1030	1130	1230	1330	1430	1530	1630	1730	1830	1930	2030
Oct 4	Normal to beam		237	—	—	—	*148*	381	551	679	700	644	632	605	568	451	312	*121*	—	—	—
	N	Beam	0	—	—	—	*0*	0	0	0	0	**0**	**0**	0	0	0	0	*0*	—	—	—
Sunrise: 06:22		Diffuse	29	—	—	—	*12*	30	52	71	84	**94**	**94**	90	76	53	31	*13*	—	—	—
Sunset: 17:38	NE	Beam	13	—	—	—	*83*	**155**	105	0	0	0	0	0	0	0	0	*0*	—	—	—
		Diffuse	32	—	—	—	*36*	**66**	71	85	86	94	94	90	76	53	31	*12*	—	—	—
	E	Beam	68	—	—	—	*145*	351	**436**	412	267	84	0	0	0	0	0	*0*	—	—	—
		Diffuse	41	—	—	—	*47*	107	**132**	135	124	104	106	90	76	53	31	*12*	—	—	—
	SE	Beam	127	—	—	—	*122*	341	512	**611**	569	433	308	165	23	0	0	*0*	—	—	—
		Diffuse	51	—	—	—	*42*	105	146	**165**	169	168	148	123	75	57	31	*12*	—	—	—
	S	Beam	153	—	—	—	*28*	132	287	452	537	**528**	518	465	378	235	108	*23*	—	—	—
		Diffuse	57	—	—	—	*32*	61	106	141	164	**184**	184	179	150	102	60	*33*	—	—	—
	SW	Beam	113	—	—	—	*0*	0	0	29	191	314	425	491	**511**	419	280	*100*	—	—	—
		Diffuse	51	—	—	—	*12*	30	57	69	112	148	168	184	**175**	137	98	*44*	—	—	—
	W	Beam	57	—	—	—	*0*	0	0	0	0	0	82	231	**345**	357	287	*119*	—	—	—
		Diffuse	41	—	—	—	*12*	30	52	71	84	106	104	136	**144**	125	100	*49*	—	—	—
	NW	Beam	11	—	—	—	*0*	0	0	0	0	0	0	0	0	86	**127**	*68*	—	—	—
		Diffuse	32	—	—	—	*12*	30	52	71	84	94	94	92	90	71	**64**	*38*	—	—	—
	Horiz.	Beam	99	—	—	—	*8*	68	176	295	361	359	352	312	246	144	56	*6*	—	—	—
	Horiz.	Diffuse	41	—	—	—	*24*	51	77	91	105	127	129	130	111	83	55	*25*	—	—	—
	Horiz.	Global	140	—	—	—	*32*	119	253	386	466	486	481	442	357	227	111	*31*	—	—	—
Nov 4	Normal to beam		194	—	—	—	—	*173*	428	612	693	718	642	569	463	324	*130*	—	—	—	—
	N	Beam	0	—	—	—	—	*0*	0	0	0	0	**0**	0	0	0	*0*	—	—	—	—
Sunrise: 07:21		Diffuse	17	—	—	—	—	*10*	26	45	56	64	**66**	60	46	28	*12*	—	—	—	—
Sunset: 16:39	NE	Beam	3	—	—	—	—	*47*	**43**	0	0	0	0	0	0	0	*0*	—	—	—	—
		Diffuse	18	—	—	—	—	*29*	**35**	51	56	64	66	60	46	28	*12*	—	—	—	—
	E	Beam	47	—	—	—	—	*151*	327	**359**	256	90	0	0	0	0	*0*	—	—	—	—
		Diffuse	25	—	—	—	—	*39*	93	**106**	94	72	76	60	46	28	*12*	—	—	—	—
	SE	Beam	119	—	—	—	—	*166*	420	584	**603**	527	357	198	57	0	*0*	—	—	—	—
		Diffuse	37	—	—	—	—	*41*	111	143	**145**	139	128	99	57	32	*12*	—	—	—	—
	S	Beam	155	—	—	—	—	*84*	267	468	597	**655**	586	490	353	202	*63*	—	—	—	—
		Diffuse	44	—	—	—	—	*31*	81	124	144	**158**	166	152	119	79	*35*	—	—	—	—
	SW	Beam	105	—	—	—	—	*0*	0	77	241	400	472	**495**	442	318	*125*	—	—	—	—
		Diffuse	37	—	—	—	—	*11*	30	56	92	121	147	**153**	136	106	*46*	—	—	—	—
	W	Beam	37	—	—	—	—	*0*	0	0	0	0	81	210	**272**	248	*113*	—	—	—	—
		Diffuse	25	—	—	—	—	*10*	26	45	56	73	77	102	**103**	90	*44*	—	—	—	—
	NW	Beam	2	—	—	—	—	*0*	0	0	0	0	0	0	0	**32**	*36*	—	—	—	—
		Diffuse	18	—	—	—	—	*10*	26	45	56	64	66	60	52	**37**	*32*	—	—	—	—
	Horiz.	Beam	58	—	—	—	—	*8*	68	165	241	279	250	198	125	51	*6*	—	—	—	—
	Horiz.	Diffuse	24	—	—	—	—	*21*	43	62	71	78	91	88	73	51	*24*	—	—	—	—
	Horiz.	Global	82	—	—	—	—	*29*	111	227	312	357	341	286	198	102	*30*	—	—	—	—
Dec 4	Normal to beam		149	—	—	—	—	—	*208*	456	592	624	591	533	410	*187*	—	—	—	—	—
	N	Beam	0	—	—	—	—	—	*0*	0	0	0	**0**	0	0	*0*	—	—	—	—	—
Sunrise: 08:03		Diffuse	11	—	—	—	—	—	*13*	29	42	49	**50**	41	29	*13*	—	—	—	—	—
Sunset: 15:57	NE	Beam	0	—	—	—	—	—	*7*	0	0	0	**0**	0	0	*0*	—	—	—	—	—
		Diffuse	12	—	—	—	—	—	*33*	32	42	49	**50**	41	29	*13*	—	—	—	—	—
	E	Beam	29	—	—	—	—	—	*152*	**257**	210	75	0	0	0	*0*	—	—	—	—	—
		Diffuse	17	—	—	—	—	—	*42*	**86**	84	60	57	41	29	*13*	—	—	—	—	—
	SE	Beam	94	—	—	—	—	—	*208*	443	**527**	474	348	207	72	*0*	—	—	—	—	—
		Diffuse	30	—	—	—	—	—	*50*	125	**144**	136	116	83	45	*15*	—	—	—	—	—
	S	Beam	130	—	—	—	—	—	*142*	369	535	**595**	563	482	332	*127*	—	—	—	—	—
		Diffuse	37	—	—	—	—	—	*41*	110	145	**157**	156	135	102	*41*	—	—	—	—	—
	SW	Beam	91	—	—	—	—	—	*0*	80	230	368	**449**	474	398	*187*	—	—	—	—	—
		Diffuse	29	—	—	—	—	—	*15*	47	88	117	**135**	134	116	*50*	—	—	—	—	—
	W	Beam	26	—	—	—	—	—	*0*	0	0	0	71	189	**231**	*137*	—	—	—	—	—
		Diffuse	17	—	—	—	—	—	*13*	29	42	56	61	80	**81**	*42*	—	—	—	—	—
	NW	Beam	0	—	—	—	—	—	0	0	0	0	**0**	0	0	*7*	—	—	—	—	—
		Diffuse	12	—	—	—	—	—	*13*	29	42	49	**50**	41	32	*33*	—	—	—	—	—
	Horiz.	Beam	32	—	—	—	—	—	*12*	74	141	173	164	127	67	*11*	—	—	—	—	—
	Horiz.	Diffuse	17	—	—	—	—	—	*25*	48	63	70	72	63	48	*25*	—	—	—	—	—
	Horiz.	Global	49	—	—	—	—	—	*37*	122	204	243	236	190	115	*36*	—	—	—	—	—

* Mean over hour centred at stated solar time

Note: italicised values are calculated for time halfway between sunrise and the end of the sunrise hour or halfway between the beginning of the sunset hour and sunset; the figures shown in bold type, when added together, give the peak total irradiance (i.e. beam plus diffuse) for the stated orientation.

Table 2.31 Design 97.5 percentile of beam and diffuse irradiance on vertical and horizontal surfaces: Manchester (Aughton) (1981–1992) — refer to CD-ROM

Table 2.32 Design 97.5 percentile of beam and diffuse irradiance on vertical and horizontal surfaces: Edinburgh (Mylnefield) (1981–1992) — refer to CD-ROM

2.7.3.3 Daily mean and hourly solar irradiance (latitudes 0–60°)

In many design situations, observed hourly global and diffuse irradiation data for horizontal surfaces are not available and a theoretical approach has to be adopted. For building services engineering design, maximum or near maximum cooling loads are associated with cloudless or near cloudless days, particularly in the case of heavily glazed buildings. Table 2.33 provides daily mean and hourly values of clear sky short-wave irradiance for latitudes from 60°N to 60°S. This table is included on the CD-ROM that accompanies this Guide.

Details of the computer algorithms used to generate Table 2.33 and notes on how the values may be adapted to local conditions of climate and geography are given in CIBSE Guide J: *Weather, solar and illuminance data*(1).

Table 2.33 Predicted clear day beam and diffuse irradiances on vertical and horizontal surfaces on specified days for latitudes 0–60°N/S (©J K Page; used with permission) — refer to CD-ROM

2.7.4 Sol-air temperatures for the UK

The procedure for estimating long-wave radiation loss and the calculation of sol-air temperatures are given in CIBSE Guide J: *Weather, solar and illuminance data*(1). Tables 2.34 to 2.36 provide and hourly air and sol-air temperatures for three UK locations: London area (Bracknell), Manchester (Aughton) and Edinburgh (Mylnefield)†. For Tables 2.35 and 2.36, refer to the CD-ROM that accompanies this Guide.

These tables were derived from the 97.5 percentile daily global irradiation exceedence data, see Tables 2.30 to 2.32, in combination with the associated month by month mean hour ending synoptic data. The associated hourly means of the synoptic data were extracted for the same days used to determine the 97.5 percentile irradiation data sets. It was assumed that synoptic time and LAT were identical.

Standardised values for the various parameters were assumed, as follows:

— short-wave surface absorptance: α_{rad} = 0.9 (dark coloured surface), 0.5 (light coloured surface)

— long-wave surface absorptance/emittance: $\alpha_1 = \varepsilon_1$ = 0.9

The long-wave radiation was calculated using sunshine data when the solar altitude was above 6° and cloud data when the solar altitude was 6° or less. The solar radiation was set to zero when the solar altitude was below 1°. For further details see CIBSE Guide J: *Weather, solar and illuminance data*(1).

† Sol-air temperatures are based on solar radiation data and air temperature data for the following combinations of sites: London area: Bracknell (radiation) and Heathrow (air temperature); Manchester: Aughton (radiation) and Ringway (air temperature); Edinburgh: Mylnefield/Strathallan (radiation) and Turnhouse (air temperature)

Table 2.34 Air and sol-air temperatures: London area (Bracknell) (1981–1992)

(*a*) January 29

Hour*	Air temp.	Sol-air temperature (/°C) for stated orientation and surface colour																	
		Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	2.5	0.2	0.2	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
02	2.0	–0.2	–0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2
03	1.9	–0.4	–0.4	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
04	1.4	–1.1	–1.1	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6
05	1.2	–1.4	–1.4	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8
06	0.9	–1.6	–1.6	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1
07	0.7	–1.9	–1.9	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4
08	0.7	–2.0	–2.0	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4	–1.4
09	1.2	1.3	0.1	0.2	–0.3	1.8	0.6	13.9	7.4	17.9	9.7	11.9	6.3	0.2	–0.3	0.2	–0.3	0.2	–0.3
10	2.3	5.6	2.9	2.7	1.8	2.7	1.8	18.7	10.8	28.7	16.5	24.0	13.8	6.9	4.1	2.7	1.8	2.7	1.8
11	3.7	9.6	5.7	4.9	3.8	4.9	3.8	16.7	10.4	32.7	19.5	32.7	19.5	16.7	10.4	4.9	3.8	4.9	3.8
12	4.7	11.5	7.3	6.2	5.0	6.2	5.0	10.3	7.3	29.6	18.2	35.1	21.4	24.2	15.1	6.2	5.0	6.2	5.0
13	5.5	12.2	8.0	7.0	5.8	7.0	5.8	7.0	5.8	24.5	15.6	35.0	21.7	29.7	18.6	11.0	8.0	7.0	5.8
14	5.9	10.9	7.5	7.1	6.0	7.1	6.0	7.1	6.0	16.7	11.4	29.6	18.7	29.6	18.7	16.7	11.4	7.1	6.0
15	5.8	8.5	6.2	6.3	5.4	6.3	5.4	6.3	5.4	9.3	7.1	21.5	14.0	24.8	15.9	17.6	11.8	6.3	5.4
16	5.3	5.0	4.2	4.3	3.9	4.3	3.9	4.3	3.9	4.3	3.9	11.2	7.8	14.7	9.8	12.3	8.4	5.3	4.5
17	4.3	1.9	1.9	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3
18	3.7	1.2	1.2	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7
19	3.3	1.0	1.0	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4
20	3.0	10.7	0.7	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1	1.1
21	2.7	0.4	0.4	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
22	2.2	–0.2	–0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2
23	1.8	–0.6	–0.6	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1
24	1.8	–0.6	–0.6	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2
Mean:	2.9	2.5	1.6	1.7	1.4	1.8	1.5	3.6	2.5	6.9	4.4	8.5	5.2	6.2	4.0	3.1	2.2	1.8	1.4

* Hour ending

Table 2.34 Air and sol-air temperatures: London area (Bracknell) (1981–1992) — *continued*

(*b*) February 26

Hour*	Air temp.	Sol-air temperature (/ °C) for stated orientation and surface colour																	
		Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	2.3	–0.2	–0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2
02	2.0	–0.6	–0.6	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1	–0.1
03	1.9	–0.8	–0.8	–0.3	–0.3	–0.3	–0.3	–0.3	–0.3	–0.3	–0.3	–0.3	–0.3	–0.3	–0.3	–0.3	–0.3	–0.3	–0.3
04	1.5	–1.2	–1.2	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6
05	1.4	–1.2	–1.2	–0.7	–0.7	–0.7	–0.7	–0.7	–0.7	–0.7	–0.7	–0.7	–0.7	–0.7	–0.7	–0.7	–0.7	–0.7	–0.7
06	1.2	–1.4	–1.4	–0.9	–0.9	–0.9	–0.9	–0.9	–0.9	–0.9	–0.9	–0.9	–0.9	–0.9	–0.9	–0.9	–0.9	–0.9	–0.9
07	1.1	–1.5	–1.5	–1.0	–1.0	–1.0	–1.0	–1.0	–1.0	–1.0	–1.0	–1.0	–1.0	–1.0	–1.0	–1.0	–1.0	–1.0	–1.0
08	1.4	0.8	–0.2	0.2	–0.2	4.5	2.1	10.4	5.5	10.5	5.5	4.7	2.3	0.2	–0.2	0.2	–0.2	0.2	–0.2
09	2.5	6.3	3.4	3.3	2.3	7.2	4.4	21.2	12.3	24.9	14.5	16.4	9.6	3.3	2.3	3.3	2.3	3.3	2.3
10	3.8	10.9	6.6	5.9	4.5	5.9	4.5	22.6	13.9	31.1	18.7	25.4	15.5	8.2	5.8	5.9	4.5	5.9	4.5
11	5.2	14.7	9.3	7.9	6.3	7.9	6.3	19.7	12.9	33.3	20.7	32.4	20.2	17.4	11.7	7.9	6.3	7.9	6.3
12	6.3	17.3	11.2	9.5	7.8	9.5	7.8	13.8	10.2	32.0	20.4	36.9	23.3	26.3	17.2	9.5	7.8	9.5	7.8
13	7.1	17.5	11.7	10.1	8.5	10.1	8.5	10.1	8.5	26.0	17.4	36.1	23.1	31.4	20.5	14.2	10.8	10.1	8.5
14	7.7	16.5	11.4	10.1	8.6	10.1	8.6	10.1	8.6	19.1	13.7	33.4	21.8	34.3	22.3	21.3	14.9	10.1	8.6
15	7.7	14.0	10.0	9.3	8.1	9.3	8.1	9.3	8.1	11.4	9.3	27.9	18.5	33.4	21.7	25.2	17.0	9.3	8.1
16	7.2	10.0	7.6	7.5	6.7	7.5	6.7	7.5	6.7	7.5	6.7	19.2	13.3	27.0	17.7	23.6	15.8	11.0	8.6
17	6.1	5.6	4.8	5.1	4.7	5.1	4.7	5.1	4.7	5.1	4.7	8.4	6.6	12.7	9.0	12.7	9.0	8.3	6.5
18	5.0	2.6	2.6	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1
19	4.4	1.8	1.8	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3	2.3
20	3.8	1.2	1.2	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7
21	3.3	0.6	0.6	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2	1.2
22	3.0	0.2	0.2	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
23	2.7	–0.1	–0.1	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4
24	2.5	–0.3	–0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
Mean:	3.8	4.7	3.1	3.1	2.7	3.5	2.8	5.7	4.1	8.6	5.7	10.3	6.7	8.4	5.6	5.4	3.9	3.4	2.8

* Hour ending

(*c*) March 29

Hour*	Air temp.	Sol-air temperature (/ °C) for stated orientation and surface colour																	
		Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	4.8	1.7	1.7	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4	2.4
02	4.3	1.4	1.4	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0
03	4.1	1.1	1.1	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7
04	3.5	0.2	0.2	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9
05	3.2	–0.2	–0.2	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6	0.6
06	3.0	–0.3	–0.3	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5
07	3.2	3.1	1.7	2.0	1.4	12.4	7.3	17.5	10.2	13.9	8.1	3.3	2.1	2.0	1.4	2.0	1.4	2.0	1.4
08	4.6	10.6	6.6	6.3	4.9	19.8	12.5	32.1	19.6	29.9	18.3	14.1	9.3	6.3	4.9	6.3	4.9	6.3	4.9
09	6.6	17.0	11.0	9.7	7.8	18.4	12.7	36.0	22.8	38.5	24.2	24.7	16.3	9.7	7.8	9.7	7.8	9.7	7.8
10	8.2	22.4	14.7	12.5	10.3	13.3	10.8	34.4	22.7	42.4	27.3	33.5	22.2	12.5	10.3	12.5	10.3	12.5	10.3
11	9.7	26.5	17.7	15.0	12.5	15.0	12.5	29.1	20.5	42.2	28.0	39.8	26.6	23.0	17.0	15.0	12.5	15.0	12.5
12	11.1	29.6	20.0	16.8	14.3	16.8	14.3	21.9	17.1	40.0	27.4	44.5	30.0	33.2	23.5	16.8	14.3	16.8	14.3
13	12.1	30.0	20.7	17.7	15.2	17.7	15.2	17.7	15.2	33.5	24.1	44.5	30.4	40.1	27.9	22.6	17.9	17.7	15.2
14	12.6	28.6	20.2	17.6	15.2	17.6	15.2	17.6	15.2	25.3	19.6	41.7	28.9	44.1	30.3	31.3	23.0	17.6	15.2
15	12.7	26.2	18.8	16.7	14.7	16.7	14.7	16.7	14.7	16.7	14.7	37.0	26.2	45.7	31.1	37.9	26.7	17.5	15.1
16	12.6	21.6	16.2	15.1	13.4	15.1	13.4	15.1	13.4	15.1	13.4	28.9	21.2	41.9	28.7	39.6	27.3	23.2	18.0
17	12.1	16.1	12.9	12.8	11.7	12.8	11.7	12.8	11.7	12.8	11.7	19.5	15.4	33.2	23.2	35.1	24.3	24.4	18.3
18	10.9	10.3	9.3	9.6	9.2	9.6	9.2	9.6	9.2	9.6	9.2	10.6	9.7	18.2	14.0	20.8	15.5	17.2	13.4
19	9.7	6.6	6.6	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3
20	8.6	5.6	5.6	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2	6.2
21	7.7	4.6	4.6	5.3	5.3	5.3	5.3	5.3	5.3	5.3	5.3	5.3	5.3	5.3	5.3	5.3	5.3	5.3	5.3
22	6.8	3.4	3.4	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2
23	5.8	2.4	2.4	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2
24	5.6	2.2	2.2	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0
Mean:	7.6	11.3	8.3	7.9	7.0	9.3	7.8	12.4	9.6	14.9	11.0	15.8	11.5	14.5	10.7	12.0	9.3	9.1	7.7

* Hour ending

Table continues

Table 2.34 Air and sol-air temperatures: London area (Bracknell) (1981–1992) — *continued*

(*d*) April 28

Hour*	Air temp.	Sol-air temperature (/ °C) for stated orientation and surface colour																	
		Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	7.1	3.9	3.9	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7	4.7
02	6.4	3.2	3.2	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0	4.0
03	6.0	2.9	2.9	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6
04	5.6	2.3	2.3	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1	3.1
05	5.3	2.0	2.0	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7
06	5.7	5.1	3.9	8.7	6.2	18.4	11.7	20.1	12.7	12.7	8.5	4.3	3.8	4.3	3.8	4.3	3.8	4.3	3.8
07	7.4	12.3	8.6	10.3	8.2	29.8	19.3	36.7	23.3	27.7	18.1	8.4	7.2	8.4	7.2	8.4	7.2	8.4	7.2
08	9.4	19.0	13.4	12.2	10.5	29.8	20.4	41.0	26.8	35.9	23.9	16.9	13.1	12.2	10.5	12.2	10.5	12.2	10.5
09	11.1	25.6	17.9	15.5	13.4	27.4	20.1	43.1	29.1	42.9	29.0	26.9	19.8	15.5	13.4	15.5	13.4	15.5	13.4
10	12.8	30.6	21.5	18.3	15.9	22.0	18.0	39.9	28.1	45.0	31.1	34.8	25.2	18.3	15.9	18.3	15.9	18.3	15.9
11	14.3	33.7	23.9	20.3	17.8	20.3	17.8	33.7	25.4	43.5	31.0	39.9	28.9	24.8	20.3	20.3	17.8	20.3	17.8
12	15.2	35.3	25.3	21.5	19.0	21.5	19.0	25.9	21.4	39.6	29.2	42.5	30.9	33.3	25.6	21.5	19.0	21.5	19.0
13	16.0	36.0	26.0	22.4	19.8	22.4	19.8	22.4	19.8	34.0	26.4	43.2	31.6	40.2	29.9	26.7	22.2	22.4	19.8
14	16.4	34.2	25.2	22.0	19.7	22.0	19.7	22.0	19.7	26.1	22.0	40.1	29.9	43.4	31.8	34.3	26.6	22.0	19.7
15	16.4	31.6	23.7	21.1	19.0	21.1	19.0	21.1	19.0	21.1	19.0	35.5	27.1	44.5	32.2	40.0	29.7	24.3	20.8
16	16.1	27.6	21.3	19.6	17.8	19.6	17.8	19.6	17.8	19.6	17.8	28.8	23.0	42.0	30.5	42.2	30.6	29.3	23.3
17	15.5	23.0	18.5	17.6	16.2	17.6	16.2	17.6	16.2	17.6	16.2	21.4	18.4	37.0	27.2	41.3	29.6	32.0	24.3
18	14.6	17.5	15.1	16.4	15.0	15.1	14.2	15.1	14.2	15.1	14.2	15.1	14.2	27.8	21.4	33.9	24.8	29.3	22.2
19	13.2	12.7	11.9	14.8	13.3	12.1	11.8	12.1	11.8	12.1	11.8	12.1	11.8	17.2	14.6	21.7	17.1	20.7	16.6
20	11.8	9.5	9.5	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9
21	10.8	8.3	8.3	8.8	8.8	8.8	8.8	8.8	8.8	8.8	8.8	8.8	8.8	8.8	8.8	8.8	8.8	8.8	8.8
22	9.7	7.2	7.2	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7	7.7
23	9.1	6.4	6.4	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9
24	8.3	5.5	5.5	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1
Mean:	11.0	16.5	12.8	12.4	11.2	14.9	12.6	17.8	14.3	18.8	14.8	17.8	14.3	17.8	14.2	16.6	13.6	14.1	12.2

* Hour ending

(*e*) May 29

Hour*	Air temp.	Sol-air temperature (/ °C) for stated orientation and surface colour																	
		Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	9.8	6.4	6.4	7.2	7.2	7.2	7.2	7.2	7.2	7.2	7.2	7.2	7.2	7.2	7.2	7.2	7.2	7.2	7.2
02	9.2	5.7	5.7	6.6	6.6	6.6	6.6	6.6	6.6	6.6	6.6	6.6	6.6	6.6	6.6	6.6	6.6	6.6	6.6
03	8.7	5.2	5.2	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1
04	8.3	4.9	4.9	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7
05	8.1	5.8	5.4	6.5	6.1	6.5	6.1	6.5	6.1	6.5	6.1	6.2	6.0	6.2	6.0	6.2	6.0	6.2	6.0
06	9.2	12.1	9.5	17.9	13.3	31.5	21.0	32.5	21.6	20.4	14.7	9.5	8.6	9.5	8.6	9.5	8.6	9.5	8.6
07	10.9	18.6	13.9	17.4	14.0	35.5	24.3	40.6	27.2	30.1	21.2	13.1	11.6	13.1	11.6	13.1	11.6	13.1	11.6
08	12.7	25.0	18.3	16.5	14.6	35.5	25.3	44.6	30.5	37.8	26.6	18.4	15.6	16.5	14.6	16.5	14.6	16.5	14.6
09	14.4	30.8	22.4	19.5	17.2	32.7	24.7	45.4	32.0	43.2	30.7	27.2	21.6	19.5	17.2	19.5	17.2	19.5	17.2
10	16.0	35.7	25.9	22.1	19.6	27.7	22.8	42.7	31.3	45.6	33.0	34.9	26.8	22.1	19.6	22.1	19.6	22.1	19.6
11	17.4	38.6	28.1	23.9	21.4	23.9	21.4	36.6	28.6	44.1	32.8	39.8	30.3	25.9	22.5	23.9	21.4	23.9	21.4
12	18.4	40.2	29.5	25.2	22.7	25.2	22.7	29.4	25.0	40.4	31.3	42.5	32.4	34.4	27.8	25.2	22.7	25.2	22.7
13	19.3	40.6	30.1	25.9	23.5	25.9	23.5	25.9	23.5	34.9	28.5	42.9	33.0	40.9	31.9	30.0	25.7	25.9	23.5
14	19.8	39.7	29.8	25.9	23.6	25.9	23.6	25.9	23.6	27.9	24.7	41.0	32.1	45.1	34.4	38.0	30.4	25.9	23.6
15	20.1	37.3	28.6	25.4	23.3	25.4	23.3	25.4	23.3	25.4	23.3	36.7	29.6	46.4	35.1	43.8	33.6	30.4	26.1
16	20.0	33.5	26.4	24.1	22.3	24.1	22.3	24.1	22.3	24.1	22.3	30.6	25.9	44.4	33.7	46.3	34.8	35.4	28.6
17	19.6	29.1	23.8	22.4	20.9	22.4	20.9	22.4	20.9	22.4	20.9	23.9	21.7	40.2	30.9	46.0	34.2	38.3	29.8
18	18.8	24.0	20.6	23.4	20.9	20.1	19.1	20.1	19.1	20.1	19.1	20.1	19.1	33.1	26.4	41.3	31.0	37.4	28.8
19	17.4	18.7	17.1	22.8	19.8	17.3	16.7	17.3	16.7	17.3	16.7	17.3	16.7	24.4	20.7	32.5	25.2	31.8	24.9
20	15.7	13.9	13.7	14.4	14.1	14.2	14.1	14.2	14.1	14.2	14.1	14.2	14.1	14.4	14.1	14.4	14.2	14.4	14.2
21	14.3	11.8	11.8	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4
22	13.1	10.5	10.5	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1
23	12.1	9.2	9.2	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9	9.9
24	11.4	8.2	8.2	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0
Mean:	14.4	21.1	16.9	16.7	15.2	19.2	16.7	21.7	18.1	21.8	18.1	20.3	17.2	21.0	17.6	20.7	17.5	18.5	16.2

* Hour ending

Table continues

Table 2.34 Air and sol-air temperatures: London area (Bracknell) (1981–1992) — *continued*

(*f*) June 21

Hour*	Air temp.	Sol-air temperature (/ °C) for stated orientation and surface colour																	
		Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	13.9	10.5	10.5	11.4	11.4	11.4	11.4	11.4	11.4	11.4	11.4	11.4	11.4	11.4	11.4	11.4	11.4	11.4	11.4
02	13.1	9.4	9.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4
03	12.4	8.8	8.8	9.6	9.6	9.6	9.6	9.6	9.6	9.6	9.6	9.6	9.6	9.6	9.6	9.6	9.6	9.6	9.6
04	12.0	8.5	8.5	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3
05	12.2	10.3	9.6	11.0	10.3	11.2	10.4	11.1	10.4	11.0	10.3	10.6	10.1	10.6	10.1	10.6	10.1	10.6	10.1
06	13.3	17.8	14.3	24.9	19.0	40.7	28.1	41.4	28.5	26.7	20.0	14.1	12.9	14.1	12.9	14.1	12.9	14.1	12.9
07	15.0	25.7	19.4	24.3	19.7	46.9	32.7	52.5	36.0	38.8	28.0	18.0	16.1	18.0	16.1	18.0	16.1	18.0	16.1
08	16.8	33.4	24.6	21.5	19.2	46.5	33.5	57.3	39.8	48.0	34.4	22.7	19.9	21.5	19.2	21.5	19.2	21.5	19.2
09	18.5	39.8	29.0	24.7	22.1	41.6	31.7	56.3	40.2	53.0	38.3	33.1	26.8	24.7	22.1	24.7	22.1	24.7	22.1
10	20.1	44.9	32.6	27.4	24.6	34.8	28.7	51.8	38.5	54.5	40.1	41.5	32.6	27.4	24.6	27.4	24.6	27.4	24.6
11	21.5	47.8	35.0	29.3	26.5	29.3	26.5	44.3	34.9	52.3	39.5	47.0	36.5	31.0	27.4	29.3	26.5	29.3	26.5
12	22.7	49.3	36.3	30.6	27.8	30.6	27.8	35.4	30.4	47.6	37.4	49.7	38.6	40.6	33.4	30.6	27.8	30.6	27.8
13	23.6	49.3	36.8	31.3	28.5	31.3	28.5	31.3	28.5	41.0	34.0	49.9	39.0	47.9	37.9	35.9	31.1	31.3	28.5
14	24.4	48.1	36.4	31.5	28.8	31.5	28.8	31.5	28.8	33.0	29.7	47.6	37.9	52.4	40.7	45.1	36.5	31.5	28.8
15	24.5	44.8	34.7	30.8	28.3	30.8	28.3	30.8	28.3	30.8	28.3	42.5	34.9	53.3	41.1	51.1	39.8	36.9	31.7
16	24.4	40.5	32.2	29.4	27.2	29.4	27.2	29.4	27.2	29.4	27.2	35.8	30.8	51.3	39.6	53.9	41.1	42.4	34.6
17	24.2	36.1	29.6	27.8	26.0	27.8	26.0	27.8	26.0	27.8	26.0	28.7	26.5	47.4	37.1	54.4	41.1	46.3	36.4
18	23.5	30.8	26.4	30.1	26.7	25.6	24.2	25.6	24.2	25.6	24.2	25.6	24.2	40.6	32.7	50.7	38.4	46.5	36.0
19	22.5	25.2	22.8	30.4	26.2	22.8	22.0	22.8	22.0	22.8	22.0	22.8	22.0	31.6	26.9	42.2	32.9	41.7	32.6
20	20.9	19.5	19.0	19.9	19.4	19.6	19.3	19.6	19.3	19.6	19.3	19.6	19.3	19.9	19.4	19.9	19.4	19.9	19.5
21	19.3	16.6	16.6	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1	17.1
22	17.9	15.1	15.1	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7
23	16.8	13.7	13.7	14.4	14.4	14.4	14.4	14.4	14.4	14.4	14.4	14.4	14.4	14.4	14.4	14.4	14.4	14.4	14.4
24	15.7	12.5	12.5	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2
Mean:	18.7	27.4	22.2	21.9	20.1	25.1	21.9	27.9	23.5	27.6	23.3	25.4	22.1	26.4	22.6	26.3	22.5	23.9	21.2

* Hour ending

(*g*) July 4

Hour*	Air temp.	Sol-air temperature (/ °C) for stated orientation and surface colour																	
		Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	14.8	11.5	11.5	12.3	12.3	12.3	12.3	12.3	12.3	12.3	12.3	12.3	12.3	12.3	12.3	12.3	12.3	12.3	12.3
02	14.2	10.7	10.7	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5
03	13.6	10.1	10.1	10.9	10.9	10.9	10.9	10.9	10.9	10.9	10.9	10.9	10.9	10.9	10.9	10.9	10.9	10.9	10.9
04	13.2	9.6	9.6	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4	10.4
05	13.3	11.1	10.6	11.8	11.3	11.9	11.3	11.9	11.3	11.8	11.3	11.5	11.1	11.5	11.1	11.5	11.1	11.5	11.1
06	14.3	18.1	15.0	24.5	19.2	39.1	27.6	39.9	28.0	26.4	20.3	14.8	13.8	14.8	13.8	14.8	13.8	14.8	13.8
07	15.9	25.3	19.7	23.9	19.9	44.3	31.6	49.6	34.6	37.4	27.5	18.4	16.8	18.4	16.8	18.4	16.8	18.4	16.8
08	17.7	32.5	24.6	22.1	20.0	44.1	32.5	53.9	38.2	45.8	33.5	23.5	20.7	22.1	20.0	22.1	20.0	22.1	20.0
09	19.5	38.5	28.8	25.2	22.8	40.2	31.3	53.6	39.0	50.8	37.4	33.0	27.2	25.2	22.8	25.2	22.8	25.2	22.8
10	21.0	42.7	31.9	27.6	25.0	34.0	28.6	49.2	37.3	51.7	38.7	40.4	32.2	27.6	25.0	27.6	25.0	27.6	25.0
11	22.5	45.5	34.2	29.6	27.0	29.6	27.0	42.8	34.4	50.0	38.5	45.4	35.9	31.4	27.9	29.6	27.0	29.6	27.0
12	23.4	46.7	35.3	30.8	28.0	30.8	28.0	35.0	30.4	45.8	36.6	47.7	37.6	39.7	33.1	30.8	28.0	30.8	28.0
13	24.2	48.0	36.4	31.7	29.0	31.7	29.0	31.7	29.0	40.8	34.1	49.0	38.8	47.1	37.7	36.1	31.4	31.7	29.0
14	24.8	46.8	35.9	31.7	29.1	31.7	29.1	31.7	29.1	33.3	30.0	46.8	37.7	51.2	40.2	44.3	36.2	31.7	29.1
15	25.3	44.4	34.8	31.3	29.0	31.3	29.0	31.3	29.0	31.3	29.0	42.7	35.3	52.8	41.1	50.5	39.8	37.0	32.1
16	25.4	40.5	32.7	30.1	28.1	30.1	28.1	30.1	28.1	30.1	28.1	36.5	31.7	51.1	40.0	53.4	41.3	42.4	35.0
17	25.0	35.8	29.9	28.2	26.6	28.2	26.6	28.2	26.6	28.2	26.6	29.2	27.2	46.7	37.0	53.1	40.7	45.3	36.3
18	24.4	30.6	26.7	30.0	27.1	26.1	24.9	26.1	24.9	26.1	24.9	26.1	24.9	39.6	32.5	48.5	37.6	44.7	35.4
19	23.3	25.2	23.3	29.5	26.2	23.3	22.7	23.3	22.7	23.3	22.7	23.3	22.7	30.8	26.9	39.6	31.9	39.1	31.6
20	21.7	20.1	19.8	20.6	20.2	20.4	20.1	20.4	20.1	20.4	20.1	20.4	20.1	20.5	20.2	20.6	20.3	20.6	20.3
21	20.1	17.6	17.6	18.1	18.1	18.1	18.1	18.1	18.1	18.1	18.1	18.1	18.1	18.1	18.1	18.1	18.1	18.1	18.1
22	18.9	16.1	16.1	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7
23	17.5	14.4	14.4	15.2	15.2	15.2	15.2	15.2	15.2	15.2	15.2	15.2	15.2	15.2	15.2	15.2	15.2	15.2	15.2
24	16.7	13.4	13.4	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2	14.2
Mean:	19.6	27.3	22.6	22.4	20.7	25.3	22.4	27.8	23.8	27.6	23.7	25.8	22.6	26.7	23.1	26.5	23.0	24.2	21.8

* Hour ending

Table continues

Table 2.34 Air and sol-air temperatures: London area (Bracknell) (1981–1992) — *continued*

(*h*) August 4

Hour*	Air temp.	Sol-air temperature (/ °C) for stated orientation and surface colour: Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	15.7	12.5	12.5	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2	13.2
02	15.0	11.6	11.6	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4	12.4
03	14.4	10.9	10.9	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7	11.7
04	13.9	10.3	10.3	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1	11.1
05	13.4	10.4	10.2	11.1	10.9	11.1	11.0	11.1	11.0	11.1	10.9	11.0	10.9	11.0	10.9	11.0	10.9	11.0	10.9
06	14.0	15.1	13.3	19.1	16.0	29.4	21.8	30.7	22.6	22.4	17.9	13.7	12.9	13.7	12.9	13.7	12.9	13.7	12.9
07	15.6	22.3	17.8	20.3	17.5	39.5	28.4	45.5	32.0	35.9	26.3	17.2	15.8	17.2	15.8	17.2	15.8	17.2	15.8
08	17.4	30.4	23.3	21.3	19.2	42.0	31.0	53.6	37.8	47.0	33.9	25.3	21.5	21.3	19.2	21.3	19.2	21.3	19.2
09	19.4	37.6	28.2	25.0	22.5	39.1	30.5	55.1	39.7	53.9	39.0	36.0	28.7	25.0	22.5	25.0	22.5	25.0	2.5
10	21.2	42.2	31.7	27.7	25.1	32.7	27.9	50.7	38.1	55.2	40.7	44.0	34.3	27.7	25.1	27.7	25.1	27.7	25.1
11	22.6	45.7	34.3	29.7	27.0	29.7	27.0	44.2	35.2	53.9	40.8	49.7	38.3	33.5	29.1	29.7	27.0	29.7	27.0
12	23.9	46.8	35.6	31.2	28.4	31.2	28.4	35.8	31.0	49.2	38.6	51.9	40.2	42.6	34.9	31.2	28.4	31.2	28.4
13	24.8	47.0	36.0	31.9	29.2	31.9	29.2	31.9	29.2	43.0	35.5	52.0	40.6	49.4	39.1	36.5	31.8	31.9	29.2
14	25.4	45.9	35.6	32.0	29.4	32.0	29.4	32.0	29.4	35.5	31.3	49.8	39.5	53.6	41.7	45.0	36.7	32.0	29.4
15	25.8	43.2	34.3	31.2	29.0	31.2	29.0	31.2	29.0	31.2	29.0	45.1	36.8	54.9	42.4	51.0	40.1	35.4	31.3
16	25.7	38.8	31.8	29.8	27.9	29.8	27.9	29.8	27.9	29.8	27.9	38.2	32.6	51.7	40.3	52.6	40.8	40.4	33.8
17	25.2	33.6	28.7	27.7	26.2	27.7	26.2	27.7	26.2	27.7	26.2	30.5	27.8	45.6	36.3	50.2	38.9	42.0	34.3
18	24.3	28.0	25.2	27.0	25.2	25.1	24.1	25.1	24.1	25.1	24.1	25.1	24.1	37.0	30.8	43.3	34.4	39.3	32.1
19	23.0	23.1	22.1	25.3	23.6	22.4	22.0	22.4	22.0	22.4	22.0	22.4	22.0	27.0	24.6	31.5	27.0	30.8	26.6
20	21.3	19.0	18.9	19.4	19.3	19.4	19.3	19.4	19.3	19.4	19.3	19.4	19.3	19.4	19.3	19.4	19.4	19.4	19.4
21	20.0	17.4	17.4	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9	17.9
22	18.9	16.2	16.2	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7	16.7
23	18.0	15.1	15.1	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7	15.7
24	17.1	14.1	14.1	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7
Mean:	19.8	26.6	22.3	21.8	20.4	24.5	21.9	27.5	23.7	28.2	24.0	26.9	23.3	26.8	23.3	25.8	22.7	23.4	21.3

* Hour ending

(*i*) September 4

Hour*	Air temp.	Sol-air temperature (/ °C) for stated orientation and surface colour: Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	11.6	8.9	8.9	9.4	9.4	9.4	9.4	9.4	9.4	9.4	9.4	9.4	9.4	9.4	9.4	9.4	9.4	9.4	9.
02	11.1	8.5	8.5	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0
03	10.5	7.8	7.8	8.3	8.3	8.3	8.3	8.3	8.3	8.3	8.3	8.3	8.3	8.3	8.3	8.3	8.3	8.3	8.3
04	9.9	6.8	6.8	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5	7.5
05	9.5	6.4	6.4	7.1	7.1	7.1	7.1	7.1	7.1	7.1	7.1	7.1	7.1	7.1	7.1	7.1	7.1	7.1	7.1
06	9.4	6.6	6.4	7.3	7.1	7.3	7.1	7.3	7.1	7.3	7.1	7.2	7.1	7.2	7.1	7.2	7.1	7.2	7.1
07	10.2	12.0	9.9	10.2	9.4	23.8	17.1	29.7	20.4	24.6	17.5	11.1	9.9	10.2	9.4	10.2	9.4	10.2	9.4
08	12.1	18.9	14.5	13.9	12.5	28.8	20.9	41.2	28.0	38.2	26.3	21.2	16.6	13.9	12.5	13.9	12.5	13.9	12.5
09	14.0	25.0	18.8	17.5	15.6	26.3	20.5	42.5	29.8	44.3	30.8	30.9	23.1	17.5	15.6	17.5	15.6	17.5	15.6
10	15.5	30.7	22.7	20.5	18.2	21.8	18.9	41.9	30.3	49.3	34.6	40.4	29.5	20.5	18.2	20.5	18.2	20.5	18.2
11	16.9	35.1	25.8	22.7	20.2	22.7	20.2	37.2	28.4	50.4	36.0	47.7	34.4	30.4	24.5	22.7	20.2	22.7	20.2
12	18.0	36.9	27.3	24.1	21.6	24.1	21.6	29.0	24.3	46.2	34.1	50.5	36.6	39.6	30.3	24.1	21.6	24.1	21.6
13	18.7	36.2	27.3	24.4	22.0	24.4	22.0	24.4	22.0	38.9	30.1	49.0	35.9	45.0	33.7	29.0	24.6	24.4	22.0
14	18.9	34.6	26.5	23.9	21.7	23.9	21.7	23.9	21.7	30.7	25.6	46.0	34.3	48.4	35.6	36.7	28.9	23.9	21.7
15	19.1	31.9	25.1	22.9	21.1	22.9	21.1	22.9	21.1	22.9	21.1	40.9	31.3	49.1	36.0	42.3	32.1	23.9	21.7
16	18.9	28.3	23.0	21.7	20.2	21.7	20.2	21.7	20.2	21.7	20.2	34.1	27.1	46.7	34.3	45.0	33.3	29.8	24.7
17	18.5	23.3	20.0	19.9	18.7	19.9	18.7	19.9	18.7	19.9	18.7	25.3	21.7	37.8	28.8	40.0	30.0	30.8	24.8
18	17.5	18.6	17.1	17.4	16.8	17.4	16.8	17.4	16.8	17.4	16.8	18.1	17.2	28.7	23.1	32.7	25.4	28.0	22.7
19	16.0	13.8	13.6	14.2	14.1	14.2	14.0	14.2	14.0	14.2	14.0	14.2	14.0	14.3	14.1	14.3	14.1	14.3	14.1
20	14.7	12.0	12.0	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5	12.5
21	13.6	10.9	10.9	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5	11.5
22	12.8	10.0	10.0	10.6	10.6	10.6	10.6	10.6	10.6	10.6	10.6	10.6	10.6	10.6	10.6	10.6	10.6	10.6	10.6
23	12.2	9.5	9.5	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0	10.0
24	11.6	8.8	8.8	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3	9.3
Mean:	14.2	18.4	15.3	14.8	13.9	16.4	14.8	19.5	16.6	21.7	17.8	22.2	18.1	21.0	17.4	18.8	16.2	16.1	14.6

* Hour ending

Table continues

Table 2.34 Air and sol-air temperatures: London area (Bracknell) (1981–1992) — *continued*

(*j*) October 4

Hour*	Air temp.	Sol-air temperature (/ °C) for stated orientation and surface colour: Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	9.0	5.7	5.7	6.4	6.4	6.4	6.4	6.4	6.4	6.4	6.4	6.4	6.4	6.4	6.4	6.4	6.4	6.4	6.4
02	8.6	5.3	5.3	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1	6.1
03	8.2	5.0	5.0	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7	5.7
04	7.8	4.5	4.5	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2	5.2
05	7.5	4.1	4.1	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9
06	7.4	4.2	4.2	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9	4.9
07	7.8	5.1	4.9	5.6	5.5	5.8	5.6	5.8	5.6	5.8	5.6	5.8	5.6	5.6	5.5	5.6	5.5	5.6	5.5
08	9.2	11.5	9.1	9.1	8.3	20.3	14.6	33.4	22.1	32.8	21.8	18.6	13.6	9.1	8.3	9.1	8.3	9.1	8.3
09	11.4	19.0	14.2	13.0	11.6	20.3	15.7	41.7	28.0	46.3	30.7	32.3	22.5	13.0	11.6	13.0	11.6	13.0	11.6
10	13.4	25.0	18.5	16.3	14.6	16.3	14.6	40.2	28.2	51.1	34.6	42.4	29.5	18.0	15.5	16.3	14.6	16.3	14.6
11	14.8	28.3	21.0	18.4	16.6	18.4	16.6	32.9	24.8	48.6	33.8	47.0	32.9	28.8	22.5	18.4	16.6	18.4	16.6
12	15.9	29.2	22.1	19.8	17.8	19.8	17.8	24.3	20.4	42.2	30.6	47.0	33.3	36.2	27.1	19.8	17.8	19.8	17.8
13	16.4	29.4	22.4	20.2	18.3	20.2	18.3	20.2	18.3	36.3	27.4	46.9	33.4	42.2	30.7	24.6	20.8	20.2	18.3
14	16.6	28.4	21.9	20.1	18.3	20.1	18.3	20.1	18.3	29.1	23.3	44.5	32.1	45.9	32.9	32.5	25.3	20.1	18.3
15	16.8	25.6	20.5	19.2	17.7	19.2	17.7	19.2	17.7	20.7	18.5	39.0	28.9	45.7	32.7	37.3	27.9	19.2	17.7
16	15.9	21.0	17.5	17.1	16.0	17.1	16.0	17.1	16.0	17.1	16.0	30.3	23.4	40.1	29.0	36.8	27.1	22.1	18.8
17	14.9	16.2	14.5	14.7	14.0	14.7	14.0	14.7	14.0	14.7	14.0	21.0	17.5	30.3	22.8	30.7	23.1	22.0	18.1
18	13.6	11.2	11.0	11.6	11.4	11.6	11.4	11.6	11.4	11.6	11.4	11.7	11.5	11.7	11.5	11.7	11.5	11.7	11.5
19	12.8	9.7	9.7	10.3	10.3	10.3	10.3	10.3	10.3	10.3	10.3	10.3	10.3	10.3	10.3	10.3	10.3	10.3	10.3
20	11.9	8.9	8.9	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.5
21	11.1	8.0	8.0	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6	8.6
22	10.5	7.3	7.3	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0	8.0
23	9.8	6.6	6.6	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3	7.3
24	9.5	6.3	6.3	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9	6.9
Mean:	11.7	13.6	11.4	11.2	10.6	12.0	11.0	15.2	12.9	18.3	14.6	19.6	15.3	17.1	13.9	14.2	12.2	11.7	10.9

* Hour ending

(*k*) November 4

Hour*	Air temp.	Sol-air temperature (/ °C) for stated orientation and surface colour: Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	5.2	2.6	2.6	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2
02	5.0	2.3	2.3	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8	2.8
03	4.5	2.0	2.0	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5
04	4.3	1.5	1.5	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1
05	3.9	1.0	1.0	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7
06	3.7	0.7	0.7	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4
07	3.7	0.6	0.6	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3
08	3.9	1.5	1.3	2.0	1.8	2.1	1.9	2.1	1.9	2.2	1.9	2.1	1.9	2.0	1.8	2.0	1.8	2.0	1.8
09	5.4	7.1	5.0	5.0	4.4	8.1	6.1	26.9	16.7	32.8	20.1	23.0	14.5	5.0	4.4	5.0	4.4	5.0	4.4
10	7.0	12.6	8.7	7.8	6.7	7.8	6.7	29.3	18.9	42.0	26.2	35.4	22.4	12.5	9.4	7.8	6.7	7.8	6.7
11	8.4	16.3	11.5	9.9	8.7	9.9	8.7	23.8	16.5	41.9	26.9	41.6	26.7	23.1	16.1	9.9	8.7	9.9	8.7
12	9.5	18.7	13.3	11.4	10.1	11.4	10.1	16.3	12.8	38.8	25.6	45.1	29.3	32.4	21.9	11.4	10.1	11.4	10.1
13	9.9	18.4	13.3	11.9	10.6	11.9	10.6	11.9	10.6	30.7	21.1	42.1	27.7	36.4	24.4	16.3	13.0	11.9	10.6
14	10.2	16.8	12.6	11.6	10.4	11.6	10.4	11.6	10.4	22.4	16.4	37.5	25.0	37.7	25.2	23.0	16.8	11.6	10.4
15	10.0	13.9	10.9	10.6	9.6	10.6	9.6	10.6	9.6	14.0	11.5	30.0	20.6	34.7	23.3	25.7	18.1	10.6	9.6
16	9.1	9.7	8.3	8.6	8.0	8.6	8.0	8.6	8.0	8.6	8.0	18.8	13.8	24.4	16.9	21.0	15.0	10.4	9.1
17	7.9	5.5	5.3	5.9	5.8	5.9	5.8	5.9	5.8	5.9	5.8	6.0	5.9	6.0	5.9	6.0	5.9	6.0	5.9
18	7.3	4.6	4.6	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1
19	6.7	4.2	4.2	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6	4.6
20	6.3	3.8	3.8	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2	4.2
21	5.9	3.4	3.4	3.8	3.8	3.8	3.8	3.8	3.8	3.8	3.8	3.8	3.8	3.8	3.8	3.8	3.8	3.8	3.8
22	5.7	3.1	3.1	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6	3.6
23	5.4	2.7	2.7	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2	3.2
24	5.0	2.5	2.5	2.9	2.9	2.9	2.9	2.9	2.9	2.9	2.9	2.9	2.9	2.9	2.9	2.9	2.9	2.9	2.9
Mean:	6.4	6.5	5.2	5.3	4.9	5.4	5.0	7.9	6.4	11.7	8.6	13.5	9.6	10.7	8.0	7.1	6.0	5.4	5.0

* Hour ending

Table continues

Table 2.34 Air and sol-air temperatures: London area (Bracknell) (1981–1992) — *continued*

(*l*) December 4

Hour*	Air temp.	Sol-air temperature (/ °C) for stated orientation and surface colour																	
		Horizontal		North		North-east		East		South-east		South		South-west		West		North-west	
		Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light	Dark	Light
01	1.3	–1.9	–1.9	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1
02	1.1	–2.1	–2.1	–1.3	–1.3	–1.3	–1.3	–1.3	–1.3	–1.3	–1.3	–1.3	–1.3	–1.3	–1.3	–1.3	–1.3	–1.3	–1.3
03	0.7	–2.5	–2.5	–1.7	–1.7	–1.7	–1.7	–1.7	–1.7	–1.7	–1.7	–1.7	–1.7	–1.7	–1.7	–1.7	–1.7	–1.7	–1.7
04	0.2	–3.2	–3.2	–2.4	–2.4	–2.4	–2.4	–2.4	–2.4	–2.4	–2.4	–2.4	–2.4	–2.4	–2.4	–2.4	–2.4	–2.4	–2.4
05	0.0	–3.7	–3.7	–2.7	–2.7	–2.7	–2.7	–2.7	–2.7	–2.7	–2.7	–2.7	–2.7	–2.7	–2.7	–2.7	–2.7	–2.7	–2.7
06	–0.2	–3.9	–3.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9
07	–0.4	–3.8	–3.8	–3.0	–3.0	–3.0	–3.0	–3.0	–3.0	–3.0	–3.0	–3.0	–3.0	–3.0	–3.0	–3.0	–3.0	–3.0	–3.0
08	–0.3	–3.8	–3.8	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9	–2.9
09	–0.1	–2.5	–2.9	–2.0	–2.2	–1.7	–2.1	–1.7	–2.1	–1.6	–2.1	–1.7	–2.1	–2.0	–2.2	–2.0	–2.2	–2.0	–2.2
10	1.2	3.5	1.2	1.2	0.4	1.2	0.4	18.3	10.1	30.0	16.8	25.4	14.2	6.7	3.5	1.2	0.4	1.2	0.4
11	2.7	7.9	4.1	3.5	2.4	3.5	2.4	17.4	10.2	37.2	21.6	37.7	21.9	18.7	11.0	3.5	2.4	3.5	2.4
12	4.0	10.2	5.9	5.0	3.8	5.0	3.8	9.9	6.6	34.3	20.5	41.3	24.6	27.9	16.8	5.0	3.8	5.0	3.8
13	4.9	10.8	6.7	5.9	4.7	5.9	4.7	5.9	4.7	27.6	17.0	40.2	24.3	33.6	20.4	10.6	7.3	5.9	4.7
14	5.4	9.6	6.2	5.8	4.7	5.8	4.7	5.8	4.7	19.2	12.3	36.1	22.0	35.6	21.7	18.1	11.7	5.8	4.7
15	5.1	6.2	4.2	4.5	3.8	4.5	3.8	4.5	3.8	9.1	6.4	24.5	15.1	28.3	17.2	18.6	11.7	4.5	3.8
16	4.1	1.9	1.5	2.3	2.0	2.3	2.0	2.3	2.0	2.3	2.0	2.5	2.1	2.5	2.1	2.5	2.1	2.4	2.1
17	3.3	0.2	0.2	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9	0.9
18	2.7	–0.5	–0.5	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.2
19	2.2	–0.8	–0.8	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2	–0.2
20	1.8	–1.2	–1.2	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6	–0.6
21	1.6	–1.5	–1.5	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8	–0.8
22	1.4	–1.8	–1.8	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1	–1.1
23	1.1	–1.8	–1.8	–1.2	–1.2	–1.2	–1.2	–1.2	–1.2	–1.2	–1.2	–1.2	–1.2	–1.2	–1.2	–1.2	–1.2	–1.2	–1.2
24	0.7	–2.5	–2.5	–1.8	–1.8	–1.8	–1.8	–1.8	–1.8	–1.8	–1.8	–1.8	–1.8	–1.8	–1.8	–1.8	–1.8	–1.8	–1.8
Mean:	1.9	0.5	–0.3	0.2	–0.1	0.2	–0.1	1.7	0.7	5.6	3.0	7.6	4.1	5.4	2.8	1.5	0.6	0.2	–0.1

* Hour ending

Table 2.35 Air and sol-air temperatures: Manchester (Aughton) (1981–1992): refer to CD-ROM

Table 2.36 Air and sol-air temperatures: Edinburgh (Mylnefield) (1981–1992): refer to CD-ROM

2.7.5 Illuminance data

Quantitative daylight illuminance data are needed for daylighting design calculations including the sizing of windows, choice of glazing materials and the design of window shading systems. Daylighting and electric lighting systems must be designed to operate interactively. The quantitative estimation of the energy consumption of artificial lighting with different control systems needs knowledge of the statistical availability of the horizontal components of daylight, the global horizontal illuminance and the diffuse horizontal illuminance from the sky vault. A means of assessing the effect of orientation on vertical surface daylight availability is also required. These issues are considered in detail in CIBSE Lighting Guide LG10: *Daylighting and window design*(13). The required climatic data are discussed in detail in CIBSE Guide J: *Weather, solar and illuminance data*(1).

2.7.5.1 Cumulative frequency data

Cumulative frequency distributions of horizontal illuminance are required in order to assess lighting system design. UK data have been derived using luminous efficacy algorithms for global and diffuse illumination estimation(14). These algorithms have been based on short time series of simultaneously observed values of irradiation and illuminance(15). The algorithms have been applied to the recent observed UK hourly irradiation time-series data sets to estimate the corresponding hourly illuminance time-series. These derived illuminance time-series were then analysed statistically to produce annual cumulative frequency illuminance data.

The cumulative frequency analysis must be based on defined working day lengths. Hunt(16) defined the standard UK working day for illumination design purposes as 09:00 to 17:30 LAT in winter and 08:00 to 16:30 LAT for the period between April and October to allow for British Summer Time. For practical purposes, the difference between LAT and GMT is negligible for the sites analysed. The cumulative data are based on the above standard working year.

Annual global cumulative frequency data(14) for a range of UK sites are given in Table 2.37 and the corresponding diffuse cumulative frequency data are given in Table 2.38. The tables are arranged in order of increasing latitude. In general, increase of latitude decreases the annual availability of daylight. However, because northern climates are much clearer, the impact on the horizontal beam illuminance of lower solar altitudes in mid-summer during cloudless clear hours is offset by greater mid-summer clarity.

Table 2.37 Proportion of year for which stated global horizontal illuminance is exceeded (based on measured global horizontal irradiation and luminous efficacy algorithms)

Illuminance / klux	London (51° 52′N)	Aberporth (52° 13′N)	Aughton (53° 55′N)	Dunstaffanage (56° 47′N)	Aberdeen (57° 20′N)	Stornaway (58° 52′N)
0.0 (axis)†	0.971	0.970	0.967	0.959	0.957	0.954
1.0	0.938	0.944	0.916	0.898	0.931	0.898
2.0	0.905	0.908	0.890	0.862	0.861	0.866
3.0	0.882	0.883	0.865	0.828	0.827	0.834
4.0	0.859	0.857	0.840	0.794	0.797	0.802
5.0	0.834	0.834	0.817	0.762	0.770	0.774
6.0	0.806	0.808	0.793	0.733	0.743	0.746
7.0	0.784	0.784	0.770	0.704	0.717	0.720
8.0	0.761	0.762	0.748	0.677	0.693	0.695
9.0	0.736	0.739	0.728	0.652	0.670	0.672
10.0	0.714	0.718	0.705	0.628	0.649	0.647
15.0	0.613	0.624	0.606	0.528	0.553	0.545
20.0	0.522	0.541	0.519	0.447	0.472	0.455
25.0	0.422	0.470	0.450	0.381	0.403	0.383
30.0	0.379	0.407	0.386	0.326	0.341	0.320
35.0	0.321	0.351	0.332	0.276	0.285	0.262
40.0	0.268	0.300	0.282	0.231	0.235	0.210
45.0	0.218	0.254	0.236	0.191	0.188	0.164
50.0	0.174	0.214	0.194	0.159	0.150	0.130
55.0	0.139	0.178	0.156	0.131	0.118	0.103
60.0	0.106	0.147	0.122	0.106	0.091	0.080
65.0	0.080	0.120	0.094	0.085	0.068	0.062
70.0	0.058	0.095	0.070	0.065	0.050	0.046
75.0	0.039	0.073	0.050	0.049	0.035	0.033
80.0	0.025	0.054	0.033	0.034	0.022	0.022
85.0	0.014	0.038	0.019	0.022	0.012	0.014
90.0	0.006	0.024	0.010	0.011	0.005	0.007
95.0	0.000	0.012	0.004	0.004	0.000	0.002
100.0	0.000	0.004	0.001	0.001	0.000	0.000
105.0	0.000	0.000	0.000	0.000	0.000	0.000

† Values at 0 klux estimated algorithmically

Note: London/Aberporth/Dunstaffanage/Aberdeen: 1975–1994; Aughton/Stornaway: 1982–1994

Table 2.38 Proportion of year for which stated diffuse horizontal illuminance is exceeded (based on measured diffuse irradiation and luminous efficacy algorithms)

Illuminance / klux	London (51°52′N)	Aberporth (52°13′N)	Aughton (53°55′N)	Stornaway (58°52′N)
0.0 (axis)†	0.971	0.960	0.967	0.954
1.0	0.945	0.947	0.916	0.915
2.0	0.911	0.909	0.888	0.871
3.0	0.877	0.875	0.856	0.826
4.0	0.848	0.845	0.830	0.791
5.0	0.817	0.823	0.802	0.758
6.0	0.786	0.793	0.772	0.720
7.0	0.755	0.763	0.742	0.686
8.0	0.719	0.734	0.712	0.656
9.0	0.687	0.705	0.682	0.627
10.0	0.658	0.674	0.652	0.599
15.0	0.513	0.526	0.515	0.466
20.0	0.385	0.392	0.396	0.349
25.0	0.266	0.284	0.284	0.248
30.0	0.168	0.195	0.190	0.164
35.0	0.095	0.126	0.119	0.098
40.0	0.043	0.078	0.066	0.052
45.0	0.015	0.046	0.032	0.025
50.0	0.005	0.026	0.013	0.011
55.0	0.000	0.013	0.004	0.004
60.0	0.000	0.006	0.001	0.000
65.0	0.000	0.000	0.000	0.000

† Values at 0 klux estimated algorithmically

Note: London/Aberporth: 1975–1994; Aughton/Stornaway: 1982–1994

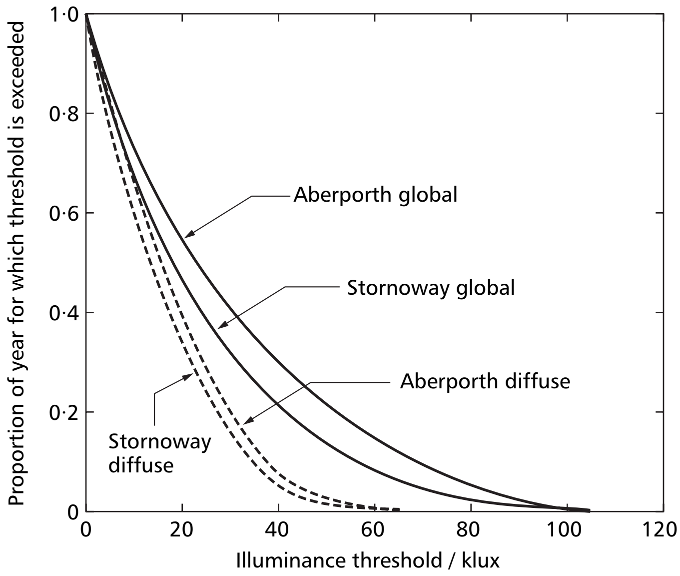


Figure 2.13 Proportion of year for which global and diffuse horizontal illuminances are exceeded for Aberporth and Stornoway

The maximum diffuse illuminances are about 60 klux, except in central London where the illuminance is affected by pollution. Figure 2.13 shows the range of both the annual global and diffuse data for Aberporth and Stornoway. The influence of latitude is greater on annual global cumulative illuminance than on annual diffuse cumulative illuminance. This is because the high levels of cloud in summer in the north increases the diffuse illuminance which offsets the substantial reduction in diffuse illumination in winter. Values for other sites can be estimated from Tables 2.37 and 2.38 using linear interpolation based on latitude. However, the effects of urban pollution mean that the data for central London should not be used for latitude interpolation.

2.7.5.2 Orientation factor for UK

The effect of orientation on the availability of diffuse illuminance can be taken into account by means of an orientation factor, see Table 2.39.

The diffuse illuminance received on the horizontal plane in a room with CIE sky daylight factor DF (%) is given by:

$$E_v = E_{vd}(c)\,(\mathrm{DF}/100)\,f_o \tag{2.1}$$

where E_v is the diffuse illuminance received on the horizontal plane (klux), $E_{vd}(c)$ is the diffuse horizontal irradiance available for proportion c (%) of the year (klux), DF is the daylight factor (%) and f_o is the orientation factor.

$E_{vd}(c)$ is obtained from Table 2.38 using linear interpolation based on latitude.

The orientation factor applies only to the annual cumulative diffuse illuminance values.

Table 2.39 Orientation factor for availability of diffuse illuminance (vertical apertures)

Orientation	Factor, f_o
North	0.97
East	1.15
South	1.55
West	1.21

2.8 Wind data

2.8.1 Measurement of wind data

Wind data are measured with an anemometer and wind vane mounted, where possible, 10 m above ground level, or in some cases, above the roof of a building. Most sites are in exposed open situations such as airports; very few data are available from city centres. The wind speed data in this Guide are derived from measurements from anemometers designed to record high wind speeds. However, as these have a relatively high starting speed, low speed data (i.e. below 2 m·s^{-1}) are less reliable.

The hourly values used to derive the statistics quoted in the tables are hourly mean speeds and median direction, derived either from anemograph charts or from a dedicated logging system which records data at 1-minute intervals.

2.8.2 Sources of published data

Comprehensive data on wind speed and direction for the UK are published by the Met Office and include tabulated values for about 100 stations[17] and contour maps of hourly mean wind speeds exceeded for percentages of time between 0.1% and 75% of the time[18]. The *European Wind Atlas*[19] gives detailed wind statistics for 22 UK stations and a large number of European locations.

2.8.3 Wind climate in the UK

Table 2.40 summarises the wind climate for 20 locations in the UK shown. The following information is given for each station:

— Name and location of the station and the terrain category of its surroundings (see Table 2.47).

— Hourly mean wind speed (m·s^{-1}).

— Parameters of the Weibull distribution fitted to the hourly wind speeds. These can be used to calculate the frequency of occurrence, $f(v)$, of any hourly wind speed, v, with a bandwidth of 1 m·s^{-1}, as follows:

$$f(v) = (k / A)\,(v / A)^{k-1} \exp\,[-(v / A)^k] \tag{2.2}$$

where $f(v)$ is the frequency of occurrence of wind speed v; k and A are Weibull coefficients and v is the wind speed to the nearest metre per second (m·s^{-1}).

The probability of the wind speed exceeding v is given by:

$$F(v) = \exp\,[-(v / A)^k] \tag{2.3}$$

For example, for Heathrow A = 3.25 and k = 1.45. For v = 5 m·s^{-1}, equation 2.2 gives $f(v)$ = 0.083, i.e. the wind speed will be between 4.5 and 5.5 m·s^{-1} for 8% of the time. Equation 2.3 gives $F(v)$ = 0.15, i.e. the wind speed will exceed 5 m·s^{-1} for 15% of the time.

— The percentiles of the distribution of wind speed. For example, at Heathrow the wind speed will be less than or equal to 1.53 m·s^{-1} for 20% of the time and less than or equal to 6.11 m·s^{-1} for 90% of the time.

Table 2.40 Summary of wind climate at selected UK locations (approx. 1982–2002; see Table 2.1)

Parameter	Belfast (Aldergrove)	Birmingham (Elmdon)	Cardiff (Rhoose)	Edinburgh (Turnhouse)	Glasgow (Abbotsinch)	London (Heathrow)	Manchester (Ringway)	Plymouth (Mount Batten)
Terrain category	I	III	II	II	I	III	II	I
Mean speed (m·s^{-1})	4.54	3.72	5.10	4.35	4.42	3.30	4.12	5.23
Weibull coeffs.	A = 4.69	A = 3.81	A = 5.35	A = 4.49	A = 4.51	A = 3.25	A = 4.22	A = 5.51
	k = 1.59	k = 1.65	k = 1.61	k = 1.57	k = 1.44	k = 1.45	k = 1.51	k = 1.53
Percentile	Maximum wind speed (/ m·s^{-1}) for stated percentile							
5	1.02	1.02	1.02	1.02	1.02	1.02	1.02	1.02
10	1.02	1.02	1.53	1.02	1.02	1.02	1.02	1.02
20	2.04	1.53	2.55	1.53	1.53	1.53	2.04	2.04
30	3.05	2.55	3.05	2.55	2.55	2.04	2.55	3.05
40	3.56	3.05	4.07	3.05	3.05	2.55	3.05	4.07
50	4.07	3.56	4.58	4.07	4.07	3.05	3.56	4.58
60	5.09	4.07	5.60	4.58	4.58	3.56	4.58	5.60
70	5.60	4.58	6.62	5.60	5.50	4.07	5.09	6.62
80	6.62	5.60	7.64	6.62	6.62	5.09	6.11	7.64
90	8.14	6.62	9.16	8.14	8.65	6.11	7.64	9.67
95	9.16	7.64	10.69	9.16	10.18	7.13	8.65	11.71

Note: values converted from measured data in knots

Note that the percentiles given are derived directly from the data, whereas the Weibull coefficients are derived from a smoothed curve through the whole wind spectrum. Therefore, there will not be exact agreement between the two methods, especially at low wind speeds.

2.8.4 Wind speed, direction and external temperature

To aid ventilation design, frequency distributions of average wind speed by both direction and external temperature are shown for the whole year for London (Heathrow), Manchester (Ringway) and Edinburgh (Turnhouse) in Tables 2.41 to 2.46. Seasonal averages are given on the CD-ROM that accompanies this Guide, along with annual and seasonal averages for Belfast, Birmingham, Cardiff, Glasgow and Plymouth (Tables 2.47 to 2.56).

The binned data are derived from the original observed data as follows:

— *wind speed*: recorded in whole knots, which have been converted to m·s^{-1} the bin labels therefore give only a guide to the values they contain

— *wind direction*: recorded to the nearest 10° with north represented by 360°; 'calm' (i.e. a wind speed of zero) and 'variable' (i.e. a wind speed of 2 knots or where no clear direction existed over the hour) have been combined

— *dry bulb temperature*: recorded to the nearest 0.1 °C.

Frequencies greater than zero but less than 0.005% are shown as '0.00'.

2.8.5 Influence of height and environment on mean wind speed

2.8.5.1 General

The mean wind speed is a function of the local environment (topography, ground roughness, nearby obstacles). Mean wind speeds, observed at two different sites of similar altitude within about ten kilometres, will not differ significantly if both sites have a similar local ground cover environment. In the opposite case, to obtain the reference regional wind, v_{ref}, from observations over a site, or to obtain an estimate of wind conditions over a site from the reference wind, the mean wind speed should be corrected using the following relationship (in the case where there are no near obstacles):

$$v_z = v_{ref}\, C_R(z)\, C_T \qquad (2.4)$$

where v_z the wind speed at height z (m·s^{-1}), v_{ref} is the reference regional wind speed (m·s^{-1}), $C_R(z)$ is the roughness coefficient at height z and C_T is the topography coefficient.

2.8.5.2 Roughness coefficient

The roughness coefficient accounts for the variability of mean wind speed at the site due to:

— height above the ground

— roughness of the terrain over which the wind has passed.

The roughness coefficient at height z (where $z \geq z_{min}$) is given by:

$$C_R(z) = K_R \ln (z / z_o) \qquad (2.5)$$

where K_R is the terrain factor, z_o is the roughness length (m) and z_{min} is the minimum height (m), see Table 2.57.

It is not possible to make realistic estimates of wind speed at heights below z_{min} and locally measured data should be used wherever possible.

The parameters depend on the terrain category, see Table 2.57. If there is a change of roughness upwind of a site within a kilometre, the smoothest terrain category in the upwind direction should be used.

Table 2.41 Annual and seasonal* average percentage frequencies of hourly wind speed by direction: London (Heathrow) (1982–2002)

Direction / deg.	Percentage frequency for stated range of wind speed / m.s^{-1}													
	0–2	2–4	4–6	6–8	8–10	10–12	12–14	14–16	16–18	18–20	20–22	22–24	24–26	All speeds
Calm/variable	0.11													0.11
350 to 010	3.09	2.77	1.34	0.30	0.05	0.01	0.00							7.56
020 to 040	2.32	3.29	1.87	0.48	0.07	0.01	0.00							8.04
050 to 070	0.89	1.88	1.40	0.60	0.11	0.01								4.89
080 to 100	0.79	1.85	1.61	0.67	0.12	0.01	0.00							5.05
110 to 130	1.17	1.82	0.96	0.28	0.03	0.00								4.26
140 to 160	1.69	1.75	0.88	0.33	0.07	0.01	0.00							4.72
170 to 190	2.64	3.83	2.41	0.99	0.28	0.07	0.01	0.00		0.00				10.24
200 to 220	3.52	5.30	4.52	2.44	0.78	0.21	0.04	0.01	0.00					16.82
230 to 250	3.22	4.74	3.86	1.73	0.41	0.07	0.03	0.00	0.00	0.00				14.06
260 to 280	3.63	4.16	2.16	0.70	0.19	0.04	0.01		0.00					10.89
290 to 310	3.17	2.17	1.04	0.25	0.05	0.01	0.00							6.68
320 to 340	2.98	2.49	0.98	0.21	0.02									6.68
All directions	29.23	36.03	23.02	8.98	2.18	0.44	0.09	0.01	0.01	0.00		0.01	0.00	100.00

* for seasonal averages refer to CD-ROM

Table 2.42 Annual and seasonal* average percentage frequencies of hourly wind speed by temperature: London (Heathrow) (1982–2002)

Temperature / °C	Percentage frequency for stated range of wind speed / m.s^{-1}													
	0–2	2–4	4–6	6–8	8–10	10–12	12–14	14–16	16–18	18–20	20–22	22–24	24–26	All speeds
–10.0 to –8.1	0.01	0.01	0.00											0.01
–8.0 to –6.1	0.02	0.02	0.01	0.00										0.05
–6.0 to –4.1	0.11	0.05	0.03	0.01	0.00									0.20
–4.0 to –2.1	0.32	0.13	0.08	0.05	0.03	0.00	0.00							0.61
–2.0 to –0.1	0.97	0.45	0.18	0.10	0.04	0.00	0.00							1.74
0.0 to 1.9	1.77	1.10	0.40	0.19	0.04	0.01	0.00							3.51
2.0 to 3.9	2.61	2.01	1.04	0.43	0.08	0.01	0.00							6.17
4.0 to 5.9	2.97	3.08	1.69	0.56	0.16	0.02	0.00							8.47
6.0 to 7.9	3.26	3.79	2.26	0.87	0.22	0.04	0.00	0.00	0.00					10.44
8.0 to 9.9	3.13	4.33	2.96	1.23	0.36	0.08	0.02	0.00	0.00					12.11
10.0 to 11.9	3.24	4.47	3.02	1.49	0.44	0.13	0.03	0.00	0.00					12.83
12.0 to 13.9	3.35	4.17	2.53	1.16	0.35	0.10	0.03	0.00	0.00	0.00				11.70
14.0 to 15.9	2.87	3.99	2.46	0.82	0.18	0.03	0.00	0.00						10.36
16.0 to 17.9	2.07	3.16	2.20	0.70	0.12	0.01	0.00							8.26
18.0 to 19.9	1.25	2.17	1.64	0.53	0.07	0.01								5.67
20.0 to 21.9	0.67	1.34	1.07	0.35	0.04	0.00	0.00							3.48
22.0 to 23.9	0.34	0.83	0.71	0.21	0.02	0.00								2.11
24.0 to 25.9	0.19	0.51	0.39	0.13	0.01									1.23
26.0 to 27.9	0.06	0.28	0.21	0.08	0.01									0.64
28.0 to 29.9	0.01	0.11	0.09	0.05	0.01									0.26
30.0 to 31.9	0.00	0.04	0.04	0.02	0.00									0.11
32.0 to 33.9	0.00	0.01	0.01	0.00										0.03
34.0 to 35.9		0.00	0.00											0.00
All temps.	29.23	36.03	23.02	8.98	2.18	0.44	0.09	0.01	0.01	0.00		0.01	0.00	100.00

* for seasonal averages refer to CD-ROM

Table 2.43 Annual and seasonal* average percentage frequencies of hourly wind speed by direction: Manchester (Ringway) (1983–2002)

Direction / deg.	Percentage frequency for stated range of wind speed / m.s^{-1}													
	0–2	2–4	4–6	6–8	8–10	10–12	12–14	14–16	16–18	18–20	20–22	22–24	24–26	All speeds
Calm/variable	0.24	0.00												0.24
350 to 010	0.67	1.13	0.70	0.29	0.09	0.02	0.00							2.91
020 to 040	0.66	1.68	1.39	0.67	0.18	0.05	0.00							4.63
050 to 070	1.32	2.09	1.62	1.00	0.40	0.13	0.04	0.01	0.00					6.62
080 to 100	1.63	2.06	1.51	0.91	0.32	0.10	0.02	0.00						6.54
110 to 130	1.45	1.81	1.09	0.44	0.10	0.02	0.00	0.00						4.91
140 to 160	1.81	2.69	2.11	1.05	0.38	0.12	0.02	0.00						8.18
170 to 190	2.94	5.51	5.28	3.02	1.21	0.43	0.11	0.02	0.00					18.52
200 to 220	2.40	3.12	3.27	2.26	0.99	0.38	0.12	0.05	0.02	0.01	0.00			12.62
230 to 250	2.20	2.69	2.43	1.53	0.60	0.25	0.08	0.04	0.01	0.00	0.00			9.84
260 to 280	2.03	3.29	2.96	1.66	0.74	0.28	0.07	0.03	0.01	0.00				11.08
290 to 310	1.33	3.16	2.75	1.05	0.29	0.08	0.02	0.00	0.00					8.68
320 to 340	1.27	2.25	1.19	0.39	0.09	0.02	0.00	0.00	0.00					5.23
All directions	19.95	31.48	26.30	14.28	5.39	1.88	0.49	0.15	0.05	0.02	0.00			100.00

* for seasonal averages refer to CD-ROM

Table 2.44 Annual and seasonal* average percentage frequencies of hourly wind speed by temperature: Manchester (Ringway) (1983–2002)

Temperature / °C	Percentage frequency for stated range of wind speed / m.s^{-1}													
	0–2	2–4	4–6	6–8	8–10	10–12	12–14	14–16	16–18	18–20	20–22	22–24	24–26	All speeds
–12.0 to –10.1	0.00	0.00												0.00
–10.0 to –8.1	0.01	0.00												0.01
–8.0 to –6.1	0.04	0.01	0.01											0.06
–6.0 to –4.1	0.15	0.03	0.01	0.02	0.00									0.21
–4.0 to –2.1	0.53	0.16	0.04	0.02	0.01	0.01								0.77
–2.0 to –0.1	1.04	0.68	0.22	0.12	0.09	0.03	0.00							2.18
0.0 to 1.9	1.50	1.58	0.74	0.34	0.17	0.05	0.02	0.00	0.00					4.40
2.0 to 3.9	1.88	2.50	1.76	0.81	0.32	0.12	0.03	0.01	0.00					7.43
4.0 to 5.9	2.10	3.28	2.71	1.48	0.60	0.20	0.05	0.01						10.44
6.0 to 7.9	2.15	3.57	3.22	2.07	0.83	0.28	0.07	0.02	0.00					12.20
8.0 to 9.9	2.30	3.75	3.50	2.29	0.95	0.33	0.09	0.03	0.01	0.00	0.00			13.25
10.0 to 11.9	2.38	3.95	3.45	2.11	0.94	0.43	0.15	0.06	0.03	0.01	0.00			13.51
12.0 to 13.9	2.12	4.06	3.45	1.79	0.63	0.22	0.06	0.02	0.00	0.00	0.00			12.35
14.0 to 15.9	1.64	3.18	2.93	1.44	0.41	0.12	0.02	0.00	0.00	0.00				9.74
16.0 to 17.9	1.01	2.07	2.12	0.95	0.23	0.04	0.00	0.00						6.41
18.0 to 19.9	0.54	1.26	1.06	0.42	0.11	0.02	0.01							3.40
20.0 to 21.9	0.29	0.69	0.58	0.21	0.05	0.01	0.00							1.84
22.0 to 23.9	0.18	0.41	0.27	0.10	0.02	0.00								0.99
24.0 to 25.9	0.07	0.20	0.12	0.06	0.01	0.00								0.48
26.0 to 27.9	0.03	0.07	0.08	0.03	0.01	0.00								0.23
28.0 to 29.9	0.01	0.02	0.02	0.02	0.00									0.08
30.0 to 31.9	0.00	0.01	0.01	0.00										0.01
32.0 to 33.9		0.00	0.00											0.01
All temps.	19.95	31.48	26.30	14.28	5.39	1.88	0.49	0.15	0.05	0.02	0.00			100.00

* for seasonal averages refer to CD-ROM

Table 2.45 Annual and seasonal* average percentage frequencies of hourly wind speed by direction: Edinburgh (Turnhouse) (1983–2002)

Direction / deg.	Percentage frequency for stated range of wind speed / m.s^{-1}													
	0–2	2–4	4–6	6–8	8–10	10–12	12–14	14–16	16–18	18–20	20–22	22–24	24–26	All speeds
Calm/variable	0.70	0.00												0.70
350 to 010	1.28	0.68	0.43	0.26	0.09	0.02	0.00							2.76
020 to 040	2.31	2.75	1.75	0.67	0.19	0.04	0.02	0.00						7.74
050 to 070	1.80	3.73	3.61	1.81	0.57	0.15	0.04	0.00						11.71
080 to 100	1.13	1.61	1.54	0.85	0.28	0.07	0.02	0.00						5.50
110 to 130	0.93	0.79	0.59	0.29	0.07	0.01								2.68
140 to 160	0.85	0.63	0.41	0.17	0.04	0.01	0.01	0.00						2.12
170 to 190	1.05	0.85	0.78	0.44	0.19	0.08	0.02	0.01	0.00					3.44
200 to 220	2.04	2.63	3.52	2.84	1.40	0.52	0.18	0.04	0.01					13.18
230 to 250	4.58	7.77	8.34	7.00	3.26	1.20	0.39	0.11	0.02	0.01	0.00	0.00		32.68
260 to 280	2.81	2.32	2.24	2.08	0.99	0.40	0.13	0.03	0.01	0.00				11.01
290 to 310	1.50	0.96	0.57	0.27	0.06	0.03	0.00	0.00						3.39
320 to 340	1.28	0.80	0.65	0.29	0.07	0.01	0.00							3.11
All directions	22.24	25.54	24.42	16.97	7.22	2.54	0.82	0.20	0.04	0.01	0.00	0.00		100.00

* for seasonal averages refer to CD-ROM

Table 2.46 Annual and seasonal* average percentage frequencies of hourly wind speed by temperature: Edinburgh (Turnhouse) (1983–2002)

Temperature / °C	Percentage frequency for stated range of wind speed / m.s^{-1}													
	0–2	2–4	4–6	6–8	8–10	10–12	12–14	14–16	16–18	18–20	20–22	22–24	24–26	All speeds
–16.0 to –14.1	0.00													0.00
–14.0 to –12.1	0.01													0.01
–12.0 to –10.1	0.02													0.02
–10.0 to –8.1	0.07	0.01												0.07
–8.0 to –6.1	0.16	0.01												0.17
–6.0 to –4.1	0.38	0.04	0.01	0.00	0.00	0.00								0.44
–4.0 to –2.1	0.75	0.17	0.02	0.00	0.01	0.00	0.00							0.95
–2.0 to –0.1	1.61	0.64	0.21	0.09	0.03	0.01	0.00	0.00						2.60
0.0 to 1.9	2.34	1.57	1.10	0.60	0.23	0.07	0.02	0.00						5.92
2.0 to 3.9	2.51	2.28	1.99	1.51	0.67	0.21	0.07	0.01	0.00					9.26
4.0 to 5.9	2.68	2.74	2.66	1.81	0.86	0.33	0.11	0.03	0.01	0.01	0.00			11.24
6.0 to 7.9	2.52	3.14	3.18	2.30	1.05	0.41	0.12	0.03	0.01	0.00	0.00			12.76
8.0 to 9.9	2.57	3.38	3.53	2.59	1.17	0.50	0.16	0.04	0.01	0.00		0.00		13.96
10.0 to 11.9	2.51	3.49	3.40	2.36	1.17	0.48	0.21	0.06	0.01					13.68
12.0 to 13.9	2.02	3.31	3.07	2.16	0.84	0.27	0.09	0.02	0.00					11.78
14.0 to 15.9	1.21	2.26	2.36	1.61	0.64	0.14	0.03	0.00						8.26
16.0 to 17.9	0.57	1.33	1.57	1.13	0.36	0.09	0.01	0.00						5.07
18.0 to 19.9	0.20	0.65	0.76	0.54	0.13	0.03	0.01							2.32
20.0 to 21.9	0.08	0.31	0.35	0.16	0.04	0.00								0.94
22.0 to 23.9	0.03	0.14	0.14	0.05	0.00	0.00								0.37
24.0 to 25.9	0.01	0.05	0.04	0.03	0.00									0.13
26.0 to 27.9	0.00	0.01	0.01	0.01	0.00									0.03
28.0 to 29.9		0.00	0.01	0.00	0.00									0.01
All temps.	22.24	25.54	24.42	16.97	7.22	2.54	0.82	0.20	0.04	0.01	0.00	0.00		100.00

* for seasonal averages refer to CD-ROM

Table 2.47 Annual and seasonal average percentage frequencies of hourly wind speed by direction: Belfast (Aldergrove) (1983–2002) — refer to CD-ROM

Table 2.48 Annual and seasonal average percentage frequencies of hourly wind speed by temperature: Belfast (Aldergrove) (1983–2002) — refer to CD-ROM

Table 2.49 Annual and seasonal average percentage frequencies of hourly wind speed by direction: Birmingham (Elmdon) (1982–2002) — refer to CD-ROM

Table 2.50 Annual and seasonal average percentage frequencies of hourly wind speed by temperature: Birmingham (Elmdon) (1982–2002) — refer to CD-ROM

Table 2.51 Annual and seasonal average percentage frequencies of hourly wind speed by direction: Cardiff (Rhoose/St Athan) (1982–2002) — refer to CD-ROM

Table 2.52 Annual and seasonal average percentage frequencies of hourly wind speed by temperature: Cardiff (Rhoose/St Athan) (1982–2002) — refer to CD-ROM

Table 2.53 Annual and seasonal average percentage frequencies of hourly wind speed by direction: Glasgow (Abbotsinch) (1983–2002) — refer to CD-ROM

Table 2.54 Annual and seasonal average percentage frequencies of hourly wind speed by temperature: Glasgow (Abbotsinch) (1983–2002) — refer to CD-ROM

Table 2.55 Annual and seasonal average percentage frequencies of hourly wind speed by direction: Plymouth (Plymouth Weather Centre) (1982–2002) — refer to CD-ROM

Table 2.56 Annual and seasonal average percentage frequencies of hourly wind speed by temperature: Plymouth (Plymouth Weather Centre) (1982–2002) — refer to CD-ROM

Table 2.57 Terrain categories and related parameters

Terrain category	Description	K_R	z_o / m	z_{min} / m
I	Rough open sea; lake shore with at least 5 km fetch up-wind and smooth flat country without obstacles	0.17	0.01	2
II	Farm land with boundary hedges, occasional small farm structures, houses or trees	0.19	0.05	4
III	Suburban or industrial areas and permanent forests	0.22	0.3	8
IV	Urban areas in which at least 15% of surface is covered with buildings of average height exceeding 15 m	0.24	1	16

Table 2.58 Effective length of upwind slope, L_e

Slope ($\Phi = H / L_u$)	L_e value
Shallow ($0.05 < \Phi < 0.3$)	L_u
Steep ($\Phi > 0.3$)	$H / 0.3$

2.8.5.3 Topography coefficient

The topography coefficient accounts for the increase in mean wind speed over isolated hills and escarpments (not undulating and mountainous regions) and is related to the wind speed upwind to the hill. It should be taken into account for locations that are:

— more than halfway up the slope of the hill

— within 1.5 times the height of the cliff from the base of the cliff.

It is defined as follows:

$$C_T = 1 \text{ for } \Phi < 0.05 \quad (2.6)$$

$$C_T = (1 + 2\,s\,\Phi) \text{ for } 0.05 < \Phi < 0.3 \quad (2.7)$$

$$C_T = (1 + 0.6\,s) \text{ for } \Phi > 0.3 \quad (2.8)$$

where C_T is the topography coefficient, s is a slope factor (obtained from Figures 2.14 and 2.15) and Φ is the upwind slope in the wind direction (H / L_u).

The slope factor s must be scaled to the length of the upwind or downwind slopes, see Figure 2.16, where L_u is the actual length of the upwind slope in the wind direction (m), L_d is the actual length of the downwind slope (m), H is the effective height of the feature (m), Φ is the upwind slope (H / L_u) in the wind direction, L_e is the effective length of the upwind slope (m), as defined in Table 2.58, x is the

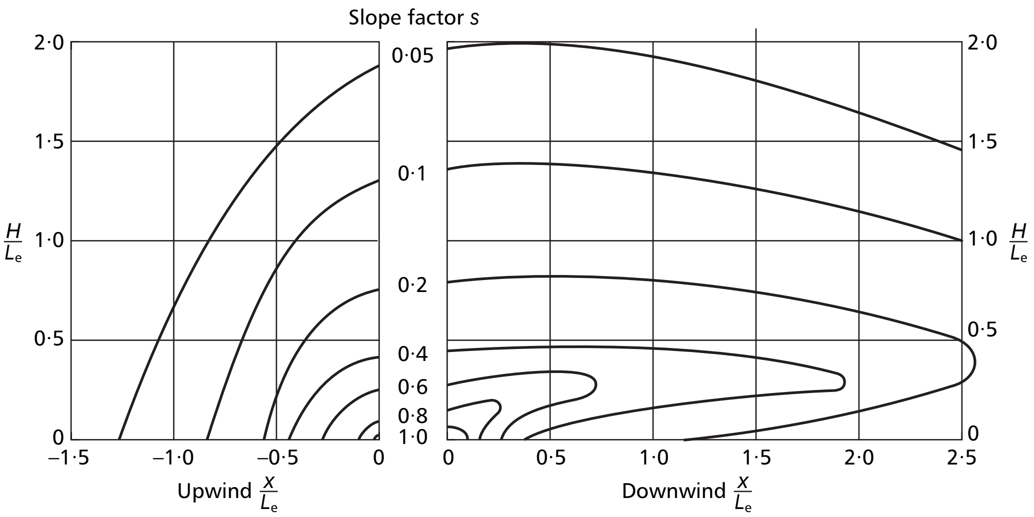


Figure 2.14 Slope factor s for cliffs and escarpments; see Table 2.58 for values of L_e

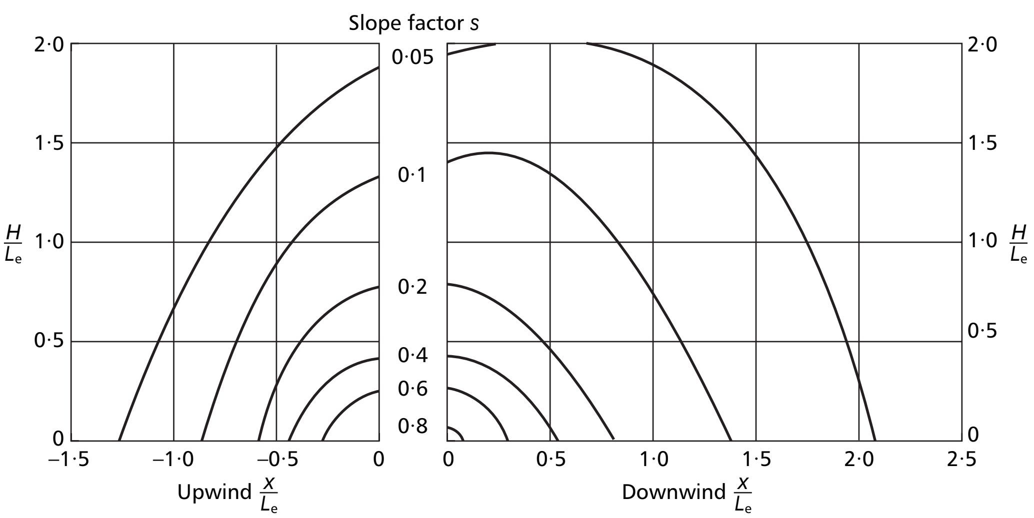


Figure 2.15 Slope factor s for hills and ridges; see Table 2.58 for values of L_e

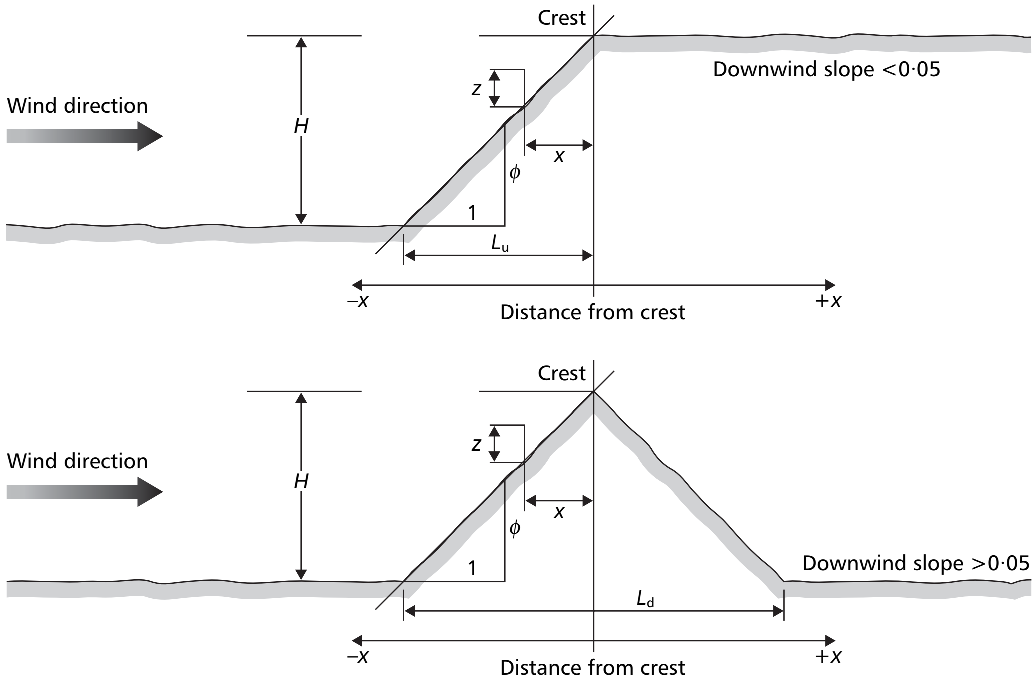


Figure 2.16 Definition of topographic features

horizontal distance of the site from the top of the crest (m) and z is the vertical distance from the ground level of the site (m).

Example 2.2

It is required to estimate the wind at 15 metres above ground in a suburban area, three quarters of the way up the slope of a hill that rises 120 m over 1 km in the direction of the prevailing wind, and is then flat. The mean wind measured at an adjacent airfield site is 4.5 m·s^{-1}.

The roughness coefficient is given by equation 2.5, using the values given in Table 2.57 for terrain category III, i.e:

$C_R(15) = 0.22 \log_e (15 / 0.3) = 0.861$

From Figure 2.16, the parameters that define the topography coefficient are as follows:

— $L_u = 1000$ m
— $L_d = \infty$
— $H = 120$ m
— $\Phi = 120 / 1000 = 0.12$
— $L_e = L_u = 1000$ (see Table 2.58)
— $H / L_e = 120 / 1000 = 0.12$
— $x = -250$ m
— $x / L_e = -0.25$

Hence, from Figure 2.14, $s = 0.6$.

Therefore, from equation 2.7, the topography coefficient is:

$$C_T = 1 + (2 \times 0.6 \times 0.12) = 1.144$$

From equation 2.4, the desired wind speed is:

$$v_z = 4.5 \times 0.861 \times 1.144 = 4.43 \text{ m·s}^{-1}$$

2.9 Climate change

There is ample evidence from the geological and historical record that the climate of the earth, and the UK in particular, has changed significantly in the past[20]. Since the major climate amelioration at the ending of the last Ice Age about 10 000 years ago, there have been a number of relatively minor, but still significant, fluctuations such as the warm period during Roman times and the colder period between the seventeenth and nineteenth centuries. Over recent decades, however, it has become clear that there is a global trend towards warmer temperatures. These meteorological observations are backed up by an increasing range of evidence for regional warming from various geophysical (e.g. glacier retreat) and biological (e.g. longer growing seasons) indicators. After considerable scientific research and debate, there is now strong evidence that this trend is associated with the rising concentration of 'greenhouse gases' due to production from energy generation, transport and industrial processes and reduction in sinks due to deforestation. Current thinking on causes and impacts is summarised in the latest report from the Intergovernmental Panel on Climate Change (IPCC)[21] and the UK Climate Impacts Programme (UKCIP)[22,23]. A recent report from the Foundation for the Built Environment (FBE)[24] and recent CIBSE publications[25,26] consider the impacts on buildings and services.

The Tyndall Centre for Climate Change Research (www.tyndall.ac.uk) is the UK's current national centre for inter-disciplinary research into sustainable responses to climate change. Although the Centre has a global reach, its four research themes have a strong UK focus.

The UK Climate Impacts Programme (www.ukcip.org.uk) helps UK organisations assess how they might be affected by climate change, so they can prepare for its impact.

Most of the climate modelling used by UKCIP to produce scenarios has been done by the Hadley Centre, which is part of the UK Met Office. (www.metoffice.com/research/hadleycentre/).

Development of future climate scenarios depends on:

(*a*) making assumptions about the emissions of CO_2 and other greenhouse gasses over the next century; the IPCC has published a range of standard scenarios[27], which depend on complex economic, technological and demographic factors

(*b*) calculating the relationships between greenhouse gas emissions and greenhouse gas concentrations in the atmosphere

(*c*) which particular global climate model is used to determine the geographic variation of changes; one of the leading models, widely used for UK climate scenarios, is the global Hadley Climate Model 3 (HadCM3); for regional scenarios, the Regional Model (HadRM3) has been used, relying on outputs from larger-scale models for its boundary conditions.

In early 2002, the UKCIP published four possible future climate scenarios[22], produced within this framework. These were felt to cover a large, but not exhaustive, part of the range of possible future changes. The change in global mean temperature (ΔT) and future CO_2 concentration are summarised for each scenario during three 30-year time periods centred on the 2020s (2011 to 2040), the 2050s (2041 to 2070), and the 2080s (2071 to 2100) in Table 2.59.

These future climate scenarios are not comprehensive predictions of future climate, but plausible estimates of future changes spanning a reasonable range of outcomes. In the present state of knowledge, it is possible to assign the following hierarchy of confidence to the different variables, see Figure 2.17.

Climate variable — **Level of confidence**

Atmospheric CO_2 concentration — High
Global mean sea level
Global mean temperature
Regional seasonal temperature
Regional temperature extremes
Regional seasonal precipitation
Regional cloud cover
Regional potential evapotranspiration
Changes in wind speed
Changes in climatic variability — Low

Figure 2.17 Hierarchy of confidence for climate variables

2.9.1 Predicted future trends

The UKCIP02 scenarios suggest that the following changes are likely for the UK[23].

2.9.1.1 Temperature

The UK climate will become warmer. By the 2080s the annual average UK temperature may rise by between 2 °C and 3.5 °C, with greater warming in the south east than the north west and greater in the summer and autumn than the winter and spring. High summer temperatures will become more frequent and very cold winters increasingly rare. A very hot August, such as 1995, which was 3.4 °C warmer than average, may occur one year in five by the 2050s and as often as three years in five by the 2080s.

2.9.1.2 Precipitation

Summers will become drier and winters wetter in all parts of the UK. The largest changes are expected in the south and east where summer precipitation may decrease by 50% and winter precipitation increase by 30% by the 2080s. Snowfall amounts will decrease significantly in all areas. Heavy winter precipitation events will become more frequent and more intense, increasing the risk of fluvial and flash flooding.

2.9.1.3 Sea level

Relative sea level will continue to rise around the UK shoreline, with faster rises in the south east than the north west because of continuing isostatic adjustments after the last ice age. This, together with more common storm surges, will increase the risk of coastal flooding.

2.9.1.4 Humidity

Vapour pressures will increase in summer and winter in all parts of the UK, however the rising temperatures mean that relative humidity will fall in all parts of the UK, except northern Scotland, with the highest falls likely in the summer in the south of England.

2.9.1.5 Solar radiation

Cloud cover will decrease in the summer, especially in the south, leading to increases in solar radiation totals. Changes in winter cloud cover are expected to be smaller with increases of only 2–3% likely over most of the country; this will lead to a correspondingly small fall in solar radiation levels. The peak levels of solar radiation are unlikely to change significantly.

Table 2.59 Summary of the characteristics of the four UKCIP02 scenarios[22]

Scenario	Climatic changes for stated period					
	2020s		2050s		2080s	
	$\Delta\theta$ / K	CO_2 / ppm	$\Delta\theta$ / K	CO_2 / ppm	$\Delta\theta$ / K	CO_2 / ppm
Low Emissions	0.79	422	1.41	489	2.00	525
Medium-Low Emissions	0.88	422	1.64	489	2.34	562
Medium-High Emissions	0.88	435	1.87	551	3.29	715
High Emissions	0.94	437	2.24	593	3.88	810

2.9.1.6 Wind speed

It is very difficult to predict future changes in wind speeds with the present climate models. Work is under way to improve the predictions, however at present the best estimate is that future wind speeds will be similar to the present.

2.9.1.7 Depressions

The number of depressions crossing the UK in the winter may increase as depression tracks shift south. This would mean an increase in the frequency of high winds, particularly in the south of England. These is however no evidence that the frequency of severe storms will change.

2.9.1.8 Soil moisture content

Soil moisture content is expect to fall sharply, especially in the summer in the south of England. This will lead to shrinking of some clays affecting the stability of buildings.

2.9.1.9 Effect on buildings and building services

The major impacts of these changes relevant to building services are summarised in Table 2.60.

There may also be more complex 'knock-on' effects from climate change impacts on transportation and water resources which are likely to affect both building and maintenance.

Climatic data customarily refer to a period assumed to be long enough to contain statistically dependable magnitudes and year-to-year variabilities, and recent enough to be considered representative of the current climate. Most of the data in the Guide are based on the latest available 20-year period, 1983–2002. This traditional use of climatic data in most fields provides a sound basis for engineering decisions provided the climate remains stable between the historic, predictor period and the future, predicted period. However, most climate change models indicate significant climate changes that need to be considered by designers. CIBSE TM34[25] provides UK design data for manual calculations for future decades, based on UKCIP scenarios. These correspond to the values given in this Guide and CIBSE Guide J[1] for the historical climate and may be used as an alternative for design, with the caveat that they are only predictions.

2.9.2 Climate change data

Tables 2.61 and 2.62 show 1-day near-extreme data selected on daily mean dry bulb temperature (1% exceedence (summer) and 99% exceedence (winter)) for London (Heathrow). Figures 2.18 and 2.19 show these data in graphical form.

Tables 2.63 to 2.66 and Figures 2.20 to 2.23 (refer to CD-ROM that accompanies this Guide) show similar data for Manchester (Ringway) and Edinburgh (Turnhouse). These data come from results of the Hadley Regional Model (HadRM3)[23] on future climate. Data for the 1970s are from HadRM3 (three runs) for the period 1960–1990.

Data for 2080s under the Medium-High (also referred to as the A2) scenario are from HadRM3 (three runs) for the period 2070–2100. For all other data, i.e. those for other time periods and scenarios, pattern scaling factors from *Climate Change Scenarios for the United Kingdom: The UKCIP02 Briefing Report*[23] were used.

The percentage exceedence values are calculated using daily average temperatures.

Using sinusoidal fits, hourly temperature profiles are produced from the average, minimum and maximum daily temperatures given in the Hadley data[28]. The 99% exceedence profiles are disjointed because of the fitting to both the mean and the extremes.

The data from HadRM3 is for 50 km by 50 km grid boxes. The boxes containing the referenced sites (Heathrow, Manchester and Edinburgh) were selected. However, the data are representative of that 2500 km^2 box and not just the single point site. As the data are representative of the 2500 km^2 boxes they should be used in comparisons only; i.e. 1970s compared to 2020s, 2050s and 2080s. It would also be misleading to use the data for comparison with site data observations.

Table 2.60 Major impacts of climate change on building services

Climate change	Impact	Consequences for building services
Rising summer temps.	Internal temps./comfort	Mean internal temperatures will increase causing problems with overheating in the summer; careful design will be needed to avoid use of air conditioning.
	Internal pollution	Higher temperatures could increase problems of outgassing of pollutants from the structure and furnishing of buildings and the ground, adversely affecting indoor air quality.
	Legionella risk	Warmer and more humid internal environments and use of air conditioning could lead to more *legionella* problems, especially poorly maintained domestic systems.
Rising winter temps.	Energy consumption	Winter energy consumption for heating will reduce but this reduction may be offset by increased use of air conditioning in the summer.
Rising vapour pressures	Condensation and mould growth	Condensation and mould growth may increase unless temperature rise outweighs this effect.
Decrease in summer rainfall	Limited water supply and drying ground	Limited water supply in some areas and disruption from increased subsidence due to drier clay soils.
More intense winter rainfall and rising sea level		Increased flooding, both from rivers due to more frequent heavy rain and in coastal areas due to sea level rise and more frequent storm surges.

Table 2.61 Climate change dry bulb temperatures (1% exceedence) for London (Heathrow)

Hour	Dry bulb temperature (/ °C) for stated scenario*					
	1970s	2020s (A2 and B2)	2050s (B2)	2050s (A2)	2080s (B2)	2080s (A2)
0	18.2	19.6	20.8	21.1	21.8	23.3
1	18.7	19.9	21.0	21.3	21.9	23.3
2	18.2	19.4	20.4	20.7	21.3	22.6
3	16.8	18.0	19.1	19.4	20.0	21.3
4	17.2	18.4	19.5	19.8	20.4	21.7
5	18.3	19.5	20.6	21.0	21.6	23.0
6	19.9	21.2	22.4	22.7	23.5	24.9
7	21.8	23.3	24.6	24.9	25.7	27.4
8	23.3	25.0	26.6	27.0	27.9	29.8
9	24.5	26.4	28.1	28.6	29.7	31.8
10	25.5	27.7	29.6	30.1	31.3	33.6
11	26.5	28.8	30.8	31.4	32.6	35.1
12	27.1	29.6	31.7	32.4	33.6	36.3
13	27.6	30.1	32.3	33.0	34.3	37.0
14	27.7	30.3	32.5	33.2	34.5	37.3
15	27.7	30.3	32.4	33.1	34.4	37.2
16	27.7	30.1	32.3	32.9	34.2	36.9
17	27.4	29.8	31.8	32.4	33.6	36.2
18	26.8	29.1	31.0	31.5	32.7	35.1
19	25.8	27.9	29.7	30.2	31.3	33.5
20	24.4	26.3	28.0	28.5	29.5	31.5
21	22.7	24.4	26.0	26.4	27.3	29.3
22	20.9	22.5	23.9	24.3	25.2	27.0
23	19.3	20.8	22.1	22.5	23.3	24.9
24	18.2	19.6	20.8	21.1	21.8	23.3

* A1 = High emissions; A2 = Medium-High emissions; B1 = Low emissions; B2 = Medium-Low emissions

Table 2.62 Climate change dry bulb temperatures (99% exceedence) for London (Heathrow)

Hour	Dry bulb temperature (/ °C) for stated scenario*					
	1970s	2020s (A2 and B2)	2050s (B2)	2050s (A2)	2080s (B2)	2080s (A2)
0	–1.6	–0.8	–0.2	0.0	0.4	1.2
1	–2.7	–1.8	–1.0	–0.8	–0.4	0.6
2	–4.2	–3.2	–2.3	–2.0	–1.4	–0.2
3	–4.2	–3.2	–2.4	–2.1	–1.6	–0.6
4	–4.2	–3.2	–2.4	–2.1	–1.6	–0.6
5	–4.2	–3.2	–2.4	–2.1	–1.6	–0.6
6	–4.2	–3.2	–2.4	–2.1	–1.6	–0.6
7	–4.2	–3.2	–2.4	–2.1	–1.6	–0.6
8	–4.1	–3.1	–2.3	–2.0	–1.5	–0.5
9	–3.8	–2.8	–2.0	–1.7	–1.2	–0.1
10	–3.3	–2.3	–1.5	–1.2	–0.7	0.4
11	–2.7	–1.7	–0.9	–0.6	–0.1	1.0
12	–1.9	–1.0	–0.1	0.1	0.6	1.7
13	–1.1	–0.2	0.6	1.2	2.1	3.5
14	3.0	3.4	3.7	3.8	4.0	4.3
15	2.7	3.1	3.5	3.6	3.8	4.2
16	1.9	2.4	2.9	3.0	3.3	3.8
17	0.8	1.5	2.0	2.2	2.6	3.3
18	–0.2	0.6	1.3	1.5	1.9	2.7
19	–1.0	–0.1	0.7	0.9	1.4	2.3
20	–1.3	–0.4	0.4	0.6	1.1	2.0
21	–1.3	–0.4	0.3	0.5	1.0	1.9
22	–1.1	–0.4	0.3	0.5	0.9	1.7
23	–1.2	–0.4	0.2	0.4	0.8	1.6
24	–1.6	–0.8	–0.2	0.0	0.4	1.2

* A1 = High emissions; A2 = Medium-High emissions; B1 = Low emissions; B2 = Medium-Low emissions

Table 2.63 Climate change dry bulb temperatures (1% exceedence) for Manchester (Ringway) — refer to CD-ROM

Table 2.64 Climate change dry bulb temperatures (99% exceedence) for Manchester (Ringway) — refer to CD-ROM

Table 2.65 Climate change dry bulb temperatures (1% exceedence) for Edinburgh (Turnhouse) — refer to CD-ROM

Table 2.66 Climate change dry bulb temperatures (99% exceedence) for Edinburgh (Turnhouse) — refer to CD-ROM

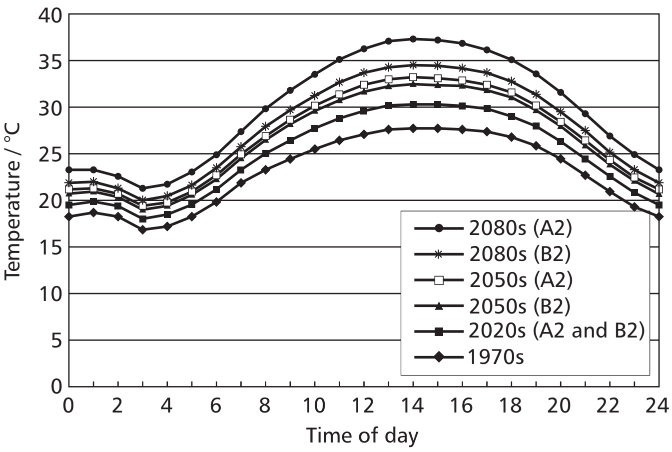


Figure 2.18 Climate change dry bulb temperatures (1% exceedence) for London (Heathrow)

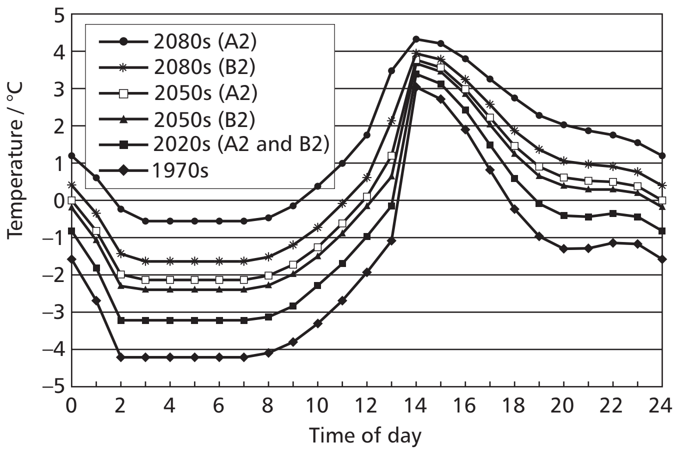


Figure 2.19 Climate change dry bulb temperatures (99% exceedence) for London (Heathrow)

Figure 2.20 Climate change dry bulb temperatures (1% exceedence) for Manchester (Ringway) — refer to CD-ROM

Figure 2.21 Climate change dry bulb temperatures (99% exceedence) for Manchester (Ringway) — refer to CD-ROM

Figure 2.22 Climate change dry bulb temperatures (1% exceedence) for Edinburgh (Turnhouse) — refer to CD-ROM

Figure 2.23 Climate change dry bulb temperatures (99% exceedence) for Edinburgh (Turnhouse) — refer to CD-ROM

2.9.3 Climate prediction uncertainty and the Hadley Model

Predictions of future climate change depend not only on estimating the range of changes in future greenhouse gas emissions, but also importantly on which climate model is used. Predictions of climate over the next 40 or so years are largely insensitive to the choice of emissions scenario, but are sensitive to the choice of model. Climate predictions for the latter part of this century depend both on the choice of emissions scenario and on the choice of model.

The data presented in this Guide all derive from climate models developed and run in the Hadley Centre in the UK. While this is a relatively good climate model and generates climate information at relatively high (50 km) resolution, we do not know a priori whether it is more believable than climate models from other countries or centres. It is therefore important to be at least aware of the range of results from other climate models. One simple measure that summarises model performance is called the 'climate sensitivity'. This is a measure of how sensitive a particular model is to rising concentrations of greenhouse gases or, more formally, 'the equilibrium rise in global surface air temperature for a doubling of atmospheric greenhouse gas concentration'. The true value of climate sensitivity is not known but the IPCC Third Assessment Report[23] suggested that its likely range is 1.5–4.5 K. The Hadley Centre global model has a sensitivity of about 3 K, and so falls roughly in the middle of this range.

Figure 2.24 shows another way of presenting this information[27,29]. The points on the *x*-axis show the climate sensitivity of the eight leading climate models used in the IPCC report[23], with the Hadley Model shown as the vertical line at 3 K. Clearly, if this Guide had adopted a model with a higher sensitivity, larger changes in climate would have resulted; conversely, a model with a lower sensitivity would have resulted in smaller changes.

Additionally, the curve shows one estimate of the probability density function (PDF) of the climate sensitivity derived from observations alone. Again, the Hadley Model falls roughly in the middle of this PDF.

2.9.4 The future of climate prediction

Over the next few years it will become possible to develop probability-based predictions of future climate rather than the discrete 'deterministic' predictions that exist at present. These probabilistic predictions will be based on sampling and using a much larger group of climate models, thus reducing the need for making subjective or political judgements about which single model one should use.

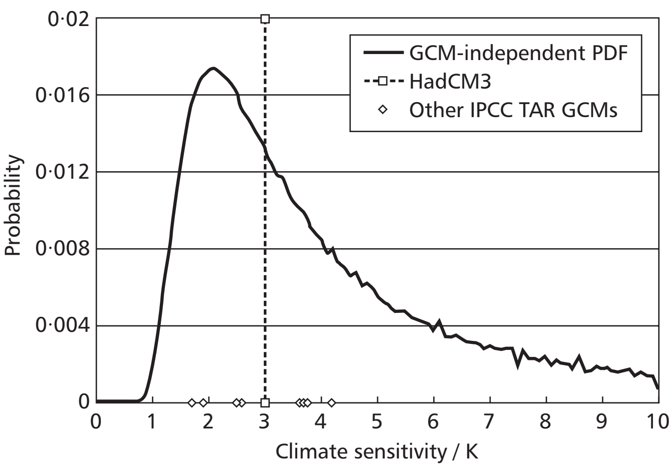


Figure 2.24 Climate sensitivity of models in terms of temperature[27,29]

On the other hand, the uncertainty about future emissions of greenhouse gases will remain, and so subjective choices about which emissions scenario to use will also remain. Thus by the time that the next IPCC Assessment is published in 2007, it is likely that for any chosen emissions scenario it will be possible to make statements such as: 'For the A2 emissions scenario, the probability of the maximum temperature in southern England by the 2020s exceeding 40 °C in any given year is 0.03.'

This hypothetical example is further illustrated in Figure 2.25[30], which shows the cumulative probability of the maximum air temperature recorded in any given year in the 2020s in southern England for four different emissions scenarios. This result is not dependent on any single model, but the probabilities are objectively derived. In a few years time, this type of information will be available for many different climatic variables and regions.

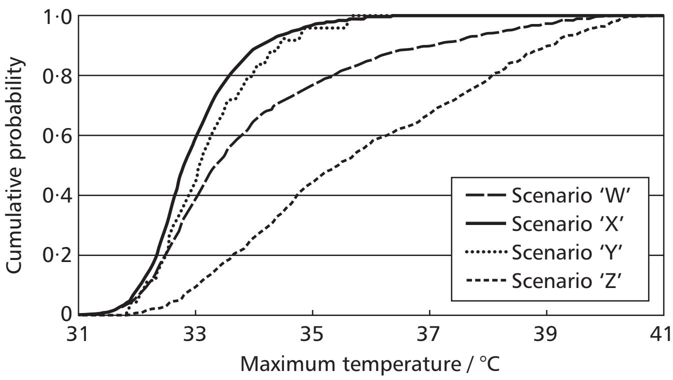


Figure 2.25 Hypothetical example of probability-based temperature prediction (southern England, 2020s)[30]

2.10 Heat island effect

The average air temperature in urban areas is generally higher than that in surrounding areas. This results from a multiplicity of differences between urban and rural areas, but the more important reasons are that urban areas have:

— greater heat capacity

— more effective absorption of solar radiation in 'street gorges' and less effective long-wave radiative cooling

— reduced wind speeds

— less vegetation.

Anthropogenic heat flux can also be important in some areas.

2.10.1 London heat island

In London, the maximum heat island reaches 8 K more than the rural area[31], see Figure 2.26, although such intense heat islands are relatively uncommon, see Figure 2.27. Such maximum values almost invariably occur at

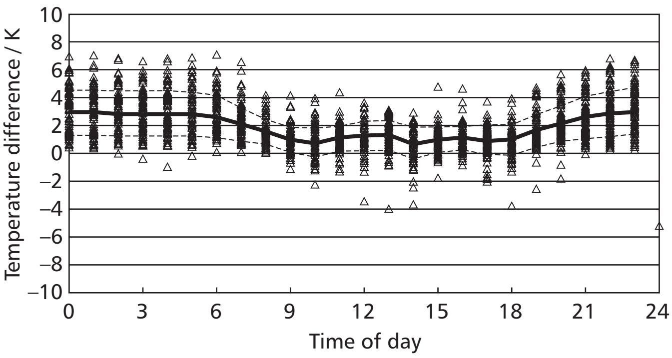


Figure 2.26 Heat island effect for London; summer 2000

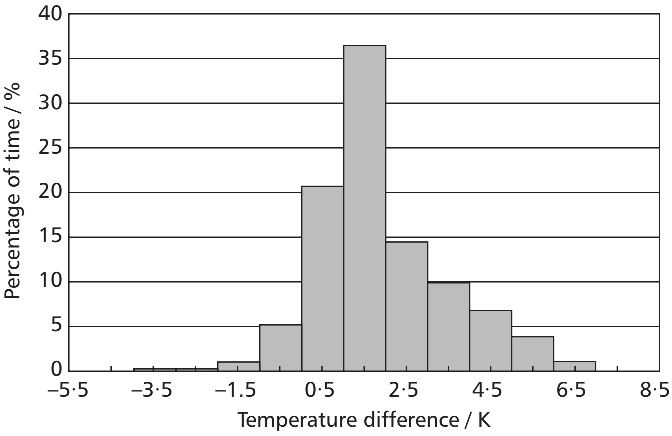


Figure 2.27 Frequency of occurrence of heat island intensities for London; summer 2000

night-time because of the heat storage effects of the urban fabric, and the reduced radiative cooling and air speed compared to rural areas. Higher night-time temperatures reduce the potential for passive night-time cooling.

The heat island in London has been measured in considerable detail in the 1960s(32) using Stephenson screens and more recently using a large array of stations measuring hourly data at a slightly higher level of 6 m(31). These studies showed that the location of the warmest place in the city moves many kilometres in line with the wind direction.

In a study of the London heat island, it was shown that the total annual load for heating and cooling of a standard air conditioned office building rose towards the centre of the city to 8.5% more than at a rural location, and then reduced at the most over-shadowed sites(33).

In sizing heating and cooling plant, the effect of any heat island can be allowed for but there is no precise method currently available. The reason for this is that the air temperature at a point in a city depends on two main factors: (*a*) the location's relative position between the urban centre and the rural surroundings (its radial distance) and (*b*) its immediate micro-environment. There is a good correlation between radial distance of a site from an urban centre and its air temperature; as the radial distance decreases the temperature rises. Overlaid upon this general trend is the effect of the local surroundings, e.g. proximity to a park will result in lower temperatures, and streets with darker surfaces will be warmer. There are many other factors that affect local temperature and this makes allowing for micro-environment effects highly complex.

Of the two main factors, allowing for radial distance is much the simpler. Where meteorological temperature data are available for the centre of a city and a rural location, the difference in mean temperatures (urban centre to rural) can be proportionally added to the rural temperature to approximate the air temperature at any point in the city. This can be done for night-time data or daytime data. The urban effects of over-shadowing, thereby reducing direct solar gain and cooling load, and the change in wind speed, should be allowed for separately.

In London, hourly air temperature data exist for 80 locations arranged on a radial grid pattern(34).

Guidance on reducing the heat island effect in UK cities is available elsewhere(34).

2.10.2 Seasonal variation in heat island effect

The intensity of the urban heat island effect is often greater in the summer than in the winter because of the greater solar energy input and lower wind speeds, and urban surfaces remaining dry (since there is less rainfall) for more of the time than in winter. However, this is not always true as much depends on a city's seasonal weather pattern, and the anthropogenic heat flux which is more important in the winter when solar input is lower. In London, the mean monthly heat island intensity increases by about 30% from 1.5 K in the winter to 2 K in the summer.

The intensity of a heat island increases as the wind speed drops. Figures 2.28 and 2.29 show the effect of wind speed on heat island intensity in London. The contour map covers Greater London (15–20 km radius) The distribution on the windy day is less pronounced (smaller gradients), and stretched out in line with the wind direction. The shift in thermal centre with wind direction can also be seen.

2.10.3 Heat island adjustments for London

The effect of the heat island on peak cooling loads can be allowed for by appropriately increasing the external temperatures used in the calculations or simulation. This can be done for air and sol-air temperature data.

Indicative values for correction in London based on the radial distance from the centre (taken here as the British Museum) are given in Table 2.67. These can be applied to design data used for the summertime (July and August). Six days with the highest global solar radiation for the months July and August 1999 and 2000 were selected to generate annular design data adjustments to use with Heathrow data. The radii used for the annular 'break points' were chosen according to the approximate position where a notable change occurred. It must be remembered that these mean data present an indicative picture in a simple form.

The indicated temperature difference should be added to the London area temperatures in Tables 2.34(g) and 2.34(h).

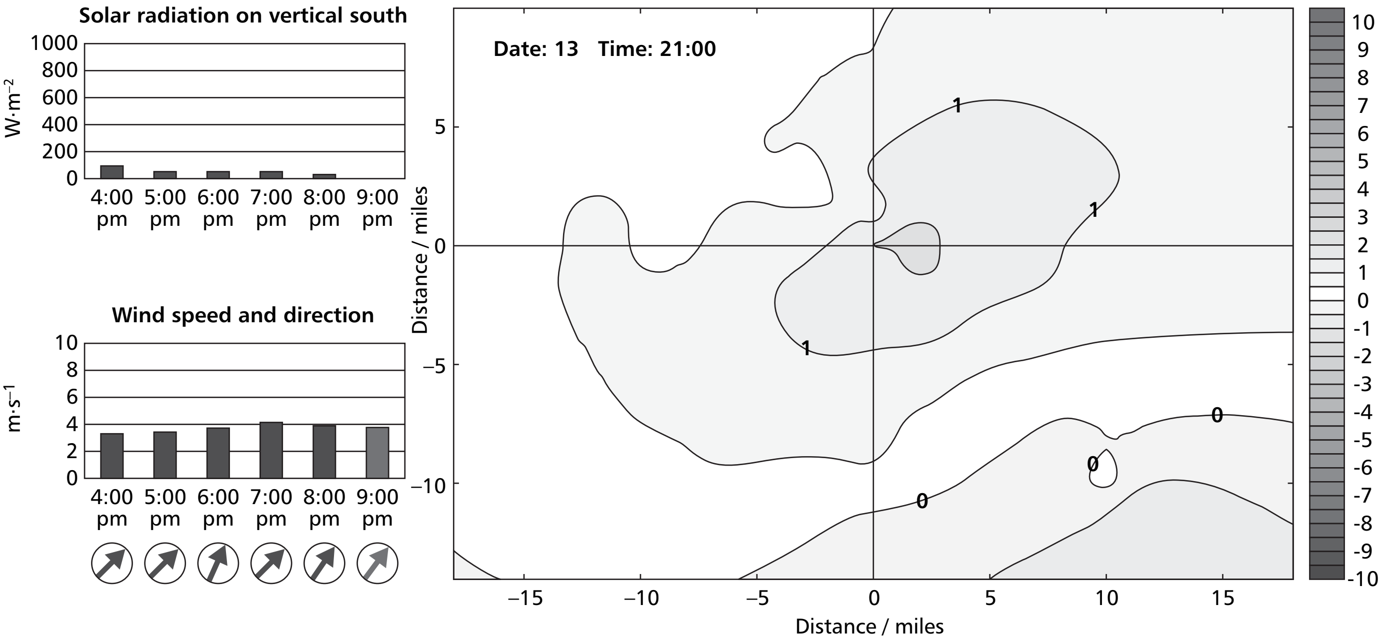


Figure 2.28 Heat island intensity distribution on a windy night; London, 13 August 1999

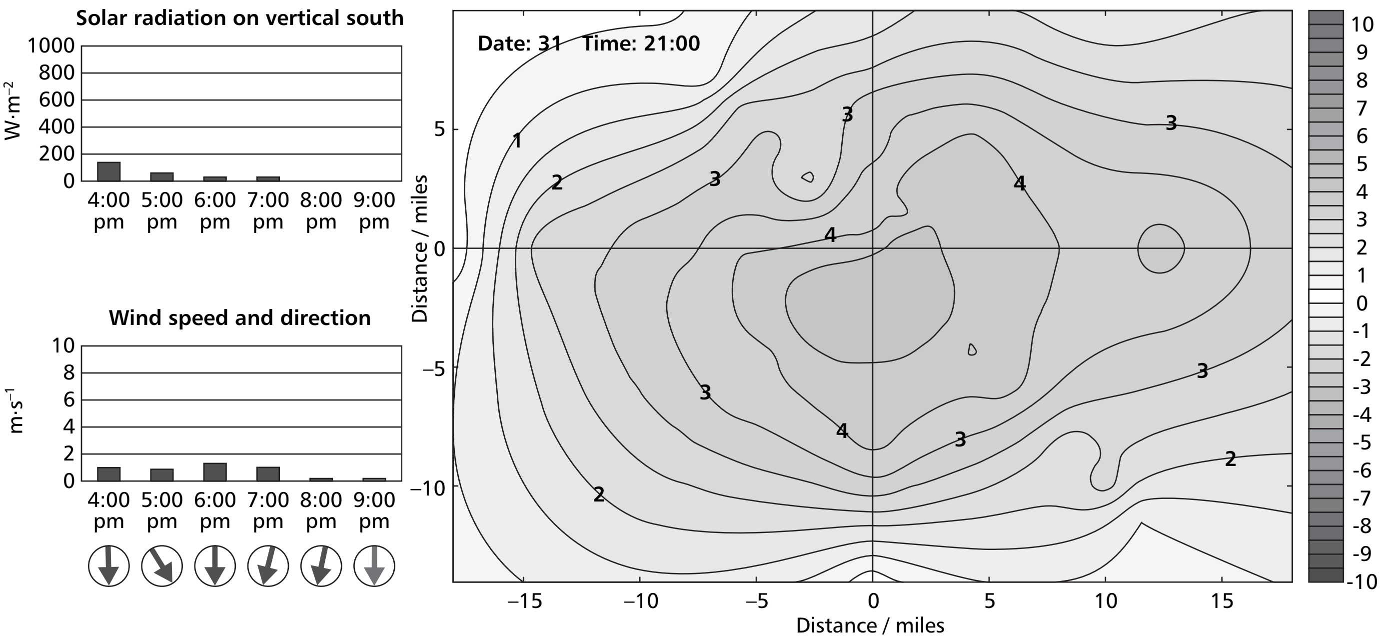


Figure 2.29 Heat island intensity distribution on a relatively calm night; London, 31 August 1999

Table 2.67 Summer temperature differences (compared to Heathrow) for annular regions centred on the British Museum

Time (GMT)	Mean temperature difference (/ K) with respect to the London area at stated distance from British Museum			Time (GMT)	Mean temperature difference (/ K) with respect to the London area at stated distance from British Museum		
	0–3 km	3–10 km	10–23 km		0–3 km	3–10 km	10–23 km
00:00	1.9	1.0	–0.9	12:00	1.7	2.1	1.3
01:00	1.9	1.1	–0.9	13:00	1.6	2.2	1.4
02:00	1.9	1.1	–0.7	14:00	0.6	1.1	0.4
03:00	1.7	0.9	–0.7	15:00	0.9	1.4	0.5
04:00	1.5	0.8	–0.7	16:00	0.9	1.4	0.4
05:00	1.4	0.7	–0.6	17:00	0.7	1.0	0.2
06:00	1.7	1.3	–0.4	18:00	0.4	0.8	0.0
07:00	1.5	1.9	0.8	19:00	0.4	0.4	–0.4
08:00	1.7	2.0	1.3	20:00	0.6	0.3	–0.9
09:00	1.8	2.2	1.5	21:00	1.0	0.3	–1.4
10:00	1.6	2.0	1.4	22:00	1.5	0.6	–1.3
11:00	1.4	1.8	1.3	23:00	1.5	0.6	–1.3

Table 2.68 Effect of London heat island on degree-day totals

Radial distance from British Museum	Decimal fraction of Heathrow degree-days
0–3 km	0.91
3–10 km	0.97
10–23 km	0.99

For an indication of the effect of the London heat island on heating demand, the fractions of the degree-days (to base temperature of 15.5 °C) of central London to London (Heathrow) have been calculated for the whole year. These are given in Table 2.68.

References

1 *Weather, solar and illuminance data* CIBSE Guide J (London: Chartered Institution of Building Services Engineers) (2002)

2 *Fundamentals* ASHRAE Handbook (Atlanta GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers) (2005)

3 Muneer T *Solar radiation and daylight models* (Oxford: Elsevier) (2004)

4 Jamieson H Meteorological data and design temperatures *J. Inst. Heating Ventilating Eng.* **22** 465–495 (1955)

5 *Reference data* CIBSE Guide C (London: Chartered Institution of Building Services Engineers) (2001)

6 *Degree-days: a practical introduction* Good Practice Guide GPG310 (The Carbon Trust) (2002) (www.thecarbontrust.co.uk)

7 *Surface airways meteorological and solar observing network (SAMSON) data set* (Ashville NC: National Climatic Data Centre) (1991)

8 *Principal station data PSD 1 to 134* (Downsview, Ontario: Atmospheric Environment Service) (1983–1987)

9 Lott J N, Baldwin R and Jones P *The FCC Integrated Surface Hourly Database — a new resource of global climate data* Report 2001-01 (Ashville NC: National Climatic Data Centre) (2001)

10 *Data documentation for data set 3505 (DSI-3505) — integrated surface hourly (ISH) data* (Ashville NC: National Climatic Data Centre) (2003)

11 *Updating the climatic design conditions in the ASHRAE Handbook of Fundamentals* Research Report RP-1273 (Atlanta GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers) (2004)

12 Page J K Estimating cumulative illuminance data from month mean daily illumination and sunshine *Private communication*

13 *Daylighting and window design* CIBSE Lighting Guide LG10 (London: Chartered Institution of Building Services Engineers) (1999)

14 Kinghorn D and Muneer T Daylight illuminance frequency distribution; review of computational techniques and new data for UK locations *Lighting Res. Technol.* **30**(4) 139–150 (1998)

15 Muneer T and Kinghorn D Luminous efficacy of solar irradiance: improved models *Lighting Res. Technol.* **29**(4) 185–191 (1997)

16 Hunt D R G *Availability of daylight — designing for natural and artificial lighting* (Garston: Building Research Establishment) (1979)

17 Shellard H C *Tables of surface wind speed and direction over the United Kingdom* Met O 792 (Bracknell: Meteorological Office) (1968)

18 Caton P G F *Maps of hourly mean wind speed over the UK 1965–1973* Meteorological Office Climatological Memorandum 79 (Bracknell: Meteorological Office) (1976)

19 Troen I and Petersen E L *European Wind Atlas* (Roskilde, Denmark: Risø National Laboratory) (1989)

20 Hulme M and Barrow E (eds.) *Climates of the British Isles, present, past and future* (London: Routledge) (1997)

21 Houghton J T, Ding Y, Griggs D J, Noguer M, van der Linden P J, Dai X, Maskell K and Johnson C A (eds.) *Climate change 2001: the scientific basis — Contribution of WG1 to the IPCC Third Assessment* (Cambridge: Cambridge University Press) (2001)

22 Hulme M, Jenkins G J, Lu X, Turnpenny J R, Mitchell T D, Jones R G, Lowe J, Murphy J M, Hassell D, Boorman P, McDonald R and Hill S *Climate change scenarios for the United Kingdom: The UKCIP02 Scientific Report* (Norwich: Tyndall Centre for Climate Change Research, University of East Anglia) (2002)

23 Hulme M, Jenkins G J, Lu X, Turnpenny J R, Mitchell T D, Jones R G, Lowe J, Murphy J M, Hassell D, Boorman P, McDonald R and Hill S *Climate change scenarios for the United Kingdom: The UKCIP02 Briefing Report* (Norwich: Tyndall Centre for Climate Change Research, University of East Anglia) (2002)

24 Graves H M and Phillipson M C *Potential implications of climate change in the built environment* FBE Report (Garston: BRE) (2000)

25 *Weather data with climate change scenarios* CIBSE TM34 (London: Chartered Institution of Building Services Engineers) (2004)

26 *Climate change and the internal environment* CIBSE TM36 (London: Chartered Institution of Building Services Engineers) (2005)

27 *Special report on emissions scenarios (SRES): A special report of Working Group III of the Intergovernmental Panel on Climate Change* (Cambridge: Cambridge University Press) (2000)

28 Chow D H C *The effects of future climate change and near-extreme weather on office buildings in the UK* PhD thesis (Manchester: Manchester University) (2005)

29 Gregory J M, Stouffer R J, Raper S C B, Stott P A and Rayner N A An observationally based estimate of the climate sensitivity *J. Climate* **15** 3117–3121 (2002)

30 Hulme M *Private communication* (April 2004)

31 Watkins R *The impact of the urban environment on the energy used for cooling buildings* PhD thesis (Uxbridge: Brunel University (2002)

32 Chandler T J *The Climate of London* (London: Hutchinson) (1965)

33 Watkins R, Palmer J, Kolokotroni M and Littlefair P The balance of the annual heating and cooling demand within the London urban heat island *Proc. Int. Conf. Climate Change and the Built Environment, Manchester, 8–9 April 2002*

34 Graves H, Watkins R, Westbury P and Littlefair P *Cooling buildings in London — Overcoming the heat island* (London: CRC) (2001)

3 Thermal properties of building structures

3.1 Introduction

3.1.1 General

A major revision of CIBSE Guide A3 was published in 1999, with substantial changes to the guidance on heat losses through thermally bridged constructions, through solid and suspended ground floors and through windows. The present edition (2005), is an update primarily reflecting changes in British and European standards.

Since publication of the previous edition, European standards specifying test methods for thermal properties have been published, replacing the previous British standards. These are BS EN 12664[1], BS EN 12667[2] and BS EN 12939[3] for thermal conductivity measurement, and BS EN ISO 8990[4] and BS EN ISO 12567-1[5] for hot-box determination of the thermal transmittance of building elements.

3.1.2 Calculation of heat losses/gains

The thermal transmittance (*U*-value) of the building envelope is the principal factor in the determination of the steady-state heat losses/gains. Hence, the capacity of the heating or cooling system required to maintain specified inside design conditions under design external conditions. Design internal temperatures are considered in chapter 1: *Environmental criteria for design* and design external temperatures in chapter 2: *External design data*.

There are many different calculation procedures for determining the dynamic thermal behaviour of building structures. The parameters required for one of these, the admittance procedure, are given in Appendix 3.A6. The mathematical basis of the admittance procedure is given in chapter 5: *Thermal response and plant sizing*. The estimation of plant capacity, using both steady state and dynamic calculation procedures, is also dealt with in chapter 5.

3.1.3 Building Regulations

Part L of Schedule 1 to the Building Regulations 2000[6], which apply in England and Wales, requires that reasonable provision shall be made for the conservation of fuel and power in buildings. Ways of showing compliance with this requirement are given in Approved Document L1[7] for dwellings and Approved Document L2[8] for other buildings, although designers may choose to demonstrate compliance using other methods which are acceptable to the building control authority as providing equivalent performance. In terms of the thermal performance of the building structure, the aim of this part of the Regulations is to limit the heat loss and, where appropriate, maximise the heat gains through the fabric of the building.

In meeting this objective, Approved Documents L1 and L2 convey standards of fabric insulation which are set having regard to national standards of cost effectiveness, the need to avoid unacceptable technical risks and the need to provide flexibility for designers. However, for individual buildings, better standards of insulation can often be justified to clients using their own economic criteria and some guidance is given in BS 8207[9] and BS 8211[10].

In Scotland, the Building Standards (Scotland) Regulations 2004[11] apply and the relevant guidance on achieving the standards are given in Section 6[12]: the standards are broadly similar to those applicable in England and Wales, although there are some important technical differences. In Northern Ireland the Building Regulations (Northern Ireland) 1994[13] apply, the relevant technical standards are given in Technical Booklet F[14].

The thermal regulations are under review. New provisions, which will apply in England and Wales from 6 April 2006 take account of the Energy Performance of Buildings Directive and set the pass criterion for new buildings in terms of a limitation of carbon dioxide emissions. Thermal performance of the building fabric will remain a vital part of compliance assessment, both in terms of its contribution to attaining the overall emissions target and because limits on heat loss that apply in addition to the emissions target. Similar revisions are expected to apply in Northern Ireland later in 2006, and in Scotland in 2007.

It is essential for designers to be fully conversant with current statutory requirements, and the various means of compliance, and be aware of any proposed amendments that may be relevant to design work already in hand. This is very important since the designer may be called upon to certify compliance with the building regulations.

3.1.4 Prediction of moisture condensation

In addition to demonstrating compliance with the Building Regulations, accurate assessment of the thermal properties of the building structure is essential for the prediction of moisture condensation in the structure. Detailed guidance on methods of prediction of condensation is given in chapter 7: *Moisture transfer and condensation*.

3.1.5 Calculation of energy demands

The energy demand of a building and its services is only partly determined by the thermal transmittance. Other considerations, such as ventilation and ventilation heat loss, the effect of solar radiation falling on the building, especially that falling on the windows, and fortuitous heat gains from people, electrical equipment and lights, must be taken into account.

3.2 Notation

The symbols used within this section are defined as follows.

A_f Area of floor (m^2)
A_{fg} Area of floor in contact with ground (m^2)
A_g Projected area of glazing (m^2)
A_m, A_n Areas of elements composed of materials m, n (m^2)
A_p, A_q Areas of elements composed of materials p, q (m^2)
A_{wf} Projected area of window frame or sash (m^2)
A_1 Area of exposed surface(s) (m^2)
A_2 Area of non-exposed surface(s) (m^2)
B' Characteristic dimension of floor (m)
b_a Breadth of air space (m)
d Thickness of material (m)
d_a Thickness of air space (m)
d_{ef} Total equivalent thickness of floor (m)
d_{eg} Total equivalent thickness of ground (m)
d_{ei}' Additional equivalent thickness of floor due to edge insulation (m)
d_{fi} Thickness of (floor) edge insulation (or foundation) (m)
d_g Total thickness of glass panes (m)
d_{gb} Average thickness of bounding panes (m)
D_i Depth of vertical (floor) edge insulation (m)
d_m, d_n Thickness of elements composed of materials m, n (m)
d_p, d_q Thickness of elements composed of materials p, q (m)
d_s Thickness of window spacer (m)
d_w Thickness of wall surrounding ground floor (m)
E Emissivity factor
F Surface factor
f Decrement factor
f_s Factor related to thermal transmittance of window frame
f_w Wind shielding factor
h_c Convective heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
h_r Radiative heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
K Constant related to room geometry
L Length of a thermal bridge (m)
L_f Characteristic dimension of floor (m)
p_f Exposed perimeter of floor (m)
P_m, P_n Proportions of total surface area occupied by elements composed of materials m, n
P_p, P_q Proportions of total surface area occupied by elements composed of materials p, q
p_{wf} Length of inner perimeter of window frame or sash (m)
q_s Rate of heat transfer to/from surface ($W \cdot m^{-2}$)
R Thermal resistance ($m^2 \cdot K \cdot W^{-1}$)
R_A Combined thermal resistance of materials in plane of pitched roof (including outside surface resistance) ($m^2 \cdot K \cdot W^{-1}$)
R_a Thermal resistance of air space ($m^2 \cdot K \cdot W^{-1}$)
R_B Combined thermal resistance of materials in plane of ceiling (including inside surface resistance) ($m^2 \cdot K \cdot W^{-1}$)
R_b Thermal resistance of bridged leaf ($m^2 \cdot K \cdot W^{-1}$)
R_{bi} Thermal resistance of internal blind or curtain ($m^2 \cdot K \cdot W^{-1}$)
R_{bm} Thermal resistance of mid-pane blind ($m^2 \cdot K \cdot W^{-1}$)
R_{b1}, R_{b2} Thermal resistances of bridged structures 1, 2 ($m^2 \cdot K \cdot W^{-1}$)
R_f Thermal resistance of floor ($m^2 \cdot K \cdot W^{-1}$)
R_{fi} Thermal resistance of floor edge insulation (or foundation) ($m^2 \cdot K \cdot W^{-1}$)
r_g Thermal resistivity of glass ($m \cdot K \cdot W^{-1}$)
R_h Thermal resistance of homogeneous leaf ($m^2 \cdot K \cdot W^{-1}$)
R_i' Additional thermal resistance due to edge insulation (or foundation) ($m^2 \cdot K \cdot W^{-1}$)
R_{ig} Thermal resistance of insulation between floor and ground ($m^2 \cdot K \cdot W^{-1}$)
R_L Lower limit of thermal resistance ($m^2 \cdot K \cdot W^{-1}$)
R_m, R_n Thermal resistances of isotropic layers of sections m, n ($m^2 \cdot K \cdot W^{-1}$)
R_p, R_q Thermal resistances of isotropic layers of sections p, q ($m^2 \cdot K \cdot W^{-1}$)
R_s Surface resistance ($m^2 \cdot K \cdot W^{-1}$)
r_s Average thermal resistivity of window spacer material and sealant ($m \cdot K \cdot W^{-1}$)
R_{se} External surface resistance ($m^2 \cdot K \cdot W^{-1}$)
$R_{s(p)}$ Surface resistance of non-planar element ($m^2 \cdot K \cdot W^{-1}$)
R_{si} Internal surface resistance ($m^2 \cdot K \cdot W^{-1}$)
R_{s1}, R_{s2} Surface resistances at surfaces 1, 2 ($m^2 \cdot K \cdot W^{-1}$)
R_U Upper limit of thermal resistance ($m^2 \cdot K \cdot W^{-1}$)
R_v Thermal resistance of roof void ($m^2 \cdot K \cdot W^{-1}$)
R_1, R_2 Thermal resistances of components 1, 2 ($m^2 \cdot K \cdot W^{-1}$)
T_s Surface temperature (K)
U Thermal transmittance ($W \cdot m^{-2} \cdot K^{-1}$)
U_{eu} Equivalent thermal transmittance for heat flow through wall surrounding underfloor space and by ventilation of underfloor space ($W \cdot m^{-2} \cdot K^{-1}$)
U_f Thermal transmittance of (uninsulated) floor ($W \cdot m^{-2} \cdot K^{-1}$)
U_{fg} Thermal transmittance for heat flow through ground ($W \cdot m^{-2} \cdot K^{-1}$)
U_{fi} Thermal transmittance of edge-insulated floor ($W \cdot m^{-2} \cdot K^{-1}$)
U_{fs} Combined thermal transmittance of uninsulated suspended floor structure ($W \cdot m^{-2} \cdot K^{-1}$)
U_{fsi} Thermal transmittance of insulated suspended floor ($W \cdot m^{-2} \cdot K^{-1}$)
U_g Thermal transmittance of glazing ($W \cdot m^{-2} \cdot K^{-1}$)
U_{gb}' Thermal transmittance of glazing corrected for mid-pane blind ($W \cdot m^{-2} \cdot K^{-1}$)
U_{gg}' Thermal transmittance of glazing corrected for resistivity of glass ($W \cdot m^{-2} \cdot K^{-1}$)
U_{gs} Thermal transmittance of glazing with 12 mm spacer (normal exposure) ($W \cdot m^{-2} \cdot K^{-1}$)
U_r Thermal transmittance of roof ($W \cdot m^{-2} \cdot K^{-1}$)
U_u Thermal transmittance of wall surrounding underfloor space above ground level ($W \cdot m^{-2} \cdot K^{-1}$)
U_w Thermal transmittance of window ($W \cdot m^{-2} \cdot K^{-1}$)
U_{wb}' Thermal transmittance of window corrected for internal blind or curtain ($W \cdot m^{-2} \cdot K^{-1}$)
U_{wf} Thermal transmittance of window frame or sash ($W \cdot m^{-2} \cdot K^{-1}$)

v	Air velocity at surface ($m \cdot s^{-1}$)
v_w	Average wind velocity at 10 m height above ground level ($m \cdot s^{-1}$)
W_i	Width of horizontal edge (floor) insulation (m)
Y	Thermal admittance ($W \cdot m^{-2} \cdot K^{-1}$)
z_f	Height of floor above external ground level (m)
α	Area of ventilation openings per unit perimeter of underfloor space ($m^2 \cdot m^{-1}$)
β	Angle of pitch of roof
ε	Emissivity of surface
ε_1	Emissivity of exposed surface(s)
ε_2	Emissivity of non-exposed surface(s)
λ	Thermal conductivity ($W \cdot m^{-1} \cdot K^{-1}$)
λ_g	Thermal conductivity of ground ($W \cdot m^{-1} \cdot K^{-1}$)
λ_m, λ_n	Thermal conductivity of materials m, n ($W \cdot m^{-1} \cdot K^{-1}$)
λ_p, λ_q	Thermal conductivity of materials p, q ($W \cdot m^{-1} \cdot K^{-1}$)
σ	Stefan-Boltzmann constant (5.67×10^{-8}) ($W \cdot m^{-2} \cdot K^{-4}$)
Φ_f	Rate of heat loss through floor (W)
ϕ	Time lag associated with decrement factor (h)
Ψ	Linear thermal transmittance of a thermal bridge ($W \cdot m^{-1} \cdot K^{-1}$)
Ψ_{fi}	Factor related to (floor) edge insulation ($W \cdot m^{-1} \cdot K^{-1}$)
Ψ_s	Linear thermal transmittance due to window spacer ($W \cdot m^{-1} \cdot K^{-1}$)
ψ	Time lag associated with surface factor (h)
ω	Time lead associated with thermal admittance (h)

3.3 Heat losses from buildings

3.3.1 General

Heat is lost from buildings by transmission through the building fabric and by ventilation. (For ventilation losses see chapter 4: *Air infiltration and natural ventilation*.)

This section provides methods for the calculation of the rate of heat loss through individual components of the envelope of a building. In most cases the thermal properties of a building component are represented by its thermal transmittance, U (in $W \cdot m^{-2} \cdot K^{-1}$). The U-value multiplied by the area of the component gives the rate of heat loss through the component per unit of temperature difference between inside and outside.

Often it is convenient to characterise the whole building by a transmission heat loss coefficient, H_t, i.e:

$$H_t = \Sigma (A\ U) + \Sigma (L\ \Psi) \tag{3.1}$$

where H_t is the transmission heat loss coefficient (W·K), $\Sigma (A\ U)$ is the sum over all the components of the building (i.e. roof, walls, floor, windows) of the product of the area of each component and its U-value ($W \cdot m^{-2} \cdot K^{-1}$) and $\Sigma (L\ \Psi)$ is the sum over all thermal bridges of the product of the length of each thermal bridge (m) and its linear thermal transmittance ($W \cdot m^{-1} \cdot K^{-1}$).

Repeating thermal bridges (which occur at fixed intervals in the element, such as mortar joints or timber studding) are taken into account in the calculation of the U-value of the component (see section 3.3.11) and no further allowance is needed. Linear thermal transmittances (Ψ-values) arise at junctions between different components; for further guidance see section 3.7.

The transmission heat loss through a component is modified if there is an unheated space between the internal and external environments. One method of allowing for this is given in BS EN ISO 13789[16].

3.3.2 Dimensions for heat loss calculations

In the past, the areas used in conjunction with U-values to calculate heat losses have sometimes been based on internal dimensions, as shown by the lower case letters in Figure 3.1. These dimensions omit the areas of walls, floors and ceilings covered by intermediate floors and internal walls, which in total may be up to 15% of the total area. These losses have been shown to be significant, particularly for well-insulated houses[17]. If measurements are based on internal dimensions, the areas should be increased by an allowance to account for the thickness of internal elements. For dwellings (including flats), an allowance of 15% is appropriate.

A more satisfactory approach is to use the overall internal dimensions, measured between finished internal faces of external elements of the building and including the

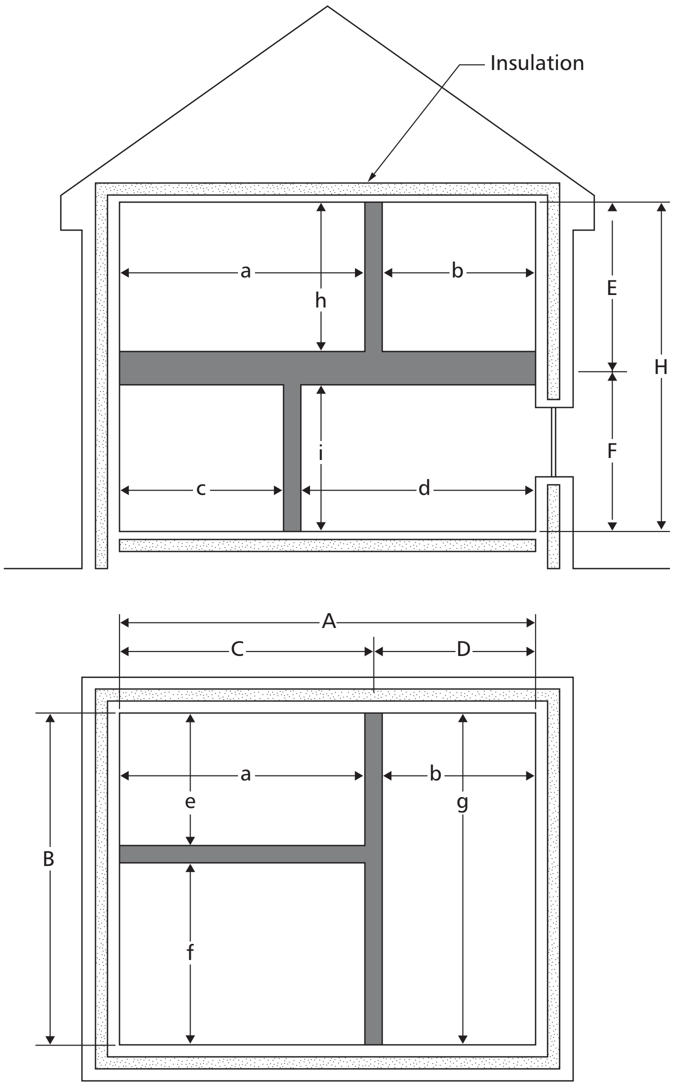


Figure 3.1 Dimensions for heat loss calculations

thickness of internal elements, as indicated by the capital letters in Figure 3.1. This is the basis used in current Building Regulations[7,8,12,14].

3.3.3 Application of thermal insulation

There are a number of factors that must be considered in both the design and application of thermal insulation. Thermal insulation is most effective when applied as a continuous and even layer, without penetrations or breaks. Penetrations form thermal bridges and breaks permit air flow within and through the insulating layer, both of which reduce its effectiveness.

It is not always possible to achieve the maximum effectiveness of the insulation due to other influences such as structural considerations and weather proofing. This applies particularly where insulation is being added to an existing structure. Therefore, in the design of new buildings, it is essential that thermal insulation is considered at the earliest stages of design.

Where penetrations and breaks in the insulating layer are unavoidable, the effects of the resulting thermal bridges should be taken into account in the calculation of the *U*-value. Other deviations from the ideal, such as the effects of gaps or air spaces on the warm side of the insulation layer, may degrade further its performance.

3.3.4 Thermal resistance of materials

The thermal properties of a material are expressed in terms of its thermal resistance. For homogeneous, isotropic materials through which heat is transmitted by conduction only, the thermal resistance is directly proportional to the thickness and is given by:

$$R = d / \lambda \tag{3.2}$$

where R is the thermal resistance ($m^2 \cdot K \cdot W^{-1}$), d is the thickness of material (m) and λ is the thermal conductivity ($W \cdot m^{-1} \cdot K^{-1}$).

In other materials, especially insulating materials, heat may be transmitted by a combination of solid and gaseous conduction, convection and radiation. For such materials, the thermal properties are not determined solely by the thermal conductivity. However, where materials are thermally homogeneous, an effective thermal conductivity can be ascribed and equation 3.2 may then be applied.

For thermally non-homogeneous materials, the thermal resistance may not be directly proportional to the thickness and equation 3.2 does not apply. This is the case for materials of low density in which the radiant heat transfer is a significant factor. For this reason, some manufacturers, particularly of insulation products, define the thermal properties of these materials by means of thermal resistance (*R*-value) rather than thermal conductivity. The value of thermal resistance applies to a particular thickness of material and cannot be extrapolated for other thicknesses. For a more detailed discussion on thermal resistance, see ISO 8302[18].

3.3.5 Thermal conductivity of masonry materials

For many types of masonry materials, including clay products and various types of aggregate and aerated concrete, the thermal conductivity is related to the bulk density and moisture content. Table 3.1 provides values of thermal conductivity for a range of densities at 'standard' values of moisture content, as defined in Table 3.2. 'Protected' includes internal partitions, inner leaves separated from outer leaves by a continuous air space, masonry protected by tile hanging, sheet cladding or other such protection, separated by a continuous air space. 'Exposed' covers rendered or unrendered masonry directly exposed to rain.

Particular masonry products can have thermal conductivities significantly lower than the corresponding values given in Table 3.1, depending on the materials used in manufacture. For such materials, manufacturers may quote thermal conductivity less than those given in Table 3.1. These values should be accepted, and used in place of the tabulated values, provided the tests have been performed in accordance with the requirements laid down in Appendix 3.A2.

If the moisture content of the sample under test differed from the appropriate standard value, a correction should be applied to the measured thermal conductivity using the procedure described in Appendix 3.A1.

3.3.6 Thermal conductivity of materials other than masonry

Thermal conductivity data for several common building materials for design purposes are given in BS EN 12524[20]. Appendix 3.A7 gives more detailed information compiled from a number of sources. In situations where the thermal conductivity of a particular material is critical, e.g. where it provides the bulk of the thermal resistance of a structure, certified test data, obtained in accordance with Appendix 3.A2, should be used. Measured data are also appropriate where the density of the materials to be used, or their working temperatures, are outside the ranges given in the tables.

3.3.7 Declared and design values

The declared value (of thermal conductivity or thermal resistance) for thermal insulation products is that provided by a manufacturer. It applies under reference conditions of temperature (usually 10 °C), moisture content (usually corresponding to equilibrium with an atmosphere of 23 °C and 50% RH) and, where relevant, ageing. Declared values are usually based on measurements carried out on the materials concerned, and they take account of statistical variations between different samples of the product such that 90% of the production has properties at least as good as those declared.

Design values apply under particular conditions of use and are used in the calculation of thermal performance. In many cases the design value may be taken as equal to the declared value as the design conditions are often similar to those for the declared value. However, in cases where the

Table 3.1 Thermal conductivity of homogeneous masonry materials at 'standard' moisture content

Material	Dry density / kg.m^{-3}	Thermal conductivity / W·m^{-1}·K^{-1}	
		Protected	Exposed
Brick (fired clay)	1200	0.36	0.50
	1300	0.40	0.54
	1400	0.44	0.60
	1500	0.47	0.65
	1600	0.52	0.71
	1700	0.56	0.77
	1800	0.61	0.83
	1900	0.66	0.90
	2000	0.70	0.96
Brick (calcium silicate)	1700	0.77	1.05
	1800	0.89	1.22
	1900	1.01	1.38
	2000	1.16	1.58
	2100	1.32	1.80
	2200	1.51	2.06
Dense aggregate concrete	1700	1.04	1.12
	1800	1.13	1.21
	1900	1.22	1.31
	2000	1.33	1.43
	2100	1.46	1.56
	2200	1.59	1.70
	2300	1.75	1.87
	2400	1.93	2.06
Blast furnace slag concrete	1000	0.19	0.20
	1100	0.24	0.25
	1200	0.27	0.29
	1300	0.32	0.35
	1400	0.38	0.41
	1500	0.45	0.48
	1600	0.53	0.56
	1700	0.60	0.65

Material	Dry density / kg.m^{-3}	Thermal conductivity / W·m^{-1}·K^{-1}	
		Protected	Exposed
Pyro-processed colliery material concrete	1100	0.39	0.42
	1200	0.41	0.44
	1300	0.44	0.47
	1400	0.46	0.49
	1500	0.48	0.52
Pumice aggregate concrete	500	0.16	0.17
	600	0.18	0.19
	700	0.20	0.22
	800	0.24	0.25
	900	0.27	0.29
	1000	0.31	0.34
	1100	0.36	0.38
	1200	0.40	0.43
	1300	0.46	0.49
Autoclaved aerated concrete	400	0.12	0.13
	500	0.15	0.16
	600	0.18	0.19
	700	0.20	0.22
	800	0.24	0.25
	900	0.27	0.29
Other lightweight aggregate concrete	600	0.20	0.22
	700	0.24	0.25
	800	0.28	0.30
	900	0.31	0.34
	1000	0.36	0.38
	1100	0.40	0.43
	1200	0.46	0.49
	1300	0.52	0.55
	1400	0.57	0.61
	1500	0.63	0.67
	1600	0.71	0.76

Note: these data have been derived from the values for the 90% fractile given in BS EN 1745[19] for the standard moisture contents given in Table 3.2. The value for mortar may be taken as 0.88 W·m^{-1}·K^{-1} (protected) and 0.94 W·m^{-1}·K^{-1} (exposed).

Table 3.2 'Standard' moisture contents for masonry

Material	Moisture content	
	Protected	Exposed
Brick (fired clay)	1% (by volume)	5% (by volume)
Brick (calcium silicate)	1% (by volume)	5% (by volume)
Dense aggregate concrete	3% (by volume)	5% (by volume)
Blast furnace slag concrete	3% (by weight)	5% (by weight)
Pumice aggregate concrete	3% (by weight)	5% (by weight)
Other lightweight aggregate concrete	3% (by weight)	5% (by weight)
Autoclaved aerated concrete	3% (by weight)	5% (by weight)

Note: % (by volume) = % (by weight) × density/1000

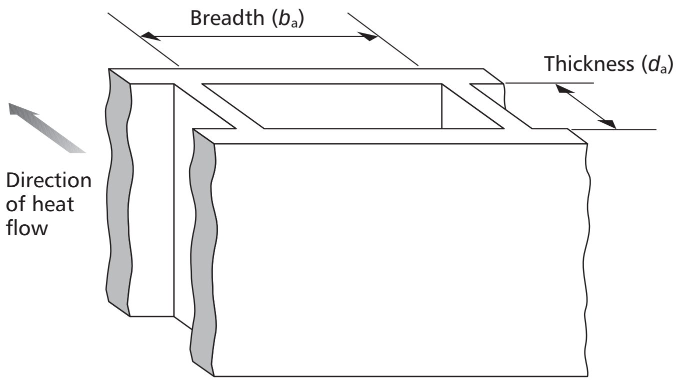


Figure 3.2 Divided vertical air space

design conditions are markedly different, thermal conductivity values should be converted before being used in calculations. Conversion coefficients for a range of insulating materials are given in BS EN ISO 10456[21].

3.3.8 Thermal resistance of air spaces

Heat transfer across an air space is approximately proportional to the difference between the temperatures of the boundary surfaces. However, the thermal resistance depends upon various other factors such as the dimensions of the air space, the direction of heat flow, the emissivities of the inner surfaces and the extent to which the airspace is ventilated.

The thermal resistance of tall, continuous vertical air spaces increases with the thickness of the air space, up to a thickness of about 25 mm. The thermal resistance is virtually constant for greater thicknesses.

If the air space is not continuous in the horizontal direction but divided into vertical strips or slots, see Figure 3.2, the thermal resistance of the air space also depends upon the breadth of the strip.

Tables 3.3 and 3.4 provide standardised values of thermal resistance for both continuous and divided air spaces. For air spaces of thicknesses greater than 25 mm, the thermal resistance should be taken as 0.18 m^2·K·W^{-1} if the breadth of the air space is greater than ten times its thickness. Otherwise, the following equation should be used:

$$R_a = \frac{1}{1.25 + 2.32\,(1 + \sqrt{(1 + d_a^2 / b_a^2)} - d_a / b_a\,)} \qquad (3.3)$$

where R_a is the thermal resistance of the air space ($m^2 \cdot K \cdot W^{-1}$), d_a is the thickness of the air space (m) and b_a is the breadth of the air space (m).

Horizontal air spaces present greater resistance to downward heat flow because downward convection is small. For an air space incorporating multiple layers of aluminium foil insulation, the thermal resistance to downward heat flow can be as high as 2 $m^2 \cdot K \cdot W^{-1}$. Such constructions can play an important role in reducing summer heat gains through roofs.

A small inclination of an air space has only a minor effect on the convective heat transfer and therefore little effect on the thermal resistance.

Radiation accounts for about two-thirds of the total heat transfer and most building materials have high emissivities (between 0.9 and 0.95). Therefore, air spaces lined with low emissivity materials, such as aluminium foil, have a higher thermal resistance because radiative heat transfer is reduced. However, unless it is known that the air space is to be lined with such a material, high emissivity should be assumed.

The temperature difference across an air space has little effect on either the radiative or convective heat transfer coefficients. Therefore, no allowance need be made unless the air space resistance forms a major proportion of the total thermal resistance of the structure.

The influence of ventilation on the thermal resistance of air spaces can be estimated from heat transfer theory if the rate and distribution of the air movement is known. This varies according to the prevailing conditions and cannot be determined easily. Where there is an airspace that is well-ventilated to the outside, such as below tiles on a pitched roof, it is recommended that all layers to the outside of it are disregarded, and that the normal value of external surface resistance is replaced by a value corresponding to still air. Specific criteria are given in BS EN ISO 6946[15].

Values for the thermal resistance of roof spaces are given in Table 3.5. Note that these values include an allowance for the thermal resistance of the roof construction but do not include the external surface resistance (R_{se}).

Table 3.3 Thermal resistances for continuous unventilated air spaces

Air space thickness / mm	Surface emissivity†	Thermal resistance (/ $m^2 \cdot K \cdot W^{-1}$) for heat flow in stated direction‡		
		Horizontal	Upward	Downward
5	High	0.11	0.11	0.11
	Low§	0.17	0.17	0.17
≥ 25	High	0.18	0.16	0.19
	Low§	0.44	0.34	0.50

† High emissivity: $\varepsilon > 0.8$; low emissivity: $\varepsilon \leq 0.2$
‡Normal to the surface in the direction of heat flow
§ Assumes that the air space is bounded by one low emissivity surface and one high emissivity surface

Table 3.4 Thermal resistances for divided air spaces for horizontal heat flow

Air space thickness / mm	Thermal resistance (/ $m^2 \cdot K \cdot W^{-1}$) for air space of stated breadth / mm				
	≥ 200	100	50	20	≤ 10
5	0.11	0.11	0.11	0.11	0.12
6	0.12	0.12	0.12	0.13	0.13
7	0.13	0.13	0.13	0.14	0.15
8	0.14	0.14	0.14	0.15	0.16
10	0.15	0.15	0.16	0.17	0.18
12	0.16	0.17	0.17	0.19	0.20
15	0.17	0.18	0.19	0.21	0.23
20	0.18	0.20	0.21	0.24	0.26
25	0.18	0.20	0.21	0.24	0.27

Notes:
(1) Applies to heat flow in horizontal direction only for air spaces of dimensions shown in Figure 3.2.
(2) Calculated in accordance with BS EN ISO 6946[17].
(3) Assumes high emissivity at surfaces.

Table 3.5 Thermal resistances for roof spaces (reproduced from BS EN ISO 6946[17] by permission of the British Standards Institution)

Item	Description	Thermal resistance / $m^2 \cdot K \cdot W^{-1}$
1	Tiled roof with no felt, boards or similar	0.06
2	Sheeted roof or tiled roof with felt or boards or similar under the tiles	0.2
3	As 2 but with aluminium cladding or other low emissivity surface at underside of roof	0.3
4	Roof lined with boards and felt	0.3

3.3.9 Surface resistance

The inside and outside surface resistances are determined by the processes of heat transfer which occur at the boundary between a structural element and the air. Heat is transferred both by radiation interchange with other surfaces and by convective heat transfer at the air/surface interface.

3.3.9.1 Heat transfer by radiation

Heat transfer by radiation is a complex process which depends upon the shape, temperature and emissivity of both the radiating surface and the surface or environment to which it radiates. A detailed description is contained in Guide C, chapter 3: *Heat transfer*[22]. However, for practical purposes, the heat transfer by radiation is characterised by an emissivity factor, E, and a radiative heat transfer coefficient, h_r.

The emissivity factor depends upon the geometry of the room and the emissivities of the surfaces. However, for a cubical room with one exposed surface, all the internal surfaces having high emissivity, E may be expressed thus (see Appendix 3.A3):

$$E = K\,\varepsilon \qquad (3.4)$$

where E is the emissivity factor, K is a constant related to room geometry and ε is the emissivity of the surface.

Most common building materials have high emissivities ($\varepsilon > 0.9$). See Appendix 3.A3 for cases of unusual room geometry or for internal surfaces with low emissivities.

The radiative heat transfer coefficient depends upon the absolute temperatures of both the radiating surface and the surface or environment receiving the radiation. The following is an adequate approximation for most building services applications:

$$h_r = 4\,\sigma\,T_s^3 \tag{3.5}$$

where h_r is the radiative heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$), σ is the Stefan-Boltzmann constant (5.67 × 10^{-8} $W \cdot m^{-2} \cdot K^{-4}$) and T_s is the surface temperature (K).

Values of the radiative heat transfer coefficient for a range of surface temperatures are given in Table 3.6. It should be noted that, for night time clear skies, the difference between the surface temperature and the 'sky' temperature can be very large, leading to underestimation of h_c in these circumstances. This is particularly important in the prediction of condensation.

Table 3.6 Radiative heat transfer coefficient, h_r

Mean temperature of surfaces / °C	Radiative heat transfer coefficient, h_r / $W \cdot m^{-2} \cdot K^{-1}$
–10	4.1
0	4.6
10	5.1
20	5.7

3.3.9.2 Heat transfer by convection

Heat transfer by convection is characterised by a heat transfer coefficient, h_c. This depends upon the temperature difference between the surface and the air, the surface roughness, the air velocity and the direction of heat flow. For still air conditions, values of the convective heat transfer coefficient are given in Table 3.7.

Table 3.7 Convective heat transfer coefficient, h_c

Direction of heat flow	Convective heat transfer coefficient†, h_c / $W \cdot m^{-2} \cdot K^{-1}$
Horizontal	2.5
Upward	5.0
Downward	0.7

† Assumes still air conditions, i.e. air speed at the surface is not greater than 0.1 $m \cdot s^{-1}$

Where significant air movement occurs, heat transfer by convection is more complex and reference should be made to Guide C, chapter 3: *Heat transfer*(22). Approximate values may be obtained using the following empirical equation but this should not be used where the air velocity is less than 1 m/s.

$$h_c = 4 + 4\,v \tag{3.6}$$

where h_c is the convective heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$) and v is the air velocity at the surface ($m \cdot s^{-1}$).

3.3.9.3 Internal surface resistance

Appendix 3.A3 shows how the emissivity factor may be combined with the above coefficients to give an internal surface resistance. Because of the complex nature of the heat transfer processes involved, the internal surface resistance is assumed to be independent of surface roughness, temperature differences between radiating surfaces, differences between surface and air temperatures etc. The internal surface resistance is given by:

$$R_{si} = \frac{1}{{}^6\!/_5\,E\,h_r + h_c} \tag{3.7}$$

where R_{si} is the internal surface resistance ($m^2 \cdot K \cdot W^{-1}$), E is the emissivity factor, h_r is the radiative heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$) and h_c is the convective heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$).

The values for internal surface resistance used in BS EN ISO 6946(15) are shown in Table 3.8. These values represent a simplification of the heat transfer processes that occur at surfaces.

Table 3.8 Internal surface resistance, R_{si}

Building element	Direction of heat flow	Surface resistance / $m^2 \cdot K \cdot W^{-1}$
Walls	Horizontal	0.13
Ceilings or roofs (flat or pitched), floors	Upward	0.10
Ceilings or floors	Downward	0.17

3.3.9.4 External surface resistance

Appendix 3.A3 also shows how the emissivity factor may be combined with the radiative and convective coefficients to give an external surface resistance. Again, the external surface resistance is assumed independent of surface roughness, temperature differences between radiating surfaces, differences between surface and air temperatures, etc. The external surface resistance is given by:

$$R_{se} = \frac{1}{E\,h_r + h_c} \tag{3.8}$$

where R_{se} is the external surface resistance ($m^2 \cdot K \cdot W^{-1}$), E is the emissivity factor, h_r is the radiative heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$) and h_c is the convective heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$).

The values for external surface resistance used in BS EN ISO 6946(15) are shown in Table 3.9, which are recommended for most design purposes. As with the Table 3.8, these values represent a simplification of the heat transfer processes that occur at surfaces.

Table 3.9 External surface resistance, R_{se}

Building element	Direction of heat flow	Surface resistance / $m^2 \cdot K \cdot W^{-1}$		
		BS EN ISO 6946 (normal design value)	Sheltered	Exposed
Wall	Horizontal	0.04	0.06	0.02
Roof	Upward	0.04	0.06	0.02
Floor	Downward	0.04	0.06	0.02

For well insulated structures, the effect of exposure is small and may be ignored for opaque elements. For glazing, additional values are given for 'sheltered' and 'severe' conditions of exposure. The conditions of exposure are defined thus:

— *normal*: most suburban and rural buildings, fourth to eighth floors of buildings in city centres (wind speed 5 $m \cdot s^{-1}$)

— *sheltered*: up to third floor of buildings in city centres (wind speed 2 $m \cdot s^{-1}$)

— *severe*: buildings on coastal or hill sites, floors above the fifth in suburban or rural districts, floors above the ninth in city centres (wind speed 9 $m \cdot s^{-1}$).

3.3.9.5 Elements with non-planar surfaces

Finned external elements constructed of masonry (concrete or brickwork), such as the extension of a floor slab to form a balcony, may increase the heat loss compared with the same building element without the fin. The effect of such fins on the total heat loss can be ignored provided that the thermal conductivity of the fin material is not greater than 2 $W \cdot m^{-1} \cdot K^{-1}$, or external insulation is carried around the fin.

For fins constructed of materials having a thermal conductivity greater than 2 $W \cdot m^{-1} \cdot K^{-1}$ and where the fin is not insulated, BS EN ISO 6946[15] requires the surface resistance (internal or external) to be modified as follows:

$$R_{s(p)} = R_s \, (A_p \, / \, A) \tag{3.9}$$

where $R_{s(p)}$ is the surface resistance of the non-planar element ($m^2 \cdot K \cdot W^{-1}$), R_s is the surface resistance of the element calculated without the fin ($m^2 \cdot K \cdot W^{-1}$), A_p is the projected area of the fin (m^2) and A is the exposed surface area of the fin (m^2).

The areas A and A_p are illustrated in Figure 3.3.

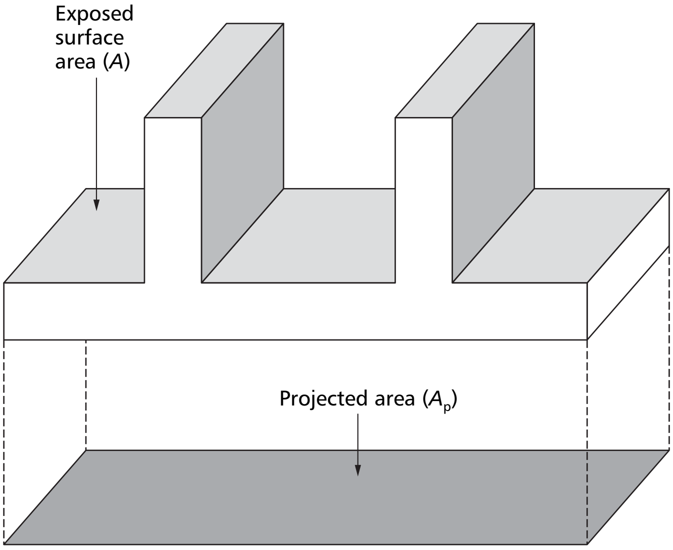


Figure 3.3 Actual and projected areas for non-planar elements

3.3.9.6 Elements containing metal components

Guidance has been published for determining *U*-values for particular types of element containing metal components. These include:

— *light steel-frame construction*: BRE Digest 465[23]

— *metal cladding*: BRE[24] and SCI[25]

— *curtain walling*: Centre for Window and Cladding Technology[26,27].

3.3.10 Thermal transmittance for elements composed of plane homogenous layers

The thermal transmittance of a building element is obtained by combining the thermal resistances of its component parts and the adjacent air layers. Thermal transmittances of simple walls and roofs composed of parallel slabs are obtained by adding the thermal resistances and taking the reciprocal of the sum, thus:

$$U = 1 \, / \, (R_{si} + R_1 + R_2 + \ldots + R_a + R_{se}) \tag{3.10}$$

where U is the thermal transmittance ($W \cdot m^{-2} \cdot K^{-1}$), R_{si} is the internal surface resistance ($m^2 \cdot K \cdot W^{-1}$), R_1 and R_2 are the thermal resistances of components 1 and 2 ($m^2 \cdot K \cdot W^{-1}$), R_a is the thermal resistance of the air spaces ($m^2 \cdot K \cdot W^{-1}$) and R_{se} is the external surface resistance ($m^2 \cdot K \cdot W^{-1}$).

Example 3.1: Calculation of U-value for structure composed of plane homogeneous layers

An external wall consisting of 105 mm brickwork, 70 mm cavity completely filled with insulation having a thermal conductivity of 0.04 $W \cdot m^{-1} \cdot K^{-1}$, 100 mm lightweight concrete block finished internally with 13 mm lightweight plaster. Table 3.10 summarises the construction.

The total thermal resistance is calculated, from which the *U*-value is then determined:

$$U = 1 \, / \, \Sigma \, R = 0.38 \ W \cdot m^{-2} \cdot K^{-1}$$

Note that for the purposes of the above example, the thermal conductivity of the mortar is taken to be the same as that bulk material for both inner and outer leaves, and the effect of the wall ties is not included. These effects usually need to be allowed for, as discussed in sections 3.3.11 and 3.3.12.

Table 3.10 Example 1: thickness, thermal conductivity and thermal resistance of materials

Element	Thickness / mm	Thermal conductivity / $W \cdot m^{-2} \cdot K^{-1}$	Thermal resistance / $m^2 \cdot K \cdot W^{-1}$
Outer surface	—	—	0.04
External brickwork	105	0.77	0.136
Cavity insulation	70	0.04	1.75
Blockwork	100	0.19	0.526
Plaster	13	0.16	0.072
Inner surface	—	—	0.13
			$\Sigma \, R$ = 2.655

3.3.11 Thermal transmittance for elements composed of bridged layers

3.3.11.1 General

The method of calculation of thermal transmittance given in section 3.3.10 assumes that the direction of heat flow is perpendicular to the plane of the structure. This is true when the layers are of uniform thickness and the thermal conductivity is isotropic along this plane. Dissimilar thermal conductivities and thicknesses mean that heat flows are not unidirectional and thermal bridges are formed.

Rigorous calculation of thermal bridges and their effect on the average *U*-value requires two- and three-dimensional heat flow analysis. However, for most constructions simpler calculation procedures give satisfactory results. The applicability of calculation methods is discussed further in BR 443[28].

Note that if the difference between the thermal resistances of the bridged and non-bridged layers is less than 0.1 $m^2 \cdot K \cdot W^{-1}$, the effect of thermal bridging is small and may be neglected.

3.3.11.2 Combined method

The basis of the combined method is to calculate the upper and lower limits of the thermal resistance of the bridged part of the structure. The lower limit of thermal resistance is that which imposes no resistance to sideways flow of heat, and the upper limit is that which imposes an infinite resistance to sideways heat flow. In practice, the heat flow pattern lies between these limits and the arithmetic mean gives the thermal resistance to a reasonable approximation.

The thermal resistance of the bridged structure is given by:

$$R_b = {}^{1}/_{2}\,(R_L + R_U) \tag{3.11}$$

where R_b is the thermal resistance of the bridged structure ($m^2 \cdot K \cdot W^{-1}$), R_L is the lower limit of thermal resistance ($m^2 \cdot K \cdot W^{-1}$) and R_U is the upper limit of thermal resistance ($m^2 \cdot K \cdot W^{-1}$).

3.3.11.3 Structural elements with one bridged layer

Figure 3.4 shows a single leaf composed of material m, bridged repetitively. For simplicity, the leaf is shown bridged by two elements only, composed of materials n and p. The leaf is divided into layers numbered from 1 to *z*; the bridged layer is shown as layer 2.

(*a*) Lower limit

The lower limit of thermal resistance is determined as follows:

$$R_L = R_{se} + R_1 + \frac{A_m + A_n + A_p}{(A_m / R_{m2}) + (A_n / R_{n2}) + (A_p / R_{p2})} + R_3 \cdots + R_z + R_{si} \tag{3.12}$$

where R_L is the lower limit of thermal resistance ($m^2 \cdot K \cdot W^{-1}$), R_1 .... R_z are the thermal resistances of (unbridged) layers 1 .... *z* ($m^2 \cdot K \cdot W^{-1}$), A_m, A_n and A_p are the areas of elements composed of materials m, n and p respectively (m^2), R_{m2}, R_{n2} and R_{p2} are the thermal resistances of elements composed of materials m, n and p respectively for (bridged) layer 2 ($m^2 \cdot K \cdot W^{-1}$), R_{se} is the external surface resistance ($m^2 \cdot K \cdot W^{-1}$) and R_{si} is the internal surface resistance ($m^2 \cdot K \cdot W^{-1}$).

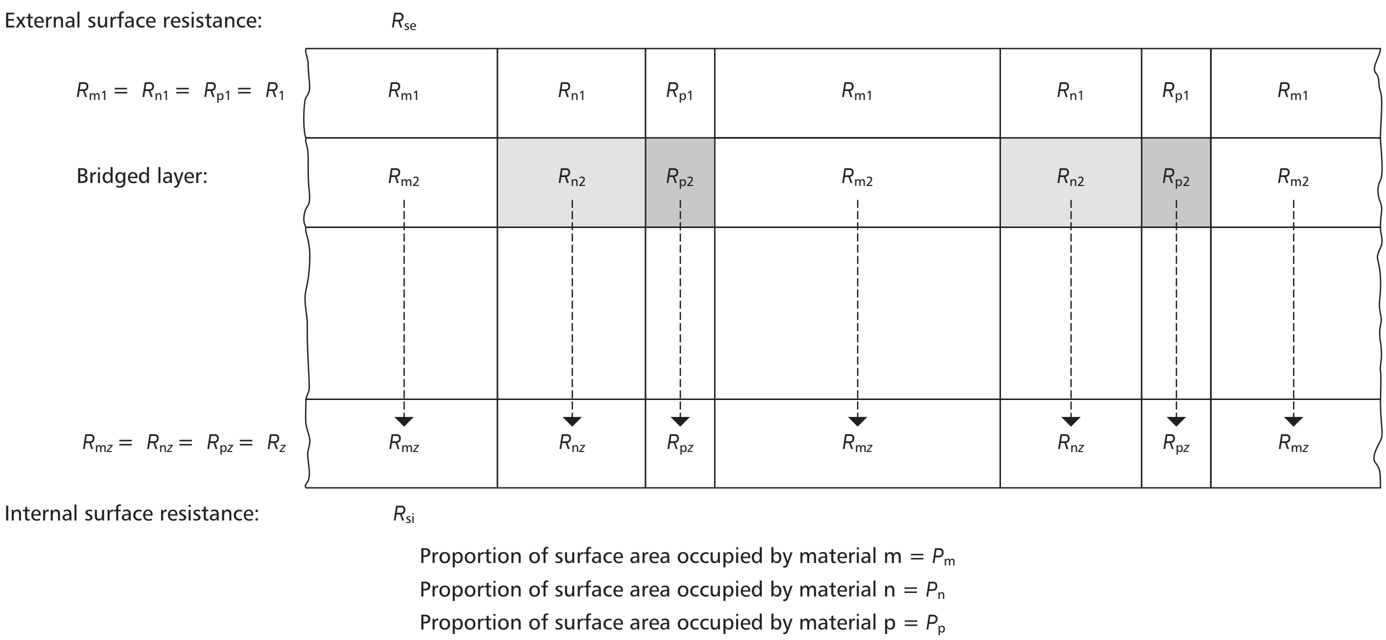


Figure 3.4 Single bridged layer, bridged repetitively by two elements composed of different materials

In terms of proportions of the total surface area:

$$R_L = R_{se} + R_1 + \frac{1}{(P_m/R_{m2}) + (P_n/R_{n2}) + (P_p/R_{p2})} + R_3 + \ldots R_z + R_{si} \qquad (3.13)$$

where P_m, P_n and P_p are the proportions of the total surface area occupied by elements composed of materials m, n and p.

(*b*) Upper limit

The upper limit of thermal resistance, R_U, is determined as follows.

$$R_U = (A_m + A_n + A_p)\left(\frac{A_m}{R_{se} + R_{m2} + (R_1 \ldots + R_z) + R_{si}} + \frac{A_n}{R_{se} + R_{n2} + (R_1 \ldots + R_z) + R_{si}} + \frac{A_p}{R_{se} + R_{p2} + (R_1 \ldots + R_z) + R_{si}}\right)^{-1} \qquad (3.14)$$

where R_U is the upper limit of thermal resistance ($m^2 \cdot K \cdot W^{-1}$), R_1 .... R_z are the thermal resistances of (unbridged) layers 1 .... z etc. ($m^2 \cdot K \cdot W^{-1}$), A_m, A_n and A_p are the areas of elements composed of materials m, n and p (m^2), R_{m2} , R_{n2} and R_{p2} are the thermal resistances of elements composed of materials m, n and p for (bridged) layer 2 ($m^2 \cdot K \cdot W^{-1}$), R_{se} is the external surface resistance ($m^2 \cdot K \cdot W^{-1}$). and R_{si} is the internal surface resistance ($m^2 \cdot K \cdot W^{-1}$).

In terms of proportions of the total surface area:

$$R_U = \left(\frac{P_m}{R_{se} + R_{m2} + (R_1 \ldots + R_z) + R_{si}} + \frac{P_n}{R_{se} + R_{n2} + (R_1 \ldots + R_z) + R_{si}} + \frac{P_p}{R_{se} + R_{p2} + (R_1 \ldots + R_z) + R_{si}}\right)^{-1} \qquad (3.15)$$

where P_m, P_n and P_p are the proportions of the total surface area occupied by elements composed of materials m, n and p.

The thermal resistance of the bridged element is the average of the upper and lower limiting values, i.e:

$$U = \frac{1}{{}^1/_2\,(R_L + R_U)} \qquad (3.16)$$

Where more than one layer is bridged, the relative positions of the bridging within each bridged layer are not generally known. Each different heat flow path (from internal to external environments) must be considered when calculating the upper limit of thermal resistance. The thermal resistances for each of the paths are combined in parallel, in proportion to their areas. The proportions for each flow path are obtained by multiplying together the proportions appropriate to each material comprising the flow path.

Examples 3.2, 3.3 and 3.4 show the use of the combined method to determine the thermal transmittance of typical bridged constructions.

Example 3.2: Calculation of U-value for masonry wall, inner leaf bridged by mortar joints

See Figure 3.5. The dimensions and thermal properties of the components of the wall are given in Table 3.11. Isothermal planes are taken at the inside air and external air. Since the thermal conductivities of brick and mortar are approximately equal, the outer leaf is regarded as thermally homogenous.

(*a*) Mean thermal resistance of bridged structure

The lower limit of thermal resistance is obtained from equation 3.13:

$$R_L = 0.04 + 0.136 + 0.18 + \left(\frac{1}{(0.933/1.136) + (0.067/0.142)}\right) + 0.072 + 0.13$$

$$= 1.331\ m^2 \cdot K \cdot W^{-1}$$

The upper limit of thermal resistance is obtained from equation 3.15:

$$R_U = \left(\frac{0.933}{0.04 + 1.136 + 0.18 + 0.136 + 0.072 + 0.13} + \frac{0.067}{0.04 + 0.136 + 0.18 + 0.142 + 0.072 + 0.13}\right)^{-1}$$

$$= 1/(0.5508 + 0.0957)$$

$$= 1.547\ m^2 \cdot K \cdot W^{-1}$$

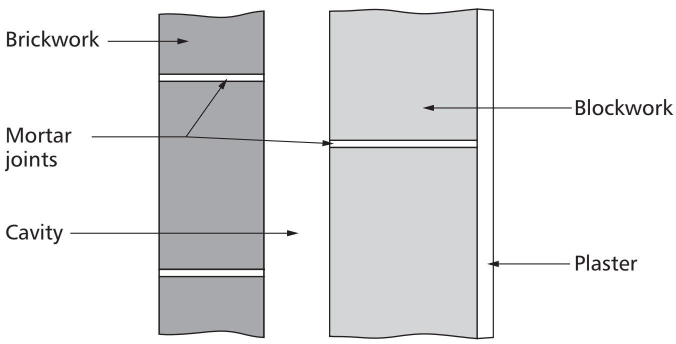


Figure 3.5 Example 3.2: cavity wall construction

Table 3.11 Example 3.2: dimensions and thermal properties of materials

Element	Thickness (mm)	Proportion of surface area	Thermal conductivity / $W{\cdot}m^{-1}{\cdot}K^{-1}$	Thermal resistance / $m^2{\cdot}K{\cdot}W^{-1}$
Outer surface (R_{se})	—	—	—	0.04
External brickwork	105	—	0.77	0.136
Cavity	—	—	—	0.18
Blockwork	125	0.933	0.11	1.136
Mortar (inner leaf)	125	0.067	0.88	0.142
Plaster	13	1.0	0.18	0.072
Inner surface (R_{si})	—	—	—	0.13

Using equation 3.11, these limiting values are averaged to give the thermal resistance of the bridged structure:

$$R_b = {}^1/_2\,(1.331 + 1.547) = 1.439\ m^2{\cdot}K{\cdot}W^{-1}$$

(*b*) Thermal transmittance of structure

The thermal transmittance of the whole structure is obtained from equation 3.16:

$$U = \frac{1}{{}^1/_2\,(1.331 + 1.547)} = 0.69\ W{\cdot}m^{-2}{\cdot}K^{-1}$$

For comparison, if the thermal bridging of the inner leaf by the mortar joints is ignored, the thermal resistance is the sum of the thermal resistances given in Table 3.11 minus that of the mortar joints. Hence, $U = 0.59\ W{\cdot}m^{-2}{\cdot}K^{-1}$.

Example 3.3: Calculation of U-value for timber frame wall

See Figure 3.6. The dimensions and thermal properties of the components of the wall are given in Table 3.12. Thermal bridging takes place at the inner leaf where the timber frame bridges the insulation. Isothermal planes are taken at the inside air and outside air. Since the thermal properties of the brick and mortar are similar, the outer leaf is regarded as homogenous.

The timber frame consists of studs, noggings, top and bottom plates and additional framing around windows and doors. For a typical wall with windows and doors, this represents about 15% of the surface area. (Note that for walls without windows and doors, the frame accounts for about 10% of the surface area while, for narrow walls with doors and windows or walls with bay windows, the frame may account for 20% of the surface area.)

(*a*) Mean thermal resistance of bridged structure

The lower limit of thermal resistance is obtained from equation 3.13:

$$R_L = 0.04 + 0.136 + 0.18 + 0.077 + \frac{1}{\left(\dfrac{0.85}{2.225} + \dfrac{0.15}{0.742}\right)} + 0.060 + 0.13$$

$$= 2.335\ m^2{\cdot}K{\cdot}W^{-1}$$

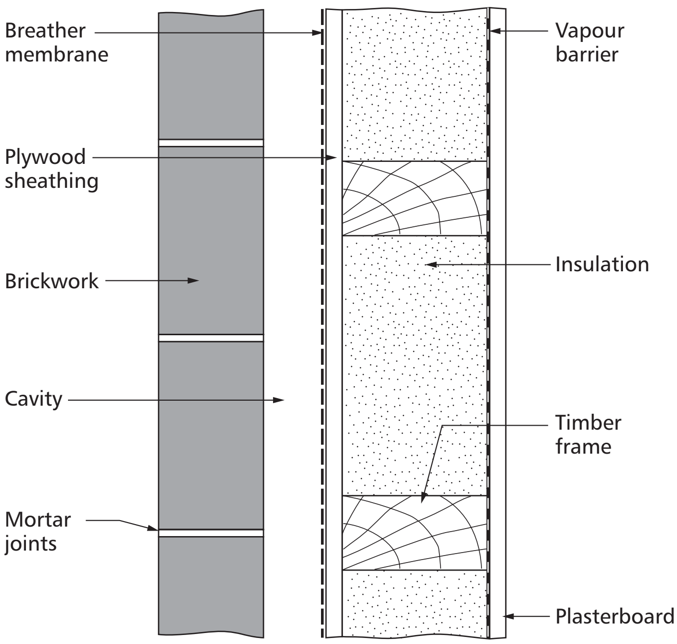


Figure 3.6 Example 3.3: timber frame wall construction

Table 3.12 Example 3.3: dimensions and thermal properties of materials

Element	Thickness (mm)	Proportion of surface area	Thermal conductivity / $W{\cdot}m^{-1}{\cdot}K^{-1}$	Thermal resistance / $m^2{\cdot}K{\cdot}W^{-1}$
Outer surface	—	—	—	0.04
External brickwork	105	—	0.77	0.136
Cavity	—	—	—	0.18
Plywood sheathing	10	—	0.13	0.077
Insulation between studs	89	0.85	0.04	2.225
Timber frame	89	0.15	0.13	0.742
Vapour control layer	—	—	—	—
Plasterboard	12.5	—	0.21	0.060
Inner surface	—	—	—	0.13

The upper limit of thermal resistance is obtained from equation 3.15:

$$R_U = \left(\frac{0.85}{0.04 + 0.136 + 0.18 + 0.077 + 2.225 + 0.060 + 0.13} + \frac{0.15}{0.04 + 0.136 + 0.18 + 0.077 + 0.742 + 0.060 + 0.13}\right)^{-1}$$

$$= 1 / (0.2985 + 0.1099) = 2.449\ \mathrm{m^2{\cdot}K{\cdot}W^{-1}}$$

Using equation 3.11, these limiting values are averaged to give the thermal resistance of the bridged structure:

$$R_b = {}^1/_2\,(2.334 + 2.449) = 2.392\ \mathrm{m^2{\cdot}K{\cdot}W^{-1}}$$

(*b*) Thermal transmittance of structure

The thermal transmittance of the whole structure is then obtained from equation 3.16:

$$U = \frac{1}{{}^1/_2\,(2.334 + 2.449)} = 0.42\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$$

3.3.11.4 Structural elements consisting of several layers, one or more of which is bridged

For structural elements consisting of several layers in close thermal contact and one or more of the layers is bridged, the isothermal planes will include all the layers. The lower limit is determined by adding together the thermal resistances of the various layers, each being calculated separately using equation 3.13. Determination of the upper limit is more complicated since this requires the division of each layer into isotropic sections, the thermal resistances of which are then added together on an area-weighted basis, as shown in equation 3.15. Appendix 3.A5 shows how this procedure may be applied to the case of a foam-filled masonry block.

3.3.11.5 Effect of bridging on calculated *U*-value

The relative effect of thermal bridging becomes increasingly important the greater the thickness of insulation that is interrupted by the thermal bridging. Table 3.13 illustrates the calculated effects of thermal bridging on the *U*-value on a timber frame wall. With 25 mm insulation the effect of the bridging is negligible, but may increase the *U*-value of the wall, compared with that of the centre panel, by up to 30% for 140 mm insulation.

3.3.12 Correction for air gaps, wall ties and roof fixings

3.3.12.1 Correction for air voids

Where relevant the *U*-value of a component should be corrected to allow for the effect of air voids:

$$U_c = U + \Delta U\,(R_I / R_T)^2 \qquad (3.17)$$

where U_c is the thermal transmittance corrected for thermal air gaps ($\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$), U is the (uncorrected) thermal transmittance of the element ($\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$), ΔU ($\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$) depends on the nature of the air voids as indicated below, R_I is the thermal resistance of the layer containing the air gaps ($\mathrm{m^2{\cdot}K{\cdot}W^{-1}}$), and R_T is the total thermal resistance of the element ($\mathrm{m^2{\cdot}K{\cdot}W^{-1}}$).

The values of ΔU are:

— $\Delta U = 0$, if there are no air voids

— $\Delta U = 0.01\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$, if there gaps bridging between the hot and cold side of the insulation (due to dimensional tolerances or the practicalities of fitting the insulation), but not causing air circulation between the warm and cold side of the insulation

— $\Delta U = 0.04\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$, if there are both gaps bridging the hot and face sides of the insulation, combined with cavities resulting in air circulation between the warm and cold sides of the insulation.

The value of $\Delta U = 0.01\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$ is usually appropriate if insulation is not continuous (e.g. fitted within a frame). Further guidance is given in BR 443[28] and Annex D of BS EN ISO 6946[15].

3.3.12.2 Correction for wall ties or roof fixings

Where a layer of insulation is penetrated by metal wall ties, roof fixings or similar, the *U*-value should be corrected as follows to account for thermal bridging by the ties/fixings[29]:

$$U_c = U + 0.8\,\lambda_f\,n_f\,A_f\,(R_I / R_T)^2 / d_f \qquad (3.18)$$

where U_c is the thermal transmittance corrected for thermal bridging by wall ties/roof fixings ($\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$), U is the (uncorrected) thermal transmittance of the element ($\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$), λ_f is the thermal conductivity of the tie/fixing ($\mathrm{W{\cdot}m{\cdot}K^{-1}}$), n_f is the number of ties/fixings per unit area ($\mathrm{m^{-2}}$), A_f is the cross-sectional area of a single tie/fixing ($\mathrm{m^2}$), R_I is the thermal resistance of the layer containing the fixings ($\mathrm{m^2{\cdot}K{\cdot}W^{-1}}$), and R_T is the total thermal resistance of the element ($\mathrm{m^2{\cdot}K{\cdot}W^{-1}}$).

Table 3.13 Effect of thermal bridging on *U*-value of timber frame wall with brick cladding using the combined method

Wall	Proportion of area bridged / %	*U*-value (/ $\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$) for stated thickness of timber (/ mm) and insulation (/ mm)				
		89 and 25	89 and 50	89 and 89	119 and 119	140 and 140
Centre panel	0	0.70	0.49	0.35	0.28	0.24
Wall with no openings	10	0.70	0.52	0.40	0.32	0.28
Typical wall with openings	15	0.71	0.53	0.42	0.34	0.30
Narrow wall (less than 5 m wide) with openings	20	0.71	0.55	0.45	0.36	0.32

No correction is required in the following cases:

- for wall ties across an empty cavity
- for wall ties between a masonry leaf and timber studs
- where the thermal conductivity of the tie/fixing is less than 1 $W \cdot m^{-1} \cdot K^{-1}$.

Where both ends of the tie/fixing are in contact with metal sheets its effect should be assessed by numerical analysis using the criteria in BS EN ISO 10211[30].

3.4 Roofs

3.4.1 Pitched roofs

For simple pitched roofs, the *U*-value is calculated normal to the plane of the roof. However, if the pitched roof includes a horizontal ceiling and an unheated loft, the *U*-value is defined with respect to the plane of the ceiling.

For heated lofts, the *U*-value of the roof is calculated in the normal way. However, in calculating the heat losses, the full surface area of the roof must be considered rather than the plan area.

Note that Building Regulations require that roof voids be ventilated. Standardised thermal resistances for loft spaces are given in Table 3.5.

3.4.2 Flat roofs

The thermal transmittance of flat roofs is calculated using the methods given in sections 3.3.10 and 3.3.11.

Where components are tapered to achieve a fall, the average thickness may be used in calculating the thermal transmittance of the roof. If the main insulation layer is tapered, the *U*-value should be calculated at intervals across the roof and the average determined. Alternatively, the method given in Annex C of BS EN ISO 6946[15] may be used.

3.5 Ground floors and basements

3.5.1 General

The heat loss through floors in contact with the ground is more complicated than that through above ground components. The heat flow is three-dimensional and the thermal performance is affected by various factors including the size and shape of the floor, the thickness of the surrounding wall and the presence of all-over or edge insulation. However, research has shown that the building dimensions affect the *U*-values of ground floors predominantly through the ratio of exposed perimeter of the floor to its area[31,32]. This allows the *U*-value of a ground floor to be readily evaluated for a floor of any size or shape.

The information on ground floors is consistent with BS EN ISO 13370[33] and provides data for different soil types.

It should be noted that the heat flux density varies over the area of a floor, in general being greatest at the edges and least at the centre. The thermal transmittance of a floor is suitable for calculating the total heat transfer through the floor but cannot be used to obtain the heat flux density at a point on the floor surface or to calculate the surface temperature of the floor.

3.5.2 Thermal transmittance of solid ground floors

Tables 3.15, 3.16 and 3.17 give *U*-values for solid ground floors in contact with the earth, calculated using equations 3.19 to 3.22, for the three types of soil in Table 3.14. The tables assume a wall thickness of 0.3 m. The *U*-values are given as a function of the ratio of exposed perimeter to floor area and the thermal resistance of the floor construction, R_f (R_f = 0 for an uninsulated floor). Linear interpolation may be used for values intermediate between those given in the tables.

Table 3.14 Thermal conductivity of soils

Soil type	Thermal conductivity, λ_g / $W \cdot m^{-1} \cdot K^{-1}$
Clay or silt	1.5
Sand or gravel	2.0
Homogeneous rock	3.5

The *U*-value of a solid floor in contact with the ground, see Figure 3.7, depends on a 'characteristic dimension' of the floor, B', and the 'total equivalent thickness', d_{ef}, of the factors that, in combination, restrict the heat flow (i.e. wall thickness, surface resistances, thermal insulation).

The characteristic dimension is defined thus:

$$B' = \frac{A_{fg}}{0.5\, p_f} \tag{3.19}$$

where B' is the characteristic dimension of the floor (m), A_{fg} is the area of floor in contact with the ground (m^2) and p_f is the exposed perimeter of the floor (m).

The total equivalent thickness is given by:

$$d_{ef} = d_w + \lambda_g (R_{si} + R_f + R_{se}) \tag{3.20}$$

where d_{ef} is the total equivalent thickness of the floor (m), d_w is the thickness of the wall surrounding the ground floor (m), λ_g is the thermal conductivity of the ground ($W \cdot m^{-1} \cdot K^{-1}$), is the inside surface resistance ($m^2 \cdot K \cdot W^{-1}$), is the external surface resistance ($m^2 \cdot K \cdot W^{-1}$) and R_f is the thermal resistance of the floor ($m^2 \cdot K \cdot W^{-1}$). For calculation of heat losses through floors, R_{si} and R_{se} take values of 0.17 $m^2 \cdot K \cdot W^{-1}$ and 0.04 $m^2 \cdot K \cdot W^{-1}$ respectively .

For an infinitely long floor, B' is the width of the floor and, at the other limiting case of a square floor, B' is half the length of one side.

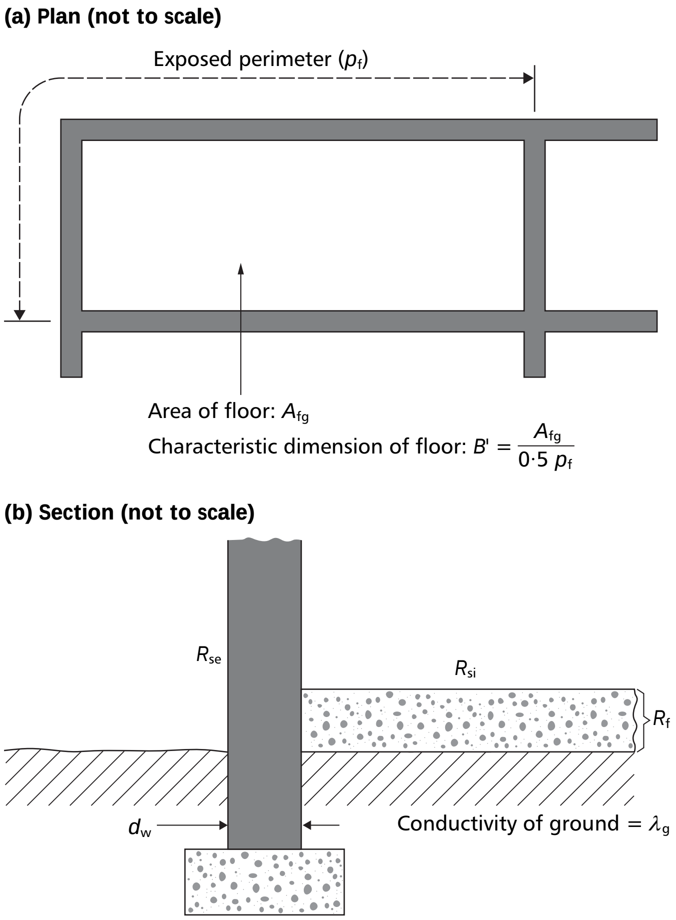


Figure 3.7 Characteristic dimension and total equivalent thickness of solid ground floor

R_f includes the thermal resistance of any all-over insulation layers above, below or within the floor slab, and that of any floor covering. The thermal resistance of dense concrete slabs and thin floor coverings may normally be neglected. Hardcore below the slab is assumed to have the same thermal conductivity as the ground and its thermal resistance is therefore not included.

For $d_{ef} < B'$, as is usually the case, the thermal transmittance is given by:

$$U_f = \frac{2\,\lambda_g}{\pi\,B' + d_{ef}} \log_e [(\pi\,B' / d_{ef}) + 1] \tag{3.21}$$

If $d_{ef} \geq B'$, as may occur for a small, well-insulated floor:

$$U_f = \lambda_g / (0.457\,B' + d_{ef}) \tag{3.22}$$

The exposed perimeter, p_f, should be interpreted as the total length of the external wall separating the heated building from the external environment or from an unheated space outside the insulated fabric. Thus for a complete building p_f is the total perimeter of the building and A_{fg} is its total ground-floor area.

To calculate the heat loss from part of a building (e.g. for each individual dwelling in a row of terraced houses), p_f includes the lengths of external walls separating the heated space from the external environment but excludes the lengths of party walls separating the part under consideration from other spaces heated to a similar internal temperature. In such cases, A_{fg} is the ground-floor area for the part of the building under consideration.

Unheated spaces outside the insulated fabric of the building (e.g. porches, attached garages or storage areas) are excluded when determining p_f and A_{fg}, but the length of the wall between the building and the unheated space must be included in the perimeter. The ground heat losses are assessed as if the unheated spaces were not present.

The thermal conductivity of the ground, λ_g, depends on several factors including density, degree of water saturation, particle size, type of mineral constituting the particles and whether the ground is frozen or unfrozen. Consequently, the thermal properties vary from one location to another and at different depths at any particular location. They may also vary with time due to

Table 3.15 *U*-values for solid ground floors on clay soil (λ_g = 1.5 W·m^{-1}·K^{-1})

Ratio p_f / A_{fg}	*U*-value (/ W·m^{-2}·K^{-1}) for stated thermal resistance, R_f (/ m^2·K.W^{-1})			
	0	0.5	1.0	2.0
0.05	0.13	0.11	0.10	0.08
0.10	0.22	0.18	0.16	0.13
0.15	0.30	0.24	0.21	0.17
0.20	0.37	0.29	0.25	0.19
0.25	0.44	0.34	0.28	0.22
0.30	0.49	0.38	0.31	0.23
0.35	0.55	0.41	0.34	0.25
0.40	0.60	0.44	0.36	0.26
0.45	0.65	0.47	0.38	0.27
0.50	0.70	0.50	0.40	0.28
0.55	0.74	0.52	0.41	0.28
0.60	0.78	0.55	0.43	0.29
0.65	0.82	0.57	0.44	0.30
0.70	0.86	0.59	0.45	0.30
0.75	0.89	0.61	0.46	0.31
0.80	0.93	0.62	0.47	0.32
0.85	0.96	0.64	0.47	0.32
0.90	0.99	0.65	0.48	0.32
0.95	1.02	0.66	0.49	0.33
1.00	1.05	0.68	0.50	0.33

Table 3.16 *U*-values for solid ground floors on sand or gravel (λ_g = 2.0 W·m^{-1}·K^{-1})

Ratio p_f / A_{fg}	*U*-value (/ W·m^{-2}·K^{-1}) for stated thermal resistance, R_f (/ m^2·K.W^{-1})			
	0	0.5	1.0	2.0
0.05	0.16	0.14	0.12	0.10
0.10	0.28	0.22	0.19	0.16
0.15	0.38	0.30	0.25	0.20
0.20	0.47	0.36	0.30	0.23
0.25	0.55	0.41	0.33	0.25
0.30	0.63	0.46	0.37	0.26
0.35	0.70	0.50	0.39	0.28
0.40	0.76	0.53	0.42	0.29
0.45	0.82	0.56	0.43	0.30
0.50	0.88	0.59	0.45	0.31
0.55	0.93	0.62	0.47	0.31
0.60	0.98	0.64	0.48	0.32
0.65	1.03	0.66	0.49	0.33
0.70	1.07	0.68	0.50	0.33
0.75	1.12	0.70	0.51	0.34
0.80	1.16	0.72	0.52	0.34
0.85	1.19	0.73	0.53	0.35
0.90	1.23	0.75	0.54	0.35
0.95	1.27	0.76	0.54	0.35
1.00	1.30	0.77	0.55	0.35

Table 3.17 U-values for solid ground floors on homogeneous rock (λ_g = 3.5 W·m^{-1}·K^{-1})

Ratio p_f / A_{fg}	*U*-value (/ W·m^{-2}·K^{-1}) for stated thermal resistance, R_f (/ m^2·K.W^{-1})			
	0	0.5	1.0	2.0
0.05	0.27	0.21	0.18	0.15
0.10	0.45	0.34	0.28	0.22
0.15	0.61	0.43	0.35	0.26
0.20	0.74	0.51	0.40	0.28
0.25	0.86	0.58	0.44	0.30
0.30	0.97	0.63	0.47	0.32
0.35	1.07	0.68	0.50	0.33
0.40	1.16	0.72	0.52	0.34
0.45	1.25	0.75	0.53	0.35
0.50	1.33	0.78	0.55	0.35
0.55	1.40	0.80	0.56	0.36
0.60	1.47	0.82	0.58	0.37
0.65	1.53	0.84	0.59	0.37
0.70	1.59	0.86	0.60	0.37
0.75	1.64	0.87	0.61	0.38
0.80	1.69	0.89	0.62	0.38
0.85	1.74	0.91	0.62	0.38
0.90	1.79	0.92	0.63	0.39
0.95	1.83	0.93	0.64	0.39
1.00	1.87	0.95	0.64	0.39

changes in moisture content or due to freezing and thawing. Table 3.14 gives representative values of λ_g for three broad categories of ground. Clay-type soils are the most prevalent in Britain and should be assumed in the absence of more specific information.

3.5.3 Heat losses through ground floors

Heat losses through ground floors are affected by the large mass of earth in thermal contact with the floor. A full treatment of heat losses through ground floors would need to take account of the steady-state component, related to annual average temperature, and the annual periodic component resulting from the annual cyclical variation of inside and outside temperatures. Such an approach allows the determination of the ground losses for each month of the year, leading to the peak loss for design purposes and the appropriate average over the heating season for energy calculations.

In practice, for the purposes of calculating plant sizes, it has been usual to apply the same temperature difference to all the elements of the structure, including the floor. This tends to overestimate the heat losses through the floor.

For seasonal energy calculations, a more accurate assessment of the ground losses is obtained using the annual average temperature difference between inside and outside, rather than the average difference over the heating season. This is discussed in Appendix 3.A4.

Example 3.4: Calculation of U-value for L-shaped solid ground floor

Figure 3.8 shows a solid ground floor on clay-type soil with surrounding wall of thickness 300 mm.

The length of perimeter of floor is:

$$p_f = 30 + 30 + 20 + 24 + 10 + 6 = 120 \text{ m}$$

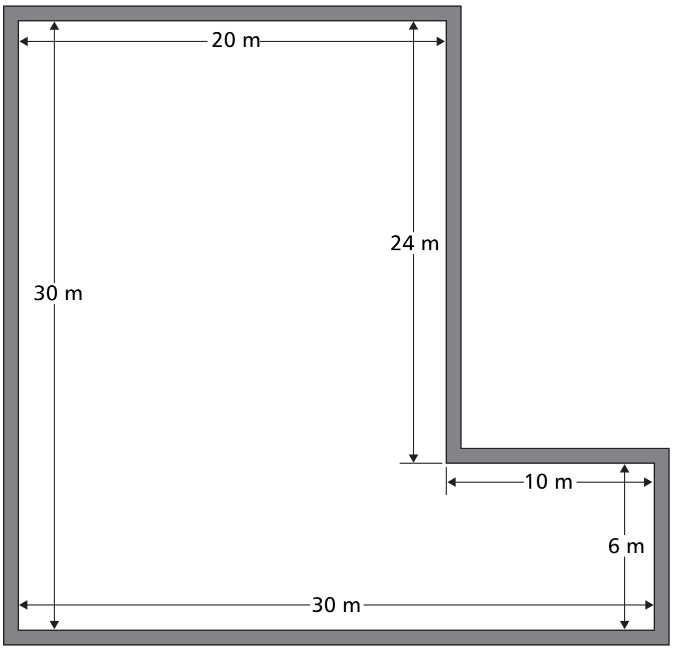


Figure 3.8 Example 3.4: plan of solid ground floor

The area of the floor in contact with the ground is:

$$A_{fg} = (20 \times 30) + (10 \times 6) = 660 \text{ m}^2$$

Hence:

$$B' = 660 / (0.5 \times 120) = 11.0 \text{ m}$$

For an uninsulated floor $R_f = 0$. Therefore:

$$d_{ef} = 0.3 + 1.5\,(0.17 + 0.04) = 0.62 \text{ m}$$

Since d_{ef} is less than B', using equation 3.21:

$$U_f = \frac{2 \times 1.5}{3.142 \times 11.0 + 0.62} \log_e\left(\frac{3.142 \times 11.0}{0.62} + 1\right)$$

$$= 0.34 \text{ W·m}^{-2}\text{·K}^{-1}$$

Example 3.5: Calculation of the U-value for solid ground floor for semi-detached dwelling with attached unheated garage

The plan is shown in Figure 3.9. The wall thickness is 300 mm, and insulation of thermal resistance 0.7 $\text{m}^2\text{·K·W}^{-1}$ is applied to the floor.

The exposed perimeter is:

$$p_f = 6 + 8 + 6 = 20 \text{ m}$$

The area of floor in contact with the ground is:

$$A_{fg} = 6 \times 8 = 48 \text{ m}^2$$

Hence:

$$B' = 48 / (0.5 \times 20) = 4.8 \text{ m}$$

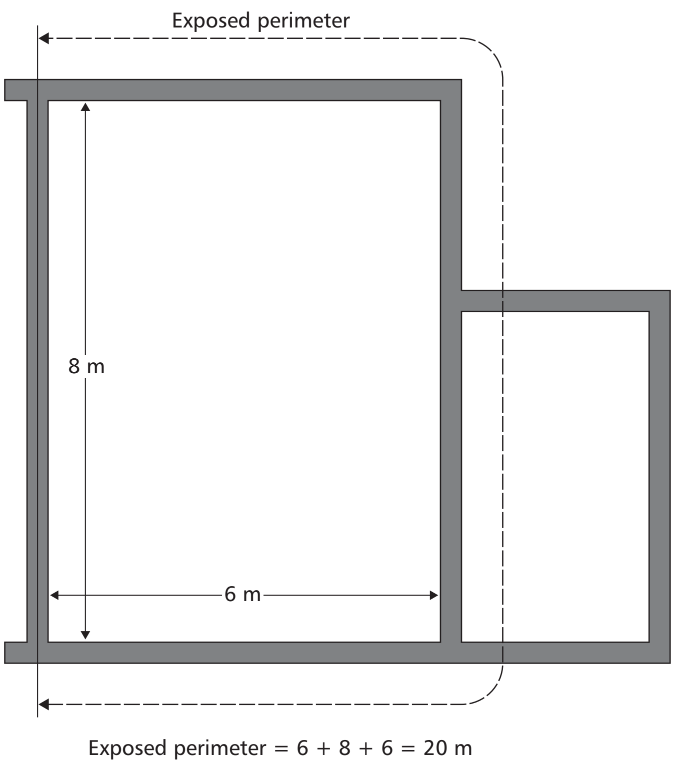


Figure 3.9 Example 3.5: plan of solid ground floor

$$d_{ef} = 0.3 + 1.5\,(0.17 + 0.7 + 0.04) = 1.67\ \text{m}$$

Again, d_{ef} is less than B', so using equation 3.20:

$$U_f = \frac{2 \times 1.5}{3.142 \times 4.8 + 1.67} \log_e \left(\frac{3.142 \times 4.8}{1.67} + 1 \right)$$

$$= 0.41\ \text{W·m}^{-2}\text{·K}^{-1}$$

3.5.4 Thermal transmittance of solid ground floors with edge insulation

A slab-on-ground floor may be insulated by means of edge insulation placed either horizontally or vertically round the perimeter of the floor, see Figures 3.10 and 3.11. The following equations are valid provided that no significant thermal bridging is introduced(34).

The U-value of an edge-insulated floor is given by:

$$U_{fi} = U_f + 2\,\Psi_{fi} / B' \tag{3.23}$$

where U_{fi} is the thermal transmittance of the edge-insulated floor (W·m^{-2}·K^{-1}), U_f is the thermal transmittance of the same floor without insulation (W·m^{-2}·K^{-1}), Ψ_{fi} is a factor related to the floor edge insulation (W·m^{-1}·K^{-1}) and B' is the characteristic dimension of the floor (m), see equation 3.19.

The edge insulation factor, Ψ_{fi}, depends on the thermal resistance of the edge insulation, on whether the edge insulation is placed horizontally or vertically and on its width (if horizontal) or depth (if vertical). Low-density foundations, of thermal conductivity less than that of the soil, are treated as vertical edge insulation.

The equations for the determination of the edge factor depend upon the additional equivalent thickness resulting from the edge insulation, d_{ei}', i.e:

$$d_{ei}' = R_i'\,\lambda_g \tag{3.24}$$

where d_{ei}' is the additional equivalent thickness of the floor due to edge insulation (m), R_i' is the additional thermal resistance (m^2·K·W^{-1}) due to edge insulation (or foundation) and λ_g is the thermal conductivity of the ground (W·m^{-1}·K^{-1}).

R_i' is the difference between the thermal resistance of the edge insulation and that of the soil (or slab) it replaces, i.e:

$$R_i' = R_{fi} - (d_{fi} / \lambda_g) \tag{3.25}$$

where R_{fi} is the thermal resistance of the floor edge insulation (or foundation) (m^2·K·W^{-1}), d_{fi} is the thickness of the floor edge insulation (or foundation) (m) and λ_g is the thermal conductivity of the ground (W·m^{-1}·K^{-1}).

3.5.4.1 Horizontal edge insulation

For insulation placed horizontally round the perimeter of the floor, see Figure 3.10, the following equation applies:

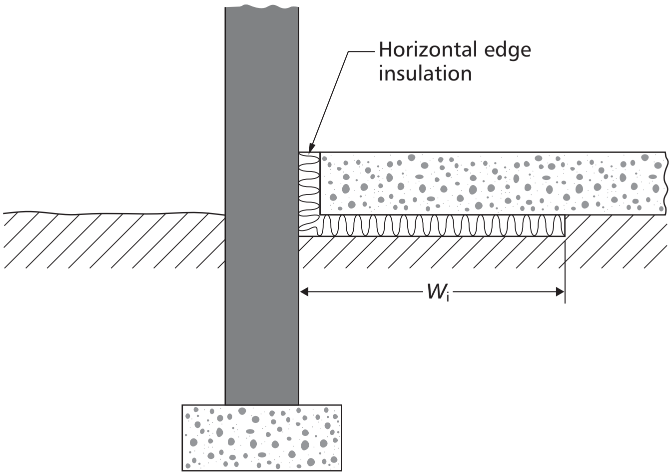


Figure 3.10 Solid ground floor with horizontal edge insulation

$$\Psi_{fi} = -\frac{\lambda_g}{\pi}\left[\log_e\left(\frac{W_i}{d_{ef}} + 1\right) - \log_e\left(\frac{W_i}{d_{ef} + d_{ei}'} + 1\right)\right] \tag{3.26}$$

where Ψ_{fi} is a factor related to the floor edge insulation (W·m^{-1}·K^{-1}), λ_g is the thermal conductivity of the ground (W·m^{-1}·K^{-1}), W_i is the width of the horizontal edge (floor) insulation (m), d_{ef} is the total equivalent thickness of the floor (in the absence of edge insulation) (m) and d_{ei}' is the additional equivalent thickness of the floor due to edge insulation (m).

Values of Ψ_{fi} for floors with horizontal edge insulation only are given in Table 3.18. The wall thickness is taken as 0.3 m.

Table 3.18 Edge insulation factor, Ψ_{fi}, for horizontal edge insulation

Soil type	Width of horizontal edge (floor) insulation, W_i / m	Edge insulation factor, Ψ_{fi} / W·m^{-1}·K^{-1}, for stated additional thermal resistance value, R_i' / m^2·K·W^{-1}: 0.5	1.0	1.5	2.0
Clay/silt	0.50	–0.13	–0.18	–0.21	–0.22
	1.00	–0.20	–0.27	–0.32	–0.34
	1.50	–0.23	–0.33	–0.39	–0.42
Sand/gravel	0.50	–0.17	–0.23	–0.25	–0.27
	1.00	–0.26	–0.35	–0.40	–0.43
	1.50	–0.31	–0.43	–0.50	–0.54
Homogeneous rock	0.50	–0.25	–0.32	–0.35	–0.37
	1.00	–0.41	–0.53	–0.59	–0.62
	1.50	–0.52	–0.68	–0.76	–0.81

3.5.4.2 Vertical edge insulation

For insulation placed vertically below ground around the perimeter of the floor, see Figure 3.11, and for foundations of material of lower thermal conductivity than the ground, see Figure 3.12, the following equation applies:

$$\Psi_{fi} = -\frac{\lambda_g}{\pi}\left[\log_e\left(\frac{2\,D_i}{d_{ef}} + 1\right) - \log_e\left(\frac{2\,D_i}{d_{ef} + d_{ei}'} + 1\right)\right] \tag{3.27}$$

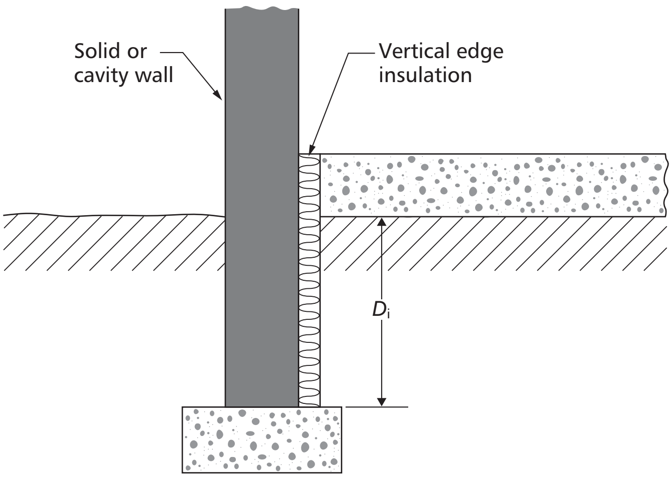


Figure 3.11 Solid ground floor with vertical edge insulation

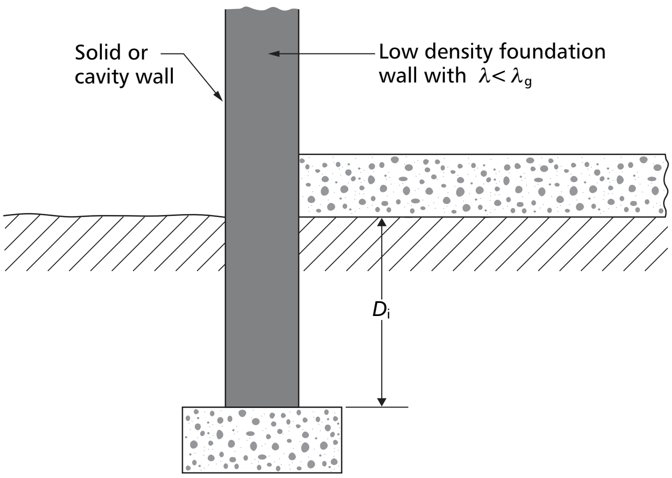


Figure 3.12 Solid ground floor with foundation wall having thermal conductivity less than that of the ground

where D_i is the depth of the vertical edge (floor) insulation (m), λ_g is the thermal conductivity of the ground (W·m^{-1}·K^{-1}), d_{ef} is the total equivalent thickness of the floor (in the absence of edge insulation) (m) and d_{ei}' is the additional equivalent thickness of the floor due to edge insulation (m).

Values of Ψ_{fi} for floors with vertical edge insulation only are given in Table 3.19. The wall thickness is taken as 0.3 m.

Table 3.19 Edge insulation factor, Ψ_{fi}, for vertical edge insulation

Soil type	Depth of vertical edge (floor) insulation, D_i / m	Edge insulation factor, Ψ_{fi} / W·m^{-1}·K^{-1}, for stated additional thermal resistance value, R_i' / m^2·K·W^{-1}			
		0.5	1.0	1.5	2.0
Clay/silt	0.25	–0.13	–0.18	–0.21	–0.22
	0.50	–0.20	–0.27	–0.32	–0.34
	0.75	–0.23	–0.33	–0.39	–0.42
	1.00	–0.26	–0.37	–0.43	–0.48
Sand/gravel	0.25	–0.17	–0.23	–0.25	–0.27
	0.50	–0.26	–0.35	–0.40	–0.43
	0.75	–0.31	–0.43	–0.50	–0.54
	1.00	–0.35	–0.49	–0.57	–0.62
Homogeneous rock	0.25	–0.25	–0.32	–0.35	–0.37
	0.50	–0.41	–0.53	–0.59	–0.62
	0.75	–0.52	–0.68	–0.76	–0.81
	1.00	–0.59	–0.79	–0.89	–0.95

3.5.5 Thermal transmittance of suspended ground floors

A suspended floor is any type of floor not in contact with the ground, e.g. timber or concrete beam-and-block floors, see Figure 3.13.

Heat is transferred through a suspended floor to the under floor space, from which it is then transferred to the external environment by three mechanisms:

— through the ground
— through the wall of the under floor space
— by ventilation of the under floor space.

The thermal transmittance, allowing for the combination of these mechanisms, is given by:

$$U_{fs} = [(1 / U_f) + 1 / (U_{fg} + U_{eu})]^{-1} \qquad (3.28)$$

where U_{fs} is the combined thermal transmittance of the uninsulated suspended floor structure (W·m^{-2}·K^{-1}), U_f is the thermal transmittance of the (uninsulated) floor (W·m^{-2}·K^{-1}), U_{fg} is the thermal transmittance for heat flow through the ground (W·m^{-2}·K^{-1}) and U_{eu} is the equivalent thermal transmittance for heat flow through the walls surrounding the underfloor space and by ventilation of the underfloor space (W·m^{-2}·K^{-1}).

U_f may be determined by the methods given in section 3.3.10 or 3.3.11, taking the surface resistances on both sides of the floor as 0.17 m^2·K·W^{-1}.

The thermal transmittance for heat flow through the ground, U_{fg}, should be calculated as for a solid ground floor, using equation 3.21 or 3.22 as appropriate, but substituting a total equivalent thickness for the ground, d_{eg}, in place of d_{ef}. This equivalent thickness is given by:

$$d_{eg} = d_w + \lambda_g (R_{si} + R_{ig} + R_{se}) \qquad (3.29)$$

where d_{eg} is the total equivalent thickness of the ground (m), d_w is the thickness of wall surrounding the ground floor (m), λ_g is the thermal conductivity of the ground (W·m^{-1}·K^{-1}), R_{si} is the internal surface resistance (m^2·K·W^{-1}), R_{ig} is the thermal resistance of the insulation between the floor and the ground (m^2·K·W^{-1}) and R_{se} is the external surface resistance (m^2·K·W^{-1}).

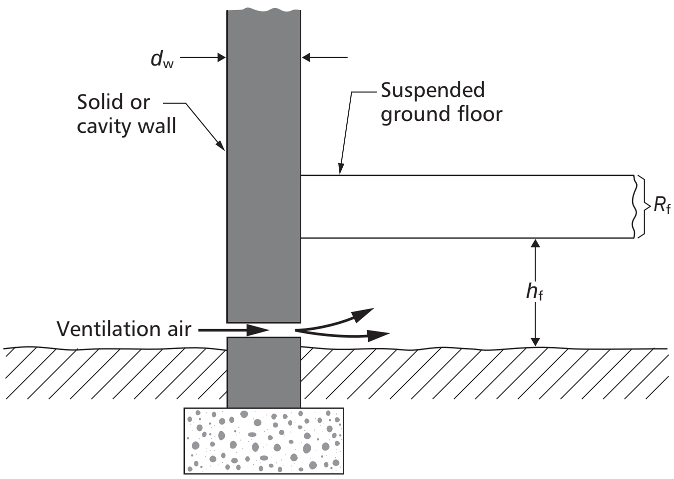


Figure 3.13 Schematic diagram of suspended ground floor

The general expression for U_{eu} is:

$$U_{eu} = (2\, h_f\, U_u / B') + 1450\, \alpha\, v_w\, f_w / B' \qquad (3.30)$$

where U_{eu} is the equivalent thermal transmittance for heat flow through the walls surrounding the underfloor space and by ventilation of the underfloor space ($W{\cdot}m^{-2}{\cdot}K^{-1}$), h_f is the height of the floor above external ground level (m), U_u is the thermal transmittance of the walls surrounding the underfloor space above ground level ($W{\cdot}m^{-2}{\cdot}K^{-1}$), B' is the characteristic dimension of the floor (m) (see equation 3.19), α is the area of ventilation openings per unit perimeter of underfloor space (m^2), v_w is the average wind speed at 10 m height above ground level ($m{\cdot}s^{-1}$) and f_w is the wind shielding factor.

If h_f varies around the perimeter of the floor, the average value should be used.

The wind shielding factor relates the wind speed at 10 m height above ground level (assumed unobstructed) to that near ground level, allowing for shielding by adjacent buildings etc. Representative values are as follows:

— sheltered location (city centre): $f_w = 0.02$

— average location (suburban): $f_w = 0.05$

— exposed location (rural): $f_w = 0.10$.

3.5.5.1 Uninsulated suspended floors

Table 3.20 gives *U*-values for uninsulated suspended floors for the following values of the relevant parameters:

— thermal resistance of floor: $R_f = 0.2\ m^2{\cdot}K{\cdot}W^{-1}$

— ventilation opening: $\alpha = 0.0015$ or $0.003\ m^2{\cdot}m^{-1}$

— average wind velocity: $v_w = 3\ m{\cdot}s^{-1}$

— wind shielding factor (average exposure): $f_w = 0.05$

— uninsulated under floor walls: $U_u = 1.7\ W{\cdot}m^{-2}{\cdot}K^{-1}$

— height of floor above external ground level: $h_f = 0.5$ m.

Table 3.20 may be used in most cases to obtain the *U*-value of a suspended floor. However, if the parameters of the actual design differ significantly from the above values, the *U*-value should be calculated using the equations given above.

3.5.5.2 *U*-values of insulated suspended floors

For floors having thermal resistances other than $0.2\ m^2{\cdot}K{\cdot}W^{-1}$, the *U*-value can be obtained from:

$$U_{fsi} = [(1 / U_{fs}) - 0.2 + R_f]^{-1} \qquad (3.31)$$

where U_{fsi} is the thermal transmittance of the insulated suspended floor ($W{\cdot}m^{-2}{\cdot}K^{-1}$), U_{fs} is the combined thermal transmittance of the uninsulated floor (obtained from Table 3.20) ($W{\cdot}m^{-2}{\cdot}K^{-1}$) and R_f is the thermal resistance of the floor excluding surface resistances ($m^2{\cdot}K{\cdot}W^{-1}$).

Example 3.6: U-value for insulated suspended timber floor

A detached building has a ground floor plan of 9.5 m by 8.2 m with a suspended timber floor consisting of 19 mm chipboard on joists (50 mm by 100 mm at 400 mm centres) with 100 mm of insulation between the joists. The thermal conductivity of the insulation is $0.04\ W{\cdot}m^{-1}{\cdot}K^{-1}$ and the thermal conductivity of both the chipboard and the joists is $0.14\ W{\cdot}m^{-1}{\cdot}K^{-1}$. The area of the ventilation openings per unit perimeter length is $0.0015\ m^2{\cdot}m^{-1}$. The floor is over clay-type soil and of average exposure.

The perimeter is 35.4 m and the plan area of the ground floor is 77.9 m². Thus $(p_f / A_{fg}) = 0.45\ m^{-1}$. From Table 3.20, the *U*-value for an uninsulated suspended floor of this size is $0.69\ W{\cdot}m^{-2}{\cdot}K^{-1}$.

Table 3.20 *U*-values for uninsulated suspended floors

Ratio p_f / A_{fg} ($/ m^{-1}$)	*U*-value ($/ W{\cdot}m^{-2}{\cdot}K^{-1}$) for stated soil type and ventilation opening, α ($/ m^2{\cdot}m^{-1}$)					
	Clay/silt		Sand/gravel		Homogeneous rock	
	0.0015	0.003	0.0015	0.003	0.0015	0.003
0.05	0.16	0.17	0.19	0.20	0.27	0.28
0.10	0.27	0.29	0.32	0.33	0.43	0.44
0.15	0.36	0.38	0.42	0.43	0.54	0.55
0.20	0.44	0.46	0.49	0.51	0.63	0.64
0.25	0.50	0.52	0.56	0.58	0.70	0.71
0.30	0.56	0.58	0.62	0.64	0.76	0.77
0.35	0.61	0.63	0.67	0.69	0.81	0.82
0.40	0.65	0.68	0.72	0.74	0.85	0.87
0.45	0.69	0.72	0.76	0.78	0.89	0.91
0.50	0.73	0.76	0.79	0.82	0.92	0.94
0.55	0.76	0.79	0.83	0.85	0.95	0.97
0.60	0.79	0.83	0.86	0.88	0.98	1.00
0.65	0.82	0.85	0.88	0.91	1.00	1.02
0.70	0.85	0.88	0.91	0.94	1.03	1.05
0.75	0.87	0.91	0.93	0.96	1.05	1.07
0.80	0.90	0.93	0.95	0.98	1.06	1.09
0.85	0.92	0.95	0.97	1.00	1.08	1.11
0.90	0.94	0.97	0.99	1.02	1.10	1.12
0.95	0.96	0.99	1.01	1.04	1.11	1.14
1.00	0.98	1.01	1.03	1.06	1.13	1.15

The thermal resistance of the floor is the sum of the thermal resistance of the homogeneous chipboard and the combined thermal resistance of the joists and intervening insulation.

The thermal resistance of the insulation, bridged by the joists, is given by:

$$R_b = [(P_m / R_m) + (P_n / R_n)]^{-1}$$

The proportions of surface area and thermal resistances are:

— insulation: $P_m = (350 / 400)$, $R_m = (0.1 / 0.04)$

— joists: $P_n = (50 / 400)$, $R_n = (0.1 / 0.14)$

Therefore, the thermal resistance of the floor is:

$$R_f = \frac{0.019}{0.14} + \left(\frac{350}{400} \times \frac{0.04}{0.1} + \frac{50}{400} \times \frac{0.14}{0.1}\right)^{-1}$$

$$= 2.04\ \mathrm{m^2{\cdot}K{\cdot}W^{-1}}$$

Hence, using equation 3.31:

$$U_{fsi} = [(1 / 0.69) - 0.2 + 2.04]^{-1} = 0.3\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$$

3.5.6 Thermal transmittance of basement floors and walls

This section gives data for the thermal transmittance of basement floors and basement walls. The *U*-value for the floor depends upon the ratio of the perimeter of the floor to its area, and on the depth of the basement floor below ground level. The *U*-value for the walls depends on the depth of the basement and the properties of the materials used in the wall construction.

A *U*-value may also be defined for the basement as a whole, as follows:

$$U_b = \frac{A_b\, U_{bf} + h_b\, p_{bf}\, U_{bw}}{A_b + h_b\, p_{bf}} \tag{3.32}$$

where U_b is the average thermal transmittance of the basement ($\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$), A_b is the area of the basement floor ($\mathrm{m^2}$), U_{bf} is the thermal transmittance of the basement floor ($\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$), h_b is the depth of the basement below ground level (m), p_{bf} is the perimeter of the basement (m) and U_{bw} is the thermal transmittance of the basement wall ($\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$). Values of U_{bf} and U_{bw} are given in Tables 3.21 and 3.22, respectively.

Sometimes a more precise value may be required, e.g. when comparing thermal transmittances to determine the optimum thickness of insulation. In such cases the more detailed method described in BS EN ISO 13370[33] may be used.

The basement floor area, A_b, is measured between the finished internal faces of the walls bounding the basement. The perimeter, p_b, is measured along the finished internal faces, The basement depth, h_b, is measured between the outside ground level and the finished internal surface of the basement floor, see Figure 3.14(a). The depth, *h*, will often be less than the internal height of the basement storey, in which case the *U*-value obtained by equation 3.32 will apply to the floor and the area of basement wall below ground level. Any wall above ground level should be assessed using the methods for walls given in sections 3.3.10 and 3.3.11.

Table 3.21 *U*-values for uninsulated basement floors

Ratio, p_b / A_b (/ $\mathrm{m^{-1}}$)	*U*-value of basement floor, U_{bf} (/ $\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$), for stated basement depth, h_b (/ m)				
	0.5	1.0	1.5	2.0	2.5
0.1	0.20	0.19	0.18	0.17	0.16
0.2	0.34	0.31	0.29	0.27	0.26
0.3	0.44	0.41	0.38	0.35	0.33
0.4	0.53	0.48	0.44	0.41	0.38
0.5	0.61	0.55	0.50	0.46	0.43
0.6	0.68	0.61	0.55	0.50	0.46
0.7	0.74	0.65	0.59	0.53	0.49
0.8	0.79	0.70	0.62	0.56	0.51
0.9	0.84	0.73	0.65	0.58	0.53
1.0	0.89	0.77	0.68	0.60	0.54

Table 3.22 *U*-values for basement walls

Thermal resistance of basement walls, R_{bw} (/ $\mathrm{m^2.K.W^{-1}}$)	*U*-value of basement walls, U_{bw} (/ $\mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$), for stated basement depth, h_b (/ m)				
	0.5	1.0	1.5	2.0	2.5
0.2	1.55	1.16	0.95	0.81	0.71
0.5	0.98	0.78	0.66	0.58	0.52
1.0	0.61	0.51	0.45	0.40	0.37
2.0	0.35	0.30	0.27	0.25	0.24
2.5	0.28	0.25	0.23	0.21	0.20

To obtain a *U*-value for split-level basements and basements on sloping sites the average depth of the basement below ground level should be used, averaged around its perimeter. For the simple case shown in Figure 3.14(b):

$$h_b = (h_{b1} + h_{b2}) / 2 \tag{3.33}$$

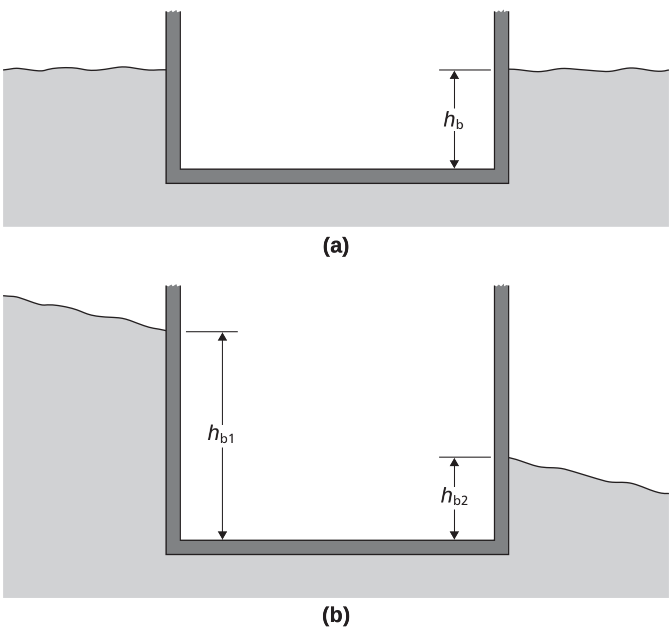


Figure 3.14 Depth of basement; (a) building on level site, (b) building on sloping site

3.5.6.1 Uninsulated basement floors

Table 3.21 gives U-values for uninsulated basement floors in terms of the ratio of basement perimeter to floor area, p_b / A_b, and the depth of the basement, h_b. Linear interpolation may be used to determine intermediate values.

3.5.6.2 Insulated basement floors

An approximate U-value for insulated basement floors may be obtained by using Table 3.21 to determine the value for an uninsulated basement floor of the same dimensions and modifying the value obtained as follows:

$$U_{bi} = [(1 / U_b) + R_{fbi}]^{-1} \tag{3.34}$$

where U_{bi} is the thermal transmittance of insulated basement floor (W·m^{-2}·K^{-1}), U_b is the thermal transmittance of uninsulated basement floor of same dimensions (W·m^{-2}·K^{-1}) and R_{fbi} is the thermal resistance of the insulation layers incorporated into basement floor (m^2·K·W^{-1}).

The thermal resistance, R_{fbi}, should not include the thermal resistance of the floor slab or any surface resistances since these are already incorporated into the U-value for the uninsulated basement floor, U_b. If the insulation has bridged layers, R_{fbi} should be calculated using the method given in section 3.3.11.

3.5.6.3 Basement walls

Table 3.22 gives U-values for basement walls as a function of the basement depth, h_b, and the thermal resistance of the basement walls, R_{bw}. The inside and outside surface resistances are taken into account in the tabulated U-values and are therefore not to be included when calculating R_{bw}.

If any insulating layers are bridged, e.g. by timber studding or mortar joints in the case of low-density concrete blockwork, the overall thermal resistance should be calculated using the combined method, see section 3.3.11.2.

3.6 Windows

3.6.1 General

The thermal transmittance of windows is made up of three components:

— centre-pane U-value of the glazing

— frame or sash

— interaction between glazing and frame, include the effect of the glazing spacer bars in multiple glazing (see section 3.6.4).

These components are determined separately as shown in the following sections. The overall U-value of the window is given by:

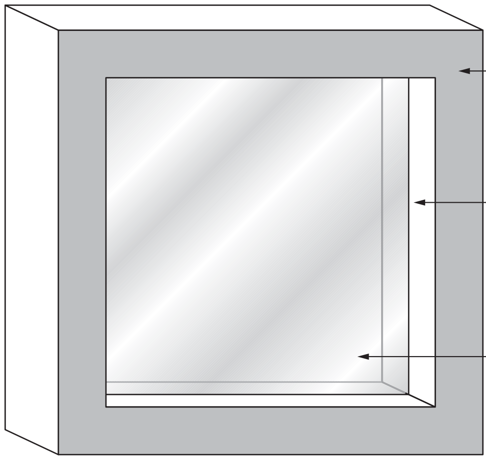


Figure 3.15 Dimensions of window for calculation purposes

$$U_w = \frac{\Sigma\,(A_g\,U_g) + \Sigma\,(A_{wf}\,U_{wf}) + \Sigma\,(p_{wf}\,\Psi_s)}{\Sigma\,A_g + \Sigma\,A_{wf}} \tag{3.35}$$

where U_w is the thermal transmittance of the window (W·m^{-2}·K^{-1}), A_g is the projected area of the glazing (m^2), A_{wf} is the projected area of the window frame or sash (m^2), U_g is the thermal transmittance of glazing (W·m^{-2}·K^{-1}), U_{wf} is the thermal transmittance of frame or sash (W·m^{-2}·K^{-1}), p_{wf} is the length of inner perimeter of frame or sash (m) and Ψ_s is the linear thermal transmittance for the glazing/frame (W·m^{-1}·K^{-1}).

The dimensions defined are shown in Figure 3.15.

3.6.2 Glazing (excluding frame or sash)

Tables 3.23 and 3.24 provide U-values for vertical, horizontal and near horizontal sloping glazing combinations for conventional and low-emissivity coated glasses, and for glazing units filled with argon gas. These values were calculated using the method given in BS EN 673[35].

Low emissivity coatings are highly transparent in the visual and solar parts of the spectrum but are reflective to radiation in the wavelength range 5 to 50 mm (i.e. far-infrared). To achieve a worthwhile improvement the emissivity must be less than 0.2 and accredited data must be obtained from the manufacturer to confirm that a particular product achieves this performance.

Values have been calculated for the surface resistance values adopted in BS EN ISO 6946[15] and for those corresponding to the conditions of exposure defined in section 3.3.5.

For the purposes of Building Regulations, ‘vertical’ includes glazing up to 20° from the vertical.

The data are given to three significant figures to illustrate the trends, whereas U-values are usually quoted to two significant figures elsewhere in this Guide.

The tabulated values are based on a temperature difference of 15 K between the outer and inner glass surfaces and a mean temperature of 10 °C. It is recommended that this basis be used for comparison purposes and in normal building calculations in the UK. For colder climates, a greater temperature difference and a lower mean temperature may be considered but the quoted values are sufficiently accurate for most cases.

The thermal conductivity of glass is approximately 1.0 $W \cdot m^{-1} \cdot K^{-1}$.

3.6.3 Frames and sashes (excluding glazing)

The *U*-values given in Table 3.25 are based on data given in BS EN ISO 10077-1[36]. Alternatively the *U*-value of window frames can be calculated by software conforming with BS EN ISO 10077-2[37].

3.6.4 Spacer between panes (multiple glazing units)

In multiple glazing units, the thermal transmittance is increased due to interaction between the glazing and the frame, including the effect of the spacer bars. This allowed for by a linear thermal transmittance related to the perimeter length of the glazing.

The linear transmittance for particular glazing and frame combinations can be calculated by software conforming with BS EN ISO 10077-2[37]. Table 3.26 gives default values that can be used in the absence of detailed information.

Table 3.23 *U*-values for vertical glazing

Type of glazing	Spacing / mm	*U*-value (/ $W \cdot m^{-2} \cdot K^{-1}$) for stated exposure of panes†		
		Normal (0.13/0.04)	Sheltered (0.13/0.06)	Severe (0.13/0.02)
Single	—	5.75	5.16	6.49
Double	25	2.76	2.60	2.90
	20	2.74	2.60	2.90
	16	2.73	2.59	2.90
	12	2.85	2.70	3.02
	9	3.01	2.84	3.20
	6	3.28	3.08	3.51
Triple	25	1.72	1.67	1.78
	20	1.71	1.66	1.77
	16	1.78	1.72	1.84
	12	1.89	1.83	1.97
	9	2.04	1.96	2.12
	6	2.29	2.19	2.40
Coated double ε = 0.2	20	1.85	1.78	1.92
	16	1.82	1.76	1.89
	12	2.02	1.95	2.11
	9	2.29	2.19	2.39
	6	2.71	2.57	2.87
Coated double ε = 0.1	20	1.60	1.55	1.65
	16	1.57	1.53	1.63
	12	1.80	1.74	1.87
	9	2.10	2.01	2.19
	6	2.57	2.44	2.71
Coated double ε = 0.05	20	1.45	1.41	1.49
	16	1.42	1.38	1.46
	12	1.67	1.61	1.72
	9	1.98	1.91	2.06
	6	2.48	2.37	2.61
Coated double argon-filled ε = 0.2	20	1.65	1.60	1.71
	16	1.63	1.58	1.69
	12	1.76	1.70	1.82
	9	1.98	1.90	2.06
	6	2.35	2.24	2.46
Coated double argon-filled ε = 0.1	20	1.38	1.34	1.42
	16	1.36	1.32	1.40
	12	1.50	1.46	1.55
	9	1.75	1.69	1.81
	6	2.16	2.07	2.26
Coated double argon-filled ε = 0.05	20	1.21	1.18	1.24
	16	1.19	1.16	1.22
	12	1.34	1.31	1.38
	9	1.61	1.56	1.66
	6	2.05	1.97	2.14

† Internal and external surface resistances ($m^2 \cdot K \cdot W^{-1}$) respectively are given in parentheses

Table 3.24 *U*-values for horizontal and roof glazing

Type of glazing	Spacing / mm	*U*-value (/ $W \cdot m^{-2} \cdot K^{-1}$) for stated exposure of panes†		
		Normal (0.13/0.04)	Sheltered (0.13/0.06)	Severe (0.13/0.02)
Single	—	6.94	6.10	8.07
Double	25	3.32	3.11	3.55
	20	3.34	3.23	3.58
	16	3.37	3.16	3.61
	12	3.41	3.19	3.66
	9	3.44	3.22	3.70
	6	3.63	3.39	3.92
Triple	25	2.06	1.98	2.15
	20	2.09	2.00	2.18
	16	2.11	2.02	2.20
	12	2.14	2.05	2.23
	9	2.17	2.08	2.27
	6	2.46	2.35	2.59
Coated double ε = 0.2	20	2.51	2.39	2.65
	16	2.56	2.43	2.69
	12	2.61	2.48	2.75
	9	2.67	2.53	2.82
	6	2.95	2.79	3.14
Coated double ε = 0.1	20	2.3	2.2	2.41
	16	2.35	2.24	2.46
	12	2.41	2.3	2.53
	9	2.47	2.35	2.6
	6	2.78	2.64	2.95
Coated double ε = 0.05	20	2.18	2.09	2.28
	16	2.22	2.13	2.33
	12	2.29	2.19	2.4
	9	2.35	2.25	2.47
	6	2.68	2.55	2.84
Coated double argon-filled ε = 0.2	20	2.20	2.10	2.30
	16	2.23	2.14	2.33
	12	2.28	2.18	2.39
	9	2.33	2.22	2.44
	6	2.52	2.40	2.66
Coated double argon-filled ε = 0.1	20	1.95	1.88	2.03
	16	2.00	1.91	2.07
	12	2.04	1.96	2.13
	9	2.09	2.02	2.19
	6	2.32	2.21	2.42
Coated double argon-filled ε = 0.05	20	1.80	1.74	1.87
	16	1.84	1.78	1.91
	12	1.90	1.83	1.97
	9	1.96	1.88	2.04
	6	2.19	2.10	2.29

† Internal and external surface resistances ($m^2 \cdot K \cdot W^{-1}$) respectively are given in parentheses

Table 3.25 Thermal transmittances for various types of window frame and sash

Material	Description	U-value / $W \cdot m^{-2} \cdot K^{-1}$
Wood	Average thickness 30 mm	2.30
	Average thickness 40 mm	2.15
	Average thickness 50 mm	2.02
	Average thickness 60 mm	1.90
	Average thickness 70 mm	1.78
	Average thickness 80 mm	1.67
	Average thickness 90 mm	1.57
	Average thickness 100 mm	1.48
Plastic	Without metal reinforcement:	
	— polyurethane	2.8
	— PVC, two hollow chambers	2.2
	— PVC, three hollow chambers	2.0
Aluminium	Thermal barrier† with:	
	— 4 mm thermal break	4.4
	— 8 mm thermal break	3.9
	— 12 mm thermal break	3.5
	— 16 mm thermal break	3.2
	— 20 mm thermal break	3.0
Aluminium or steel	Without thermal barrier	6.9

† Thermal barrier must be continuous and totally isolate the interior side of the frame or frame sections from the exterior side

Table 3.26 Linear thermal transmittance, Ψ_s, for conventional sealed multiple glazing units

Frame type	Linear thermal transmittance (/ $W \cdot m^{-1} \cdot K^{-1}$) for stated glazing type	
	Double or triple glazing uncoated glass, air or gas filled	Double or triple glazing, low-emissivity glass (1 pane coated for double glazing or 2 panes coated for triple glazing), air or gas filled
Wood or PVC	0.06	0.08
Metal with thermal break	0.08	0.11
Metal without thermal break	0.02	0.05

3.6.5 Effect of blinds and curtains

3.6.5.1 Internal blinds and curtains

Internal roller blinds or curtains provide additional insulation due to the air enclosed between the window and the blind. The degree of insulation depends strongly on the level of enclosure achieved[38].

Good levels of air entrapment are almost impossible to achieve with curtains. Roller blinds can achieve effective entrapment provided they run in side channels and are sealed at the top and bottom. With well-sealed blinds, further improvement can be achieved by using a material which has a low emissivity surface protected by layer transparent to infrared radiation.

The thermal transmittance of the window can be corrected for the effect of blinds or curtains, as follows:

$$U_{wb}' = [(1 / U_w) + R_{bi}]^{-1} \qquad (3.36)$$

where U_{wb}' is the thermal transmittance of the window corrected for an internal blind or curtain ($W \cdot m^{-2} \cdot K^{-1}$), U_w is the thermal transmittance of the window ($W \cdot m^{-2} \cdot K^{-1}$), R_{bi} is the thermal resistance of the internal blind or curtain ($m^2 \cdot K \cdot W^{-1}$).

Values for the thermal resistance of internal blinds and curtains are given in Table 3.27.

Table 3.27 Thermal resistance of blinds and curtains

Description	Thermal resistance / $m^2 \cdot K \cdot W^{-1}$
Conventional roller blind, curtain or venetian blind (vertical slats)	0.05
Closely fitting curtain with pelmet	0.07
Roller blind:	
— bottom only sealed	0.09
— sides only sealed in channels	0.11
— sides and top sealed	0.15
— sides and bottom sealed	0.16
— fully sealed	0.18
Low emissivity roller blind, fully sealed	0.44

3.6.5.2 Mid-pane blinds

Coupled windows are usually fitted with horizontal slatted or roller blinds between the panes. The effect on the insulation of the window with the blind lowered can be estimated by adding the thermal resistance of the blind to that of the glazing, as follows:

$$U_{gb}' = [(1 / U_g) + R_{bm}]^{-1} \qquad (3.37)$$

where U_{gb}' is the thermal transmittance of the glazing corrected for a mid-pane blind ($W \cdot m^{-2} \cdot K^{-1}$), U_g is the thermal transmittance of the glazing ($W \cdot m^{-2} \cdot K^{-1}$) and R_{bm} is the thermal resistance of the mid-pane blind ($m^2 \cdot K \cdot W^{-1}$).

Values of thermal resistance for mid-pane blinds are given in Table 3.28.

Table 3.28 Thermal resistance of mid-pane blinds

Description	Thermal resistance / $m^2 \cdot K \cdot W^{-1}$
Venetian blind:	
— slats horizontal	0.02
— slats vertical	0.07
Roller blind	0.07

3.6.6 Indicative *U*-values for conceptual design

At the concept design stage, it is convenient to use indicative U-values for typical window configurations to enable an initial evaluation of the heat losses and energy consumption of the proposed building. Table 3.29 provides such values for these purposes.

Table 3.29 Indicative *U*-values for windows for conceptual design

Type	Indicative *U*-value / $W{\cdot}m^{-2}{\cdot}K^{-1}$	
	Glazing only	Window (including frame or sash)
Single	5.7	5.0
Double	2.8	3.0
Double (low emissivity)	1.8	2.2
Triple	1.8	2.2

3.6.7 Indicative *U*-values for energy rating

Tables 3.30, 3.31 and 3.32, based on tables given in *The Government's Standard Assessment Procedure for Energy Rating of Dwellings*[39] (SAP2005), provide indicative *U*-values for windows, doors and rooflights for the purposes of energy rating. Table 3.30 applies to windows and rooflights with wood or PVC-U frames, and to solid wooden doors. Table 3.31 applies to windows with metal frames, to which (if applicable) the adjustments for thermal breaks and rooflights in Table 3.32 should be applied. The tables do not apply to curtain walling or to other structural glazing not fitted in a frame.

The *U*-value of a window or rooflight containing low-E glazing is influenced by the emissivity, ε_n, of the low-E coating. Low-E coatings are of two principal types, known as 'hard' and 'soft'. Hard coatings generally have emissivities in the range 0.15 to 0.2, and the data for ε_n = 0.2 should be used for hard coatings if the emissivity is not specified, or if the glazing is stated to be low-E but the type of coating is not specified. Soft coatings generally have emissivities in the range 0.05 to 0.1. The data for ε_n = 0.1 should be used for a soft coating if the emissivity is not specified.

3.6.8 Measurement of thermal transmittance of windows

The thermal transmittance of windows may be determined by direct measurement. This may be necessary where the design of the window is such that the foregoing methods are not applicable or where the manufacturer believes that the performance of the window is better than that predicted. BS EN ISO 12567-1[5] gives a precise methodology for the measurement of the *U*-values of windows using hot-box techniques.

Table 3.30 Indicative *U*-values (/ $W{\cdot}m^{-2}{\cdot}K^{-1}$) for windows and rooflights with wood or PVC-U frames, and doors (Crown copyright, reproduced with the permission of the Controller of Her Majesty's Stationery Office and the Queen's Printer for Scotland)

Item	Indicative *U*-value (/ $W{\cdot}m^{-2}{\cdot}K^{-1}$) for stated gap between panes			Adjustment for rooflights in dwellings†
	6 mm	12 mm	16 mm or more	
Single glazing	4.8	—	—	+0.3
Double glazing (air filled):	3.1	2.8	2.7	+0.2
— low-E, ε_n = 0.2[1]	2.7	2.3	2.1	+0.2
— low-E, ε_n = 0.15	2.7	2.2	2.0	+0.2
— low-E, ε_n = 0.1	2.6	2.1	1.9	+0.2
— low-E, ε_n = 0.05	2.6	2.0	1.8	+0.2
Double glazing (argon filled[2]):	2.9	2.7	2.6	+0.2
— low-E, ε_n = 0.2	2.5	2.1	2.0	+0.2
— low-E, ε_n = 0.15	2.4	2.0	1.9	+0.2
— low-E, ε_n = 0.1	2.3	1.9	1.8	+0.2
— low-E, ε_n = 0.05	2.3	1.8	1.7	+0.2
Triple glazing:	2.4	2.1	2.0	+0.2
— low-E, ε_n = 0.2	2.1	1.7	1.6	+0.2
— low-E, ε_n = 0.15	2.0	1.7	1.5	+0.2
— low-E, ε_n = 0.1	2.0	1.6	1.5	+0.2
— low-E, ε_n = 0.05	1.9	1.5	1.4	+0.2
Triple glazing (argon filled[2]):	2.2	2.0	1.9	+0.2
— low-E, ε_n = 0.2	1.9	1.6	1.5	+0.2
— low-E, ε_n = 0.15	1.8	1.5	1.4	+0.2
— low-E, ε_n = 0.1	1.8	1.4	1.3	+0.2
— low-E, ε_n = 0.05	1.7	1.4	1.3	+0.2
Solid wooden door[3]	3.0	—	—	

† No correction need be applied to rooflights in buildings other than dwellings

Notes:

[1] The emissivities quoted are normal emissivities. (Corrected emissivity is used in the calculation of glazing *U*-values.) Uncoated glass is assumed to have a normal emissivity of 0.89.

[2] The gas mixture is assumed to consist of 90% argon and 10% air.

[3] For doors which are half-glazed the *U*-value of the door is the average of the appropriate window *U*-value and that of the non-glazed part of the door (e.g. 3.0 $W{\cdot}m^{-2}{\cdot}K^{-1}$ for a wooden door).

Table 3.31 Indicative U-values (/ $W \cdot m^{-2} \cdot K^{-1}$) for windows and fully-glazed doors with metal frames (4 mm thermal break) (Crown copyright, reproduced with the permission of the Controller of Her Majesty's Stationery Office and the Queen's Printer for Scotland)

Item	Indicative U-value (/ $W \cdot m^{-2} \cdot K^{-1}$) for stated gap between panes		
	6 mm	12 mm	16 mm or more
Single glazing	5.7	—	—
Double glazing (air filled):	3.7	3.4	3.3
— low-E, ε_n = 0.2[1]	3.3	2.8	2.6
— low-E, ε_n = 0.15	3.3	2.7	2.5
— low-E, ε_n = 0.1	3.2	2.6	2.4
— low-E, ε_n = 0.05	3.1	2.5	2.3
Double glazing (argon filled[2]):	3.5	3.3	3.2
— low-E, ε_n = 0.2	3.0	2.6	2.5
— low-E, ε_n = 0.15	3.0	2.5	2.4
— low-E, ε_n = 0.1	2.9	2.4	2.3
— low-E, ε_n = 0.05	2.8	2.3	2.1
Triple glazing:	2.9	2.6	2.5
— low-E, ε_n = 0.2)	2.6	2.1	2.0
— low-E, ε_n = 0.15)	2.5	2.1	2.0
— low-E, ε_n = 0.1)	2.5	2.0	1.9
— low-E, ε_n = 0.05)	2.4	1.9	1.8
Triple glazing (argon filled[2]):	2.8	2.5	2.4
— low-E, ε_n = 0.2	2.4	2.0	1.9
— low-E, ε_n = 0.15	2.3	1.9	1.8
— low-E, ε_n = 0.1	2.2	1.9	1.7
— low-E, ε_n = 0.05	2.2	1.8	1.7

[1] and [2]: see footnotes to Table 3.30

Table 3.32 Adjustments to U-values in Table 3.31 for frames with thermal breaks (Crown copyright, reproduced with the permission of the Controller of Her Majesty's Stationery Office and the Queen's Printer for Scotland)

Thermal break / mm	Adjustment to U-value (/ $W \cdot m^{-2} \cdot K^{-1}$)	
	Adjustment for thermal break	Additional adjustment for rooflights angled < 70° to horizontal
0 (no break)	+0.3	+0.4
4	+0.0	+0.3
8	–0.1	+0.3
12	–0.2	+0.3
16	–0.2	+0.3
20	–0.3	+0.3
24	–0.3	+0.3
28	–0.3	+0.3
32	–0.4	+0.3
36	–0.4	+0.3

3.7 Linear thermal transmittance

Linear thermal transmittances arise typically at junctions between elements, such as junctions of floor and roof with external walls and around window and door openings. They represent the heat loss associated with junctions, over and above that obtained from the U-values and areas of the elements.

In buildings with relatively poor standards of thermal insulation, the transmission heat loss is dominated by the heat loss through the plain areas of the fabric. Thermal bridging at junctions is usually a relatively small proportion of the total, and in the past it has not usually been taken into account in calculations. For better insulated buildings, however, the proportionate effect of thermal bridging can be significant. It is, however, limited by good design of details: for example by using the details illustrated in *Robust construction details*(40) the additional effect of thermal bridging should not exceed 10–15% of the total transmission heat loss.

Linear thermal transmittance values are generally calculated by numerical analysis, using the criteria set out in BS EN ISO 10211(30). Where values are available, they can be included in calculations as indicated in section 3.3.1. Further information is given in BRE Information Paper IP 17/01(41).

3.8 Non-steady state thermal characteristics

3.8.1 Admittance procedure

There are several methods available for assessing the non-steady state or dynamic performance of a structure. One of the simplest is the admittance procedure(42) which is described in detail in chapter 5: *Thermal response and plant sizing*. The method of calculation of admittances and related parameters is defined in BS EN ISO 13786(43) and a summary is given in Appendix 3.A6.

This procedure requires the calculation of three parameters in addition to the thermal transmittance: admittance, surface factor and decrement factor. These parameters depend upon the thickness, thermal conductivity, density and specific heat capacity of the materials used within the structure and the relative positions of the various elements that make up the construction. Each of these parameters is expressed as an amplitude and an associated time lead/lag.

3.8.1.1 Thermal admittance (Y-value)

The most significant of the three parameters is the admittance. This is the rate of flow of heat between the internal surface of the structure and the environmental temperature in the space, for each degree of deviation of the space temperature about its mean value. The associated time dependency (ω) takes the form of a time lead.

For thin structures composed of a single layer, the admittance is equal in amplitude to the U-value and has a time lead of zero. The amplitude tends towards a limiting value for thicknesses greater than about 100 mm.

For multi-layered structures, the admittance is primarily determined by the characteristics of the materials in the layers nearest to the internal surface. For example, the admittance of a structure comprising heavyweight concrete slabs lined internally with insulation will be close to the value for the insulation alone. However, placing the insulation within the construction, or on the outside surface will have little or no effect on the admittance.

3.8.1.2 Decrement factor (*f*)

The decrement factor is the ratio of the rate of flow of heat through the structure to the environmental temperature in the space for each degree of deviation in external temperature about its mean value, to the steady state rate of flow of heat (*U*-value). The associated time dependency (ϕ) takes the form of a time lag.

For thin structures of low thermal capacity, the amplitude of the decrement factor is unity with a time lag of zero. The amplitude decreases and the time lag increases with increasing thickness and/or thermal capacity.

3.8.1.3 Surface factor (*F*)

The surface factor is the ratio of the variation of heat flow about its mean value readmitted to the space from the surface, to the variation of heat flow about its mean value absorbed by the surface. The associated time dependency (ψ) takes the form of a time lag.

The amplitude of the surface factor decreases and its time lag increases with increasing thermal conductivity but both are virtually constant with thickness.

3.8.1.4 Internal structural elements

For internal structural elements, such as floors and partition walls, which are not symmetrical about their mid-plane, the dynamic responses will be different for the two faces and two sets of admittances and surface factors are required. The decrement factor is the same for heat flow in either direction.

Where internal structures divide spaces in which the thermal conditions are identical, the energy transfers can be simplified by combining the admittance and surface factor with the decrement factor to give modified admittance and surface factor respectively.

3.8.2 Heat capacity

The heat capacity per unit area of a building element, c ($kJ \cdot m^{-2} \cdot K^{-1}$), is a measure of the thermal response characteristics of the element. For a sinusoidal temperature variation it is defined in terms of the maximum change in heat stored within the element during a half cycle[(43)]. It is used in simplified methods for calculating the energy use of buildings, such as that described in BS EN ISO 13790[(44)].

3.8.3 Effect of thermal bridging of dynamic characteristics

The presence of thermal bridges will affect the overall dynamic performance of a structure. However, since the admittance is mainly determined by the properties of the materials immediately adjacent to the interior spaces of the building, the presence of heat bridges within the structure will have little effect on the overall thermal performance. Therefore, it is only where the bridging material is at or near the surface temperature that it will affect the dynamic thermal performance. In cases where it is felt necessary to account for the effects of thermal bridges, an area-weighted mean approach can be used.

References

1 BS EN 12664: 2001: *Thermal performance of building materials and products. Determination of thermal resistance by means of guarded hot plate and heat flow meter methods. Dry and moist products of low and medium thermal resistance* (London: British Standards Institution) (2001)

2 BS EN 12667: 2000: *Thermal performance of building materials and products. Determination of thermal resistance by means of guarded hot plate and heat flow meter methods. Products of high and medium thermal resistance* (London: British Standards Institution) (2000)

3 BS EN 12939: 2001: *Thermal performance of building materials and products. Determination of thermal resistance by means of guarded hot plate and heat flow meter methods. Thick products of high and medium thermal resistance* (London: British Standards Institution) (2001)

4 BS EN ISO 8990: 1996: *Thermal insulation. Determination of steady-state thermal transmission properties. Calibrated and guarded hot box* (London: British Standards Institution) (1996)

5 BS EN ISO 12567-1: 2000: *Thermal performance of windows and doors. Determination of thermal transmittance by hot box methods*: Part 1: *Complete windows and doors* (London: British Standards Institution) (2000)

6 Building and Buildings. The Building Regulations 2000 Statutory Instrument 2000 No. 2531 and Building Regulations (Amendment) Regulations 2001 Statutory Instrument 2001 No. 3335 (London: The Stationery Office) (2000 and 2001)

7 *Conservation of fuel and power in dwellings* Building Regulations 2000 Approved Document L1 (2002 edn.) (London: The Stationery Office) (2001) (www.odpm.gov.uk) (under revision)

8 *Conservation of fuel and power in buildings other than dwellings* Building Regulations 2000: Approved Document L2 (2002 edn.) (London: The Stationery Office) (2001) (www.odpm.gov.uk) (under revision)

9 BS 8207: 1985 (1995): *Code of practice for energy efficiency in buildings* (London: British Standards Institution) (1985)

10 BS 8211: *Energy efficiency in housing*: Part 1: *Code of practice for energy efficient refurbishment of housing* (London: British Standards Institution) (1988)

11 Building (Scotland) Regulations 2004 Scottish Statutory Instrument 2004 No. 406 (Edinburgh: The Stationery Office) (2004) (also available from www.sbsa.gov.uk)

12 *Technical handbook: Domestic* and *Technical handbook: Non-domestic* (Scottish Building Standards Agency) (www.sbsa.gov.uk) (under revision)

13 Building Regulations (Northern Ireland) 1994 Statutory Rules of Northern Ireland 1994 No. 243 (Belfast: The Stationery Office) (1994)

14 *Conservation of Fuel and Power* Building Regulations (Northern Ireland) 1994 Technical Booklet F (Belfast: The Stationery Office) (1998) (www.dfpni.gov.uk/buildingregulations) (www.buildingregulationsni.gov.uk) (under revision)

15 BS EN ISO 6946: 1997: *Building components and building elements. Thermal resistance and thermal transmittance. Calculation method* (London: British Standards Institution) (1997) (under revision)

16 BS EN ISO 13789: 1999: *Thermal performance of buildings — Transmission heat loss coefficient. Calculation method* (London: British Standards Institution) (1999) (under revision)

17 Siviour J B Areas in heat loss calculations *Building Serv. Eng. Res. Technol.* **6** 134 (1985)

18 ISO 8302: 1991: *Thermal insulation. Determination of steady-state thermal resistance and related properties. Guarded hot plate apparatus* (Geneva: International Organization for Standardization) (1991)

19 BS EN 1745: 2002: *Masonry and masonry materials. Methods for determining design thermal values* (London: British Standards Institution) (2002)

20 BS EN 12524: 2000: *Building materials and products. Hygrothermal properties. Tabulated design values* (London: British Standards Institution) (2000)

21 BS EN ISO 10456: 1999: *Building materials and products and products. Procedures for determining declared and design thermal values* (London: British Standards Institution) (1999) (under revision)

22 *Heat transfer* CIBSE Guide C, section 3 (London: Chartered Institution of Building Services Engineers) (2002)

23 Doran S M and Gorgolewski M T *U-values for light steel-frame construction* BRE Digest 465 (Garston: Building Research Establishment) (2002)

24 Ward T I *Metal cladding: assessing the thermal performance of built-up systems that use 'Z' spacers* BRE Information Paper IP10/02 (Garston: Building Research Establishment) (2002)

25 *Metal cladding: U-value calculation — assessing thermal performance of built-up metal roof and wall cladding systems using rail & bracket spacers* Technical Information Leaflet P312 (Ascot: Steel Construction Institute) (2002)

26 *Guide to good practice for assessing glazing frame U-values* (Bath: Centre for Window and Cladding Technology) (2002)

27 *Guide to good practice for assessing heat transfer and condensation risk for a curtain wall* (Bath: Centre for Window and Cladding Technology) (2002)

28 *Conventions for U-value calculations* BRE Report BR 443: (Garston: Building Research Establishment) (2002) (under revision)

29 EN ISO 6946/prA2: Amendment 2, Annex D.3: *Correction for mechanical fasteners* (Brussels: Comité Européen de Normalisation) (available in UK through BSI) (2003)

30 BS EN ISO 10211-1: 1996: *Thermal bridges in building construction. Heat flows and surface temperatures*: Part 1: *General calculation methods* (London: British Standards Institution) (1996) (under revision)

31 Anderson B R The relationship between the *U*-value of uninsulated ground floors and the floor dimensions *Building Serv. Eng. Res. Technol.* **12** (3) 103–105 (1991)

32 Anderson B R *Calculation of the steady-state heat transfer through a slab-on-ground floor* Building and Environment **26** (4) 405–415 (1991)

33 BS EN ISO 13370: 1998: *Thermal performance of buildings. Heat transfer via the ground. Calculation methods* (London: British Standards Institution) (1998) (under revision)

34 Anderson B R The effect of edge insulation on the steady-state heat loss through a slab-on-ground floor *Building and Environment* **28** (3) 361–367 (1993)

35 BS EN 673: 1998: *Glass in building. Determination of thermal transmittance (U-value). Calculation method* (London: British Standards Institution) (1998)

36 BS EN ISO 10077-1: 2000: *Thermal performance of windows, doors and shutters. Calculation of thermal transmittance*: Part 1: *Simplified methods* (London: British Standards Institution) (2000) (under revision)

37 BS EN ISO 10077-2: 2003: *Thermal performance of windows, doors and shutters. Calculation of thermal transmittance*: Part 2: *Numerical method for frames* (London: British Standards Institution) (2003) (under revision)

38 Littler J G F and Ruyssevelt P A Heat reflecting roller blinds and methods of edge sealing *Proc. Conf. Windows in Design and Maintenance, Gothenberg, Sweden* (1984)

39 *The Government's Standard Assessment Procedure for Energy Rating of Dwellings* (SAP2005) (Garston: Building Research Establishment) (2005) (www.bre.co.uk/sap2005)

40 *Limiting thermal bridging and air leakage: Robust construction details for dwellings and similar buildings* (London: The Stationery Office) (2001)

41 Ward T I *Assess the effect of thermal bridging at junctions and around openings* BRE Information Paper 17/01 (Garston: Building Research Establishment) (2001) (under revision)

42 Milbank N O and Harrington-Lynn J Thermal response and the admittance procedure *Building Serv. Eng.* 42 38–51 (1974)

43 BS EN ISO 13786: 1999: *Thermal performance of building components. Dynamic thermal characteristics. Calculation method* (London: British Standards Institution) (1999) (under revision)

44 BS EN ISO 13790: 2004: *Thermal performance of buildings. Calculation of energy use for space heating* (London: British Standards Institution) (2004) (under revision)

Appendix 3.A1: Moisture content of masonry materials

3.A1.1 Standard moisture content

While insulating materials are generally 'air-dry' (i.e. in equilibrium with the internal environment), this is not true for masonry materials in external walls. Research[A1-1] has shown that typical moisture contents of both the inner and outer leaves of external twin-leaf masonry walls are above air-dry values and the thermal conductivities used for calculating *U*-values should be corrected to take account of the presence of moisture.

The moisture content of the structural elements of occupied buildings varies widely depending upon many factors including climate, type of masonry, thickness of wall, whether or not the wall is rendered, standards of workmanship in construction, local exposure to rain (which varies across the building) etc. Therefore, it is convenient to base *U*-value calculations on thermal conductivities at standard values of moisture content.

Typical moisture contents for UK conditions are given in Table 3.2 for masonry that is 'protected' or 'exposed'. 'Exposure' refers to the external climate (i.e. solid masonry or the outer leaf of cavity walls without protective cladding). 'Protected' refers to solid masonry or the outer leaf of cavity walls protected by cladding such as tile hanging or weather boarding, and to the inner leaf of cavity walls (whether or not the cavity is filled with an insulating material).

3.A1.2 Correction factors for thermal conductivity

The way in which the thermal conductivity of different materials increases with moisture content is shown in Table 3.33. For maximum accuracy, this variation is given in terms of either percentage by weight or percentage by volume, according to the characteristics of the particular material.

Table 3.33 Correction factors for moisture content

Material	Correction factor
Brick (fired clay)	10% per % (by volume)
Brick (calcium silicate)	10% per % (by volume)
Dense aggregate concrete	4% per % (by volume)
Blast furnace slag concrete	4% per % (by weight)
Pumice aggregate concrete	4% per % (by weight)
Other lightweight aggregate concrete	4% per % (by weight)
Autoclaved aerated concrete	4% per % (by weight)

Reference

A1-1 Arnold P J *Thermal conductivity of masonry materials* BRE CP1/70 (Garston: Building Research Establishment) (1970)

Appendix 3.A2: Thermal conductivity and thermal transmittance testing

With the increasing demand for thermal insulation and the subsequent development of new forms of insulation, guidance is required on appropriate methods of determining the thermal performance of these materials.

The thermal conductivity value used in calculations should be representative of material used on site. For masonry, BS EN 1745[20] sets out a suitable method for determining that this is so, related to the manufactured density range and based on a minimum of three tests. This is an alternative to the use of the tabulated values in Table 3.1.

The recommended method of determining the thermal conductivity of of materials is the use of either guarded or heat flow meter hot plate apparatus as laid down in BS EN ISO 12664[1] or BS EN ISO 12667[2].

There are now European product standards covering many types of insulation products. For factory made products these are BS EN 13162 to BS EN 13171 inclusive, covering respectively factory-made products of mineral wool, expanded polystyrene, extruded polystyrene, rigid polyurethane, phenolic foam, cellular glass, wood wool, expanded perlite, expanded cork and wood fibre. These standards specify that thermal conductivity and/or thermal resistance is to be declared as a limit value representing at least 90% of the production and, for foamed plastics blown other than by air, representing the average value over 25 years. Corresponding standards for in-situ formed materials are in preparation.

The thermal transmittance of a complete structure may be determined using apparatus that conforms to BS EN ISO 8990[4]. However, where masonry materials are involved, it is not usually possible to replicate the standard moisture contents in a laboratory. Therefore, the results of such tests may not reflect the actual conditions unless suitable analysis has been undertaken to convert the test results to the moisture conditions that will apply in practice.

Appendix 3.A3: Heat transfer at surfaces

3.A3.1 General

Heat is transferred to and from surfaces by radiation interchange with other surfaces and by convective heat transfer at the air/surface interface. This may be represented by:

$$\phi_s = E\, h_r\, (\theta_r - \theta_s) + h_c\, (\theta_a - \theta_s) \qquad (3.38)$$

where ϕ_s is the rate of heat transfer to/from the surface (W·m^{-2}), θ_r is the radiant temperature experienced by the surface (°C), θ_s is the surface temperature (°C), θ_a is the air temperature (°C), E is the emissivity factor, h_r is the radiative heat transfer coefficient (W·m^{-2}·K^{-1}) and h_c is the convective heat transfer coefficient (W·m^{-2}·K^{-1}).

This and subsequent equations are valid for heat flow into and out of a surface provided that the sign of q_s is taken as positive if the heat flow is into the surface.

In general, both E and θ_r are complicated functions of the geometry and emissivities of the surfaces involved. Usually equation 3.38 is considered in relation to an exposed part of the structure having surface temperature θ_s radiating to and from the other surfaces of the enclosure, all of which are assumed to be at the same temperature. Then θ_r may be taken as the temperature of these other surfaces.

The emissivity factor E depends on the room geometry and the emissivities of all the surfaces. Where the emissivity of the exposed surfaces is uniform and the emissivity of the non-exposed surfaces is also uniform, but different in value, the emissivity factor is given by:

$$\frac{1}{E\, A_1} = \frac{1}{A_1} + \frac{1-\varepsilon_1}{A_1\, \varepsilon_1} + \frac{1-\varepsilon_2}{A_2\, \varepsilon_2} \qquad (3.39)$$

where E is the emissivity factor, A_1 is the area of exposed surface(s) (m^2), ε_1 is the emissivity of the exposed surface(s), A_2 is the area of the non-exposed surface(s) (m^2) and ε_2 is the emissivity of the non-exposed surface(s).

For a cubical room with one exposed surface, $A_2 = 5\, A_1$, therefore:

$$\frac{1}{E} = 1 + \frac{1-\varepsilon_1}{\varepsilon_1} + \frac{1-\varepsilon_2}{5\, \varepsilon_2} \qquad (3.40)$$

Hence:

$$E = \varepsilon_1\, [1 + \varepsilon_1\, (1-\varepsilon_2) / 5\, \varepsilon_2)]^{-1} \qquad (3.41)$$

For most interior surfaces, the value of ε_2 is high. Taking ε_2 as 0.9 and plotting E as a function of ε_1, it may be demonstrated that E is directly proportional to ε_1 to within 2% for values of ε_1 between 0 and 1.

Hence:

$$E = K\, \varepsilon_1 \qquad (3.42)$$

where K is a constant related to room geometry.

Note that for the calculation of the conventional values of inside surface resistance given in Table 3.9, K is taken as 1, hence E = 0.9.

3.A3.2 Interior surfaces

For practical application at interior surfaces it is convenient to make two modifications to equation 3.38. The first is to use $\bar{\theta}_{ri}$, the mean temperature of all the surfaces in the enclosure, rather than θ_r since this is more easily measured and is more closely related to thermal comfort.

Strictly, $\bar{\theta}_{ri}$ varies throughout an enclosure dependent on the shape factors for all relevant surfaces at the point chosen. However, at the centre of a cubical room with one exposed wall:

$$6\, \theta_{ri} = 5\, \bar{\theta}_{ri} + \theta_{si} \qquad (3.43)$$

where θ_{ri} is the inside radiant temperature seen by the surface (°C), $\bar{\theta}_{ri}$ is the mean inside radiant temperature (°C) and θ_{si} is the inside surface temperature (°C).

Substituting in equation 3.38, and introducing the subscript 'i' to denote inside temperatures, gives:

$$\phi_s = {}^{6}/_{5}\, E\, h_r\, (\bar{\theta}_{ri} - \theta_{si}) + h_c\, (\theta_{ai} - \theta_{si}) \qquad (3.44)$$

The second modification is to combine radiant and air temperatures into environmental temperature (see chapter 5), thus:

$$\phi = (1 / R_{si})\, (\theta_{ei} - \theta_s) \qquad (3.45)$$

where R_{si} is the inside surface resistance (m^2·K·W^{-1}) and θ_{ei} is the inside environmental temperature (°C).

This is equivalent to equation 3.43 where:

$$\theta_{ei} = \frac{{}^{6}/_{5}\, E\, h_r}{{}^{6}/_{5}\, E\, h_r + h_c}\, \theta_{ri} + \frac{h_c}{{}^{6}/_{5}\, E\, h_r + h_c}\, \theta_{ai} \qquad (3.46)$$

and:

$$R_{si} = \frac{1}{({}^{6}/_{5}\, E\, h_r + h_c)} \qquad (3.47)$$

Thus θ_{ei} varies with both θ_{ri} and θ_{ai}.

3.A3.3 Exterior surfaces

As with interior surfaces, the heat transfer is represented by equation 3.44 by replacing the suffix 'i' by 'o' to denote outside temperatures. However, in this case no enclosure is involved and it is more convenient to retain θ_r. The outside environmental temperature is then given by:

$$\theta_{eo} = \frac{E\, h_r}{E\, h_r + h_c}\, \theta_{ro} + \frac{h_c}{E\, h_r + h_c}\, \theta_{ao} \qquad (3.48)$$

and the external surface resistance is given by:

$$R_{se} = \frac{1}{E_{hr} + h_c} \quad (3.49)$$

For design purposes it is usual to assume that $\theta_{ro} = \theta_{ao}$, therefore:

$$\theta_{eo} = \theta_{ao} \quad (3.50)$$

Equation 3.48 can be used to calculate heat losses from roofs, for example, where there is significant radiation to a clear sky, the temperature of which (θ_{ro}) is lower than θ_{ao}. Note that neither the outside surface resistance nor the U-value is affected by such considerations.

Solar radiation on exterior surfaces has the effect of reducing the heat loss through the component. In appropriate cases this can be allowed for by using the sol-air temperature in place of the external air temperature.

Appendix 3.A4: Seasonal heat losses through ground floors

The U-value of a ground floor (see section 3.5) relates the average heat loss over one year to the average temperature difference over the same period.

The annual variation of external temperature about its mean value gives rise to a similar variation in heat flow but out of phase with the temperature variation(A4-1). This phase difference is generally between about two weeks and about three months, and is greater for insulated floors than for uninsulated floors.

Measurements have shown(A4-1) that when an appreciable phase difference is involved, the average seasonal heat loss is approximately that calculated using the average annual internal and external temperatures, i.e:

$$\Phi_f = A_f\, U_f\, (\bar{\theta}_{ei} - \bar{\theta}_{ao}) \quad (3.51)$$

where Φ_f is the rate of heat loss through the floor (W), A_f is the area of the floor (m^2), U_f is the thermal transmittance of the floor (W·m^{-2}·K^{-1}), $\bar{\theta}_{ei}$ is the annual average inside environmental temperature (°C) and $\bar{\theta}_{ao}$ is the annual average outside air temperature (°C).

With a smaller phase difference, the average seasonal heat loss is given by:

$$\Phi_f = A_f\, U_f\, (\bar{\theta}_{ei(w)} - \bar{\theta}_{ao(w)}) \quad (3.52)$$

where $\bar{\theta}_{ei(w)}$ is the average inside environmental temperature over the heating season (°C), $\bar{\theta}_{ao(w)}$ is the average outside air temperature over the heating season (°C).

Table 3.34 Wintertime average ($\bar{\theta}_{ao(w)}$) and annual average ($\bar{\theta}_{ao}$) outside air temperatures

Location	Outside air temperature averaged over stated period (/ °C)	
	Wintertime†	Annual
London (Heathrow)	7.2	10.8
Manchester (Ringway)	6.3	9.6
Edinburgh (Turnhouse)	5.5	8.6

† October – April inclusive

Notes:
(1) Figures are 20-year means (1976–1995)
(2) Subtract 0.6 °C for each 100 m elevation above sea level

Values of $\bar{\theta}_{ei(w)}$ and $\bar{\theta}_{ao(w)}$ for three UK locations are given in Table 3.34.

In general, the actual heat losses, averaged over the heating season will fall between the values indicated by equations 3.51 and 3.52.

A more detailed procedure is given in BS EN ISO 13370(33).

References

A4-1 Spooner D C Heat loss measurements through an insulated domestic ground floor *Building Serv. Eng. Res. Technol.* **3**(4) 147–151 (1982)

A4-2 Spooner D C *Private communication*

Appendix 3.A5: Application of the combined method to multiple layer structures

The following example shows how the combined method may be used to determine the thermal resistance of complicated structural elements involving bridged layers.

Example 3.7: Calculation of thermal resistance of foam-filled masonry block

Figure 3.16 shows a hollow masonry block with a single slot, 25 mm thick, closed at one end and filled with expanded polystyrene (EPS) insulation. The thermal conductivities of the masonry and insulation are 0.51 $W{\cdot}m^{-1}{\cdot}K^{-1}$ and 0.035 $W{\cdot}m^{-1}{\cdot}K^{-1}$ respectively.

Since the thermal resistance is to be calculated for the block alone, rather than as part of a multiple-leaf construction, the surface resistances are not included. Therefore, the isothermal planes are considered to be the inner and outer surfaces of the block.

The block may be considered as consisting of three layers, the outer layers being homogeneous masonry and the middle layer being composed of an EPS slab thermally bridged on three sides by masonry. To simplify the calculation, the outer layers may be rearranged as shown in Figure 3.17. Each layer is divided into sections, each of which is composed of a single material only, as shown.

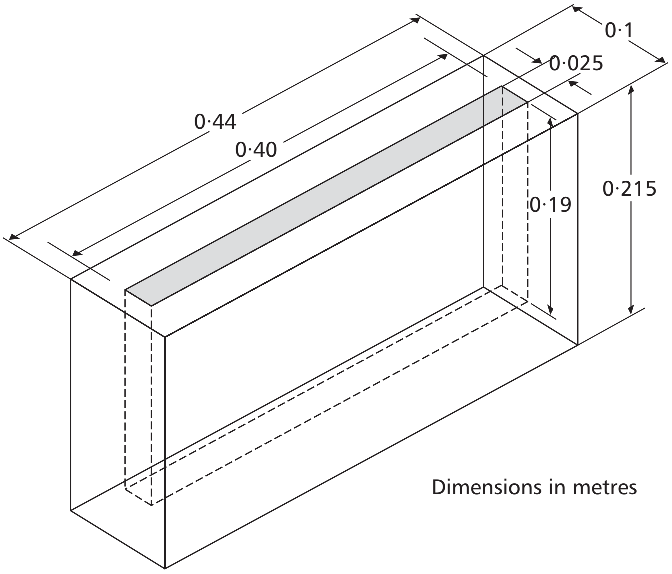


Figure 3.16 Example 3.7: foam-filled masonry block

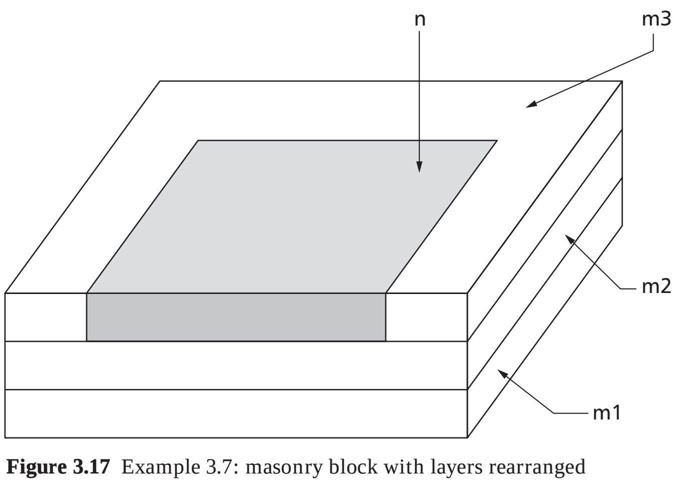


Figure 3.17 Example 3.7: masonry block with layers rearranged

The areas of the sections, and their proportions of the total area, are:

— m_1 (masonry): $A_{m1} = 0.0946$ m^2; $P_{m1} = 1$
— m_2 (masonry): $A_{m2} = 0.0946$ m^2; $P_{m2} = 1$
— m_3 (masonry): $A_{m3} = 0.0186$ m^2; $P_{m3} = 0.197$
— n (EPS): $A_n = 0.076$ m^2; $P_n = 0.803$

The thermal resistances are:

— m_1 (masonry): $R_{m1} = 0.0375 / 0.51 = 0.0735$ $m^2{\cdot}K{\cdot}W^{-1}$
— m_2 (masonry): $R_{m2} = 0.0375 / 0.51 = 0.0735$ $m^2{\cdot}K{\cdot}W^{-1}$
— m_3 (masonry): $R_{m3} = 0.025 / 0.51 = 0.049$ $m^2{\cdot}K{\cdot}W^{-1}$
— n (EPS): $R_n = 0.025 / 0.035 = 0.714$ $m^2{\cdot}K{\cdot}W^{-1}$

(*a*) Lower limit

Using equation 3.13, the thermal resistance of the bridged layer is added to the thermal resistances of the two unbridged layers:

$$R_L = \frac{1}{(P_n / R_n) + (P_{m3} / R_{m3})} + R_{m1} + R_{m2}$$

$$= \left(\frac{1}{(0.803 / 0.714) + (0.197 / 0.049)}\right) + 0.0735 + 0.0735$$

$$= (1 / 5.145) + 0.0735 + 0.0735$$

$$= 0.341 \ m^2{\cdot}K{\cdot}W^{-1}$$

(*b*) Upper limit

Using equation 3.15, the thermal resistances of each section are added together on an area-weighted basis:

$$R_U = \frac{1}{\left(\dfrac{P_n}{R_n + R_{m1} + R_{m2}}\right) + \left(\dfrac{P_{m3}}{R_{m3} + R_{m1} + R_{m2}}\right)}$$

$$= \frac{1}{(0.803/0.861) + (0.197/0.196)}$$

$$= 0.516 \ m^2{\cdot}K{\cdot}W^{-1}$$

The thermal resistance of the masonry block is the arithmetic mean of the upper and lower limits, i.e. 0.428 $m^2{\cdot}K{\cdot}W^{-1}$.

Appendix 3.A6: Calculation method for admittance, decrement factor and surface factor

The temperature distribution in a homogeneous slab subject to one dimensional heat flow is given by the diffusion equation:

$$\frac{\partial^2 \theta}{\partial x^2} = \frac{\rho\, c}{\lambda}\; \frac{\partial \theta}{\partial t} \tag{3.53}$$

where θ is temperature (°C), x is distance in direction perpendicular to surface of slab (m^2), ρ is density (kg·m^{-3}), c is specific heat capacity (J·kg^{-1}·K^{-1}), λ is thermal conductivity (W·m^{-1}·K^{-2}) and t is time (s).

For finite slabs and for sinusoidal temperature variations the temperature and energy cycles can be linked by the use of matrix algebra[41,A6.1], i.e:

$$\begin{bmatrix}\theta_1\\ q_1\end{bmatrix} = \begin{bmatrix} m_1 & m_2\\ m_3 & m_1\end{bmatrix}\begin{bmatrix}\theta_2\\ q_2\end{bmatrix} \tag{3.54}$$

where $q\ (= -\lambda\ \partial\theta/\partial x)$ is the heat flux (W·m^{-2}).

For a slab of homogenous material of thickness d (m), the coefficients of the matrix are given by:

$$m_1 = \cosh\,(p + i\,p) \tag{3.55}$$

$$m_2 = \frac{d \sinh\,(p + i\,p)}{\lambda\,(p + ip)} \tag{3.56}$$

$$m_3 = \frac{\lambda\,(p + ip) \sinh\,(p + i\,p)}{d} \tag{3.57}$$

For a 24-hour cycle:

$$p = \left(\frac{\pi\, d^2 \rho\, c}{86\,400\ \lambda}\right)^{1/2} \tag{3.58}$$

For an air gap, or a surface resistance between a layer and the air, where the diffusivity (i.e. $\lambda / \rho\, c$) is high, the coefficients of the matrix are given by:

$$m_1 = 1 \tag{3.59}$$

$$m_2 = -R_a \text{ or } -R_s \tag{3.60}$$

$$m_3 = 0 \tag{3.61}$$

Clearly, for a composite wall, the matrices of each of the layers can multiplied together to give the relation between inside and outside, as follows:

$$\begin{bmatrix}\theta_i\\ q_i\end{bmatrix} = \begin{bmatrix}1 & -R_{si}\\ 0 & 1\end{bmatrix}\begin{bmatrix}m_1 & m_2\\ m_3 & m_1\end{bmatrix}\begin{bmatrix}n_1 & n_2\\ n_3 & n_1\end{bmatrix}\cdots\begin{bmatrix}1 & -R_{se}\\ 0 & 1\end{bmatrix}\begin{bmatrix}\theta_e\\ q_e\end{bmatrix} \tag{3.62}$$

which can be written:

$$\begin{bmatrix}\theta_i\\ q_i\end{bmatrix} = \begin{bmatrix}M_1 & M_2\\ M_3 & M_4\end{bmatrix}\begin{bmatrix}\theta_e\\ q_e\end{bmatrix} \tag{3.63}$$

Note that the components of this matrix will be complex numbers.

The non-steady state parameters are now derived as follows.

Admittance (Y):

$$Y_c = -\frac{M_4}{M_2} \tag{3.64}$$

$$Y = |Y_c| \tag{3.65}$$

$$\omega = \frac{12}{\pi} \arctan\left(\frac{\mathrm{Im}\,(Y_c)}{\mathrm{Re}\,(Y_c)}\right) \tag{3.66}$$

The arctangent should be evaluated in the range 0 to π radians, thus ω is a time lead.

Decrement factor (f):

$$f_c = -\frac{1}{U M_2} \tag{3.67}$$

where U is thermal transmittance (W·m^{-2}·K^{-1}).

$$f = |f_c| \tag{3.68}$$

$$\phi = \frac{12}{\pi} \arctan \frac{\mathrm{Im}\,(f_c)}{\mathrm{Re}\,(f_c)} \tag{3.69}$$

The arctangent should be evaluated in the range $-\pi$ to 0 radians, thus ϕ is a time lag.

Surface factor (F):

$$F_c = 1 - R_{si}\, Y_c \tag{3.70}$$

$$F = |F_c| \tag{3.71}$$

$$\psi = \frac{12}{\pi} \arctan \frac{\mathrm{Im}\,(F_c)}{\mathrm{Re}\,(F_c)} \tag{3.72}$$

As with decrement factor, the arctangent should be evaluated in the range $-\pi$ to 0 radians, thus ψ is a time lag.

For internal partitions, the decrement factor is combined with the admittance and surface factor, i.e:

$$Y_{ci} = (Y_c - U f_c) = -\frac{M_4 - 1}{M_2} \tag{3.73}$$

$$F_{ci} = 1 - R_{si}\, Y_{ci} \tag{3.74}$$

Equations 3.65, 3.66, 3.71 and 3.72 can then be used as before.

page corrected: Jan 07

The heat capacity of the element per unit area is:

$$\chi = \frac{t}{2\,\pi}\left|\frac{M_4 - 1}{M_2}\right| \tag{3.75}$$

where χ is the heat capacity per unit area ($J{\cdot}K^{-1}{\cdot}m^{-2}$) and t is the period of the temperature cycle (s)

Example 3.8: Non-steady-state properties for a solid external wall

Properties of the wall are as follows:

— thickness, $d = 0.22$ m

— density, $\rho = 1700$ $kg{\cdot}m^{-3}$

— thermal conductivity, $\lambda = 0.84$ $W{\cdot}m^{-1}{\cdot}K^{-1}$

— specific heat capacity, $c = 800$ $J{\cdot}kg^{-1}{\cdot}K^{-1}$.

The thermal transmittance is:

$$U = \frac{1}{(0.13 + \frac{0.22}{0.84} + 0.04)} = 2.32\ W{\cdot}m^{-2}{\cdot}K^{-1}.$$

From equation 3.58:

$$p = 1.688$$

For manual calculation, it is convenient to express cosh $(p + ip)$ and sinh $(p + ip)$ in terms of the functions $\sin p$, $\cos p$ and e^p, i.e:

$$\cosh(p + ip) = \tfrac{1}{2}\,[(e^p + e^{-p})\cos p + i(e^p - e^{-p})\sin p] \tag{3.76}$$

$$\sinh(p + ip) = \tfrac{1}{2}\,[(e^p - e^{-p})\cos p + i(e^p + e^{-p})\sin p] \tag{3.77}$$

In the matrix, the coefficient m_1, is given by equation 3.76; m_2 and m_3 are given by:

$$\begin{aligned} m_2 = \{d\,[(e^p - e^{-p})\cos p + (e^p + e^{-p})\sin p \\ - i(e^p - e^{-p})\cos p + i(e^p + e^{-p})\sin p]\}\,/\,4\,\lambda p \end{aligned} \tag{3.78}$$

$$\begin{aligned} m_3 = \{\lambda p\,[(e^p - e^{-p})\cos p + (e^p + e^{-p})\sin p \\ + i(e^p - e^{-p})\cos p + i(e^p + e^{-p})\sin p]\}/\,2\,d \end{aligned} \tag{3.79}$$

Which gives the matrix as follows:

$$\begin{bmatrix} -0.3270 + i2.594 & -0.1918 - i0.2392 \\ 19.87 - i15.93 & -0.3270 + i2.594 \end{bmatrix}$$

Performing the matrix multiplication from left to right (which corresponds to inside to outside):

$$\begin{bmatrix} 1 & -0.13 \\ 0 & 1 \end{bmatrix}\begin{bmatrix} -0.3270 + i2.594 & -0.1918 - i0.2392 \\ 19.87 - i15.93 & -0.3270 + i2.594 \end{bmatrix}$$

$$= \begin{bmatrix} -2.910 + i4.665 & -0.1492 - i0.5764 \\ 19.87 - i15.93 & -0.3270 + i2.594 \end{bmatrix}$$

Since M_2 and M_4 only are required, it is necessary to evaluate the right-hand column only of the product matrix, i.e:

$$\begin{bmatrix} -2.910 + i4.665 & -0.1492 - i0.5764 \\ 19.87 - i15.93 & -0.3270 + i2.594 \end{bmatrix}\begin{bmatrix} 1 & -0.04 \\ 0 & 1 \end{bmatrix}$$

$$= \begin{bmatrix} * & -0.0329 - i0.7630 \\ * & -1.122 + i3.231 \end{bmatrix}$$

From equations 3.64, 3.65 and 3.66:

$$Y_c = -\frac{-1.122 + i3.231}{-0.0329 - i0.7630}$$

$$= -\frac{(-1.122 + i3.231)(-0.0329 - i0.7630)}{0.0329^2 + 0.7630^2}$$

$$= 4.16 + i1.65$$

$$Y = 4.48\ W{\cdot}m^{-2}{\cdot}K^{-1}$$

$$\omega = 1.44\ h$$

From equations 3.67, 3.68 and 3.69:

$$f_c = -\frac{1}{2.32\,(-0.0329 - i0.7630)}$$

$$= -\frac{(-0.0329 + i0.7630)}{2.32\,(0.0329^2 + 0.7630^2)}$$

$$= -0.024 - i0.5640$$

$$f = 0.56$$

$$\phi = -5.84\ h$$

From equations 3.70, 3.71 and 3.72:

$$F_c = 1 - 0.13\,(4.16 + i1.65) = 0.459 - i0.214$$

$$F = 0.51$$

$$\psi = -1.67\ h$$

From equation 3.75:

$$\chi = \frac{86400}{2 \times 3.142}\left|\frac{-2.122 + i3.231}{-0.0329 - i0.7630}\right|$$

$$= 13749.0 \times \left|\frac{(-2.122 + i3.231)(-0.0329 + i0.7630)}{0.0329^2 + 0.7630^2}\right|$$

$$= |-56468 - i40672| = 69590\ J{\cdot}K^{-1}{\cdot}m^2$$

$$= 69.6\ kJ{\cdot}K^{-1}{\cdot}m^2$$

Reference

1 Pipes L A Matrix analysis of heat transfer problems *J. Franklin Inst.* 623 195–206 (1957)

Appendix 3.A7: Properties of materials

The data tabulated in this appendix have been abstracted, with permission, from the report of a thorough review of existing data sets undertaken by the Building Environmental Performance Analysis Club (BEPAC)[A7.1]. Fourteen data sets were studied, see Table 3.35.

Tables 3.36 to 3.46 contain only a selection of the data given in the BEPAC report, these being regarded as broadly representative of the materials listed. In each case the source of the data is identified by a code which refers to one of the data sets listed in Table 3.35.

Thermal conductivity, density and specific heat capacity data for typical materials are given in Tables 3.36 to 3.39. Table 3.36 refers to impermeable materials, i.e. those which act as a barrier to water in the vapour and/or liquid states, and whose hygrothermal properties do not alter by absorbing water. Table 3.37 includes lightweight insulation materials, such as mineral wools and foamed plastics, which display water vapour permeability, zero hygroscopic water content and an apparent thermal conductivity, and which operate under conditions of air-dry equilibrium normally protected from wetting by rain. Table 3.38 deals with masonry and related materials which are inorganic, porous and may contain significant amounts of water (due to hygroscopic absorption from the air or wetting by rain) which affects their hygrothermal properties and their thermal conductivity in particular. Table 3.39 provides data for organic materials such as wood and wood-based products which are porous and strongly hygroscopic and which display a highly non-linear water vapour permeability.

Tables 3.40 to 3.42 give absorptivity and emissivity data for impermeable, inorganic porous and hygroscopic materials, respectively.

Tables 3.43 to 3.46 contain vapour resistivity values for a wide variety of materials divided into four categories as for Tables 3.36 to 3.39.

Table 3.35 Sources of data

Data source	Code
ASHRAE, USA	A[A7.2]
BS 5250, UK	BS[A7.3]
CIBSE, UK	C[A7.4]
CSTC, Belgium	T[A7.5]
DOE-2 Program, USA	D[A7.6]
ESP program, UK	E[A7.7]
Eurosol, UK	Eu
France	F[A7.8]
Germany	G[A7.9]
Italy	Y[A7.10]
India	I[A7.11]
Leeds University, UK	L[A7.12,A7.13]
Leuven University, Belgium	B[A7.14]
Netherlands	S[A7.15]

References

A7.1 Clark J A, Yanske P P and Pinney A A *The harmonisation of thermal properties of building materials* BEPAC Research Report (Reading: Building Environmental Performance Analysis Club) (1990)

A7.2 *Applications* ASHRAE Handbook (Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers) (1985)

A7.3 BS 5250: 1989 (1995): *Code of practice for the control of condensation in buildings* (London: British Standards Institution) (1989)

A7.4 *Design data* CIBSE Guide A and *Reference data* CIBSE Guide C (London: Chartered Institution of Building Services Engineers) (1986)

A7.5 NBN B62-002: *Thermische geleidbaarheid van de bouwmaterialen/Conductivities thermiques des materiaux de construction* (Brussels: Institut Belge de Normalisation) (1980)

A7.6 *DOE-2 Reference Manual* Report No. LBL-8706 (Berkeley CA, USA: Lawrence Berkeley Laboratory) (1984)

A7.7 *ESP-r Reference Manual* (Glasgow: Energy Simulation Research Unit/University of Strathclyde) (1989)

A7.8 DTU Regles Th-K77 *Cahiers du CSTB* **1478** November 1977 (Marne La Vallee: Centre Scientific et Technique du Batiment) (1977)

A7.9 DIN 4108: *Warescutz im Hochbau, Warme und feuteschutztechnische Kenwerte* (Berlin: Deutsches Institut fur Normung) (1981)

A7.10 UNI 7357: *Conduttivita termica apparente di materiali* (Milan: Ente Nazionale Italiano di Unificazione) (1974)

A7.11 IS 3792-1978: *Revised guide for heat insulation of non-industrial buildings* (New Delhi: Indian Standards Institution)

A7.12 Tinker J A *Aspects of mix proportioning and moisture content on the thermal conductivity of lightweight aggregate concretes* PhD thesis (Salford: University of Salford) (1985)

A7.13 Tinker J A and O'Rourke A Development of a low thermal conductivity building mortar *Second Europ. Conf. on Architecture, Paris, December 1989* (1990)

A7.14 *IEA Annex XIV: Condensation and Energy 1: Material properties* (Leuven: Laboratorium voor Bouwfysica/Kotholieke Universiteit Leuven) (1991)

A7.15 *Eigenschappen van bouwen isolatiematerialen* Report No. 17 (Rotterdam: Stichting Bouwresearch)

Table 3.36 Thermal conductivity, density and specific heat capacity: impermeable materials

Material	Condition/test (where known)	Source	Thermal conductivity / $W \cdot m^{-1} \cdot K^{-1}$	Density / $kg \cdot m^{-3}$	Specific heat / $J \cdot kg^{-1} \cdot K^{-1}$
Asphalt		C	0.50	1700	1000
		E	1.20	2300	1700
— poured		T	1.20	2100	920
— reflective coat		E	1.20	2300	1700
— roofing, mastic		E	1.15	2330	840
Bitumen					
— composite, flooring		E	0.85	2400	1000
— insulation, all types		T	0.20	1000	1700
Ceramic, glazed		T	1.40	2500	840
Glass					
— cellular sheet		T	0.048	140	840
— foam	At 50°C	I	0.056	130	750
		S	0.052	140	840
— solid (soda-lime)	At 10°C	C	1.05	2500	840
Linoleum		T	0.19	1200	1470
Metals					
— aluminium		T	230	2700	880
— aluminium cladding		D	45	7680	420
— brass		T	110	8500	390
— bronze		T	64	8150	—
— copper		T	384	8600	390
— duraluminium		T	160	2800	580
— iron		T	72	7900	530
— iron, cast		T	56	7500	530
— lead		T	35	11340	130
— stainless steel, 5% Ni		T	29	7850	480
— stainless steel, 20% Ni		T	16	8000	480
— steel		T	45	7800	480
— tin		T	65	7300	240
— zinc		T	113	7000	390
Polyvinylchloride (PVC)		E	0.16	1380	1000
— tiles		T	0.19	1200	1470
Roofing felt		E	0.19	960	840
Rubber		T	0.17	1500	1470
— expanded board, rigid		G	0.032	70	1680
— hard		E	0.15	1200	1000
— tiles		E	0.30	1600	2000

Table 3.37 Thermal conductivity, density and specific heat capacity: non-hygroscopic materials

Material	Condition/test (where known)	Source	Thermal conductivity / $W \cdot m^{-1} \cdot K^{-1}$	Density / $kg \cdot m^{-3}$	Specific heat / $J \cdot kg^{-1} \cdot K^{-1}$
Carpet/underlay:					
— with cellular rubber underlay		E	0.10	400	1360
— synthetic		E	0.06	160	2500
Foam:					
— phenol		C	0.040	30	1400
— phenol, rigid		S	0.035	110	1470
— polyisocyanate		S	0.030	45	1470
— polyurethane		T	0.028	30	1470
— polyurethane, freon-filled		S	0.030	45	1470
— polyvinylchloride		S	0.035	37	1470
— urea formaldehyde		C	0.04	10	1400
— urea formaldehyde resin		S	0.054	14	1470
Glass fibre/wool:					
— fibre quilt		C	0.040	12	840
— fibre slab		C	0.035	25	1000
— fibre, strawboard-like		S	0.085	300	2100

Table continues

Table 3.37 Thermal conductivity, density and specific heat capacity: non-hygroscopic materials — *continued*

Material	Condition/test (where known)	Source	Thermal conductivity / W·m^{-1}·K^{-1}	Density / kg·m^{-3}	Specific heat / J·kg^{-1}·K^{-1}
Glass fibre/wool (continued):					
— wool	At 10°C	Eu	0.040	10	840
	At 10°C	Eu	0.040	12	840
	At 10°C	Eu	0.037	16	840
	At 10°C	Eu	0.033	24	840
	At 10°C	Eu	0.032	32	840
	At 10°C	Eu	0.03	48	840
	At 10°C	Eu	0.031	80	840
— wool, resin bonded	At 50°C	I	0.036	24	1000
Loose fill/powders:					
— cellulosic insulation		A	0.042	43	1380
— exfoliated vermiculite	At 50°C	I	0.069	260	880
— floor/roof screed		C	0.41	1200	840
— glass, granular cellular		T	0.07	180	840
— gravel		E	0.36	1840	840
— perlite, expanded		A	0.051	100	1090
— polystyrene, moulded beads	At 10°C	A	0.036	16	1210
— roof gravel or slag		D	1.44	880	1680
— sand		I	1.74	2240	840
— stone chippings for roofs		C	0.96	1800	1000
— white dry render		E	0.50	1300	1000
Mineral fibre/wool:					
— fibre blanket, bonded	At 10°C	A	0.042	12	710
	At 10°C	A	0.036	24	710
	At 10°C	A	0.032	48	710
— fibre blanket, metal reinforced	At 37.7°C	A	0.038	140	710
	At 93.3°C	A	0.046	140	710
— fibre board, preformed		D	0.042	240	760
— fibre board, wet felted		A	0.051	290	800
— fibre board, wet moulded		A	0.061	370	590
— fibre board, resin bonded		A	0.042	240	710
— fibre, textile, organic bonded	At 10°C	A	0.043	10	710
— fibre slag, pipe insulation	At 23.8°C	A	0.036	100	710
	At 23.8°C	A	0.048	200	710
	At 93.3°C	A	0.048	100	710
	At 93.3°C	A	0.065	200	710
— wool		S	0.038	140	840
— wool, fibrous		D	0.043	96	840
— wool, resin bonded		I	0.036	99	1000
Miscellaneous materials:					
— acoustic tile		D	0.057	290	1340
— felt sheathing		E	0.19	960	950
— mineral filler for concrete		S	0.13	430	840
— perlite, bitumen bonded		S	0.061	240	840
— perlite, expanded, hard panels		T	0.055	170	840
— perlite, expanded, pure		T	0.046	65	840
— plastic tiles		E	0.50	1050	840
— polyurethane, expanded		D	0.023	24	1590
— polyurethane, unfaced	At 10°C	G	0.023	32	1590
Carpet/underlay:					
— polyurethane board, cellular		A	0.023	24	1590
— polyisocyanurate board		A	0.020	32	920
— polyisocyanurate board,					
— foil-faced, glass-fibre reinforced	At 10°C	A	0.019	32	920
— polystyrene, expanded (EPS)		S	0.035	23	1470
— polystyrene, extruded (EPS)		S	0.027	35	1470
— polyvinylchloride (PVC), expanded		E	0.04	100	750
— vermiculite, expanded, panels		T	0.082	350	840
— vermiculite, expanded, pure		T	0.058	350	840
— silicon		E	0.18	700	1000
Rock wool	At 10°C	Eu	0.037	23	710
	At 10°C	Eu	0.033	60	710
	At 10°C	Eu	0.033	100	710
	At 10°C	Eu	0.034	200	710
— unbonded		I	0.047	92	840
		I	0.043	150	840

Table 3.38 Thermal conductivity, density and specific heat capacity: inorganic, porous materials

Material	Condition/test (where known)	Source	Thermal conductivity / W·m^{-1}·K^{-1}	Density / kg·m^{-3}	Specific heat / J·kg^{-1}·K^{-1}
Asbestos-related materials:					
— asbestos cement		S	1.02	1750	840
— asbestos cement building board		D	0.6	1920	840
— asbestos cement decking		C	0.36	1500	1050
— asbestos cement sheet	Conditioned	C	0.36	700	1050
— asbestos fibre	At 50°C	I	0.06	640	840
— asbestos mill board	At 50°C	I	0.25	1400	840
Brick		D	0.72	1920	840
		D	1.31	2080	921
— aerated		S	0.30	1000	840
— brickwork, inner leaf		C	0.62	1700	800
— brickwork, outer leaf		C	0.84	1700	800
— burned		S	0.75	1300	840
		S	0.85	1500	840
		S	1.00	1700	840
— mud	At 50°C	I	0.75	1730	880
— paviour		E	0.96	2000	840
— reinforced	At 50°C	I	1.10	1920	840
— tile	At 50°C	I	0.8	1890	880
Cement/plaster/mortar:					
— cement		D	0.72	1860	840
— cement blocks, cellular		T	0.33	520	2040
— cement fibreboard, magnesium oxysulphide binder		A	0.082	350	1300
— cement mortar		S	0.72	1650	920
	Dry	T	0.93	1900	840
	Moist	T	1.5	1900	840
— cement/lime plaster		S	0.8	1600	840
— cement panels, wood fibres	Dry	T	0.08	350	1890
	Moist	T	0.12	350	3040
		T	0.12	400	1470
	Dry	T	0.35	1650	840
— cement plaster		S	0.72	1760	840
		S	1.50	1900	840
— cement plaster, sand aggregate		A	0.72	1860	840
— cement screed		E	1.40	2100	650
— gypsum		E	0.42	1200	840
— gypsum plaster		S	0.51	1120	960
— gypsum plaster, perlite aggregate		A	0.22	720	1340
— gypsum plaster, sand aggregate		A	0.81	1680	840
— gypsum plasterboard		D	0.16	800	840
		S	0.65	1100	840
— gypsum plastering		S	0.80	1300	840
— limestone mortar		T	0.70	1600	840
— plaster		T	0.22	800	840
		T	0.35	950	840
		T	0.52	1200	840
— plaster ceiling tiles		E	0.38	1120	840
— plaster, lightweight aggregate		D	0.23	720	840
— plaster, sand aggregate		D	0.82	1680	840
— plasterboard		C	0.16	950	840
— render, synthetic resin,					
— exterior insulation		T	0.70	1100	900
— rendering	Moisture content 1%	E	1.13	1430	1000
	Moisture content 8%	E	0.79	1330	1000
— vermiculite plaster		E	0.20	720	840
Ceramic/clay tiles:					
— ceramic tiles	Dry	T	1.20	2000	850
— ceramic floor tiles	Dry	T	0.80	1700	850
— clay tiles		E	0.85	1900	840
— clay tile, burnt		S	1.3	2000	840
— clay tile, hollow, 10.2 mm, 1 cell		D	0.52	1120	840
— clay tile, hollow, 20.3 mm, 2 cells		D	0.623	1120	840
— clay tile, hollow, 32.5 mm, 3 cells		D	0.693	1120	840
— clay tile, pavior		D	1.803	1920	840

Table continues

Table 3.38 Thermal conductivity, density and specific heat capacity: inorganic, porous materials — *continued*

Material	Condition/test (where known)	Source	Thermal conductivity / W·m^{-1}·K^{-1}	Density / kg·m^{-3}	Specific heat / J·kg^{-1}·K^{-1}
Concrete blocks/tiles:					
— block, aerated		E	0.24	750	1000
— block, heavyweight, 300 mm		D	1.31	2240	840
— block, lightweight, 150 mm		D	0.66	1760	840
— block, lightweight, 300 mm		D	0.73	1800	840
	Dry	T	0.24	620	840
	Dry	T	0.25	670	840
	Dry	T	0.26	720	840
	Dry	T	0.30	750	840
	Dry	T	0.28	770	840
	Dry	T	0.29	820	840
	Dry	T	0.30	870	840
— block, mediumweight, 150 mm		D	0.77	1900	840
— block, mediumweight, 300 mm		D	0.83	1940	840
	Dry	T	0.31	920	840
	Dry	T	0.32	970	840
	Dry	T	0.35	1050	840
	Dry	T	0.40	1150	840
— block, hollow, heavyweight, 300 mm		D	1.35	1220	840
— block, hollow, lightweight, 150 mm		D	0.48	880	840
— block, hollow, lightweight, 300 mm		D	0.76	780	840
— block, hollow, mediumweight, 150 mm		D	0.62	1040	840
— block, hollow, mediumweight, 300 mm		D	0.86	930	840
— block, partially filled, heavyweight, 300 mm	D	1.35	1570	840	
— block, partially filled, lightweight, 150 mm		D	0.55	1170	840
— block, partially filled, lightweight, 300 mm		D	0.74	1120	840
— block, partially filled, mediumweight, 150 mm		D	0.64	1330	840
— block, partially filled, mediumweight,300 mm		D	0.85	1260	840
— block, perlite-filled, lightweight, 150 mm		D	0.17	910	840
— block, perlite-filled, mediumweight,150mm		D	0.2	1070	840
— block, with perlite, lightweight, 150 mm		D	0.33	1180	840
— block, with perlite,					
— mediumweight, 150 mm		D	0.39	1340	840
— tiles		E	1.10	2100	840
Concrete, cast:					
— aerated		E	0.16	500	840
		S	0.29	850	840
		S	0.42	1200	840
— aerated, cellular		S	0.15	400	840
		S	0.23	700	840
		S	0.70	1000	840
		S	1.20	1300	840
— aerated, cement/lime based		S	0.21	580	840
— cellular		T	0.16	480	840
	At 50°C	I	0.19	700	1050
— cellular bonded		T	0.30	520	2040
— dense		S	1.70	2200	840
— compacted,		S	2.20	2400	840
— dense, reinforced		S	1.90	2300	840
— compacted		S	2.30	2500	840
— expanded clay filling		S	0.26	780	840
		S	0.60	1400	840
— foamed	At 50°C	I	0.07	320	920
	At 50°C	I	0.08	400	920
	At 50°C	I	0.15	700	920
— foam slag		E	0.25	1040	960

Table continues

Table 3.38 Thermal conductivity, density and specific heat capacity: inorganic, porous materials — *continued*

Material	Condition/test (where known)	Source	Thermal conductivity / $W \cdot m^{-1} \cdot K^{-1}$	Density / $kg \cdot m^{-3}$	Specific heat / $J \cdot kg^{-1} \cdot K^{-1}$
Concrete, cast (continued):					
— glass reinforced		E	0.90	1950	840
— heavyweight	Dry	T	1.30	2000	840
	Moist	T	1.70	2000	840
— lightweight	Dry	T	0.20	620	840
	Dry	T	0.25	750	840
	Dry	T	0.21	670	840
	Dry	T	0.22	720	840
	Dry	T	0.23	770	840
	Dry	T	0.24	820	840
	Dry	T	0.25	870	840
	Moist	T	0.43	750	840
	Moist	T	0.38	770	840
	Moist	T	0.40	820	840
	Moist	T	0.43	870	840
		S	0.08	200	840
		S	0.12	300	840
		S	0.17	500	840
		S	0.23	700	840
— mediumweight	Dry	T	0.32	1050	840
	Dry	T	0.37	1150	840
	Dry	T	0.59	1350	840
	Dry	T	0.84	1650	840
	Dry	T	0.37	1050	840
	Dry	T	0.27	920	840
	Dry	T	0.29	980	840
	Moist	T	0.59	1050	840
		S	0.50	1000	840
		S	0.80	1300	840
		S	1.20	1600	840
		S	1.40	1900	840
— mediumweight, with lime	At 50°C	I	0.73	1650	880
— no fines		E	0.96	1800	840
— residuals of iron works		S	0.35	1000	840
		S	0.45	1300	840
		S	0.70	1600	840
		S	1.00	1900	840
— roofing slab, aerated		C	0.16	500	840
— vermiculite aggregate		E	0.17	450	840
— very lightweight		T	0.14	370	840
		T	0.15	420	840
		T	0.16	470	840
		T	0.17	520	840
		T	0.18	570	840
		T	0.12	350	840
		T	0.18	600	840
Masonry:					
— block, lightweight		T	0.19	470	840
		T	0.20	520	840
		T	0.22	570	840
		T	0.22	600	840
— block, mediumweight	Dry	T	0.60	1350	840
	Dry	T	0.85	1650	840
	Dry	T	1.30	1800	840
— heavyweight	Dry	T	0.90	1850	840
	Dry	T	0.73	1850	840
	Dry	T	0.79	1950	840
	Dry	T	0.90	2050	840
	Moist	T	0.81	1650	840
— lightweight	Dry	T	0.22	750	840
	Dry	T	0.27	850	840
	Dry	T	0.24	850	840
	Dry	T	0.27	950	840
— mediumweight	Dry	T	0.32	1050	840
	Dry	T	0.54	1300	840
	Dry	T	0.37	1150	840
	Dry	T	0.42	1250	840
	Dry	T	0.45	1350	840
	Dry	T	0.49	1450	840

Table continues

Table 3.38 Thermal conductivity, density and specific heat capacity: inorganic, porous materials — *continued*

Material	Condition/test (where known)	Source	Thermal conductivity / $W \cdot m^{-1} \cdot K^{-1}$	Density / $kg \cdot m^{-3}$	Specific heat / $J \cdot kg^{-1} \cdot K^{-1}$
Masonry (continued):					
— mediumweight	Dry	T	0.54	1550	840
— quarry-stones, calcareous	Dry	T	1.40	2200	840
Miscellaneous materials:					
— aggregate	Undried	D	1.8	2240	840
— aggregate (sand, gravel or stone)	Oven dried	A	1.3	2240	920
— building board, tile and lay-in panel		A	0.058	290	590
— calcium silicate brick		S	1.50	2000	840
— granolithic		E	0.87	2085	840
— mud phuska	At 50°C	I	0.52	1620	880
— tile bedding		E	1.40	2100	650
— tile hanging		C	0.84	1900	800
Roofing materials:					
— built-up roofing		D	0.16	1120	1470
— roof tile		C	0.84	1900	800
— tile, terracotta		T	0.81	1700	840
Soil:					
— alluvial clay, 40% sands		I	1.21	1960	840
— black cotton clay, Indore		I	0.61	1680	880
— black cotton clay, Madras		I	0.74	1900	880
— diatomaceous, Kieselguhr or infusorial earth	Moisture content 9%	E	0.09	480	180
— earth, common		E	1.28	1460	880
— earth, gravel-based		E	0.52	2050	180
Stone:					
— basalt		T	3.49	2880	840
— gneiss		T	3.49	2880	840
— granite		T	3.49	2880	840
— granite, red		E	2.9	2650	900
— hard stone (unspecified)		T	3.49	2880	840
		S	2.9	2750	840
— limestone		E	1.5	2180	720
		S	2.9	2750	840
	At 50°C	I	1.80	2420	840
— marble		S	2.9	2750	840
	Dry	T	2.91	2750	840
	Moist	T	3.49	2750	840
— marble, white		E	2	2500	880
— petit granit (blue stone)	Dry	T	2.91	2700	840
	Moist	T	3.49	2700	840
— porphyry		T	3.49	2880	840
— sandstone		E	1.83	2200	710
		T	3	2150	840
		T	1.3	2150	840
		S	5	2150	840
— sandstone tiles	Dry	T	1.2	2000	840
— slate		D	1.44	1600	1470
	At 50°C	I	1.72	2750	840
— slate shale		T	2.1	2700	840
— white calcareous stone	Firm, moist	T	2.09	2350	840
	Firm, dry	T	1.74	2350	840
	Hard, moist	T	2.68	2550	840
	Hard, dry	T	2.21	2550	840
— tufa, soft	Dry	T	0.35	1300	840
	Moist	T	0.50	1300	1260

Table 3.39 Thermal conductivity, density and specific heat capacity: organic, hygroscopic materials

Material	Condition/test (where known)	Source	Thermal conductivity / $W \cdot m^{-1} \cdot K^{-1}$	Density / $kg \cdot m^{-3}$	Specific heat / $J \cdot kg^{-1} \cdot K^{-1}$
Cardboard/paper:					
— bitumen impregnated paper		E	0.06	1090	1000
— laminated paper		A	0.072	480	1380
Cloth/carpet/felt:					
— bitumen/felt layers		C	0.50	1700	1000
— carpet, simulated wool		E	0.06	200	1360
— carpet, Wilton		E	0.06	190	1360
— felt, semi-rigid, organic	At 37.7°C	A	0.035	48	710
bonded	At 37.7°C	A	0.039	88	710
— jute felt	At 50°C	I	0.042	290	880
— jute fibre	At 50°C	I	0.067	330	1090
— wool felt underlay		E	0.04	160	1360
Cork:		E	0.04	110	1800
— board		E	0.04	160	1890
— expanded		S	0.044	150	1760
— expanded, impregnated		S	0.043	150	1760
— slab		I	0.043	160	960
		I	0.055	300	960
— tiles	Conditioned	E	0.08	530	1800
Grass/straw materials:					
— straw board	At 50°C	I	0.057	310	1300
— straw fibreboard or slab		S	0.10	300	2100
— straw thatch		E	0.07	240	180
Miscellaneous materials:					
— afzelia, minunga, meranti		T	0.29	850	2070
— ebonite, expanded		S	0.035	100	1470
— perlite board, expanded, organic bonded		A	0.052	16	1260
— glass fibre board, organic bonded		A	0.036	100	960
— weatherboard		E	0.14	650	2000
Organic materials and their derivatives:					
— coconut pith insulation board	At 50°C	I	0.06	520	1090
— coir board	At 50°C	I	0.038	97	1000
— flax shive, cement bonded board		S	0.10	520	1470
— flax shive, resin bonded board		S	0.12	500	1880
— rice husk	At 50°C	I	0.051	120	1000
— vegetable fibre sheathing		A	0.055	290	1300
Woods:					
— fir, pine		A	0.12	510	1380
— hardwood (unspecified)		T	0.05	90	2810
	Dry	T	0.17	700	1880
		S	0.23	800	1880
— maple, oak and similar hardwoods		A	0.16	720	1260
— oak, radial		E	0.19	700	2390
— oak, beech, ash, walnut	Moist	T	0.23	650	3050
meranti	Dry	T	0.17	650	2120
— pine, pitch pine	Dry	T	0.17	650	2120
	Moist	T	0.23	650	3050
— red fir, Oregon fir	Dry	T	0.14	520	2280
	Moist	T	0.17	520	3440
— resinous woods (spruce, sylvester pine)	Dry	T	0.12	530	1880
— softwood		D	0.12	510	1380
		E	0.13	630	2760
		S	0.14	550	1880
— timber	At 50°C	I	0.072	480	1680
	At 50°C	I	0.14	720	1680
— timber flooring		C	0.14	650	1200
— willow, North Canadian gaboon		T	0.12	420	2400
— willow, birch, soft beech		T	0.14	520	2280
	Moist	T	0.17	520	3440

Table continues

Table 3.39 Thermal conductivity, density and specific heat capacity: organic, hygroscopic materials

Material	Condition/test (where known)	Source	Thermal conductivity / W·m^{-1}·K^{-1}	Density / kg·m^{-3}	Specific heat / J·kg^{-1}·K^{-1}
Wood derivatives:					
— cellulosic insulation, loose fill		A	0.042	43	1380
— chipboard	At 50°C	I	0.067	430	1260
— chipboard, bonded with PF	Dry	T	0.12	650	2340
	Moist	T	0.25	650	5020
— chipboard, bonded with UF	Dry	T	0.12	630	2260
	Moist	T	0.25	630	5020
— chipboard, bonded with melamine	Dry	T	0.12	630	2260
	Moist	T	0.25	630	5020
— chipboard, perforated	At 50°C	I	0.066	350	1260
— flooring blocks		C	0.14	650	1200
— hardboard		E	0.08	600	2000
		A	0.12	880	1340
		S	0.29	1000	1680
— multiplex, beech	Dry	T	0.15	650	2300
— multiplex, North Canadian gaboon	Dry	T	0.12	450	2300
— multiplex, red fir	Dry	T	0.13	550	2300
	Moist	T	0.21	550	2300
— particle board		I	0.098	750	1300
		A	0.17	1000	1300
		A	0.12	800	1300
— plywood		A	0.12	540	1210
		E	0.15	700	1420
— sawdust	At 50°C	I	0.051	190	1000
— softboard	At 50°C	I	0.047	250	1300
— wallboard	At 50°C	I	0.047	260	1260
— wood chip board, cement bonded		S	0.15	530	1470
— wood fibres, compressed		Y	0.055	320	100
— wood (soft) fibre, loose fill		A	0.043	45	1380
— wood particle panels		T	0.09	300	1880
		T	0.12	500	1880
		T	0.14	700	1880
	Hard	T	0.29	1000	1990
	Soft	T	0.08	250	2520
— wood shingle		D	0.12	510	1260
— woodwool board, cement	At 50°C	I	0.081	400	1130
bonded	At 50°C	I	0.11	670	1130
— woodwool roofing slabs		C	0.10	500	1000
— woodwool, xylolite cement slabs		S	0.11	450	1470
— woodwool		E	0.1	500	1000

Table 3.40 Absorptivity and emissivity: impermeable materials

Material	Condition (where known)	Absorptivity	Emissivity
Aluminium	Polished	0.10–0.40	0.03–0.06
	Dull/rough polish	0.40–0.65	0.18–0.30
	Anodised	—	0.72
Aluminium surfaced roofing		—	0.216
Asphalt	Newly–laid	0.91–0.93	—
	Weathered	0.82–0.89	—
	Block	0.85–0.98	0.90–0.98
Asphalt pavement		0.852–0.928	—
Bitumen/felt roofing		0.86–0.89	0.91
Bitumen pavement		0.86–0.89	0.90–0.98
Brass	Polished	0.30–0.50	0.03–0.05
	Dull	0.40–0.065	0.20–0.30
	Anodised	—	0.59–0.61
Bronze		0.34	—
Copper	Polished	0.18–0.50	0.02–0.05
	Dull	0.40–0.065	0.20–0.30
	Anodised	0.64	0.60
Glass	Normal	*	0.88
	Hemispherical	*	0.84
Iron	Unoxidised	—	0.05
	Bright/polished	0.40–0.65	0.20–0.377
	Oxidised	—	0.736–0.74
	Red rusted	—	0.61–0.65
	Heavily rusted	0.737	0.85–0.94
Iron, cast	Unoxidised/polished	—	0.21–0.24
	Oxidised	—	0.64–0.78
	Strongly oxidised	—	0.95
Iron, galvanised	New	0.64–0.66	0.22–0.28
	Old/very dirty	0.89–0.92	0.89
Lead	Unoxidised	—	0.05–0.075
	Old/oxidised	0.77–0.79	0.28–0.281
Rubber	Hard/glossy	—	0.945
	Grey/rough	—	0.859
Steel	Unoxidised/polished/ stainless	0.20	0.074–0.097
	Oxidised	0.20	0.79–0.82
Tin	Highly polished/ unoxidised	0.10–0.40	0.043–0.084
Paint			
— aluminium		0.30–0.55	0.27–0.67
— zinc		0.30	0.95
Polyvinylchloride (PVC)		—	0.90–0.92
Tile	Light colour	0.3–0.5	0.85–0.95
Varnish		—	0.80–0.98
Zinc	Polished	0.55	0.045–0.053
	Oxidised	0.05	0.11–0.25

* See manufacturers' data

Table 3.41 Absorptivity and emissivity: inorganic, porous materials

Material	Condition (where known)	Absorptivity	Emissivity
Asbestos:			
— board		—	0.96
— paper		—	0.93–0.94
— cloth		—	0.90
— cement	New	0.61	0.95–0.96
	Very dirty	0.83	0.95–0.96
Brick	Glazed/light	0.25–0.36	0.85–0.95
	Light	0.36–0.62	0.85–0.95
	Dark	0.63–0.89	0.85–0.95
Cement mortar, screed		0.73	0.93
Clay tiles	Red, brown	0.60–0.69	0.85–0.95
	Purple/dark	0.81–0.82	0.85–0.95
Concrete		0.65–0.80	0.85–0.95
— tile		0.65–0.80	0.85–0.95
— block		0.56–0.69	0.94
Plaster		0.30–0.50	0.91
Stone:			
— granite (red)		0.55	0.90–0.93
— limestone		0.33–0.53	0.90–0.93
— marble		0.44–0.592	0.90–0.93
— quartz		—	0.90
— sandstone		0.54–0.76	0.90–0.93
— slate		0.79–0.93	0.85–0.98

Table 3.42 Absorptivity and emissivity: hygroscopic materials

Material	Condition (where known)	Absorptivity	Emissivity
Paper		—	0.091–0.94
— white, bond		0.25–0.28	—
Cloth:			
— cotton, black		0.67–0.98	—
— cotton, deep blue		0.82–0.83	—
— cotton, red		0.562	—
— wool, black		0.75–0.88	—
— felt, black		0.775–0.861	—
— fabric (unspecified)		—	0.89–0.92
Wood:			
— beach		—	0.94
— oak		—	0.89–0.90
— spruce		—	0.82
— walnut		—	0.83

Table 3.43 Vapour resistivity: impermeable materials

Material	Density / $kg \cdot m^{-3}$	Vapour resistivity / $MN \cdot s \cdot g^{-1}$
Asphalt (laid)	—	∞
Bitumen roofing sheets	—	2 000–60 000
Bituminous felt	—	15 000
Glass:		
— brick	—	∞
— cellular	—	∞
— expanded/foamed	—	∞
— sheet/mirror/window	—	∞
Linoleum	1200	9 000
Metals and metal cladding	—	∞
Paint, gloss (vapour resistant)	—	40–200
Plastic, hard	—	45 000
Polyvinylchloride (PVC) sheets on tile	—	800–1300
Rubber	1200–1500	4500
Rubber tiles	1200–1500	∞
Tiles:		
— ceramic	—	500–5000
— glazed ceramic	—	∞

Table 3.44 Vapour resistivity: non-hygroscopic materials

Material	Density / $kg \cdot m^{-3}$	Vapour resistivity / $MN \cdot s \cdot g^{-1}$
Mineral fibre/wool:		
— glass fibre/wool	—	5–7
— mineral fibre/wool	—	5–9
— rock wool	—	6.5–7.5
Phenol formaldehyde	—	19–20
Phenolic (closed cell)	—	150–750
Polyethylene foam	—	20 000
Polystyrene:		
— expanded	—	100–750
— extruded	—	600–1500
— extruded without skin	—	350–400
Polyurethylene foam	—	115–1000
Polyvinylchloride (PVC) foam, rigid	—	40–1300
Urea formaldehyde foam	—	5–20

Table 3.45 Vapour resistivity: inorganic, porous materials

Material	Density / $kg \cdot m^{-3}$	Vapour resistivity / $MN \cdot s \cdot g^{-1}$
Asbestos cement	800	70
Asbestos cement sheet, substitutes	1600–1900	185–1000
Brick:		
— blast furnace slag	1000–2000	350–500
— calcium silicate	< 1400	25–50
	> 1400	75–125
— dense	> 2000	100–250
— heavyweight	> 1700	45–70
— lightweight	< 1000	25–50
— mediumweight	> 1300	23–45
— sand lime	< 1400	25–50
	> 1500	75–200
Concrete:		
— blocks (lightweight)	—	15–150
— cast	< 1000	14–33
	> 1000	30–80
	> 1900	115–1000
— cellular	450–1300	9–50
— close textured	—	350–750
— expanded clay	500–1000	25–33
	1000–1800	33–75
— foamed steam hardened	400–800	25–50
— insulating	—	23–26
— natural pumice	500–1400	25–75
— no fines	1800	20
— polystyrene, foamed	400	80–100
— porous aggregate	1000–2000	15–50
— porous aggregate (without quartz sand)	—	25–75
— slag and Rhine sand	1500–1700	50–200
Gypsum plasterboard	—	30–60
Plaster/mortar:		
— cement based	1900–2000	75–205
— lime based	1600–1800	45–205
— gypsum	—	30–60
Stone:		
— basalt	—	∞
— bluestone	—	∞
— clay	—	75
— granite	—	150–∞
— limestone, firm	—	350–450
— limestone, soft	—	130–160
— limestone, soft tufa	—	25–50
— marble	—	150–∞
— porphyry	—	∞
— sandstone	—	75–450
— slate	—	150–450
— slatey shale	—	> 3000
Tile:		
— clay, ceramic	—	750–1500
— floor tiles, ceramic	—	115
— roof tiles, terracotta	—	180–220

Table 3.46 Vapour resistivity: organic, hygroscopic materials

Material	Density / $kg \cdot m^{-3}$	Vapour resistivity / $MN \cdot s \cdot g^{-1}$
Carpet:		
— normal backing	—	7–20
— foam backed/underlay	—	100–300
Chipboard:	—	230–500
— bonded with melanine	—	300–500
— bonded with PF	—	250–750
— bonded with UF	—	200–700
— cement-bonded	—	19–50
Cork:		
— insulation	—	25–50
— expanded	—	23–50
— expanded, impregnated	—	45–230
— expanded, bitumous binding	—	45–230
Corkboard	—	50–200
Fibreboard:	—	150–375
— bitumen-coated	—	25
— cement-based	—	19–50
— hardwood fibres	—	350
— porous wood fibres	—	25
Hardboard	—	230–1000
Mineral/vegetable fibre insulation	—	5
Multiplex	800	200–2000
— light pine	—	80
— North Canadian gaboon	—	80
— red pine	—	875–250
— triplex	700	200–500
Paper	—	500
Particle board, softwood	—	25
Plywood	—	150–2000
— decking	—	1000–6000
— marine	—	230–375
— sheathing	—	144–1000
Strawboard	—	45–70
Wood:		
— ash	—	200–1850
— balsa	—	45–265
— beech	—	200–1850
— beech, soft	—	90–700
— birch	—	90–700
— fir	—	45–1850
— North Canadian gaboon	—	45–1850
— oak	—	200–1850
— pine	—	45–1850
— pine, Northern red/Oregon	—	90–200
— pine, pitch	—	200–1850
— spruce	—	45–1850
— teak	—	185–1850
— walnut	—	200–1850
— willow	—	45–1850
Wood lath	—	4
Woodwool:		
— slab	—	15–40
— cement slab	—	15–50
— magnesia slab	—	19–50

Appendix 3.A8: Thermal properties of typical constructions

Tables 3.49 to 3.55 provide values of thermal transmittance, thermal admittance, decrement factor and surface factor for a range of constructions. The tables are provided to illustrate the thermal properties of typical constructions. For design purposes, values should be calculated for the specific construction under consideration using the methods given in this Guide. The tabulated values have been calculated making certain assumptions about the densities, thermal conductivities and specific heat capacities of the materials involved. These assumed values are given in Tables 3.47and 3.48.

Thermal transmittances in Tables 3.49 to 3.55 have been calculated:

— by the combined method described in 3.3.11, taking account of repeating thermal bridges such as mortar joints, timber framing and wall ties

— using surface resistances appropriate to the heat flow direction for the element concerned (upwards for roofs, horizontal for walls, downwards for floors).

The values of thermal admittance, decrement factor, surface factor and heat capacity in the tables:

— do not take account of thermal bridging, and so refer strictly to plain unbridged areas of the structures (see also 3.8.3)

— were calculated for average surface resistances (taken as equal to those for horizontal heat flow).

The heat capacity represents the heat stored in the construction during a half cycle and released during the subsequent half cycle. For internal partitions the heat capacity is given in the tables for one side of the partition only.

These parameters depend on heat flow direction and so, for roofs and floors, should strictly be different depending on whether heat is entering or leaving the building element (i.e. varying over the period of the cyclic temperature variation). The values shown should therefore be regarded as indicative.

Representative values for thermal conductivity (λ) have been used in the calculation of *U*-values for typical constructions. Manufacturers' products may exhibit better thermal properties than those given for the generic types of material listed in Table 3.47. Where such products are likely to be used, *U*-values may be calculated using thermal conductivity data obtained from tests provided that the data and test methods have been properly accredited, see Appendix 3.A2.

Calculated *U*-values may also under-estimate the actual thermal transmittance due to ventilation within the construction. The effectiveness of thermal insulation depends largely on preventing air movement within the insulating layer. Air flow through gaps and cracks must be avoided, particularly on the warm side of the insulation. Air flow must also be avoided between insulation boards or batts and behind wall linings. Such deficiencies in installation have been shown to add between 0.05 and 0.2 $W \cdot m^{-2} \cdot K^{-1}$ to the nominal *U*-value of the construction.

The use of unrealistic values for the thermal conductivities of building materials also contributes to the under-estimation of *U*-values. Tabulated values are usually based on laboratory measurements using small well-prepared samples. In buildings, the thermal conductivities of the same materials may differ appreciably from the laboratory measurements due to variations in quality during production, storage conditions on site and variations in construction techniques.

Table 3.47 Properties of materials used in calculation of thermal properties of typical constructions

Material	Density / $kg \cdot m^{-1}$	Thermal conductivity / $W \cdot m^{-1} \cdot K^{-1}$	Specific heat capacity / $J \cdot kg^{-1} \cdot K^{-1}$
Masonry materials:			
— sandstone	2300	1.8	1000
— brick (exposed)	1750	0.77	1000
— brick (protected)	1750	0.56	1000
— no–fines concrete	2000	1.33	1000
— concrete block (dense) (exposed)	2300	1.87	1000
— concrete block (dense) (protected)	2300	1.75	1000
— precast concrete (dense) (exposed)	2100	1.56	1000
— precast concrete (dense) (protected)	2100	1.46	1000
— cast concrete	2000	1.33	1000
— cast concrete	1800	1.13	1000
— lightweight aggregate concrete block	600	0.20	1000
— autoclaved aerated concrete block	700	0.20	1000
— autoclaved aerated concrete block	500	0.15	1000
— screed	1200	0.46	1000
— ballast (chips or paving slab)	1800	1.10	1000
Surface materials/finishes:			
— external render (lime,sand)	1600	0.80	1000
— external render (cement,sand)	1800	1.00	1000
— plaster (dense)	1300	0.57	1000
— plaster (lightweight)	600	0.18	1000
— plasterboard (standard)	700	0.21	1000
— plasterboard (fire-resisting)	900	0.25	1000
Insulation materials:			
— mineral wool (quilt)	12	0.042	1030
— mineral wool (batts)	25	0.038	1030
— expanded polystyrene (EPS)	15	0.040	1450
— extruded polystyrene	40	0.035	1400
— polyurethane foam	30	0.025	1400
— urea formaldehyde (UF) foam	10	0.040	1400
— blown fibre	12	0.040	1030
Miscellaneous materials:			
— plywood sheathing	500	0.13	1600
— timber studding	500	0.13	1600
— timber battens	500	0.13	1600
— timber decking	500	0.13	1600
— timber flooring	500	0.13	1600
— timber flooring (hardwood)	700	0.18	1600
— chipboard	600	0.14	1700
— vinyl floor covering	1390	0.17	900
— waterproof roof covering	110	0.23	1000
— wood blocks	600	0.14	1700
— floor joists	500	0.13	1600
— cement-bonded particle board	1200	0.23	1500
— carpet/underlay	200	0.6	1300
— steel	7800	50	450
— stainless steel	7900	17	460
— soil	1500	1.5	1800

Table 3.48 Values of surface and airspace resistance used in calculation of thermal properties of typical constructions

Structure	External surface resistance / $m^2 \cdot K \cdot W^{-1}$	Internal surface resistance / $m^2 \cdot K.W^{-1}$	Airspace resistance / $m^2 \cdot K \cdot W^{-1}$
External walls	0.04	0.13	0.18
Party walls and internal partitions	0.13	0.13	0.18
Roofs:			
— pitched	0.04	0.10	0.16
— flat	0.04	0.10	0.16
Ground floors	0.04	0.17	0.21
Internal floors/ceilings	0.13	0.13	0.18

Table 3.49 Thermal properties of typical wall constructions

Construction	Transmittance U / W·m^{-2}·K^{-1}	Admittance		Decrement factor		Surface factor		Heat capacity, χ / kJ·m^{-2}·K^{-1}
		Y / W·m^{-2}·K^{-1}	ω / h	f	ϕ / h	F	ψ / h	
1 Stone walls								
(*a*) 600 mm stone, 50 mm airspace, 25 mm dense plaster on laths	1.38	3.00	1.4	0.04	17.3	0.65	0.8	41
(*b*) 600 mm stone, 50 mm air-space/timber battens, 25 mm EPS insulation, 12.5 mm plasterboard	0.72	1.13	1.8	0.03	17.0	0.87	0.3	15
2 No-fines concrete walls								
(*a*) 19 mm render, 220 mm no-fines concrete, 50 mm airspace/timber battens, 12.5 mm plasterboard	1.63	2.46	0.8	0.31	7.7	0.69	0.4	39
(*b*) 19 mm render, 220 mm no-fines concrete, 50 mm mineral fibre insulation between battens, 12.5 mm plasterboard	0.67	0.89	2.6	0.24	8.3	0.91	0.3	14
(*c*) 19 mm render, 50 mm mineral wool insulation between battens, 220 mm no-fines concrete, 50 mm airspace/battens, 12.5 mm plaster-board	0.52	2.47	0.8	0.09	9.4	0.69	0.4	34
3 Solid brick walls								
(*a*) 220 mm solid brick, 13 mm dense plaster	2.09	4.49	1.3	0.42	7.4	0.49	1.6	70
(*b*) 220 mm solid brick, 50 mm airspace/battens, 12.5 mm plaster-board	1.41	2.36	1.0	0.33	8.0	0.71	0.4	37
(*c*) 220 mm solid brick, 50 mm mineral wool insulation between battens, 12.5 mm plasterboard	0.63	0.89	2.6	0.25	8.7	0.91	0.3	14
(*d*) 19 mm render, 50 mm EPS insulation, 220 mm solid brick, 13 mm dense plaster	0.54	4.23	1.4	0.12	11.1	0.52	1.5	59
4 Dense concrete walls								
(*a*) 19 mm render, 200 mm dense concrete block, 13 mm dense plaster	3.02	5.19	0.9	0.42	6.5	0.38	1.6	79
(*b*) 19 mm render, 200 mm dense concrete block, 50 mm airspace/battens, 12.5 mm plasterboard	1.78	2.51	0.8	0.34	7.0	0.68	0.4	39
(*c*) 19 mm render, 200 mm dense concrete block, 25 mm polyurethane insulation between battens, 12.5 mm plasterboard	0.90	1.01	2.1	0.26	7.5	0.89	0.3	16
(*d*) 19 mm render, 50 mm mineral wool between battens, 200 mm dense concrete block, 13 mm dense plaster	0.70	5.32	0.8	0.16	8.2	0.35	1.6	74
5 Precast concrete panel walls								
(*a*) 80 mm dense concrete, 25 mm EPS insulation, 100 mm dense concrete, 13 mm dense plaster	1.07	5.36	1.3	0.34	7.7	0.42	2.3	77
(*b*) 80 mm dense concrete, 25 mm EPS insulation, 100 mm dense concrete, 50 mm airspace/battens, 12.5 mm plasterboard	0.85	2.59	1.0	0.21	8.4	0.68	0.5	38

Table continues

Table 3.49 Thermal properties of typical wall constructions — *continued*

Construction	Transmittance U / W·m^{-2}·K^{-1}	Admittance		Decrement factor		Surface factor		Heat capacity, χ / kJ·m^{-2}·K^{-1}
		Y / W·m^{-2}·K^{-1}	ω / h	f	ϕ / h	F	ψ / h	
5 Precast concrete panel walls (continued)								
(*c*) 80 mm dense concrete, 50 mm EPS insulation, 100 mm dense concrete, 12.5 mm plasterboard	0.56	2.61	1.0	0.17	8.8	0.68	0.5	37
(*d*) 19 mm render, 80 mm dense concrete, 50 mm EPS insulation, 100 mm dense concrete, 13 mm dense plaster	0.63	5.42	1.4	0.26	9.0	0.42	2.4	77
6 Brick/brick cavity walls								
(*a*) 105 mm brick, 50 mm airspace, 105 mm brick, 13 mm dense plaster	1.44	4.38	1.5	0.35	8.8	0.53	1.7	67
(*b*) 105 mm brick, 50 mm airspace, 105 mm brick, 13 mm lightweight plaster	1.34	3.55	1.3	0.33	8.9	0.59	1.0	54
(*c*) 105 mm brick, 50 mm UF foam insulation, 105 mm brick, 13 mm dense plaster	0.59	4.57	1.5	0.25	10.1	0.51	1.8	65
(*d*) 105 mm brick, 50 mm blown wool insulation, 105 mm brick, 13 mm dense plaster	0.59	4.51	1.5	0.24	10.1	0.51	1.7	64
(*e*) 105 mm brick, 50 mm cavity, 25 mm EPS insulation, 105 mm brick, 22 mm airspace/battens, 12.5 mm plasterboard	0.67	2.39	1.2	0.17	10.6	0.71	0.5	34
7 Brick/dense concrete block cavity walls								
(*a*) 105 mm brick, 50 mm airspace, 100 mm dense concrete block, 13 mm dense plaster	1.77	5.37	1.2	0.34	8.1	0.40	2.2	80
(*b*) 105 mm brick, 50 mm UF foam insulation, 100 mm dense concrete block, 13 mm dense plaster	0.63	5.57	1.3	0.24	9.3	0.39	2.4	78
(*c*) 105 mm brick, 50 mm blown fibre insulation, 100 mm dense concrete block, 13 mm dense plaster	0.63	5.57	1.3	0.24	9.3	0.39	2.4	78
(*d*) 105 mm brick, 50 mm EPS insulation, 100 mm dense concrete block, 13 mm dense plaster	0.64	5.57	1.3	0.24	9.4	0.39	2.4	78
8 Brick/lightweight aggregate concrete block cavity walls								
(*a*) 105 mm brick, 50 mm airspace, 100 mm lightweight aggregate concrete block, 13 mm dense plaster	1.06	2.72	2.6	0.53	7.4	0.76	1.1	44
(*b*) 105 mm brick, 50 mm UF foam insulation, 100 mm lightweight aggregate concrete block, 13 mm dense plaster	0.52	2.98	2.8	0.42	8.7	0.75	1.3	44
(*c*) 105 mm brick, 50 mm blown fibre insulation, 100 mm lightweight aggregate concrete block, 13 mm dense plaster	0.52	2.97	2.7	0.41	9.2	0.75	1.3	44
(*d*) 105 mm brick, 100 mm blown fibre insulation, 100 mm lightweight aggregate concrete block, 13 mm dense plaster	0.33	3.05	2.8	0.39	9.2	0.75	1.4	44
(*e*) 105 mm brick, 50 mm EPS insulation, 100 mm lightweight aggregate concrete block, 13 mm dense plaster	0.52	2.98	2.8	0.42	8.8	0.75	1.3	44

Table continues

Table 3.49 Thermal properties of typical wall constructions — *continued*

	Construction	Transmittance U / W·m^{-2}·K^{-1}	Admittance		Decrement factor		Surface factor		Heat capacity, χ / kJ·m^{-2}·K^{-1}
			Y / W·m^{-2}·K^{-1}	ω / h	f	ϕ / h	F	ψ / h	
8	**Brick/lightweight aggregate concrete block cavity walls (continued)**								
(*f*)	105 mm brick, 25 mm airspace, 25 mm EPS insulation, 100 mm lightweight aggregate concrete block, 13 mm dense plaster	0.66	2.92	2.8	0.46	8.3	0.76	1.3	44
9	**Brick/autoclaved aerated concrete block cavity walls**								
(*a*)	105 mm brick, 50 mm airspace, 100 mm autoclaved aerated concrete block (density 700 kg·m^{-3}), 13 mm lightweight plaster	1.01	2.29	2.2	0.49	7.8	0.77	0.8	37
(*b*)	105 mm brick, 50 mm airspace, 150 mm autoclaved aerated concrete block (density 500 kg·m^{-3}), 13 mm lightweight plaster	0.79	2.07	2.4	0.42	9.0	0.80	0.8	33
(*c*)	105 mm brick, 25 mm airspace, 25 mm EPS insulation, 150 mm autoclaved aerated concrete block (density 500 kg·m^{-3}), 13 mm lightweight plaster	0.55	2.14	2.4	0.33	10.2	0.79	0.8	32
10	**Timber frame walls**								
(*a*)	105 mm brick, 50 mm airspace, 19 mm plywood sheathing, 95 mm studding, 12.5 mm plasterboard	1.14	1.58	1.7	0.67	4.9	0.82	0.4	26
(*b*)	105 mm brick, 50 mm airspace, 19 mm plywood sheathing, 95 mm studding, 95 mm mineral wool insulation between studs, 12.5 mm plasterboard	0.39	0.75	3.9	0.58	6.0	0.95	0.3	13
(*c*)	105 mm brick, 50 mm airspace, 19 mm plywood sheathing, 140 mm studding, 140 mm mineral wool insulation between studs, 12.5 mm plasterboard	0.29	0.74	4.3	0.57	6.5	0.96	0.3	12
11	**Party walls (internal)**								
(*a*)	12 mm plasterboard 22 mm airspace/battens, 100 mm lightweight aggregate concrete block, 75 mm airspace, 100 mm lightweight aggregate concrete block, 22 mm airspace/battens, 12.5 mm plasterboard	0.56	1.98	2.2	—	—	0.80	0.7	27
(*b*)	13 mm dense plaster, 215 mm brick, 13 mm dense plaster	1.45	4.61	1.5	—	—	0.50	1.8	63
(*c*)	13 mm dense plaster, 215 mm dense concrete block, 13 mm dense plaster	2.33	5.63	1.2	—	—	0.38	2.4	77

Table 3.50 Thermal properties of typical roof constructions

Construction	Transmittance U / W·m^{-2}·K^{-1}	Admittance		Decrement factor		Surface factor		Heat capacity, χ / kJ·m^{-2}·K^{-1}
		Y / W·m^{-2}·K^{-1}	ω / h	f	ϕ / h	F	ψ / h	
1 Flat concrete roofs								
(*a*) Waterproof covering, 75 mm screed, 150 mm cast concrete, 13 mm dense plaster	2.19	5.06	1.0	0.34	7.4	0.41	1.7	76
(*b*) Waterproof roof covering, 35 mm polyurethane insulation, vapour control layer, 75 mm screed, 150 mm cast concrete, 13 mm dense plaster	0.54	5.08	1.0	0.17	9.0	0.40	1.6	71
(*c*) Waterproof roof covering, 100 mm polyurethane insulation, vapour control layer, 75 mm screed, 150 mm cast concrete, 13 mm dense plaster	0.25	5.07	1.0	0.15	9.9	0.40	1.6	70
(*d*) Waterproof roof covering, 200 mm polyurethane insulation, vapour control layer, 75 mm screed, 150 mm cast concrete, 13 mm dense plaster	0.12	5.07	1.0	0.13	12.2	0.40	1.6	70
(*e*) Ballast (chips or paving slab), 50 mm extruded polystyrene insulation, waterproof roof covering, 75 mm screed, 150 mm cast concrete, 13 mm dense plaster	0.59	5.07	1.0	0.16	10.5	0.40	1.6	71
(*f*) Ballast (chips or paving slab), 100 mm extruded polystyrene insulation, waterproof roof covering, 75 mm screed, 150 mm cast concrete, 13 mm dense plaster	0.39	5.07	1.0	0.14	11.0	0.40	1.6	70
2 Flat timber roofs								
(*a*) Waterproof roof covering, 19 mm timber decking, ventilated airspace, vapour control layer, 12.5 mm plasterboard	2.35	2.14	0.7	0.99	0.7	0.73	0.3	10
(*b*) Waterproof roof covering, 19 mm timber decking, ventilated airspace, 50 mm mineral fibre insulation, vapour control layer, 12.5 mm plasterboard	0.64	0.83	2.8	0.99	1.0	0.92	0.3	10
(*c*) Waterproof roof covering, 35 mm polyurethane insulation, vapour control layer, 19 mm timber decking, unventilated airspace, 12.5 mm plasterboard	0.53	1.39	3.4	0.93	1.9	0.90	0.6	19
(*d*) Waterproof roof covering, 100 mm polyurethane insulation, vapour control layer, 19 mm timber decking, unventilated airspace, 12.5 mm plasterboard	0.23	1.52	3.9	0.88	3.0	0.91	0.7	22
(*e*) Waterproof roof covering, 200 mm polyurethane insulation, vapour control layer, 19 mm timber decking, unventilated airspace, 12.5 mm plasterboard	0.13	1.58	4.0	0.75	5.3	0.91	0.7	23

Table continues

Table 3.50 Thermal properties of typical roof constructions — *continued*

Construction	Transmittance U / W·m^{-2}·K^{-1}	Admittance		Decrement factor		Surface factor		Heat capacity, χ / kJ·m^{-2}·K^{-1}
		Y / W·m^{-2}·K^{-1}	ω / h	f	ϕ / h	F	ψ / h	
3 Pitched roofs (insulated at ceiling level)								
(*a*) 12.5 mm plasterboard, no insulation, roof space, tiling	2.30	2.05	0.6	1.00	0.5	0.74	0.2	8
(*b*) 12.5 mm plasterboard, 25 mm mineral wool quilt between ceiling joists, roof space, tiling	1.10	1.11	1.6	1.00	0.4	0.87	0.3	8
(*c*) 12.5 mm plasterboard, 50 mm mineral wool quilt between ceiling joists, roof space, tiling	0.71	0.85	2.6	1.00	0.4	0.92	0.3	8
(*d*) 12.5 mm plasterboard, 100 mm mineral wool quilt between ceiling joists, roof space, tiling	0.42	0.72	3.7	1.00	0.6	0.95	0.3	9
(*e*) 12.5 mm plasterboard, 100 mm mineral wool quilt between ceiling joists, 50 mm mineral wool quilt over joists, roof space, tiling	0.28	0.70	4.5	0.99	1.0	0.97	0.3	10
(*f*) 12.5 mm plasterboard, 100 mm mineral wool quilt between ceiling joists, 100 mm mineral wool quilt over joists, roof space, tiling	0.21	0.68	4.5	0.99	1.0	0.97	0.3	10
(*g*) 12.5 mm plasterboard, 100 mm mineral wool quilt between ceiling joists, 150 mm mineral wool quilt over joists, roof space, tiling	0.17	0.68	4.7	0.98	1.3	0.97	0.3	10
(*h*) 12.5 mm plasterboard, 100 mm mineral wool quilt between ceiling joists, 200 mm mineral wool quilt over joists, roof space, tiling	0.14	0.72	4.8	0.97	1.7	0.98	0.3	10
4 Pitched roofs (insulated at rafter level)								
(a) 12.5 mm plasterboard, 25 mm PU insulation between rafters, ventilated airspace, roofing felt, 25 mm ventilated airspace, clay tiles	0.95	0.98	2.1	1.00	0.3	0.92	0.2	8
(*b*) 12.5 mm plasterboard, 50 mm PU insulation between rafters, ventilated airspace, roofing felt, 25 mm ventilated airspace, clay tiles	0.56	0.76	3.3	1.00	0.5	0.95	0.2	9
(*c*) 12.5 mm plasterboard, 100 mm PU insulation between rafters, ventilated airspace, roofing felt, 25 mm ventilated airspace, clay tiles	0.31	0.70	4.4	0.99	0.7	0.97	0.3	9
(*d*) 12.5 mm plasterboard, 150 mm PU insulation between rafters, ventilated airspace, roofing felt, 25 mm ventilated airspace, clay tiles	0.22	0.71	4.8	0.98	1.2	0.98	0.3	10
(*e*) 12.5 mm plasterboard, 150 mm PU insulation between rafters and 50 mm over rafters, ventilated airspace, roofing felt, 25 mm ventilated airspace, clay tiles	0.15	0.73	4.9	0.96	4.9	0.98	0.3	11
5 Sheet metal construction								
(*a*) 0.4 mm inner sheet, 150 mm Z-spacer with MW insulation, 0.7 mm outer sheet	0.35	0.28	1.2	0.99	0.7	0.97	0.0	2
(*b*) 0.4 mm inner sheet, 85 mm MW insulation, 40 mm MW insulation between rails, 0.7 mm profiles outer sheet	0.35	0.31	0.5	1.00	0.3	0.96	0.0	1

Table 3.51 Thermal properties of typical internal partitions

Construction	Transmittance U / W·m^{-2}·K^{-1}	Admittance		Decrement factor		Surface factor		Heat capacity, χ / kJ·m^{-2}·K^{-1}
		Y / W·m^{-2}·K^{-1}	ω / h	f	ϕ / h	F	ψ / h	
(*a*) 13 mm lightweight plaster 105 mm brick, 13 mm lightweight plaster	1.69	3.76	2.2	0.52	5.4	0.65	1.6	52
(*b*) 13 mm lightweight plaster, 100 mm lightweight concrete block, 13 mm lightweight plaster	1.11	2.27	3.8	0.81	3.6	0.88	1.1	31
(*c*) 12.5 mm plasterboard, timber studding, 12.5 mm plasterboard	1.70	0.61	5.7	1.00	0.5	1.00	0.3	8

Table 3.52 Thermal properties of typical internal floors/ceilings for heat flow from below (values for heat flow from above shown in parentheses)

Construction	Transmittance U / W·m^{-2}·K^{-1}	Admittance		Decrement factor		Surface factor		Heat capacity, χ / kJ·m^{-2}·K^{-1}
		Y / W·m^{-2}·K^{-1}	ω / h	f	ϕ / h	F	ψ / h	
(*a*) 50 mm screed, 150 mm cast concrete, 13 mm dense plaster	2.25	5.44 (4.38)	1.4 (1.6)	0.34	7.0	0.43 (0.53)	2.5 (1.7)	75 (60)
(*b*) 25 mm wood block, 65 mm cast concrete, 50 mm airspace, 12.5 mm plasterboard ceiling	1.61	2.61 (4.12)	2.2 (2.1)	0.50	5.0	0.74 (0.62)	1.0 (1.8)	36 (57)
(*c*) 19 mm timber flooring or chipboard on 100 mm joists, 12.5 mm plasterboard ceiling	1.64	0.86 (1.00)	5.4 (5.4)	0.99	0.9	0.99 (0.99)	0.4 (0.5)	12 (14)

Table 3.53 Thermal properties of typical floors in contact with the ground

Construction	Transmittance U / W·m^{-2}·K^{-1}	Admittance		Decrement factor		Surface factor		Heat capacity, χ / kJ·m^{-2}·K^{-1}
		Y / W·m^{-2}·K^{-1}	ω / h	f	ϕ / h	F	ψ / h	
1 Solid concrete floors								
(*a*) Vinyl floor covering, 75 mm screed, 150 mm cast concrete	*	3.59	1.3	—	—	0.58	1.0	58
(*b*) 10 mm carpet/underlay, 75 mm screed, 150 mm cast concrete	*	2.35	0.9	—	—	0.71	0.4	38
(*c*) Vinyl floor covering, 75 mm screed, 50 mm extruded polystyrene insulation, 150 mm cast concrete	*	3.88	2.5	—	—	0.67	1.8	56
(*d*) Vinyl floor covering, 19 mm timber or chipboard, 50 mm extruded polystyrene insulation, 150 mm cast concrete	*	1.39	3.6	—	—	0.90	0.6	22
2 Suspended timber floors								
(*a*) Vinyl floor covering, 19 mm timber or chipboard on 100 mm joists, ventilated underfloor cavity	*	3.07	0.3	—	—	0.60	0.2	7
(*b*) Vinyl floor covering, 19 mm timber or chipboard on 100 mm joists, 100 mm mineral fibre between joists, ventilated underfloor cavity	*	1.36	4.2	—	—	0.93	0.6	19
(*c*) 10 mm carpet/underlay, 19 mm timber or chipboard on 100 mm joists, ventilated underfloor cavity	*	2.09	0.3	—	—	0.73	0.1	5
(*d*) 10 mm carpet/underlay, 19 mm timber or chipboard on 100 mm joists, 100 mm mineral fibre between joists, ventilated underfloor cavity	*	1.24	3.6	—	—	0.91	0.5	17

* Thermal transmittance of floors depends on upon the size and shape of the floor, see section 3.5.

Table 3.54 Thermal properties of typical floors exposed to outside air below

Construction	Transmittance U / W·m^{-2}·K^{-1}	Admittance		Decrement factor		Surface factor		Heat capacity, χ / kJ·m^{-2}·K^{-1}
		Y / W·m^{-2}·K^{-1}	ω / h	f	ϕ / h	F	ψ / h	
(*a*) Vinyl floor covering, 50 mm screed, 150 mm cast concrete	2.26	3.77	1.1	0.55	5.7	0.55	1.0	59
(*b*) Vinyl floor covering, 50 mm screed, 150 mm cast concrete, 50 mm mineral fibre insulation between battens, 12 mm cementitious building board on underside	0.63	3.91	1.0	0.16	8.2	0.53	1.0	55
(*c*) Vinyl floor covering, 50 mm screed, 150 mm cast concrete, 100 mm mineral fibre insulation between battens, 12 mm cementitious building board on underside	0.38	3.91	1.0	0.15	8.5	0.53	1.0	54
(*d*) 10 mm carpet/underlay, 50 mm screed, 150 mm cast concrete, 100 mm mineral fibre insulation between battens, 12 mm cementitious building board on underside	0.36	2.46	0.7	0.10	8.9	0.69	0.3	34
(*e*) Vinyl floor covering, 19 mm timber or chipboard on 100 mm joists, 12 mm cementitious building board on underside	1.59	1.97	1.1	0.99	1.0	0.76	0.4	14
(*f*) Vinyl floor covering, 19 mm timber or chipboard on 100 mm joists, 100 mm mineral fibre insulation between joists, 12 mm cementitious building board on underside	0.39	1.37	4.2	0.96	1.8	0.93	0.7	19
(*g*) 10 mm carpet/underlay, 19 mm timber or chipboard on 100 mm joists, 100 mm mineral fibre insulation between joists, 12 mm cementitious building board on underside	0.37	1.24	3.6	0.91	2.4	0.91	0.5	18

Table 3.55 Thermal properties of typical floors exposed to internal air below

Construction	Transmittance U / W·m^{-2}·K^{-1}	Admittance		Decrement factor		Surface factor		Heat capacity, χ / kJ·m^{-2}·K^{-1}
		Y / W·m^{-2}·K^{-1}	ω / h	f	ϕ / h	F	ψ / h	
(*a*) Vinyl floor covering, 50 mm screed, 150 mm cast concrete	1.74	5.73	0.8	0.15	10.7	0.32	2.0	79
(*b*) Vinyl floor covering, 50 mm screed, 150 mm cast concrete, 50 mm mineral fibre insulation between battens, 12 mm cementitious building board on underside	0.58	3.98	1.1	0.16	8.7	0.52	1.1	55
(*c*) Vinyl floor covering, 50 mm screed, 150 mm cast concrete, 100 mm mineral fibre insulation between battens, 12 mm cementitious building board on underside	0.36	3.95	1.0	0.15	9.0	0.52	1.0	54
(*d*) 10 mm carpet/underlay, 50 mm screed, 150 mm cast concrete, 100 mm mineral fibre insulation between battens, 12 mm cementitious building board on underside	0.34	2.49	0.7	0.10	9.4	0.68	0.3	34
(*e*) Vinyl floor covering, 19 mm timber or chipboard on 100 mm joists, 12 mm cementitious building board on underside	1.37	1.23	5.2	0.97	1.4	0.98	0.6	17
(*f*) Vinyl floor covering, 19 mm timber or chipboard on 100 mm joists, 100 mm mineral fibre insulation between joists, 12 mm cementitious building board on underside	0.37	1.33	5.1	0.95	2.2	0.97	0.7	18
(*g*) 10 mm carpet/underlay, 19 mm timber or chipboard on 100 mm joists, 100 mm mineral fibre insulation between joists, 12 mm cementitious building board on underside	0.35	1.23	4.6	0.91	32.8	0.95	0.6	17

4 Ventilation and air infiltration

4.1 Introduction

4.1.1 Scope

Information on ventilation and air infiltration is needed to assess:

— the design and adequacy of air change for indoor air quality purposes

— the impact of air change on heating and cooling load

— the design of ventilation methods for passive cooling.

This chapter presents a discussion on ventilation and its roles. It also outlines the basic driving mechanisms and illustrates some design parameters. A simple algorithm for preliminary design is given in Appendix 4.A1.

For details of ventilation systems, see chapter 2 of CIBSE Guide B: *Heating, ventilating, air conditioning and refrigeration*(1). There are also many recent publications associated with ventilation requirements and designs, which are listed in section 4.1.2.

4.1.2 Associated publications and Regulations

Listed below are key publications related to this topic that provide more detailed information on the principles, design and regulations concerning ventilation.

(*a*) CIBSE publications:

— AM10: *Natural ventilation in non-domestic buildings*(2)

— AM13: *Mixed mode ventilation*(3)

— TM21: *Minimising pollution at air intakes*(4)

— TM23: *Testing buildings for air leakage*(5)

— TM30 *Improved life cycle performance of mechanical ventilation systems*(6)

— TM33: *CIBSE standard tests for the assessment of building services design software*(7)

— CIBSE Guide B: *Heating, ventilating, air conditioning and refrigeration*(1), chapter 2: *Ventilation and air conditioning*

— CIBSE Guide J: *Weather, solar and illuminance data*(8)

(*b*) Building Regulations (UK):

For England and Wales, relevant Building Regulations Approved Documents covering requirements for ventilation and airtightness include:

— Part F: *Ventilation*(9)

— Part J: *Combustion appliances and fuel storage systems*(10)

— Part L1: *Conservation of fuel and power in dwellings*(11)

— Part L2: *Conservation of fuel and power in buildings other than dwellings*(12)

For Scotland:

— Scottish Building Standards Agency's *Technical Domestic* and *Non-domestic Handbook*(13)

For Northern Ireland:

— DOE Technical Booklet F: *Conservation of fuel and power*(14)

— DOE Technical Booklet K: *Ventilation*(15)

— DOE Technical Booklet L: *Heat producing appliances*(16)

(*c*) European (CEN) Standards

(*d*) Other publications

— Good Practice Guide 257: *Energy-efficient mechanical ventilation systems*(17)

— *The national air quality strategy for England, Scotland, and Northern Ireland*(18).

4.1.3 Notation

A	Area of opening (m^2)
A_1, A_2	Area of opening 1, 2 etc. (m^2)
A_b	Equivalent area for ventilation by stack effect only (m^2)
A_w	Equivalent area for ventilation by wind only (m^2)
C	Flow coefficient (m^3·s^{-1}·Pa^{-n})
C_d	Discharge coefficient
C_j	Flow coefficient for path j
C_p	Wind pressure coefficient
C_{p1}	Wind pressure coefficient on facade 1 of building
C_{p2}	Wind pressure coefficient on facade 1 of building
$C_r(z)$	Roughness coefficient for terrain at height z
C_t	Topography coefficient
c_p	Specific heat capacity of air (J·kg^{-1}·K^{-1})
F_v	Air infiltration heat loss factor per unit room volume (W·K^{-1}·m^{-3})
g	Acceleration due to gravity (9.81 m·s^{-2})
J_ϕ	Function of angle of window opening ϕ
k_l	Flow coefficient per unit length of crack (L·s^{-1}·m^{-1}·Pa^{-n})
l_c	Total length of crack or opening window joint (m)
N	Air change rate (h^{-1})
n	Flow exponent
n_j	Flow exponent of path j

p_i Inside pressure of building (Pa)
p_{oj} External pressure due to wind and temperature acting on path j (Pa)
p_w Surface pressure due to wind (Pa)
q_v Volumetric flow rate through opening ($m^3 \cdot s^{-1}$)
q_{vb} Volumetric flow rate due to stack effect only ($m^3 \cdot s^{-1}$)
q_{vc} Volumetric flow rate through crack ($L \cdot s^{-1}$)
q_{vm} Volumetric flow rate for mechanical ventilation system ($m^3 \cdot s^{-1}$)
q_{vt} Total volume flow rate ($m^3 \cdot s^{-1}$)
q_{vw} Volume flow rate due to wind only ($m^3 \cdot s^{-1}$)
v Mean wind speed ($m \cdot s^{-1}$)
v_m Mean wind speed measured in open country at datum height of 10 m ($m \cdot s^{-1}$)
v_r Mean wind speed at building roof height ($m \cdot s^{-1}$)
v_z Mean wind speed at height z ($m \cdot s^{-1}$)
z Height above ground level (m)
z_1, z_2 Heights above ground of centres of openings 1 and 2 (m)
α Wind angle (°)
ΔC_p Difference in wind pressure coefficient
Δp Pressure difference (Pa)
Δz Difference between heights z_1 and z_2 (m)
$\Delta\theta$ Difference between mean inside and outside temperatures (K)
ε Ratio of areas of openings 1 and 2
$\bar{\theta}$ Mean of inside and outside temperatures (K)
θ_f Surface temperature of internal surfaces of building fabric (°C)
θ_i Inside temperature (°C)
θ_o Outside temperature (°C)
λ Total number of flow paths
ρ Density of air ($kg \cdot m^{-3}$)
Φ_v Heat transfer by ventilation (W)
ϕ Angle of window opening (°)

Note: in compound units, the abbreviation 'L' has been used to denote 'litre'.

4.2 Role of ventilation

4.2.1 Background

Ventilation is the process by which fresh air is provided to occupants and by which concentrations of potentially harmful pollutants are diluted and removed from a space. Ventilation is also used to passively cool a space and as a mechanism to distribute thermally conditioned air to a space from heating and cooling plant.

From an energy perspective, losses resulting from ventilation and general air exchange can account for more than half of the primary energy used in a building. These losses comprise space heating and refrigerative cooling losses as well as the electrical load associated with driving mechanical services.

4.2.2 Minimum ventilation rates for air quality

See chapter 1, section 1.7 and chapter 8, section 8.4.

Ventilation is critical for minimising the concentration of harmful pollutants. It is for this reason that higher ventilation rates are usually associated with improved health.

The amount of ventilation required for air quality depends on:

— occupant density
— occupant activities
— pollutant emissions within a space.

European Standard BS EN 13779: *Ventilation for buildings. Performance requirements for ventilation and air-conditioning systems*[19] provides basic definitions of air quality standards in occupied spaces and relates these to fresh air ventilation rates required for each occupant (in terms of $L \cdot s^{-1}$ per person). These are summarised in Table 4.1.

It is important to note that these definitions relate to comfort air quality and need not necessarily reflect the purity of air with respect to health related contaminants. It is assumed that the space is relatively free from sources of pollution and that the ventilation air is itself pure.

From April 2006, Building Regulations Part F[9] will require a minimum ventilation rate of 10 $L \cdot s^{-1}$ per person for most non-domestic applications. This fits between classes IDA2 and IDA3 in Table 4.1.

For guidance on ventilation techniques, see CIBSE Guide B[1].

Table 4.1 Ventilation and indoor air quality classification (BS EN 13779)[19]

Classification	Indoor air quality standard	Ventilation range / ($L \cdot s^{-1}$/person)	Default value / ($L \cdot s^{-1}$/person)
IDA1	High	> 15	20
IDA2	Medium	10–15	12.5
IDA3	Moderate	6–10	8
IDA4	Low	< 6	5

4.2.3 Ventilation rate and metabolic carbon dioxide

Because carbon dioxide is emitted as part of the metabolic process, the resultant increase in CO_2 concentration above the ambient outdoor value can be used as an estimate of the adequacy of ventilation. In this respect, CO_2 is considered as a tracer gas and, even at concentrations associated with unacceptably low ventilation, is rarely regarded as harmful. The rate of emission of CO_2 is dependent on the degrees of physical activity. Guidelines related to CO_2 concentrations almost always refer to sedentary environments and do not apply to areas of substantial physical activity (e.g. manual labour and sports activities). Also, it takes a finite period for CO_2 to reach a steady state level thus monitoring a space shortly after occupancy commences may give an erroneously low result. Subject to the above caveats the increase in CO_2 concentration above the ambient outdoor concentration that reflect the air quality classifications of Table 4.1 are summarised in Table 4.2.

Table 4.2 Approximate maximum sedentary CO_2 concentrations associated with CEN indoor air quality standards (BS EN 13779)[19]

Classification	Rise in indoor CO_2 concentration / ppm	Default value / ppm	Range in outdoor concentration / ppm	Total indoor value* / ppm
IDA1	< 400	350	350–400	700–750
IDA2	400–600	500	350–400	850–900
IDA3	600–1000	800	350–400	1150–1200
IDA4	> 1000	1200	350–400	1550–1600

* i.e. concentration rise plus outdoor value

4.2.4 Outdoor air quality

Ventilation supply air must be clean. Although filtration for particles and gaseous pollutants is possible it is expensive and is not practicable on a population-wide basis. Within the UK, outdoor air quality is addressed by *The National Air Quality Strategy for England, Scotland and Northern Ireland*[18]. Areas of poor air quality are designated ‘air quality management areas’ in which special consideration is needed when planning developments. Information on outdoor air quality for specific locations is available from the environmental officer of the relevant local authority.

Pollution generated from very localised sources such as from adjacent exhaust stacks, polluting processes and local traffic must be dealt with by source control and by the careful placement of air intakes. Intake location is considered in CIBSE TM21[4].

4.2.5 Ventilation for cooling

In buildings without mechanical cooling, high rates of air change are used to assist in providing thermal comfort in summer. More details are given in section 4.6.3.6. There is a limit to the effectiveness of natural cooling and it is important that all sources of internal heat gain as well as solar gain are minimised. Very often, peak temperatures in standard buildings will exceed the outdoor peak temperature by three or more degrees. At best, peak indoor temperature can be maintained at a degree or so below the peak outdoor temperature but this is only possible by means of night cooling combined with thermal mass. Nevertheless, good passive design can minimise the periods during which mechanical cooling may be needed and can also substantially reduce the size of cooling plant, perhaps restricting its need to only essential locations.

Note that the heat island effect can reduce the opportunities for night cooling in urban areas, see chapter 2, section 2.10.2.

4.2.6 Ventilation to transport thermally conditioned air

The ventilation system in large commercial buildings is widely used to transport heated and cooled air. Since the volume of air required to fulfil this task is frequently several times larger than the fresh air requirement, a significant part of the ventilation air is recirculated. Where recirculation is undesirable, thermal and latent heat recovery systems are used to recapture losses and used to pre-condition the supply air stream.

4.3 Ventilation techniques

Ventilation may be provided by natural, mechanical or mixed mode methods. The fundamentals of these techniques are summarised below. More detail is presented in chapter 2 of CIBSE Guide B[1].

4.3.1 Natural ventilation

Natural ventilation is driven by the climatic forces of wind (wind effect) and temperature (stack effect). For this reason, natural ventilation is highly variable since, at any instant, both the pattern of air flow and the rate of ventilation will depend on the prevailing weather conditions. If an annual temperature span of between –10 °C to +30 °C is considered, combined with wind speeds of 0–10 $m \cdot s^{-1}$ and 8 cardinal wind directions, a system could be subjected to over 3000 different sets of driving conditions during the course of a year. Despite the possibility that this range may be reduced by prevailing wind, shelter and micro-climate as well as by the ventilation design, allowance must be made for the fact that many conditions will occur. Therefore, a natural ventilation design must be flexible so that it can respond to the variability in climate. Ideally, for detailed design, hourly temperatures, windspeeds and wind directions for key design years should be verified. Some basic calculation guidelines are given in section 4.7.4. A full review of natural ventilation configurations and examples is given in CIBSE AM10[2].

4.3.2 Mechanical ventilation

Mechanical ventilation is applied by means of driving fans and a network of ducts. In large office buildings, mechanically supplied air is usually filtered and thermally conditioned by heating and cooling. Key variations are:

(*a*) *Supply-only ventilation*: supply-only systems incorporate mechanical supply fans to provide fresh air combined with a network of passive outlets (vents) for the exhaust of air. Because the air is mechanically supplied (or ‘pushed’ into the space), the building will be at a positive pressure with respect to the ambient outdoor atmospheric pressure. The central location of supply means that the supply air can be pre-cleaned by filtration and that the risk of air entering the space from fabric leakage cracks is reduced.

(*b*) *Extract-only ventilation*: extract systems incorporate mechanical fans to extract or suck air out of a space. Make-up fresh air is provided by a network of passive supply openings. Because air is being sucked out of the space, the building is a negative pressure relative to the ambient outdoor atmospheric pressure. A central extract zone enables any pollutants generated in the space to be filtered out before discharge to the atmosphere. In addition moisture can be extracted at source rather than entering the building fabric where it may condense. This has advantages in wet or polluting spaces where the pollutants can be captured before emission into the atmosphere.

(*c*) *Balanced mechanical ventilation*: this relies on separate fan systems to provide for both the supply of fresh air and the extract of exhaust air. They offer the combined advantages of the separate systems (i.e. filtration of supply air, cleaning of extract air combined with opportunity for recovery of thermal losses from the exhaust air for supply air pre-conditioning). However, they operate at twice the mechanical cost. Also, balanced systems have an approximately neutral impact on the pressure distribution in the building. This means that infiltration can add to the total air changes whereas mechanical supply systems pressurise the building, with respect to atmospheric pressure, and extract-only systems de-pressurise the building.

4.3.3 Mixed mode ventilation

Mixed mode ventilation utilises a combination of both natural and mechanical ventilation technology. By good design this approach can result in a considerable reduction of ventilation energy load when compared with wholly mechanical systems. The basic modes of operation, defined in CIBSE AM 13[3], are:

- *Supplementary*: mechanical ventilation is applied when natural driving forces are inadequate to meet ventilation need. Usually a low energy fan is incorporated in the natural ventilation network to provide this supplementary need.
- *Complementary*: natural and mechanical ventilation work together to meet the ventilation needs of a building.
- *Alternate*: both natural and mechanical ventilation systems are separately incorporated into the building. The actual system used at any time is dependent on current climate conditions and ventilation need.

Mixed mode ventilation can be used in conjunction with passive cooling and local refrigerative cooling, as necessary to meet transient cooling need.

4.4 Ventilation estimation techniques

It is necessary to calculate ventilation and infiltration rates for the following purposes:

- to ensure that ventilation is adequate to meet occupancy or other ventilation requirements
- to determine the thermal losses or gains arising from air change in order to perform heating and cooling load calculations
- to assess the potential for using ventilation to assist in the passive cooling of a building.

Estimation of natural ventilation performance involves calculation procedures for the sizing of openings to ensure that ventilation requirements are met throughout the full range of weather conditions. Mechanical ventilation calculations, especially for systems based on supply or extract only approaches, should involve similar calculations to check the impact of climate induced pressures on flows through passive inlets and outlets. Mechanical balanced systems may require the addition of an air infiltration calculation because air infiltration will add directly to the air change through the system.

Section 4.5 provides general guidance on the theory of airflow through openings and the determination of the driving forces that produce such flow in buildings. Sections 4.6 and 4.7 cover alternative methods of predicting air infiltration and natural ventilation rates. These range from the use of empirical data to complex numerical calculation techniques. Section 4.8 considers airtightness testing.

For detailed information on designing buildings for natural ventilation reference should be made to CIBSE AM10: *Natural ventilation in non-domestic buildings*[2].

4.5 Outline of ventilation and air infiltration theory

4.5.1 General

The rate of airflow through a building depends upon the areas and resistances of the various apertures (both intentionally provided and fortuitous) and the pressure difference between one end of the flow path and the other. This pressure difference may be due to:

- wind (wind-driven natural ventilation)
- differences in density of the air due to the indoor–outdoor temperature differences (commonly referred to as 'stack effect' or stack-driven ventilation)
- pressure differences created by mechanical ventilation fans
- a combination of the three above mechanisms.

4.5.1.1 Wind-driven ventilation

The effect of wind is to drive air into the building through gaps and openings on the windward side of the building, where the surface pressure is high. The air then passes from one side of the building to the other and exits through apertures on the leeward side, where the pressure is low, see Figure 4.1. The higher the wind speed, the higher will be the air infiltration and natural ventilation rates.

Two distinct mechanisms of wind driven cross flow ventilation are now recognised, as illustrated in the Figure 4.2[20].

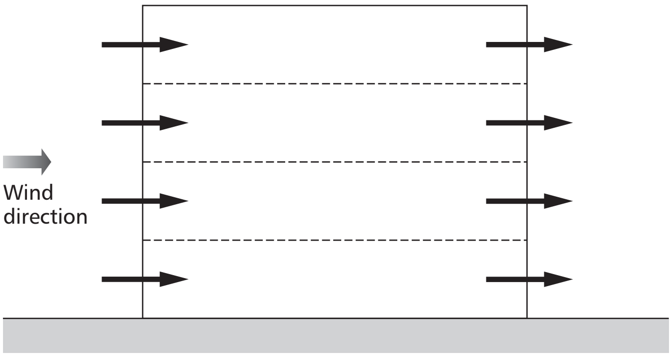


Figure 4.1 Wind driven airflow through a building

The first considers small openings such as infiltration gaps and flow through trickle ventilators. In this instance, the kinetic energy of the wind dissipates by generating a positive pressure on the external surface of the opening. The resultant pressure imbalance between the outside and inside of the opening causes air to flow through the opening. At the leeward side opening, the wind generates a suction pressure that draws air out of the building. This is the classical approach that forms the basis of many infiltration and natural ventilation flow models. Sandberg[21] suggests that this is not valid beyond a total opening area of more than 30% of a building face.

The second considers large openings such as open windows. In this instance kinetic energy dissipation is much reduced and, instead, the wind essentially passes through the building. For a given wind speed the flow rate through a unit area of opening will be higher via the second of these two mechanisms. Currently, much research is taking place on the second mechanism but there are currently no simple calculation guidelines available. This dilemma has recently been studied in considerable detail by Etheridge[22], who concludes that since the primary aim of natural ventilation design is to ensure that a sufficient number of openings are provided for occupant and/or automatic control, current solution techniques, based on classical flow theory, will provide an adequate level of accuracy.

Cool air is denser than warm air. Therefore, in a heated building in winter, gravity (i.e. 'stack effect') causes cold air from outside to enter through low level gaps and openings to displace the warmer internal air, which escapes through gaps and openings at high level, see Figure 4.3. This direction of movement would be reversed in summer if the temperature of the air inside were lower than that outside.

When both wind and stack effects apply, the magnitude and directions of air infiltration will vary depending on the relative strengths of the two forces and whether they are acting locally in the same or opposite directions, see Figure 4.4.

4.5.1.2 Combined wind and stack ventilation

The routes by which air moves within a building will depend on the presence of such features as corridors, stairwells, lift shafts, flues and mechanical ventilation systems.

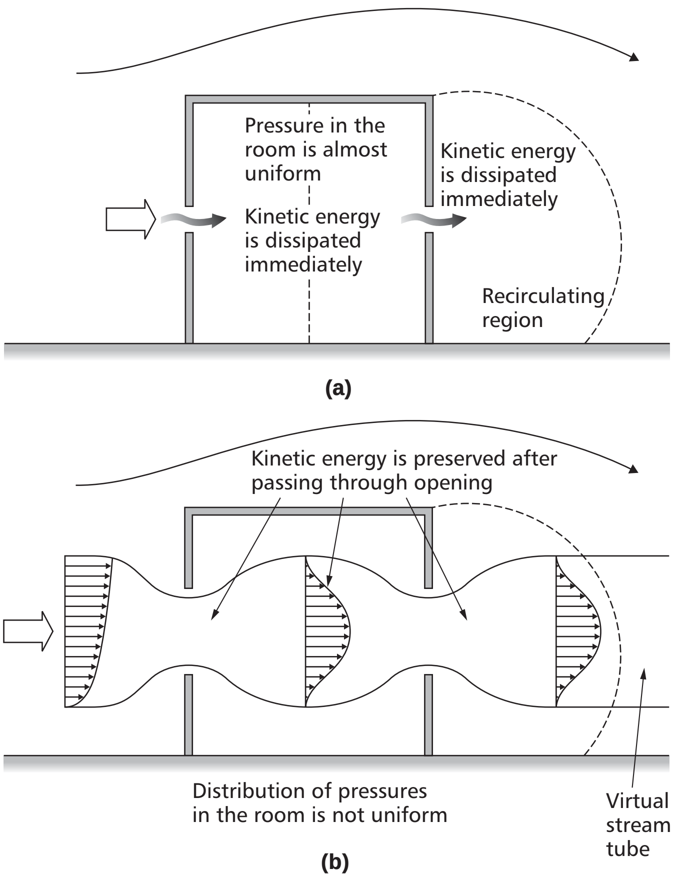


Figure 4.2 Flow configurations[20]; (a) through small openings, (b) through large openings

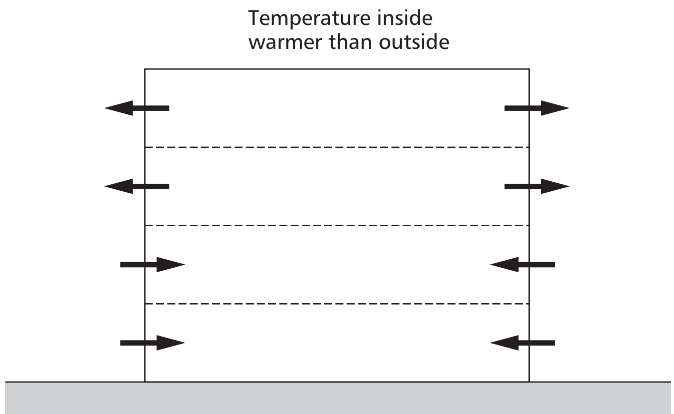


Figure 4.3 Temperature driven airflow (stack effect) through a building

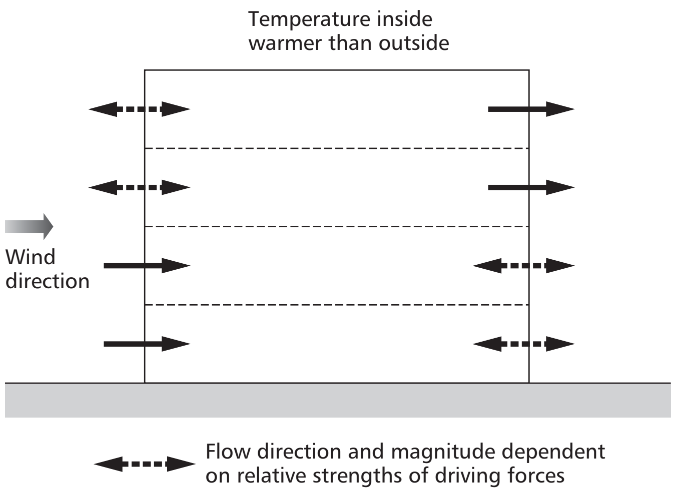


Figure 4.4 Combined wind- and temperature-driven airflow through a building

4.6 Assessing natural ventilation and air infiltration rates

The calculation of air flow is a complex topic and readers are referred to CIBSE AM10[2] for detailed guidance. However, this section outlines some basic flow principles which can give guidance on estimating natural ventilation flow rates.

4.6.1 Flow through openings

The magnitude of the airflow through small openings such as infiltration gaps and trickle ventilators is a function of the applied pressure difference across the opening and its length, cross-sectional area and internal geometry. This relationship is often described by the empirical power law:

$$q_v = C\,(\Delta p)^n \qquad (4.1)$$

where q_v is the volumetric flow rate through the opening ($m^3 \cdot s^{-1}$), C is the flow coefficient ($m^3 \cdot s^{-1} \cdot Pa^{-n}$), Δp is the pressure difference across the opening (Pa) and n is the flow exponent.

The flow coefficient (C) is related to the size of opening. The flow exponent (n) characterises the flow regime, its value varying between 0.5 for fully turbulent flow to 1.0 for laminar flow. Typical turbulent flow paths include orifice-type openings such as purpose provided air vents and open windows. Airflow through small, adventitious cracks and gaps tends to be of a more laminar nature, with typical flow exponent values between 0.6 and 0.7.

When considering infiltration through cracks, it is convenient to express the flow coefficient in terms of metre length of crack or opening. Equation 4.1 then becomes:

$$q_{vc} = l_c\,k_1\,(\Delta p)^n \qquad (4.2)$$

where q_{vc} is the volumetric flow rate through the crack ($L \cdot s^{-1}$), l_c is the total length of crack or opening (m) and k_1 is the flow coefficient per unit length of opening ($L \cdot s^{-1} \cdot m^{-1} \cdot Pa^{-n}$). Note that, since k_1 is expressed in terms of $L \cdot s^{-1}$, the units for flow rate through the opening are $L \cdot s^{-1}$.

Note: air flow paths through mechanical systems may be treated in the same way as air flow path for natural ventilation and air infiltration openings, see section 4.6.2.

Tables 4.3 and 4.4 give typical values of flow coefficient (k_1) and exponent (n) for a range of door and window types when fully closed[23]. However, in modern buildings doors and windows may not be the predominant leakage routes so other types of joint or perforation in the building envelope may need to be taken into account. Further data are given in AIVC Technical Note 44[23].

For orifice-type openings such as open windows, equation 4.1 takes the form of the common orifice flow equation:

$$q_v = C_d\,A\,(2\,\Delta p / \rho)^{0.5} \qquad (4.3)$$

Table 4.3 Flow characteristics for doors (per unit length of joint)[23]

Item	Flow coefficient per metre length of joint, k_1 / $L \cdot s^{-1} \cdot m^{-1} \cdot Pa^{-n}$, where $n = 0.6$			Size of sample
	Lower quartile	Median	Upper quartile	
External doors (weather stripped):				
— hinged	0.082	0.27	0.84	15
— sliding	—	—	—	0
— revolving (laboratory test)	1.0	1.5	2.0	4
External doors (non-weather stripped):				
— hinged	1.1	1.2	1.4	17
— sliding	—	0.20	—	1
Roller door per m^2 of surface (laboratory test)	3.3*	5.7*	10*	2*
Internal doors (non-weather stripped)	1.1	1.3	2.0	84
Loft hatches (non-weather stripped)	0.64	0.68	0.75	4

* Flow coefficient expressed in $L \cdot s^{-1} \cdot m^{-2} \cdot Pa^{-n}$

Table 4.4 Flow characteristics for windows (per unit length of joint)[23]

Item	Flow coefficient per metre length of joint, k_1 / $L \cdot s^{-1} \cdot m^{-1} \cdot Pa^{-n}$, where $n = 0.6$			Size of sample
	Lower quartile	Median quartile	Upper	
Windows (weather stripped):				
— hinged	0.086	0.13	0.41	29
— sliding	0.079	0.15	0.21	19
Windows (non-weather stripped):				
— hinged	0.39	0.74	1.1	42
— sliding	0.18	0.23	0.37	36

where q_v is the volumetric flow rate through the opening ($m^3 \cdot s^{-1}$), C_d is the discharge coefficient, A is the area of the opening (m^2), Δp is the pressure difference across the opening (Pa) and ρ is the density of air ($kg \cdot m^{-3}$).

Thus, the flow characteristics of such openings can be expressed directly in terms of their open area. This greatly simplifies the calculation procedure.

The theoretical value of C_d for a sharp-edged opening is 0.61 and it is common practice to refer other openings, for which the airflow is governed by the square root of the pressure difference, to this value. Measurements of flow rate and applied pressure difference are used to calculate an equivalent area assuming a discharge coefficient of 0.61. This approach is particularly useful where the nature of the opening makes it difficult to directly measure its geometrical area, such as in the case of trickle ventilators. Recent research[(24)], however, questions the validity of a fixed pressure coefficient value for wind directions that are at an oblique angle to an opening. At present no guidance is available but the reader should be aware that this simplification is questionable for non-normal flow directions. Ideally, manufacturers of natural ventilation intakes should provide flow data for their products for a range of wind angles.

These relationships indicate that with knowledge of the opening characteristics and the pressure drop, the airflow rate can be calculated. Alternatively, calculations can be made of the opening characteristics that are needed to produce the required airflow rate at a predetermined pressure drop.

4.6.2 Representing flow through mechanical systems

Mechanical systems create a driving force by means of a fan. The flow rate through the fan is a function of fan size, duct and component pressure losses (e.g. through filters, sound attenuators etc), diffuser and outlet characteristics and under or over pressure developed in the space itself. Fan manufacturers supply the relevant airflow versus pressure drop data.

Provided that the correct fan is selected for the intended operating pressure range then, to a first approximation, small variations about the selected pressure drop can be represented by a constant flow rate. to give a flow rate equation:

$$q_{vm} = \text{constant} \qquad (4.4)$$

where q_{vm} is the volumetric flow rate for the mechanical ventilation system ($m^3 \cdot s^{-1}$).

This is a useful approximation because it enables flow through a mechanical system to be treated as an air flow path, as for natural ventilation and air infiltration openings. This means, for example, that mechanical ventilation can be readily incorporated into simple design tools. The user must verify, however, that the given flow rate will be provided at the ultimately derived pressure. In other words a fan must be selected that will meet the pressure drop through the fan and ductwork combined with that determined by the air flow calculation.

4.6.3 Driving forces

The forces driving natural ventilation and air infiltration are maintained by the actions of wind and temperature. Unfortunately, the pattern of pressure distribution arising from these parameters is extremely complex and considerable simplification is necessary in any mathematical representation. A brief outline on the calculation of pressure distribution is presented in this section.

4.6.3.1 Wind speed

Within the lower regions of the earth's atmosphere, wind is characterised by random fluctuations in velocity which, when averaged over a fixed period of time, yields mean values of speed and direction.

Wind data are generally obtained from a meteorological station located away from an urban environment. In general this wind speed must be corrected for terrain conditions and for the height of the building relative to the height of wind measurement (usually 10 m). An approximate correction is proposed in BS 5925 and is given as:

$$v_z = v_m \, k \, z^a \qquad (4.5)$$

where v_z is the wind speed at the building height ($m \cdot s^{-1}$), v_m is the wind speed measures in open country at a height of 10 m ($m \cdot s^{-1}$), z is the building height (m), k and a are constants dependent on the terrain (see Table 4.5).

For basic design purposes wind data are provided in chapter 2, section 2.8, and in CIBSE Guide J[(8)]. These consist of summary wind speed and direction data for 8 UK sites and detailed data, consisting of average percentage frequency of wind speed, direction and temperature for three UK sites: London (Heathrow), Manchester (Ringway) and Edinburgh (Turnhouse). Comprehensive data on wind speed and direction in the UK are also published by the Met. Office.

Table 4.5 Terrain coefficients for wind speed

Terrain	k	a
Open, flat country	0.68	0.17
Country with scattered wind breaks	0.52	0.20
Urban	0.35	0.25
City	0.21	0.33

4.6.3.2 Wind pressure calculation

On impinging the surface of a rectangular building, wind deflection induces a positive pressure on the upwind face. The flow then separates at the corners, resulting in negative pressure regions being developed along the sides of the building, see Figure 4.5(a). A negative pressure distribution is also developed along the rear facing or leeward facade.

The pressure distribution on the roof varies according to pitch, see Figure 4.5(b), the pressure on the upwind face being negative for roof pitches of less than about 30°. For pitches greater than 30° the pressure on the upwind face is positive.

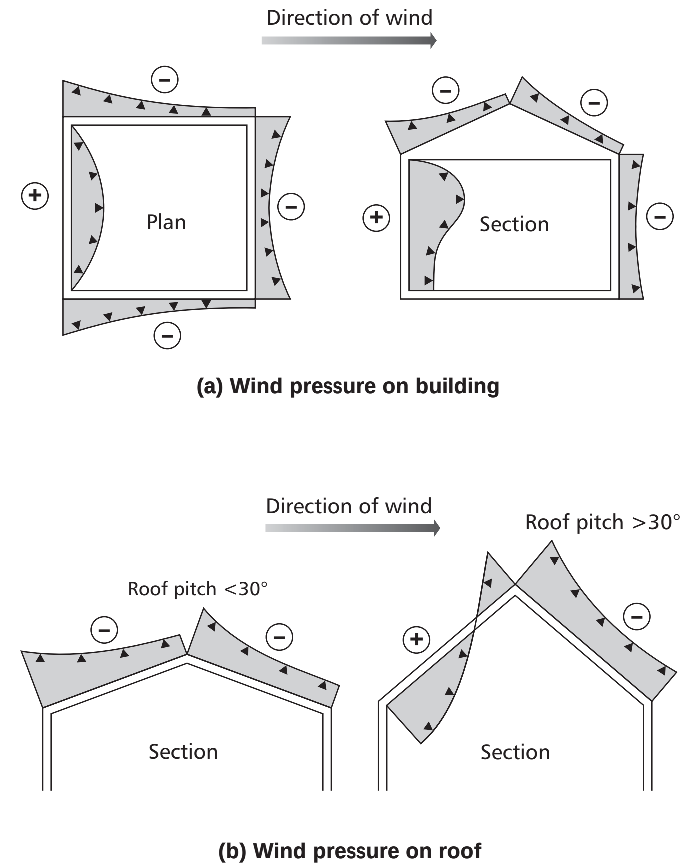


Figure 4.5 Wind pressure; (*a*) on building, (*b*) on roof

The magnitude of the wind pressure coefficient at any point for any given wind direction can generally be regarded as independent of the wind speed. Relative to the static pressure of the free wind, the time-averaged pressure acting on any point on the surface of a building may be represented by the following equation:

$$p_w = 0.5\,\rho\, C_p\, v_z^{\,2} \tag{4.6}$$

where p_w is the surface pressure due to wind (Pa), ρ is the density of air, C_p is the wind pressure coefficient at a given position on the building surface and v_z is the mean wind velocity at height z (m).

The wind pressure coefficient, C_p, is a function of wind direction and of spatial position on the building surface. However, accurate evaluation of this parameter is extremely difficult and normally involves wind tunnel tests using a scale model of the building and its surroundings.

In approximate form, the wind pressure coefficient is expressed as a single average value for each face of the building. BS 5925[25] gives average values for buildings of simple shape in exposed locations. However, these data ignore the very significant influence of surrounding obstructions in shielding the building from wind. A synthesis of façade-averaged values for buildings subjected to varying degrees of shelter and wind directions is given in AIVC Applications Guide: *A guide to energy efficient ventilation*[26]. A summary of these data for a simple, square-plan building, see Figure 4.6, which may be used for basic design calculations, is given in Table 4.6. The values in the table apply to square plan buildings of heights up to three stories.

Although averaged data provide a useful approximation, their reliability diminishes for high-rise buildings (i.e. greater than four storeys) since the pressure distribution can be significantly dependent on the height of the building. Some test data for high-rise buildings have been published by Bowen[27]. Spatial distribution is also important for single sided ventilation analysis because the wind pressure variation across a face can be considerable and this information is lost when using face averaged values. For detailed analysis, a wind tunnel study of the building and its surroundings may be necessary.

The wind speed (v_z) is always expressed as a value for a given building height (z). Since this term is raised to the power two in the pressure calculation, it is imperative that remote weather station data are corrected for terrain and building height as described by equation 4.5. The use of unadjusted weather data may result in considerable over-estimation of air infiltration or natural ventilation rates.

The procedure for calculating wind pressure at any location on the building surface may be accomplished as follows:

— determine building height

— determine nature of surrounding terrain

— correct meteorological wind speed data according to building height and terrain classification (equation 4.5)

— determine the approximate wind pressure coefficient for each face or for each opening in the building envelope (Table 4.6) or undertake wind tunnel studies if more accurate values of wind pressure coefficient are required.

Example 4.1

Tables 4.7 and 4.8 show the determination of the roof height wind speed and average wind pressure for the front and rear aspects of an 8 m high, building located in (*a*) a 'green field' site in open country and (*b*) an urban area. The roof pitch is 0°. The mean wind speed measured at a nearby meteorological station is 4 $m \cdot s^{-1}$ (measured in open country at a height of 10 m). The surfaces are defined in Figure 4.7

The meteorological wind speed is modified for building height and location by applying equation 4.5 using the coefficients given in Table 4.5.

Values of the average wind pressure coefficient are obtained from Tables 4.6(*a*) and 4.6(*c*) for a wind angle of 0°, i.e. perpendicular to wall 1.

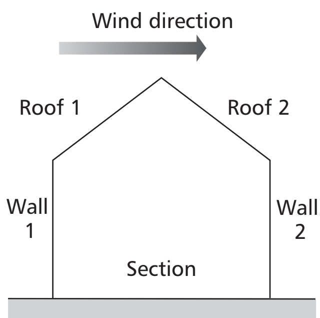


Figure 4.7 Example 4.1: definition of surfaces

Table 4.6 Approximate wind pressure coefficients for square-plan building averaged over stated surfaces[35]

(*a*) Building in open flat country

Surface	Average wind pressure coefficient, C_p, for stated wind angle α							
	0	45	90	135	180	225	270	315
Walls:								
— facade 1	0.7	0.35	–0.5	–0.4	–0.2	–0.4	–0.5	–0.35
— facade 2	–0.2	–0.4	–0.5	0.35	0.7	0.35	–0.5	–0.4
— facade 3	–0.5	0.35	0.7	0.35	–0.5	–0.4	–0.2	–0.4
— facade 4	–0.5	–0.4	–0.2	–0.4	–0.5	0.35	0.7	0.35
Roof (front):								
— pitch < 10°	–0.8	–0.7	–0.6	–0.5	–0.4	–0.5	–0.6	–0.7
— pitch 11° to 30°	–0.4	–0.5	–0.6	–0.5	–0.4	–0.5	–0.6	–0.5
— pitch > 30°	0.3	–0.4	–0.6	–0.4	–0.5	–0.4	–0.6	–0.4
Roof (rear):								
— pitch < 10°	–0.4	–0.5	–0.6	–0.7	–0.8	–0.7	–0.6	–0.5
— pitch 11° to 30°	–0.4	–0.5	–0.6	–0.5	–0.4	–0.5	–0.6	–0.5
— pitch > 30°	–0.5	–0.4	–0.6	–0.4	–0.3	–0.4	–0.6	–0.4

(*b*) Building in open country with scattered windbreaks

Surface	Average wind pressure coefficient, C_p, for stated wind angle α							
	0	45	90	135	180	225	270	315
Walls:								
— facade 1	0.4	0.1	–0.3	–0.35	–0.2	–0.35	–0.3	–0.1
— facade 2	–0.2	–0.35	–0.3	0.1	0.4	0.1	–0.3	–0.35
— facade 3	–0.3	0.1	0.4	0.1	–0.3	–0.35	–0.2	–0.35
— facade 4	–0.3	–0.35	–0.2	–0.35	–0.3	0.1	0.4	0.1
Roof (front):								
— pitch < 10°	–0.6	–0.5	–0.4	–0.5	–0.6	–0.5	–0.4	–0.5
— pitch 11° to 30°	–0.35	–0.45	–0.55	–0.45	–0.35	–0.45	–0.55	–0.45
— pitch > 30°	0.3	–0.5	–0.6	–0.5	–0.5	–0.5	–0.6	–0.5
Roof (rear):								
— pitch < 10°	–0.6	–0.5	–0.4	–0.5	–0.6	–0.5	–0.4	–0.5
— pitch 11° to 30°	–0.35	–0.45	–0.55	–0.45	–0.35	–0.45	–0.55	–0.45
— pitch > 30°	–0.5	–0.5	–0.6	–0.5	–0.3	–0.5	–0.6	–0.5

(*c*) Building in urban location

Surface	Average wind pressure coefficient, C_p, for stated wind angle α							
	0	45	90	135	180	225	270	315
Walls:								
— facade 1	0.2	0.05	–0.25	–0.3	–0.25	–0.3	–0.25	0.05
— facade 2	–0.25	–0.3	–0.25	0.05	0.2	0.05	–0.25	–0.3
— facade 3	–0.25	0.05	0.2	0.05	–0.25	–0.3	–0.25	–0.3
— facade 4	–0.25	–0.3	–0.25	–0.3	–0.25	0.05	0.2	0.05
Roof (front):								
— pitch < 10°	–0.5	–0.5	–0.4	–0.4	–0.5	–0.5	–0.4	–0.5
— pitch 11° to 30°	–0.3	–0.4	–0.5	–0.4	–0.3	–0.4	–0.5	–0.4
— pitch > 30°	0.25	–0.3	–0.5	–0.3	–0.4	–0.3	–0.5	–0.3
Roof (rear):								
— pitch < 10°	–0.5	–0.5	–0.4	–0.5	–0.5	–0.5	–0.4	–0.5
— pitch 11° to 30°	–0.3	–0.4	–0.5	–0.4	–0.3	–0.4	–0.5	–0.4
— pitch > 30°	–0.4	–0.3	–0.5	–0.3	–0.25	–0.3	–0.5	–0.3

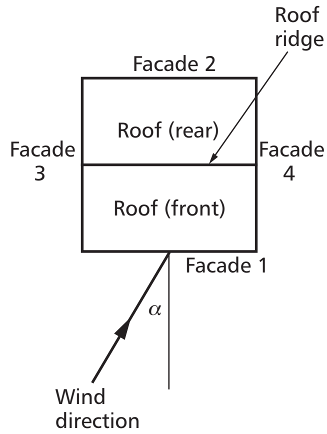


Figure 4.6 Square-plan building used in determination of wind pressure coefficients given in Table 4.6

Table 4.7 Example 4.1: determination of wind pressure coefficients

Surface	Wind pressure coefficient, C_p	
	Open country (with windbreaks)	Industrial (urban)
Facade 1	0.4	0.2
Facade 2	–0.2	–0.25
Roof 1	–0.6	–0.5
Roof 2	–0.6	–0.5

Table 4.8 Example 4.1: calculation of wind pressure

Surface	Wind pressure / Pa	
	Open country (with windbreaks)	Industrial (urban)
Facade 1	3.7	1.0
Facade 2	–1.8	–1.3
Roof 1	–5.5	–2.6
Roof 2	–5.5	–2.6

4.6.3.3 Effect of wind turbulence

Turbulent or random fluctuations of the wind can itself influence the rate of infiltration or ventilation particularly when the mean wind speed is low or when the temperature (i.e. stack) effect is minimal.

For example, in the situation shown in Figure 4.8(*a*), the average wind pressure coefficients at both openings would be equal so that no airflow would be expected across the building. In reality under turbulent conditions, an exchange of air will occur as inflow and outflow takes place intermittently. However, the turbulent effect need not be considered when estimating infiltration ventilation rates for design wind or temperature conditions, since this will be outweighed by the other driving mechanisms.

Similarly, internal airflow will be generated by turbulence when there is an opening on one side only see Figure 4.8(*b*). A formula for the quantitative assessment of the ventilation rate under such circumstances is presented in Table 4.23(*a*). Improved results are possible by considering the spatial distribution of wind pressure as discussed in 4.6.3.2 (e.g. wind tunnel analysis).

4.6.3.4 Stack effect

Stack effect arises as a result of differences in temperature, and hence air density, between the inside and outside of a building. This produces an imbalance in the pressures of the internal and external air masses, thus creating a vertical pressure gradient.

When the internal temperature is greater than the outside temperature, air enters through openings in the lower part of the building and leaves through openings at a higher level. The flow direction is reversed when the internal air temperature is lower than that of the outside air.

Assuming two vertically displaced openings (see Figure 4.9), one at level z_1 and the other at level z_2, linking air at temperatures θ_o (outside) and θ_i (inside), the pressure difference induced by the stack effect is given by:

$$\Delta p = -\rho_o \, g \, 273 \, (z_2 - z_1) \left(\frac{1}{\theta_o + 273} - \frac{1}{\theta_i + 273} \right) \quad (4.8)$$

where Δp is the pressure difference (Pa), ρ_o is the density of air at 0 °C (kg·m^{-3}), g is the acceleration due to gravity (9.81 m·s^{-2}), z_1 and z_2 are the heights above ground of openings 1 and 2 (m), θ_o is the outside temperature (°C) and θ_i is the inside temperature (°C).

For the range of temperatures found in practice, equation 4.8 approximates to:

$$\Delta p = -3455 \, (z_2 - z_1) \left(\frac{1}{\theta_o + 273} - \frac{1}{\theta_i + 273} \right) \quad (4.9)$$

Table 4.10 gives pressure differences, calculated using this expression, for a range of temperature and height differences.

Example 4.2

For the building in example 4.1, assuming openings at heights 1 m and 7 m above ground, the pressure difference created by stack effect for an internal temperature of 20 °C and with the outside at 0 °C, is:

$$\Delta p = -3455 \times 6 \, [(1 / 273) - (1 / 293)] = -5.2 \text{ Pa}$$

Comparing this value with Table 4.8 shows that in this case the pressure difference due to a wind speed of 4 m·s^{-1} is higher than that due to stack effect for an open country site, but lower for an urban site.

Table 4.10 Pressure differences due to stack effect

Temperature difference, $(\theta_i - \theta_o)$ / K	Pressure difference (/ Pa) for stated vertical height difference (/ m)				
	5	10	20	50	100
–10	2.2	4.3	8.6	22	43
0	0	0	0	0	0
10	–2.2	–4.3	–8.6	–22	–43
20	–4.3	–8.6	–17	–43	–86

Note: minus sign indicates reduction in pressure with height, i.e. flow upwards within the building

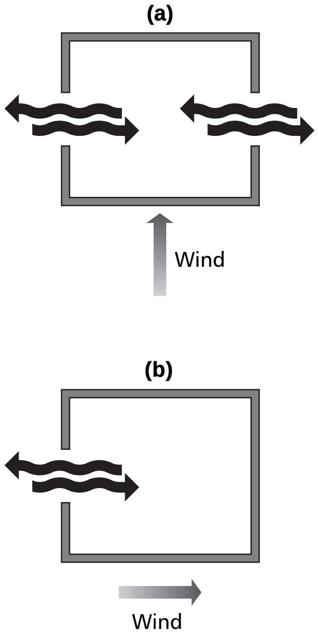


Figure 4.8 Effect of turbulent fluctuations of wind; (*a*) openings on opposite sides of enclosure, (*b*) openings on one side only

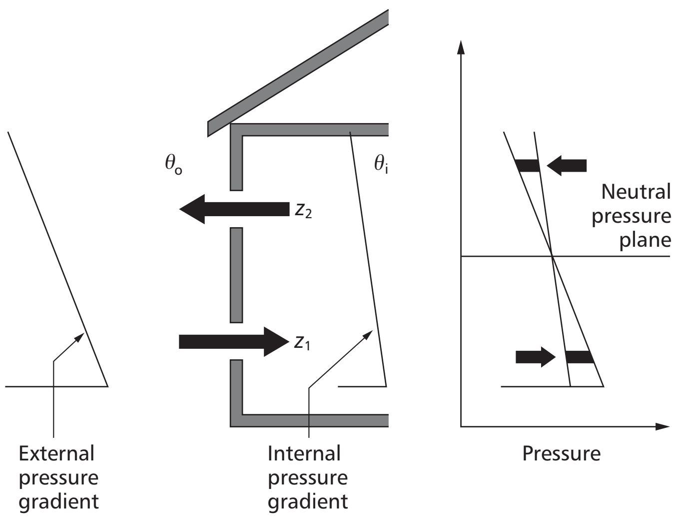


Figure 4.9 Stack effect

4.6.3.5 Combined wind and stack effects

As illustrated in Figure 4.4, in some parts of a building wind and stack effects may act in the same direction whereas in other parts they may act in opposite directions. The effects are additive in one case, and counteractive in the other. The resulting pressures may be calculated accordingly. This is illustrated in example 4.3.

Example 4.3

For the building considered in example 4.1, the only additional information required to calculate the local pressures is knowledge of the stack effect neutral plane. The position of the neutral plane is affected by the relative resistances to flow of the openings in the facade. Assuming that similar openings are located in opposite facades at heights of 1 m and 7 m, then the neutral plane will be at mid-height. Therefore, the stack pressure difference calculated above will be divided equally between the upper and lower openings.

The combined pressure distributions at the openings for open country and industrial locations are shown in Table 4.11.

Note that in the industrial location, the impact of the stack effect on the upper windward and lower leeward openings is to reverse the direction of flow compared to that due to wind alone. This is because the pressure changes from positive to negative and vice versa. More complex worked examples are given in CIBSE Applications Manual AM10[2].

4.6.3.6 Cooling by ventilation

Colder outside air may be used to cool an indoor space. This is achieved by flushing out warm air that develops in a space as a consequence of heat transfer from surfaces and other sources. There is an upper limit to cooling potential which is reached when the rate of heat transfer from sources and surfaces is at a maximum. This is covered in detail by Levermore[28], who gives the heat transfer equation as:

$$\Phi_v = {}^1/_3\, N\, V\, (\theta_f - \theta_o)\, (1 - e^{-x}) \qquad (4.10)$$

where Φ_v is the heat transfer by ventilation (W), N is the air change rate (h^{-1}), V is the room volume (m^3), θ_f is the surface temperature of the internal surfaces of the building fabric (°C), θ_o is the outdoor air temperature (°C) and exponent x is given by:

$$x = \frac{4.8\, A}{{}^1/_3\, N\, V} \qquad (4.11)$$

4.7 Estimation methods

4.7.1 General

There are a number of methods available for estimating air infiltration and natural ventilation rates and for determining the area of openings required to provide a specific ventilation rate. The choice ranges from the use of empirical data or standard equations to complex models requiring the use of computer software.

The most appropriate method for a particular application depends on the intended use of the derived data, the complexity of the building and the availability of data relevant to the chosen method. Table 4.12 provides guidance on the appropriateness of each of the methods.

4.7.2 Empirical data for air infiltration

Empirical data may be of use to provide basic guidance on possible infiltration rates that may be expected in buildings of typical construction in normal use in winter and under average annual conditions (perhaps to estimate allowances for cooling load). The data presented in this section are summarised from sample calculations using the calculation principles outlined in the previous sections.

4.7.2.1 Method 1: Empirical data based on air change rate at 50 Pa pressure test results

An estimate of the infiltration rate averaged over a year of weather data is sometimes inferred from the airtightness value given in air changes per hour at a reference pressure of 50 Pa. This approach was first suggested by Kronvall[29] who analysed data, principally of dwellings, to derive a ‘divide by 20’ rule. In other words, the estimated annual average ventilation correlates with the pressure test value at 50 Pa divided by 20. This rule was extended by Dubrul[30] to consider a wider range of building sizes, climate and exposure. This showed that for high rise buildings and buildings exposed to high winds, the ACH_{50} value needed to be divided by a value as low as 10, whereas for sheltered and low rise buildings, the divisor could approach a value of 30.

Table 4.11 Example 4.3: calculation of combined wind and stack pressures

Opening	Open country location			Industrial (urban) location		
	Wind pressure	Stack pressure	Combined pressure	Wind pressure	Stack pressure	Combined pressure
Facade 1:						
— upper opening	+3.8	–2.6	+1.2	+1.0	–2.6	–1.6
— lower opening	+3.8	+2.6	+5.0	+1.0	+2.6	+3.6
Facade 2:						
— upper opening	–1.9	–2.6	–0.7	–1.3	–2.6	–3.9
— lower opening	–1.9	+2.6	+4.5	–1.3	+2.6	+1.3

Table 4.12 Summary of estimation methods

Method	Application	Data requirements	Advantages	Disadvantages
Empirical data	Air infiltration rate assessment when little is known about the airflow characteristics of the building	Building type, height and exposure	Easy to use	Does not provide detailed predictions
Standard formulae	Estimation of natural ventilation rates for simple buildings with openings on opposite sides or on one side only	Design wind speed(s) and temperatures, wind pressure coefficients, required ventilation rates or opening areas	Easy to use	Limited to narrow range of building configurations
Theoretical calculation:				
— single zone model	Estimation of internal airflows for buildings with simple internal layouts	Design wind speed(s) and temperatures,wind pressure coefficients, location, size and flow characteristics of each opening to outside, airflow paths, building volume	Relatively easy to use, predicts magnitude and direction of airflow, calculates internal pressure, changes may be easily accommodated	Requires detailed knowledge of the building
— multi-zone model	Estimation of internal airflows for complex buildings with known characteristics	As above plus internal configuration and its airflow characteristics	As above plus internal pressure distribution	Requires extensive input data and considerable computational effort

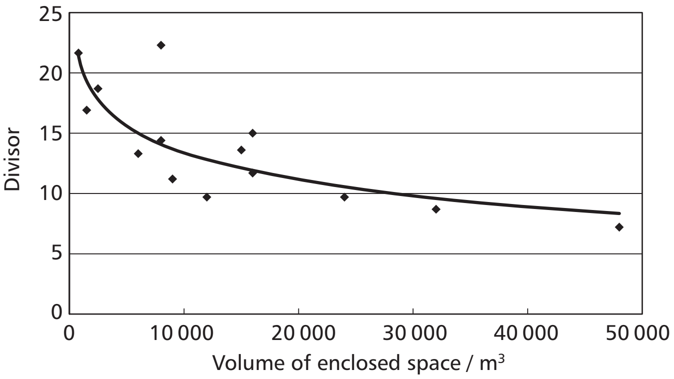


Figure 4.10 Approximate ACH_{50} divisor to obtain average annual infiltration rate

Figure 4.10 gives estimates of the divisor based on calculations using a full year of CIBSE hourly average weather data for London (Heathrow) for a selection of building sizes and exposures. This must, however, be regarded as very approximate although, nevertheless, it is consistent with the previously cited observations.

4.7.2.2 Method 2: Tabular values

The data in Tables 4.13 to 4.21 provide a very approximate estimate of the contribution that air infiltration rate makes to the overall air change rate of a building. They are based on applying the calculation technique described in section 4.7.4.1. Air permeability values as defined by Part L of the Building Regulations[11,12] are used. The definition of air permeability is given in section 4.8.1. The values are:

— 20 $m^3/(m^2{\cdot}h)$ at 50 Pa (this represents an existing 'leaky' building that does not comply with current regulations)

— 10 $m^3/(m^2{\cdot}h)$ at 50 Pa (this represents a building that complies with 2002 regulations)

— 7 $m^3/(m^2{\cdot}h)$ at 50 Pa (a moderately tight building that complies with 2005 regulations)

— 5 $m^3/(m^2{\cdot}h)$ at 50 Pa (a tight building)

— 3 $m^3/(m^2{\cdot}h)$ at 50 Pa (a very tight building).

For non-residential buildings, air leakage values (and hence calculations) are based on the single pressure testing of the entire closed space as described in TM23[5] (i.e. all internal doors are open). *Air leakage values for residential buildings (and hence calculations) are based on individual testing of each dwelling.*

These data should be used for guidance only.

The data presented in Tables 4.13 to 4.21 have the flowing format:

(*a*) 'Peak': infiltration rate for a winter design condition of approximately:

— outdoor temperature = –5 °C

— indoor temperature = 21 °C

— windspeed = 4 $m{\cdot}s^{-1}$

Applications:

— estimation of peak infiltration heat loss for sizing applications.

(*b*) 'Average': annual average infiltration rate based on:

— CIBSE hourly 20-year average wind and temperature data.

Applications:

— estimation of annual average infiltration loss for heat loss calculations

— estimation of contribution of infiltration air to fulfiling ventilation needs

— estimation of infiltration contribution to summer cooling potential or infiltration heat load.

(*c*) 'Air change at 50 Pa': the equivalent air change (for the particular building dimensions used) given the 2002 Part L[11,12] leakage value of 10 $m^3/(m^2 \cdot h)$ at 50 Pa.

(*d*) 'ACH_{50} divisor' (see also Figure 4.10): the value obtained when dividing the 50 Pa air change rate (determined from the 2002 Part L[11,12] value) by the calculated average annual infiltration rate.

Notes:

— On severely exposed sites, a 50% increase above the tabulated values should be allowed. On sheltered sites, the infiltration rate may be reduced by 33%.

— The air change rate in rooms in tall buildings may be significantly higher than the values given in Tables 4.13 to 4.21. The design of tall buildings should include barriers against vertical air movement through stairwells and shafts to minimise the stack effect. If this is not done, the balance of internal temperatures can be seriously disturbed.

— Allowance must be made in the sizing of heating or cooling plant to meet the needs of both the ventilation air and air infiltration. Where ventilation is achieved by mechanical supply only (or extract only) systems, infiltration (or ex-filtration) is partially inhibited and converted to ventilation airflow. In this case, the plant sizing may be based on the ventilation rate plus an air infiltration rate of 50% or less of the tabulated value, depending on the relative magnitude of the internal pressure generated by the mechanical system and the natural external pressures. Where ventilation is by a balanced mechanical ventilation system, infiltration is not inhibited and the values given in the table should be applied without modification.

— Office buildings are based on the definitions given in Energy Consumption Guide ECG019[31]. These are:

— Type 1: naturally ventilated, 100–3000 m^2

— Type 2: naturally ventilated, 500–4000 m^2

— Type 3: air conditioned, 2000–8000m^2

— Type 4: air conditioned HQ-type building, 4000–20000 m^2

Table 4.13 Empirical values for air infiltration rate due to air infiltration for rooms in buildings on normally-exposed sites in winter — office type 1: naturally ventilated up to 6 storeys (100–3000 m^2); partial exposure

Air permeability / ($m^3/m^2 \cdot h$ at 50 Pa)	Infiltration rate (ACH) for given building size / h^{-1}							
	1 storey; 250 m^2 (20 m × 12.5 m × 3 m)*		2 storeys; 500 m^2 (20 m × 12.5 m × 3 m)*		4 storeys; 2000 m^2 (25 m × 20 m × 3 m)*		6 storeys; 3000 m^2 (25 m × 20 m × 3 m)*	
	Peak	Ave	Peak	Ave	Peak	Ave	Peak	Ave
20.0 (leaky)	1.20	0.90	0.95	0.70	0.75	0.55	0.75	0.55
10.0 (Part L (2002))	0.60	0.45	0.50	0.35	0.40	0.30	0.40	0.30
7.0 (Part L (2005))	0.45	0.30	0.35	0.25	0.25	0.20	0.30	0.20
5.0	0.30	0.25	0.25	0.20	0.20	0.15	0.20	0.15
3.0	0.20	0.15	0.15	0.10	0.15	0.10	0.15	0.10
Air change rate at 50 Pa (/ h^{-1})		9.30		5.95		3.50		2.95
ACR_{50} divisor		22.0		16.9		13.5		11.5

* (Length × width × height for each storey)

Note: tabulated values should be adjusted for local conditions of exposure

Table 4.14 Empirical values for air infiltration rate due to air infiltration for rooms in buildings on normally-exposed sites in winter — office type 2: naturally ventilated up to 10 storeys (500–4000 m^2); partial exposure

Air permeability / ($m^3/m^2 \cdot h$ at 50 Pa)	Infiltration rate (ACH) for given building size / h^{-1}							
	2 storeys; 500 m^2 (20 m × 12.5 m × 3 m)*		2 storeys; 1000 m^2 (25 m × 20 m × 3 m)*		4 storeys; 2000 m^2 (25 m × 20 m × 3 m)*		8 storeys; 4000 m^2 (25 m × 20 m × 3 m)*	
	Peak	Ave	Peak	Ave	Peak	Ave	Peak	Ave
20.0 (leaky)	0.95	0.70	0.80	0.60	0.75	0.55	0.80	0.55
10.0 (Part L (2002))	0.50	0.35	0.40	0.30	0.40	0.30	0.40	0.30
7.0 (Part L (2005))	0.35	0.25	0.30	0.25	0.25	0.20	0.30	0.20
5.0	0.25	0.20	0.20	0.15	0.20	0.15	0.20	0.15
3.0	0.15	0.10	0.15	0.10	0.15	0.10	0.15	0.10
Air change rate at 50 Pa (/ h^{-1})		5.95		5.15		3.50		2.65
ACR_{50} divisor		16.9		17.1		13.3		9.7

* (Length × width × height) for each storey

Note: tabulated values should be adjusted for local conditions of exposure

Table 4.15 Empirical values for air infiltration rate due to air infiltration for rooms in buildings on normally-exposed sites in winter — office type 3: air conditioned up to 8 storeys (2000–8000 m^2); partial exposure

Air permeability / ($m^3/m^2 \cdot h$ at 50 Pa)	Infiltration rate (ACH) for given building size / h^{-1}							
	2 storeys; 2000 m^2 (20 m × 25 m × 4 m)*		4 storeys; 4000 m^2 (40 m × 25 m × 4 m)*		6 storeys; 6000 m^2 (40 m × 25 m × 4 m)*		8 storeys; 8000 m^2 (40 m × 25 m × 4 m)*	
	Peak	Ave	Peak	Ave	Peak	Ave	Peak	Ave
20.0 (leaky)	0.75	0.60	0.75	0.52	0.80	0.55	0.85	0.60
10.0 (Part L (2002))	0.40	0.30	0.40	0.26	0.40	0.30	0.45	0.30
7.0 (Part L (2005))	0.25	0.20	0.30	0.18	0.30	0.20	0.30	0.20
5.0	0.20	0.15	0.20	0.13	0.20	0.15	0.25	0.15
3.0	0.15	0.10	0.15	0.08	0.15	0.10	0.15	0.10
Air change rate at 50 Pa (/ h^{-1})		4.30		3.05		2.65		2.45
ACR_{50} divisor	15.4		11.7		9.7			8.7

* (Length × width × height) for each storey

Note: tabulated values should be adjusted for local conditions of exposure

Table 4.16 Empirical values for air infiltration rate due to air infiltration for rooms in buildings on normally-exposed sites in winter — office type 4: air conditioned HQ-type building up to 20 storeys; sheltered (up to 4 storeys), partial exposure (up to 12 storeys), exposed (above 12 storeys)

Air permeability / ($m^3/m^2 \cdot h$ at 50 Pa)	Infiltration rate (ACH) for given floor range / h^{-1}											
	< 2 storeys*		< 4 storeys*		< 8 storeys*		< 12 storeys*		< 16 storeys*		< 20 storeys*	
	Peak	Ave	Peak	Ave	Peak	Ave	Peak	Ave	Peak	Ave	Peak	Ave
20.0 (leaky)	0.60	0.34	0.60	0.35	0.65	0.45	0.80	0.50	0.90	0.65	0.95	0.65
10.0 (Part L (2002))	0.30	0.17	0.30	0.20	0.35	0.25	0.40	0.25	0.45	0.35	0.50	0.35
7.0 (Part L (2005))	0.20	0.12	0.25	0.15	0.25	0.15	0.30	0.20	0.35	0.25	0.35	0.25
5.0	0.15	0.08	0.15	0.10	0.20	0.15	0.20	0.15	0.25	0.20	0.25	0.20
3.0	0.10	0.05	0.10	0.05	0.10	0.10	0.15	0.10	0.15	0.10	0.15	0.10
Air change rate at 50 Pa (/ h^{-1})		3.80		2.55		1.95		1.75		1.65		1.55
ACR_{50} divisor		22.3		15.0		8.8		7.2		5.2		4.7

* (Length × width × height) = 40 m × 25 m × 4 m for each storey (all cases)

Note: tabulated values should be adjusted for local conditions of exposure

Table 4.17 Empirical values for air infiltration rate due to air infiltration for rooms in buildings on normally-exposed sites in winter — factories, warehouses, halls; partial exposure

Air permeability / ($m^3/m^2 \cdot h$ at 50 Pa)	Infiltration rate (ACH) for given building size / h^{-1}							
	500 m^2 (25 m × 20 m × 5 m)		1500 m^2 (50 m × 30 m × 10 m)		5000 m^2 (100 m × 50 m × 20 m)		10 000 m^2 (100 m × 100 m × 25 m)	
	Peak	Ave	Peak	Ave	Peak	Ave	Peak	Ave
20.0 (leaky)	1.00	0.65	0.75	0.45	0.55	0.35	0.45	0.3
10.0 (Part L (2002))	0.50	0.35	0.40	0.25	0.30	0.20	0.25	0.15
7.0 (Part L (2005))	0.30	0.25	0.25	0.15	0.20	0.15	0.15	0.10
5.0	0.20	0.20	0.20	0.15	0.15	0.10	0.15	0.10
3.0	0.15	0.10	0.15	0.10	0.10	0.05	0.10	0.05
Air change rate at 50 Pa (/ h^{-1})		5.80		3.05		1.60		1.20
ACR_{50} divisor	18.7		13.6		10			9.2

Note: tabulated values should be adjusted for local conditions of exposure

Table 4.18 Empirical values for air infiltration rate due to air infiltration for rooms in buildings on normally-exposed sites in winter — schools; partial exposure

Air permeability / ($m^3/m^2{\cdot}h$ at 50 Pa)	Infiltration rate (ACH) for given building size / h^{-1}							
	1 storey; 500 m^2 (25 m × 20 m × 4 m)*		1 storey; 1000 m^2 (40 m × 25 m × 4 m)*		2 storeys; 1000 m^2/floor (40 m × 25 m × 4 m)*		3 storeys; 1000 m^2/floor (40 m × 25 m × 4 m)*	
	Peak	Ave	Peak	Ave	Peak	Ave	Peak	Ave
20.0 (leaky)	1.05	0.70	0.90	0.65	0.65	0.5	0.65	0.45
10.0 (Part L (2002))	0.55	0.35	0.45	0.35	0.35	0.25	0.35	0.25
7.0 (Part L (2005))	0.40	0.25	0.35	0.25	0.25	0.20	0.25	0.20
5.0	0.30	0.20	0.25	0.20	0.20	0.15	0.20	0.15
3.0	0.15	0.10	0.15	0.10	0.10	0.10	0.10	0.10
Air change rate at 50 Pa (/ h^{-1})		6.80		6.30		3.80		3.00
ACR_{50} divisor	20.0		19.7		15.2		13.5	

* (Length × width × height) for each storey; each storey is nominally isolated by structural design and fire doors etc.

Note: tabulated values should be adjusted for local conditions of exposure

Table 4.19 Empirical values for air infiltration rate due to air infiltration for rooms in buildings on normally-exposed sites in winter — hospitals and health care buildings; partial exposure

Air permeability / ($m^3/m^2{\cdot}h$ at 50 Pa)	Infiltration rate (ACH) for given building size / h^{-1}							
	< 2 storeys; 500 m^2/fl. (25 m × 20 m × 4 m)*		< 4 storeys; 1000 m^2/fl. (40 m × 25 m × 4 m)*		< 8 storeys; 1000 m^2/fl. (40 m × 25 m × 4 m)*		< 12 storeys; 1000 m^2/fl. (40 m × 25 m × 4 m)*	
	Peak	Ave	Peak	Ave	Peak	Ave	Peak	Ave
20.0 (leaky)	0.75	0.60	0.65	0.45	0.65	0.45	0.85	0.60
10.0 (Part L (2002))	0.40	0.30	0.35	0.25	0.35	0.25	0.45	0.30
7.0 (Part L (2005))	0.25	0.20	0.25	0.15	0.25	0.15	0.30	0.25
5.0	0.20	0.15	0.15	0.15	0.20	0.15	0.20	0.15
3.0	0.15	0.10	0.10	0.10	0.10	0.10	0.15	0.10
Air change rate at 50 Pa (/ h^{-1})		4.3		2.55		1.95		1.75
ACR_{50} divisor	15.3		11.6		8.8		7.7	

* (Length × width × height) for each storey

Note: tabulated values should be adjusted for local conditions of exposure

Table 4.20 Empirical values for air infiltration rate due to air infiltration for rooms in buildings on normally-exposed sites in winter — hotels; partial exposure

Air permeability / ($m^3/m^2{\cdot}h$ at 50 Pa)	Infiltration rate (ACH) for given building size / h^{-1}							
	1 storey (50 m × 12 m × 3 m)*		2 storeys (50 m × 12 m × 3 m)*		5 storeys (50 m × 12 m × 3 m)*		10 storeys (50 m × 12 m × 3 m)*	
	Peak	Ave	Peak	Ave	Peak	Ave	Peak	Ave
20.0 (leaky)	1.15	0.85	0.85	0.65	0.80	0.60	0.90	0.65
10.0 (Part L (2002))	0.60	0.45	0.45	0.35	0.40	0.30	0.45	0.30
7.0 (Part L (2005))	0.40	0.30	0.30	0.25	0.30	0.20	0.35	0.25
5.0	0.30	0.20	0.25	0.20	0.20	0.15	0.25	0.15
3.0	0.20	0.15	0.15	0.10	0.15	0.10	0.15	0.10
Air change rate at 50 Pa (/ h^{-1})		8.75		5.40		3.40		2.73
ACR_{50} divisor	21.3		16.9		12.1		9.1	

* (Length × width × height) for each storey

Note: tabulated values should be adjusted for local conditions of exposure

Table 4.21 Empirical values for air infiltration rate due to air infiltration for rooms in buildings on normally-exposed sites in winter — dwellings; partial exposure

Air permeability / ($m^3/m^2{\cdot}h$ at 50 Pa)	Infiltration rate (ACH) for given building size / h^{-1}							
	1 storey (10 m × 8 m × 2.75 m)* (Height to roof: 5.5 m)		2 storeys (10 m × 8 m × 2.75 m)* (Height to roof: 8.0 m)		Apartmts (storeys 1–5) (10 m × 8 m × 2.75 m)* (Floor spacing: 3.0 m)		Apartmts (storeys 6–10) (10 m × 8 m × 2.75 m)* (Floor spacing: 3.0 m)	
	Peak	Ave	Peak	Ave	Peak	Ave	Peak	Ave
20.0 (leaky)	1.60	1.15	1.50	1.00	1.95	1.40	2.25	1.60
10.0 (Part L (2002))	0.80	0.60	0.75	0.50	1.00	0.70	1.15	0.80
7.0 (Part L (2005))	0.55	0.40	0.55	0.35	0.70	0.50	0.80	0.55
5.0	0.40	0.30	0.40	0.25	0.50	0.35	0.70	0.40
3.0	0.25	0.20	0.25	0.15	0.30	0.25	0.35	0.25
Air change rate at 50 Pa (/ h^{-1})		11.80		8.15		11.80		11.80
ACR$_{50}$ divisor		20.6		17.0		17.3		15.1

* (Length × width × height) for each storey; for apartments, air leakage is based on each apartment being pressure tested separately

Note: tabulated values should be adjusted for local conditions of exposure

4.7.3 Method 3: Natural ventilation in simple building layouts

The assumption that ventilation openings can be represented by orifice flow equations (e.g. equation 4.4) enables estimates of ventilation rates using standard formulae for simple building layouts. These layouts and associated formulae are shown in Table 4.22 for a simple building with airflow through opposite sides and in Table 4.23 for a situation with openings in one wall only. Both wind-induced and temperature-induced ventilation are given.

The values of area (A) used in the formulae should be taken as the minimum cross-sectional area perpendicular to the direction of the airflow passing through the opening. Typical C_p values are given in Table 4.3.

The formulae given in Table 4.22 illustrate a number of general characteristics of natural ventilation, as follows:

Table 4.22 Standard formulae for estimating airflow rates for simple building layouts (openings on opposite sides)

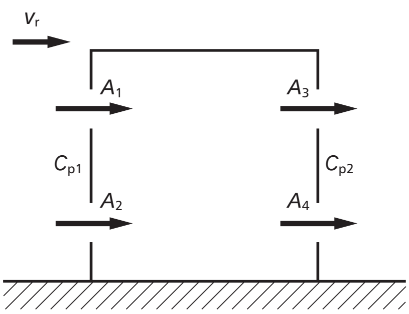
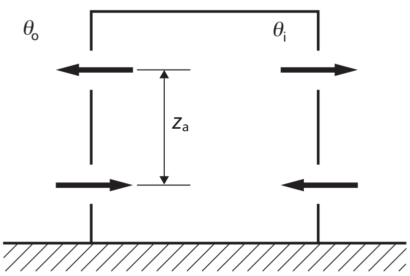
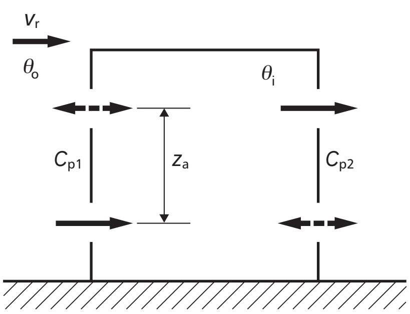
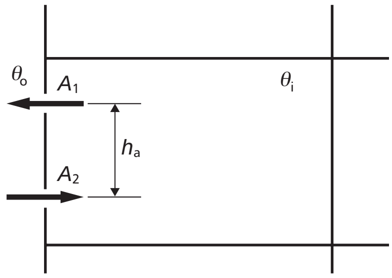
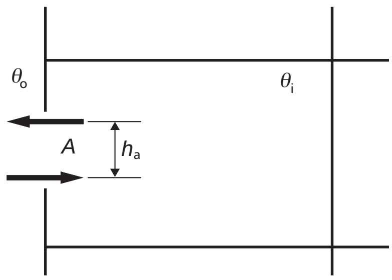
Conditions	Schematic	Equations
(a) Wind only		$Q_w = C_d\, A_w\, v_r\, (\Delta C_p)^{0.5}$ $\dfrac{1}{A_w^{\,2}} = \dfrac{1}{(A_1 + A_2)^2} + \dfrac{1}{(A_3 + A_4)^2}$
(b) Temperature difference only		$Q_b = C_d\, A_b \left(\dfrac{2\,\Delta\theta\; h_a\, g}{\bar{\theta} + 273}\right)^{0.5}$ $\dfrac{1}{A_b^{\,2}} = \dfrac{1}{(A_1 + A_3)^2} + \dfrac{1}{(A_2 + A_4)^2}$
(c) Wind and temperature difference together		$Q_t = Q_b$ for $(v_r / \sqrt{\Delta t}) < 0.26\,(A_b/A_w)\,(h_a / \Delta C_p)^{0.5}$ $Q_t = Q_w$ for $(v_r / \sqrt{\Delta t}) > 0.26\,(A_b/A_w)\,(h_a / \Delta C_p)^{0.5}$

Table 4.23 Standard formulae for estimating airflow rates for simple building layouts (openings on one side only)

Conditions	Schematic	Equations
(a) Wind only		$Q = 0.025\,A\,V_r$
(b) Temperature difference only: two openings		$Q = C_d\,(A_1 + A_2)\left(\frac{\varepsilon\sqrt{2}}{(1+\varepsilon)\,(1+\varepsilon^2)^{0.5}}\right)\left(\frac{\Delta\theta\,h_a\,g}{\bar{\theta} + 273}\right)^{0.5}$ where $\varepsilon = (A_1 / A_2)$
(c) Temperature difference only: one openings		$Q = C_d\,(A/3)\left(\frac{\Delta\theta\,h_a\,g}{\bar{\theta} + 273}\right)^{0.5}$ If opening light is present: $Q = C_d\,(A\,J_\phi\,/3)\left(\frac{\Delta\theta\,h_a\,g}{\bar{\theta} + 273}\right)^{0.5}$ where J_ϕ given by Figure 4.11

— The effective area of a number of openings combined in parallel, across which the same pressure difference is applied, can be obtained by simple addition.

— The effective area of a number of openings combined in series (across which the same pressure difference is applied) can be obtained by adding the inverse squares of the individual areas and taking the inverse of the square root of the total (see Table 4.22(*b*)).

— When wind is the dominating mechanism the ventilation rate is proportional to wind speed and to the square root of the difference in pressure coefficient. Thus, although ΔC_p may range between 0.1 and 1.0, this will result in a ratio of only about 1 to 3 in the predicted ventilation rates for the same wind speed

— When stack effect is the dominating mechanism the ventilation rate is proportional to the square root of both temperature difference and height between upper and lower openings. When wind and stack effect are of the same order of magnitude their interaction is complicated. However, for the simple case illustrated, the actual rate, to a first approximation, may be taken as equal to the larger of the rates for the two alternative approaches, considered separately. This is shown in Table 4.22(*c*).

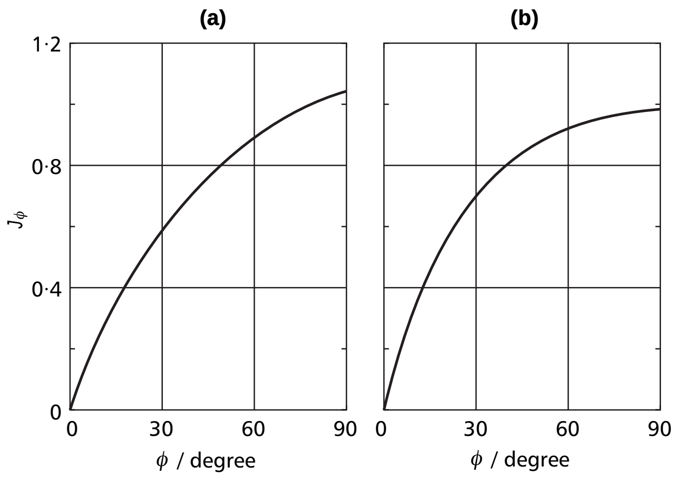


Figure 4.11 Variation of J_ϕ with angle of opening; (a) side-mounted casement window, (b) centre-pivoted windows[32]

Measurements[33,34] have shown that, with normally sized windows, the magnitude of the resulting single-sided ventilation, while smaller than cross-ventilation with similar areas of opening under comparable conditions, can be large enough to contribute to natural cooling. Table 4.23 provides formulae that enable ventilation rates to be calculated for wind and stack effect. It is suggested that calculations be carried out using both formulae and the larger value taken. The formula for wind induced infiltration represents a minimum, which will be enhanced up to threefold for certain wind directions and windows with openings that tend to deflect inwards the impinging wind.

4.7.4 Theoretical calculation methods

In principle, the airflow through a building and the ventilation rates of individual spaces within a building can be determined for a given set of weather conditions (i.e. wind speed, wind direction and external air temperature) if the following are known:

- the position and characteristics of all openings through which flow can occur
- a detailed distribution of surface mean pressure coefficients for the wind direction under consideration
- the internal air temperature(s).

In practice, equation 4.1 and its simplified forms are non-linear. Also, because the number of flow paths likely to be present in any but the simplest building will be considerable, full solutions involving the prediction of flow between individual rooms can only be obtained by computer methods. Examples of these techniques are considered in detail in the AIVC Applications Guide: *Air infiltration calculation techniques*[35]. Additionally, predictions are only as accurate as the input data on which they are based and these are often difficult to quantify.

4.7.4.1 Single-zone model

Despite the complexities of flow prediction, the flow equation may be readily solved when the interior of the building is represented as a single enclosed volume. This single cell network approach is a comparatively easy calculation to undertake and provides facilities to:

- incorporate any number of flow paths to and from outside
- take account of combinations of wind, stack and mechanically induced pressures
- identify the magnitude and direction through each of the flow paths
- calculate the internal pressure
- assess the effects of changes to location or characteristics of flow paths
- determine the size of openings required to provide adequate ventilation.

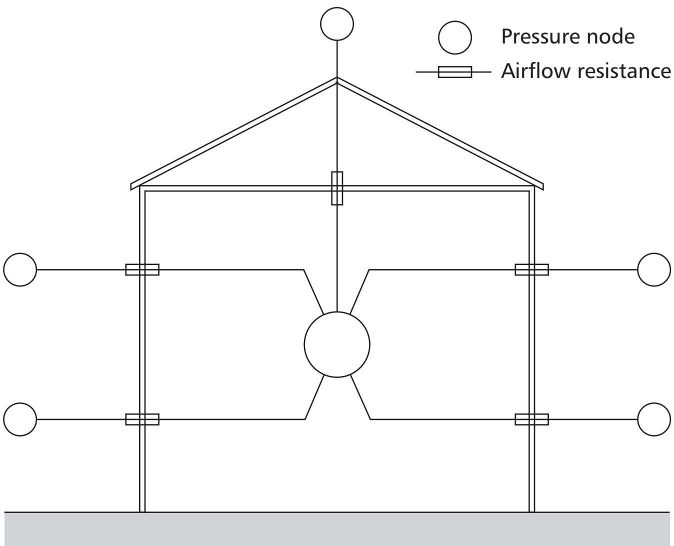


Figure 4.12 Single zone model: simple flow path network

The leakage characteristics of the building envelope are first represented by a series of flow paths linking the exterior of the building with the interior, see Figure 4.12.

Ideally, the location, size and flow characteristics of each opening should be defined. In practice this is rarely possible and an approximation or an amalgamation of flow paths is usually necessary. Flow characteristics for typical windows and doors are presented in Tables 4.3 and 4.4; a more comprehensive list of leakage openings is given in AIVC Technical Note 44[23].

Once the flow network has been constructed, equation 4.1 is applied to each opening. Since the magnitude of flow entering the building must be matched by the magnitude of air leaving, a summation of equation 4.1 for all openings must equal zero. Hence:

$$q_v = \sum_{j=\lambda}^{1} [C_j (p_{oj} - p_i)^{n_j}] = 0 \tag{4.12}$$

where λ is the total number of flow paths, C_j is the flow coefficient for path j, p_{oj} is the external pressure due to wind and temperature acting on path j (Pa), p_i is the inside pressure of building (Pa), n_j is the flow exponent of path j.

The external pressures acting on each path are derived directly from the wind and stack pressure equations (i.e. equations 4.5 and 4.6 respectively).

The only remaining unknown in equation 4.12 is the internal pressure, which is determined by iteration. An initial pressure is assumed and is repeatedly adjusted until flow balance is achieved. At this point, infiltration is represented by the sum of the ingress flows. Figure 4.12 illustrates typical infiltration characteristics for the simple network shown in Figure 4.11.

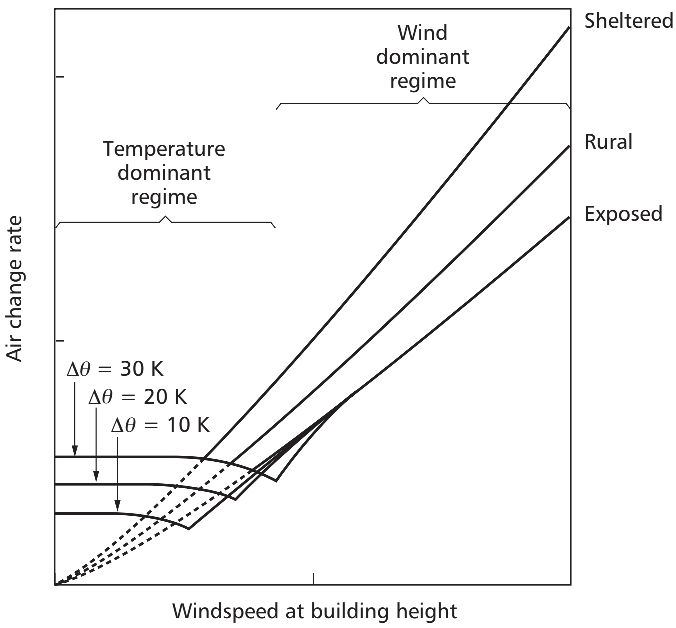


Figure 4.13 Typical infiltration characteristics for a simple flow path network[26]

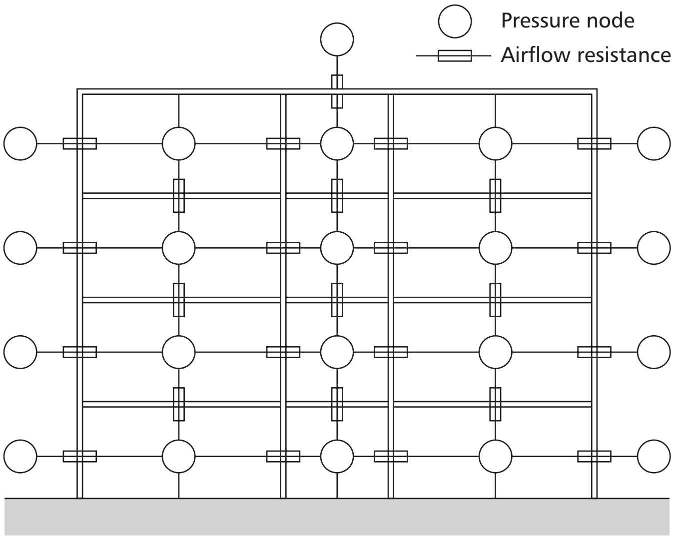


Figure 4.13 Multi-zone model

From this example, it may be deduced that:

— there is a clearly defined temperature dominant regime at low wind speeds, and a wind dominant regime at higher wind speeds

— the magnitude of the wind induced infiltration is considerably influenced by the degree of surrounding shielding.

A single zone approach, based on these concepts is presented in CIBSE AM10[2].

4.7.4.2 Multi-zone model

For buildings with internal divisions that impede the movement of air, it will be necessary to use a multi-zone model that takes account of specific internal airflow paths, see Figure 4.13. In terms of the input data requirements and the output information, single-zone and multi-zone models are similar. However, a multi-zone calculation involves many iterations to produce a solution in which there is a balance of inlet and outlet flows in all of the zones. This adds considerably to the complexity of the numerical method and the computing capacity required for its solution.

The multi-zone approach is capable of predicting air infiltration and ventilation flows in complex buildings provided data are available to numerically define the flow network and the external pressure distribution.

A suitable public domain model (CONTAM) that enables air flow and contaminant transport in complex multizone buildings to be analyse is available from the US National Institute for Standards and Testing* (NIST).

4.7.4.3 Calculating using computational fluid dynamics (CFD)

Computational fluid dynamics (CFD) is increasingly becoming a valuable design tool. In theory it can act as a 'numerical wind tunnel' and bypass the need to use data derived from tabular approximation to assess the pressure induced across building openings. Much is still needed in the way of validation and improved resolution, especially for complex building shapes and for buildings located in urban environments in which the wind regime is influenced by surrounding buildings and obstructions. Potential modes of operation include:

— as a numerical wind tunnel to derive wind pressure coefficients

— as an internal airflow model in which boundary pressure data are specified by conventional means or by the results of a numerical wind tunnel configuration

— complete building and outdoor flow regime simulation in a single model.

4.8 Airtightness testing

4.8.1 Measurement

The rate of air leakage through the fabric of buildings can be measured using an air pressurisation technique. Air is supplied to the building under test at a range of airflow rates, and the resulting pressure differential across the building envelope is measured for each rate of flow. It is recommended that the range of pressure differentials be extended to at least 50 Pa. This pressure is low enough to avoid damage to the building but sufficiently high to overcome the detrimental effects of moderate wind speeds. The format of the test is described in CIBSE TM23[5]. Two airtightness parameters are defined:

— *Air leakage index*: the results of the air leakage test are distributed about the internal surface area of the external façade and is calculated from the dimensions bordering the internal volume of the building under test. This area is given by the total of the walls, top, floor, ceiling (or underside of roof) depending on where the air barrier is and, in special circumstances, the area of the floor (i.e. when it is not ground supported, e.g. a timber floor or when there is an underground car park beneath the building.

— *Air permeability*: similar to the air leakage index but the floor area is included regardless of the underfloor construction.

To conduct a test, a fan system is temporarily coupled to a suitable doorway or similar opening in the building envelope. For a large or leaky building, this requires a high capacity fan system. For small buildings, a device known as a 'blower door' may be used. This is an assembly that includes one or more fans, some means of controlling airflow rate and instrumentation for measuring pressures. It is designed to fit into a normal doorframe, with facilities to clamp and seal it in place.

Buildings are tested with all external doors and windows closed, and with all internal doors wedged open. Any natural and mechanical ventilation openings are also sealed with polythene sheet and adhesive tape. Smoke extract fans or vents are left closed but not sealed. Other integral openings such as lift shafts are left unsealed.

* May be downloaded from the NIST website: www.bfrl.nist.gov/IAQanalysis/CONTAMWdesc.htm

The measured airflow rates and pressure differentials are related by equation 4.1.

Airtightness testing may be accompanied by smoke tests as a means of helping to identify the air leakage routes.

4.8.2 Data processing

The pressure difference across the building envelope versus the measured airflow rate is plotted on a log–log graph. A straight-line relationship will confirm that the test results are as expected and that nothing extraneous, such as a door or window left open, had occurred during the tests.

The airflow test data are further processed to take account of two factors. First, an air density correction is applied. This is determined from the air temperature and barometric pressure at the position of airflow measurement. The second correction factor takes account of any change in the volumetric airflow rate due to differences in the temperature of the supply air and that within the building. During the pressurisation test, outside air passes through the apparatus into the building and mixes with the inside air. If the temperature of the inside air is higher than that of the supply air, the supply air expands so that the volume rate of flow out of the building envelope is slightly greater than the measured airflow rate.

Following these corrections, a regression analysis is carried out on the pressure differentials across the building envelope and the corrected airflow rate to calculate the air leakage coefficient (C) and the value of the exponent (n), see equation 4.1. A correlation coefficient is also calculated to indicate the 'closeness of fit' of the measured data to the calculated relationship.

Using the calculated relationship, the airflow rate required to pressurise the building to 50 Pa (Q_{50}) is determined and then normalised with respect to the surface area (S) of the building to yield values for the quotient Q_{50} / S ($m^3{\cdot}h^{-1}){\cdot}m^{-2}$. This gives an air leakage value that may be compared with specified standards or with air leakage rates for other buildings. Other reference pressures may be used, as well as air leakage rates expressed in air changes per hour, for example.

References

1 *Heating, ventilating, air conditioning and refrigeration* CIBSE Guide B (London: Chartered Institution of Building Services Engineers) (2001/2)

2 *Natural ventilation in non-domestic buildings* CIBSE AM10 (London: Chartered Institution of Building Services Engineers) (2005)

3 *Mixed mode ventilation* CIBSE AM13 (London: Chartered Institution of Building Services Engineers) (2005)

4 *Minimising pollution at air intakes* CIBSE TM21 (London: Chartered Institution of Building Services Engineers) (1999)

5 *Testing buildings for air leakage* CIBSE TM23 (London: Chartered Institution of Building Services Engineers) (2000)

6 *Improved life cycle performance of mechanical ventilation systems* CIBSE TM30 (London: Chartered Institution of Building Services Engineers) (2003)

7 *CIBSE standard tests for the assessment of building services* CIBSE TM33 (London: Chartered Institution of Building Services Engineers) (2004)

8 *Weather, solar and illuminance data* CIBSE Guide J (London: Chartered Institution of Building Services Engineers) (2002)

9 *Ventilation* The Building Regulations 1991 Approved Document F (London: Her Majesty's Stationery Office) (1994)

10 *Combustion appliances and fuel storage systems* The Building Regulations 2000 Approved Document J (London: The Stationery Office) (2001)

11 *Conservation of fuel and power in dwellings* The Building Regulations 2000 Approved Documents L1 (London: The Stationery Office) (2001)

12 *Conservation of fuel and power in buildings other than dwellings* The Building Regulations 2000 Approved Documents L2 (London: The Stationery Office) (2001)

13 *The Scottish Building Standards — Technical Domestic and Non-domestic Handbook* (Edinburgh: The Scottish Executive (2004)

14 *Conservation of fuel and power* The Building Regulations (Northern Ireland) 1994 Technical booklet F (London: The Stationery Office) (1998)

15 *Ventilation* The Building Regulations (Northern Ireland) 1994 Technical booklet K (London: The Stationery Office) (1999)

16 *Heat producing appliances* The Building Regulations (Northern Ireland) 1990 Technical booklet L (London: Her Majesty's Stationery Office) (1991)

17 *Energy-efficient mechanical ventilation systems* Good Practice Guide GPG257 (The Carbon Trust) (www.thecarbontrust.co.uk)

18 *The national air quality strategy for England, Scotland and Northern Ireland: working together for clean air* (London: The Stationery Office) (2000)

19 BS EN 13779: *Ventilation for buildings. Performance requirements for ventilation and air-conditioning systems* (London: British Standards Institution) (2005)

20 Kato S Flow network model based on power balance as applied to cross-ventilation *Int. J. Ventilation* **2** (4) 395–408 (March 2004)

21 Sandberg M An alternative view on the theory of cross-ventilation *Int. J. Ventilation* **2**(4) 409–418 (March 2004)

22 Etheridge D W Natural ventilation through large openings — Measurements at model scale and envelope flow theory *Int. J. Ventilation* **2** (4) 325–342 (March 2004)

23 Orme M, Liddament M W and Wilson A *An analysis and summary of AIVC's numerical database* AIVC Technical Note 44 (Coventry: Air Infiltration and Ventilation Centre) (1994)

24 Sawachi T, Narita K, Kiyota N, Seto H, Nishizawa S and Ishikawa Y Wind pressure and air flow in a full-scale building model under cross-ventilation *Int. J. Ventilation* **2** (4) 343–358 (March 2004)

25 BS 5925: 1991: Code of practice for ventilation principles and designing for natural ventilation (London : British Standards Institution) (1991)

26 Liddament M W *A guide to energy efficient ventilation* (Coventry: Air Infiltration and Ventilation Centre) (1996)

27 Bowen A J *A wind tunnel investigation using simple models to obtain mean surface wind pressure coefficients for air infiltration estimates* Report LTR LA 20N (Canada: National Aeronautical Establishment) (1976)

28 Levermore G J The exponential limit to the cooling of buildings by natural ventilation *Building Serv. Eng. Res. Technol.* **23** (2) 199–125 (2002)

29 Kronvall J Testing of houses for air leakage using a pressure method *ASHRAE Trans.* **84** (1) 1978

30 Dubrul C *Inhabitants' behaviour with respect to ventilation* AIVC Technical Note 23 (Inive, Belgium: Air Infiltration and Ventilation Centre) (1988)

31 *Energy use in offices* Energy Consumption Guide ECG019 (The Carbon Trust) (www.thecarbontrust.co.uk)

32 *Principles of natural ventilation* BRE Digest 210 (Garston: Building Research Establishment) (1978)

33 Warren P R Ventilation through openings on one wall only *Proc. UNESCO International Seminar on Heat Transfer in Buildings, Dubrovnik 1977* **5** (1977)

34 Walker R R and White M K Single sided natural ventilation: how deep an office? *Building Serv. Eng. Res. Technol.* **13** (4), 231–236 (1992)

35 Liddament M W *Air infiltration calculation techniques — an application guide* (Coventry: Air Infiltration and Ventilation Centre) (1986)

Appendix 4.A1: Air infiltration development algorithm (AIDA)

4.A1.1 Introduction

This algorithm was developed by the Air Infiltration and Ventilation Centre (AIVC)* and is reproduced here by kind permission of the AIVC. The CIBSE cannot take responsibility for its accuracy. Any queries regarding the algorithm or its use should be referred to the AIVC.

AIDA is a basic infiltration and ventilation calculation procedure intended for the calculation of air change rates in single zone enclosures. It also resolves flow rates for any number of user-defined openings and calculates wind and stack pressures. The program is easy to use and provides an accurate solution to the flow balance equation. As its name suggests, this is a development algorithm that may be adapted to suit individual needs. It uses concepts outlined in chapter 12 of the AIVC's *Guide to energy efficient ventilation*[26].

4.A1.2 Program operation

AIDA is written in BASIC and a full listing is given in section 4.A1.4. Initiation of the code will be machine dependent but in the BASIC environment will normally be achieved by using the 'RUN' command. Once the response 'Welcome to AIDA' appears on the screen, the 'EXE' or 'ENTER' key is pressed sequentially in response to each input question (identified by '?').

Data entry is self-explicit. The order of data entry for each flow path is as follows:

- building volume (m^3)
- number of flowpaths
- height of flow path (m)
- flow coefficient ($m^3 \cdot s^{-1}$ at 1 Pa)
- wind pressure coefficient.

Once the flow path data have been entered, the following items of climatic data are requested:

- outdoor temperature (°C)
- internal temperature (°C)
- wind speed at building height ($m \cdot s^{-1}$)

* Air Infiltration and Ventilation Centre, University of Warwick Science Park, Sovereign Court, Sir William Lyons Road, Coventry CV4 7EZ, UK.

On completion of data entry, the computer responds with the message 'Calculation in progress'. After iteration is completed, the infiltration rate is displayed. The air change rate and request for further climatic data are automatically displayed; breakout of the program may be achieved using 'CONTROL BREAK'. At the end of a session, the most recent data remain in store and can be recovered using the 'PRINT' command followed by the variable name, e.g:

- 'PRINT Q' displays the infiltration rate
- 'PRINT F(2)' displays flow path 2.

Care must be taken when entering data since there is no error trapping and no editing facility. It will be necessary to restart the program if an error is made.

As a demonstration algorithm, the input/output routines are rudimentary and may be adapted to suit individual requirements.

4.A1.3 Solution technique

The flow balance equation is solved by iteration using a combination of 'bisection' and 'addition'. An internal pressure, known to be substantially negative with respect to the true pressure, is selected as a starting condition. For most applications, a value of –100 Pa should be satisfactory and is introduced automatically at line 320. Successive iterations improve the internal pressure value until a flow balance within 0.0001 $m^2 \cdot s^{-1}$ is achieved. The flow balance criterion is established in line 450. An understanding of the technique may be gleaned from an analysis of lines 320 to 470 of the program. While the approach adopted is not necessarily the most numerically efficient, it is extremely robust and should not fail under normal circumstances over a wide range of flow conditions and leakage characteristics.

4.A1.4 Program listing and variable names

```
20    PRINT "Welcome to AIDA"
30    PRINT "Air Infiltration Development Algorithm"
40    PRINT "M Liddament - AIVC Guide to
      Ventilation 1995"
50    DIM H(50) , C(50) , N(50) , P(50) , T(50) , W(50) ,
      S(50) , F(50)
55    PRINT:PRINT:PRINT
60    D = 1.29:REM Air density at 0 deg C
70    PRINT "Enter building data: "
```

```
80   INPUT "Building volume (m3) = ";V
85   PRINT:PRINT:PRINT
90   PRINT "Enter flow path data: "
100  INPUT "Number of flow paths = ";L
110  FOR J = 1 TO L
115  PRINT:PRINT:PRINT
120       PRINT "Height (m) (Path ";J;") = ";:INPUT H(J)
130       PRINT "Flow coef (Path ";J;") = ";:INPUT C(J)
140       PRINT "Flow exp (Path ";J;") = ";:INPUT N(J)
150       PRINT "Pres coef (Path ";J;") = ";:INPUT P(J)
160  NEXT J
170  PRINT "Enter climatic data:"
175  PRINT:PRINT:PRINT
180  INPUT "Ext temp (deg C) = " ;E
190  INPUT "Int temp (deg C) = " ;I
200  INPUT "Wind spd (bldg ht) (m/s) = " ;U
210  REM   Pressure calculation
220  FOR J = 1 TO L
230       REM Wind pressure calculation
240       W(J) = .5*D*P(J)*U*U
250       REM Stack pressure calculation
260       S(J) = -3455*H(J)*(1/(E + 273)-1/(I + 273))
270       REM Total pressure
280       T(J) = W(J) + S(J)
290  NEXT J
300  REM Calculate infiltration
305  CLS: PRINT:PRINT:PRINT
310       PRINT "Calculation in progress"
320  R = -100
330  X = 50
340  Y = 0
350  B = 0
360  R = R + X
370  FOR J = 1 TO L
380       Y = Y + 1
390       O = T(J)-R
400       IF O = 0 THEN F(J) = 0; GOTO 430
410            F(J) = C(J)*ABS(O) ^ N(J))*O/ABS(O)
420            B = B + F(J)
430  NEXT J
440  IF B < 0 THEN R + R-X: X = X/2: GOTO 350
450  IF B < .0001 THEN GOTO 470
460  GOTO 350
470  Q = 0
480  FOR J = 1 TO L
490       IF F(J) > 0 THEN Q = Q + F(J)
500  NEXT J
505  PRINT:PRINT:PRINT
520  PRINT "Infiltration rate (m3/s) = ";Q
530  A = Q*3600/V
540  PRINT "Air change rate (ach) = ";A
545  PRINT:PRINT:PRINT
550  GOTO 170
```

The names of variables are as follows:

A	Air change rate (h^{-1})
B	Flow balance
C(J)	Flow coefficient for path J
D	Air density ($kg \cdot m^{-3}$)
E	External temperature (°C)
F(J)	Calculated flow rate for flow path J ($m^3 \cdot s^{-1}$)
H(J)	Height of flow path J (m)
I	Internal temperature (°C)
J	Flow path number
L	Total number of flow paths
N(J)	Flow exponent for flow path J
O	Pressure difference across flow path (Pa)
P(J)	Wind pressure coefficient for flow path J
Q	Infiltration rate ($m^3 \cdot s^{-1}$)
R	Internal pressure (Pa)
S(J)	Stack-induced pressure for flow path J (Pa)
T(J)	Total external pressure on flow path J (Pa)
U	Wind speed at building height (m·s–1)
V	Volume of building enclosure (m^3)
W(J)	Wind-induced pressure (Pa)
X	Iteration pressure step (Pa)
Y	Iteration counter

5 Thermal response and plant sizing

5.1 Introduction

This section presents design information for the calculation of heating loads, cooling loads and heating plant capacity. Typically this will follow the process outlined in Figure 5.1. (*Note*: Figure 5.1 also indicates the relevant sections of this chapter). It is recognised that design is an iterative process and this process will therefore normally be repeated at concept, detailed proposals (scheme) and final proposals (detail) design stages. While empirical rules of thumb may be appropriate at concept stage, later design stages will usually require better accuracy and quality, less uncertainty, a clear understanding of the sensitivity to assumptions and reduced risk.

The heating and cooling requirements are determined by:

— location of the building

— design internal and external temperatures

— design internal and external humidities

— thermal characteristics of the building

— ventilation rate

— type of system

— building usage patterns.

This chapter gives the background to steady state and dynamic calculation techniques that can be used for sizing plant when the above conditions are known. At the time of publication (November 2005) there is considerable international debate on the preferred methods for load calculation and a draft European Standard that sets down compliance tests is under development[1]. In the meantime guidance is given on the various methods currently available. Data and guidance are provided on the calculation of steady state heat losses, determination of summertime temperatures (or overheating risk) for naturally and mechanically ventilated buildings and the determination of peak space cooling loads. In common with current legislation no recommendations are made as to maximum acceptable temperatures in buildings.

It is recognised that many engineers will make use of computer programs to carry out the design calculations described in this section. These programs may claim to use the methods described here or alternatives. The majority of the alternative methods use techniques that are intended to simulate the performance of an actual building and associated systems. The simulations will normally be for the purpose of determining energy consumption or overheating. Although some of the calculation techniques described in this chapter of the Guide may be used as part of an energy calculation, energy calculation methods are not specifically covered here.

This section does not describe a 'CIBSE simulation method'. However, recognising that such methods will be used, it is important that they are suitable for the purpose and so the CIBSE has developed a set of software tests, published as CIBSE TM33: *CIBSE standard tests for the assessment of building services design software*[2]. The tests described in TM33 are not exhaustive; they are intended to provide a minimum standard. The tests are intended to be used by both developers and users and set a basic performance standard for design software. Furthermore because the way the software is used can have a significant effect upon the result this section includes details of quality assurance procedures that, if followed, will help to minimise errors when using software.

CIBSE TM33[2] is intended to form part of an accreditation process for detailed thermal models for use in the 'National Calculation Methodology' as described in the Building Regulations (and the equivalent legislation for Scotland and Northern Ireland) and the European Performance of Buildings Directive. As a consequence it is currently subject to detailed revision. For this reason no direct references are made in this section to parts of TM33.

In using this Guide, it is important to understand the distinction between comfort temperature, air and surface temperatures that determine the rate of heat transfer to and from the building, and the temperatures sensed by the detectors employed to control the heating, air conditioning and ventilation systems. For design and calculation purposes these are often combined together and treated as a single index temperature. In addition to the effects of necessary simplifications in the calculation methods described here the assumed thermal properties of the building, its construction and pattern of use as represented by the calculation techniques are likely to differ from those found in reality. Therefore some discrepancies between predicted and measured performance must be expected. Reference data are provided where available and definitions of commonly used indices and parameters are also given. This revision differs from the previous version in that dry resultant temperature has been replaced by operative temperature. This is because the two indices are identical under 'normal' conditions and operative temperature is more commonly used throughout the rest of the world.

Users should also be aware that the some of the 'standard' values (such as heat transfer coefficients) used in this Guide differ slightly from those contained within European Standards. Differences are small and so the examples contained in this section use the traditional CIBSE constants.

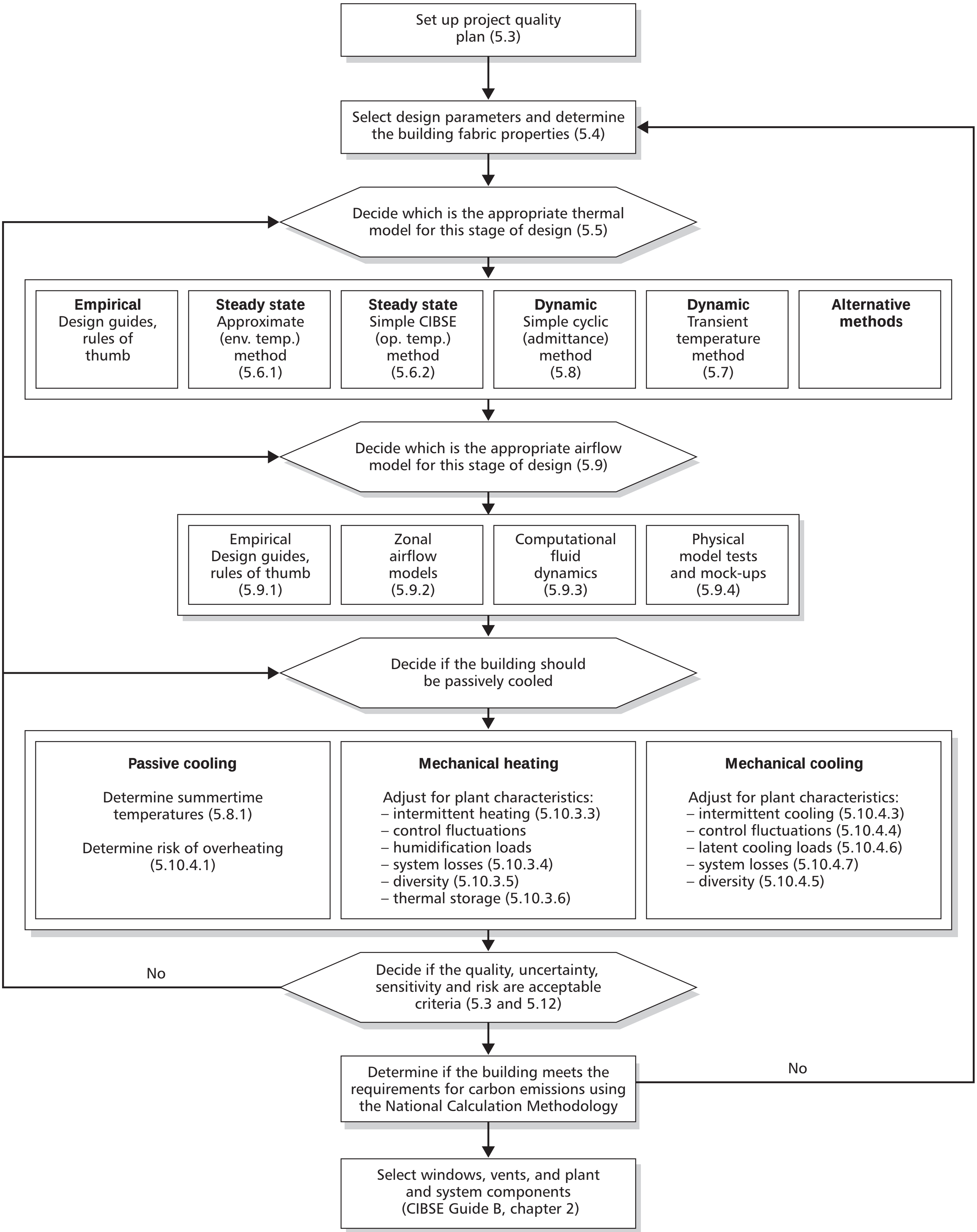


Figure 5.1 Thermal modelling and plant sizing flow diagram

The structure of this chapter is as follows:

- notation and glossary of terms
- a brief note on the need for quality control of calculations
- data requirements
- description of calculation methods with simple examples
- application of the methods
- quality control techniques to minimise errors.

Appendices provide details of background issues and theory.

5.2 Notation and glossary of terms

5.2.1 Symbols

Some of the quantities in this Guide occur in three forms: the instantaneous value, which is denoted by the appropriate letter (e.g. X); the 24-hour mean or steady state value, denoted by $\bar{X}$; and the instantaneous variation about the mean value, denoted by $\tilde{X}$. Where time lags occur, the variation symbol may be given a subscript to indicate the time at which it occurs, e.g. $\tilde{X}_q$ is the value of $\tilde{X}$ at time q. Peak values are indicated by a caret mark (^) above the symbol (e.g. $\hat{I}$). Symbols that appear only in the appendices are omitted. These symbols are defined in the appendices in which they occur.

A_h	Projected area of heat emitter (m^2)
A_n	Area of surface n (m^2)
c_p	Specific heat capacity of air (J·kg^{-1}·K^{-1})
C_v	Ventilation conductance (W·K^{-1})
F	Surface factor (—)
F_{1c}, F_{2c}	Factors related to characteristics of heat source with respect to operative temperature (—)
F_3	Correction factor for intermittent heating (—)
F_{au}	Room conduction factor with respect to air node (—)
F_{ay}	Room admittance factor with respect to air node (—)
F_{cu}	Room conduction factor with respect to operative temperature (—)
F_{cy}	Room admittance factor with respect to operative temperature (—)
f	Decrement factor (—)
f_r	Thermal response factor (—)
h_c	Convective heat transfer coefficient (W·m^{-2}·K^{-1})
h_r	Radiative heat transfer coefficient of a black body (W·m^{-2}·K^{-1})
h_{so}	Outside heat transfer coefficient (W·m^{-2}·K^{-1})
I_l	Re-radiation loss (W·m^{-2})
I_t	Total solar irradiance (W·m^{-2})
N	Number of air changes per hour (h^{-1})
q_m	Mass flow rate of air (kg·s^{-1})
q_v	Natural ventilation rate (m^3·s^{-1})
q_{vt}	Total ventilation (i.e. mechanical plus infiltration) rate (m^3·s^{-1})
R	Radiant fraction of the heat source (—)
R_{se}	Thermal resistance between the inside of a surface and the environmental temperature (m^2·K·W^{-1})
$\tilde{S}_a$	Cyclic solar gain factor at air node (—)
$\bar{S}_a$	Mean solar gain factor at air node (—)
S_c	Shading coefficient (—)
$\tilde{S}_e$	Cyclic solar gain factor at environmental node (—)
$\bar{S}_e$	Mean solar gain factor at environmental node (—)
$\tilde{S}_{eh}$	Cyclic solar gain factor at environmental node for heavyweight building (—)
$\tilde{S}_{el}$	Cyclic solar gain factor at environmental node for lightweight building (—)
t	Time (h)
t_{in}	Duration of internal heat source n (h)
t_o	Duration of plant operation including preheat (h)
t_r	Recharge time for heat storage system (h)
U_n	Thermal transmittance of material n (W·m^{-2}·K^{-1})
U'	Thermal transmittance modified for heat loss through internal partitions (W·m^{-2}·K^{-1})
V	Room volume (m^3)
Y	Thermal admittance (W·m^{-2}·K^{-1})
α	Surface absorption coefficient (—)
Φ_b	Heat loss from back of radiator (W)
Φ_c	Internal heat gain (W)
Φ_{con}	Convective component of internal gains (W)
Φ_{csa}	Total sensible cooling load to the air node (W)
Φ_f	Fabric heat gain (W)
Φ_{fa}	Fabric gain to the air node (W)
Φ_{in}	Instantaneous heat gain from internal heat source n (W)
Φ_k	Cooling load (W)
Φ_o	Plant output (W)
Φ_p	Plant size for intermittent operation (W)
Φ_r	Input rating of plant (W)
Φ_{rad}	Radiant component of internal gains (W)
Φ_{sa}	Solar heat gain to the air node (W)
Φ_{se}	Solar heat gain to the environmental node (W)
Φ_t	Total heat loss (W)
Φ_{ta}	Total gains to the air node (W)
Φ_{te}	Total gains to the environmental node (W)
Φ_v	Cooling load related to ventilation (W)
ϕ_s	Short wave heat flow (W·m^{-2})
ϕ_r	Net rate of radiant heat flow into the surface (W·m^{-2})
θ_a	Air temperature (°C)
θ_{ai}	Inside air temperature (°C)
θ_{ao}	Outside air temperature (°C)
θ_c	Operative temperature at centre of room (°C)
θ_e	Heat transfer temperature (°C)
θ_{ei}	Environmental temperature (°C)
θ_{eo}	Sol-air temperature (°C)
θ_h	Temperature of heat emitter (°C)
θ_m	Mean surface temperature (°C)
θ_r	Mean radiant temperature (°C)
θ_s	Temperature of surface (°C)
θ_{sn}	Temperature of surface n (°C)
ρ	Density of air (kg·m^{-3})
ΣA	Sum of room surface areas, unless otherwise indicated (m^2)
$\Sigma (A\,U)$	Sum of the products of surface area and corresponding thermal transmittance over surfaces through which heat flow occurs (W·K^{-1})
$\Sigma (A\,Y)$	Sum of the products of surface area and corresponding thermal admittance over all surfaces (W·K^{-1})
ϕ	Time lag associated with decrement factor (h)
ψ	Time lag associated with surface factor (h)
ω	Time lead associated with admittance (h)

5.2.2 Definitions

Air node

The hypothetical point in space at which all convective heat transfer is assumed to take place.

Air temperature (θ_a)

The temperature registered by a dry thermometer, shielded from radiation, suspended in the air.

Black body

A body which absorbs all incident radiation at all wavelengths (i.e. emissivity = 1). A black body is also an ideal radiator.

Convective heat transfer coefficient (h_c)

The coefficient relating the exchange of heat between a surface and the surrounding air. The value of h_c will depend upon the nature of the airflow over the surface (natural or forced) and the temperature of that surface. For design using the CIBSE admittance method, standardised values are used, see Table 5.1. Heat flow from surfaces is defined such that a positive heat flow is one that flows from a hot body to a cold body. For example, upward heat flow from a warm floor would be positive, as would upward heat flow into a cold ceiling.

The convective heat transfer coefficient depends on the dimensions, slope, roughness of the surface, surface temperature, and the local air temperature and air velocity. Detailed thermal models may make use of coefficients based on correlations between these variables[3,4]. The traditional CIBSE values given in Table 5.1 are for buoyancy driven flow. No simple figures are available for forced convection but an average value of 3 $W{\cdot}m^{-2}{\cdot}K^{-1}$ may be considered typical.

Table 5.1 Surface convective heat transfer coefficient, h_c

Surface	Direction of heat flow	Surface coefficient, h_c / $W{\cdot}m^{-2}{\cdot}K^{-1}$
Floor	Upwards	4.3
	Downwards	1.5
Wall	Horizontal	3.0
Ceiling	Upwards	4.3
	Downwards	1.5

Decrement factor (f)

The variation in the rate of heat flow through the structure due to variations in external heat transfer temperature from its mean value with the environmental temperature held constant, divided by the steady state transmittance.

Environmental node

The hypothetical point in a room from which radiant and convective heat is exchanged to the room surfaces. It has a temperature equal to the environmental temperature.

Environmental temperature (θ_{ei})

A hypothetical temperature that determines rate of heat flow into a room surface by convection from the room and radiation from surrounding surfaces and other radiant sources. It is the temperature at the environmental node and traditionally taken as:

$$\theta_c = {}^1/_3\ \theta_{ai} + {}^2/_3\ \theta_r \qquad (5.1)$$

G-value

See *total solar energy transmittance*.

Grey body

A body which absorbs the same fraction of incident radiation at all wavelengths (i.e. assumes that emissivity is independent of wavelength). A grey body is often used to simplify the problems of predicting radiant energy exchange between surfaces.

Heat transfer temperature (θ_e)

A temperature calculated to give a rate of heat transfer to a surface equivalent to that from a combination of radiation and convection. For example, the rate of heat flow to a surface can be expressed as:

$$\phi_s = h_c\,(\theta_a - \theta_s) + \phi_r \qquad (5.2)$$

or:

$$h_c\,(\theta_e - \theta_s) = h_c\,(\theta_a - \theta_s) + \phi_r \qquad (5.3)$$

$$\theta_e = \theta_a + \phi_r / h_c \qquad (5.4)$$

where ϕ_s is the rate of heat flow into the surface ($W{\cdot}m^{-2}$), h_c is the convective heat transfer coefficient ($W{\cdot}m^{-2}{\cdot}K^{-1}$), θ_a is the air temperature (°C), θ_s is the surface temperature (°C), ϕ_r is the net rate of radiant heat flow into the surface ($W{\cdot}m^{-2}$), θ_e is the heat transfer temperature (°C).

Inside air temperature (θ_{ai})

The average air temperature in an enclosed space.

Inside surface temperature (θ_s)

The temperature of the surface immediately adjacent to an air space. It is the driving temperature for both the convective and radiant energy transfers within the building and can be directly measured by means of a suitable contact thermometer. The temperature varies across the surface and is affected by corners, changes in construction, air movement etc. For most heat transfer calculations a single temperature is assumed for each surface.

Longwave radiation

Radiation from a 'low temperature' source (i.e. less than, say, 500 °C), e.g. walls, heating radiators. Longwave radiation exchange depends on the temperatures of the emitter and receiver.

Mean radiant temperature (θ_r)

The mean radiant temperature is a function of the respective areas, shapes and surface temperatures of the

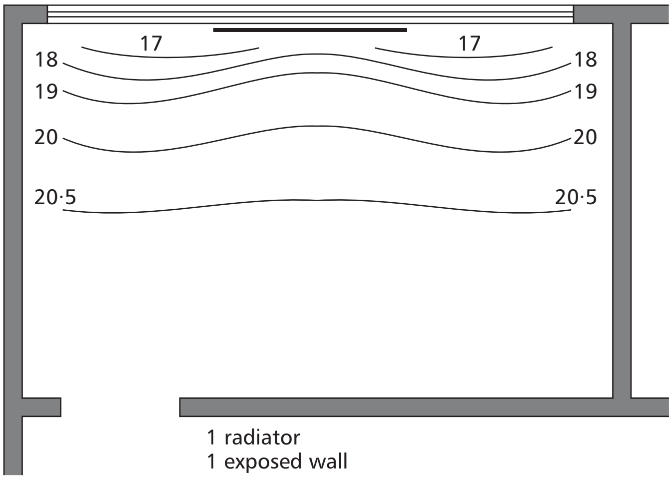


Figure 5.2 Example of the variation of mean radiant temperatures across a room (plan view)

enclosing elements as viewed from a point. For a point at the centre of a cubical room with all surfaces having an emissivity of 1 and where there are no other radiant sources, the mean radiant temperature is the same as the mean surface temperature (θ_m). Determination of precise values is complex due to the effect of surface view factors (see also Appendix 5.A3). The mean radiant temperature varies from point to point within a space. Figure 5.2 shows a computer prediction of mean radiant temperature for a specific situation in which a radiator is placed adjacent to a cold surface to minimise the variation of radiant temperature.

Mean surface temperature (θ_m)

The mean surface temperature is determined by totalling the products of the areas and temperatures of the surrounding surfaces and dividing this total by the sum of the areas, i.e:

$$\theta_m = \frac{\Sigma\,(A\,\theta_s)}{\Sigma\,(A)} \tag{5.5}$$

Method or calculation method

A procedure for using a mathematical model to define an outcome of interest. For example, a *U*-value calculation is a method that uses a 'steady state' mathematical model of heat transfer to represent the rate of heat exchange between two sides of a fabric envelope. Similarly plant sizing methods may use an underlying mathematical model representing a number of selected objects and processes to calculate the size of plant required to maintain a prescribed condition in a building.

Model

An abstraction (or representation) of real life objects and processes. For example, a physical model of a building is a physical representation of the building using normally smaller scale, tangible materials. A mathematical model consists of symbolic representations of quantities appearing in physical objects, systems and processes. A computer model is a mathematical model of objects and/or processes created by means of a computer program.

— *Steady state model*: the underlying assumption in such models is that all parameters of a model are constant and do not vary with time. The *U*-value for example is used to predict one dimensional heat transfer between two static environments (so that changes in heat storage are neglected) through a homogeneous construction.

— *Dynamic model*: a model in which a number of parameters vary with time and calculations represent behaviour over a chosen time period, e.g. of one hour or less. This allows temporal variations in thermal storage, weather, occupancy etc., to be represented.

— *Steady-cyclic or cyclic model*: a dynamic model in which the parameters are repeated at a regular interval. The CIBSE admittance method is a cyclic model in which the assumption is that weather is represented as a harmonic with a (repeated) period of 24 hours.

Operative temperature (θ_c)

The operative temperature θ_c in a real room is equal to the air temperature in an hypothetical room such that an occupant would experience the same net energy exchange with the surroundings. It is used as an index temperature for comfort where the air velocities are low.

In the hypothetical room, air temperature is uniform and equal to that of the internal surfaces. All the internal surface emissivities are unity (e.g. radiantly black).

An approximate formula for operative temperature (°C) in terms of room air temperature (°C), mean radiant temperature (°C) and coefficient *A* related to air speed v_a ($m{\cdot}s^{-1}$) is:

$$\theta_c = A\,\theta_{ai} + (1 - A)\,\theta_r \tag{5.6}$$

where *A* takes the following values:

— for $v_a < 0.2$: $A = 0.5$

— for $0.2 < v_a < 0.6$: $A = 0.6$

— for $0.6 < v_a < 1$: $A = 0.7$

In most practical instances where the air speed is below $0.2\ m{\cdot}s^{-1}$, or the mean radiant and air temperatures differ by less than 0.4°C, the operative temperature may be taken to be the average of the air temperature and mean radiant temperature:

$$\theta_c = {}^1/_2\,\theta_{ai} + {}^1/_2\,\theta_r \tag{5.7}$$

Outside air temperature (θ_{ao})

The average temperature of the air surrounding the building.

Overheating criteria

Criteria used to assess the performance of naturally and mechanically ventilated spaces during periods of hot weather. Design criteria can vary in complexity from the number of hours for which a particular temperature is exceeded to a sophisticated comfort prediction. Note that there is no legal maximum limit for temperatures within buildings. Recommended overheating criteria are considered in chapter 1, section 1.4.2.

Performance assessment methods (PAMs)

Documents setting out detailed procedures for carrying out and reporting upon common modelling tasks.

QA procedures

Formalised procedures that have been developed to assure the agreed standard is met.

Quality assurance

Process of delivering an agreed standard for the delivery of a product or service, often in accordance with BS EN ISO 9000.

Radiant heat transfer coefficient (h_r)

The coefficient relating the exchange of longwave radiation between surfaces. It is a function of the geometrical arrangement of the bodies and their temperatures. For simplified techniques a constant value of 5.7 $W \cdot m^{-2} \cdot K^{-1}$ is used (corresponding to a temperature of 20 °C and an emissivity of 1).

Radiant temperature

The apparent radiant temperature of a surface as measured by a suitable radiant thermometer. The measured temperature varies depending on the position of the thermometer to the surface. For a surface forming part of a black body (i.e. with emissivity = 1) the radiant temperature measured will equal the actual surface temperature. For a surface forming part of a grey body (i.e. with emissivity less than 1) the radiant temperature measured is a function of the temperature of the surface, the emissivity of the surface and radiation from the surroundings reflected from the surface.

Shading coefficient (S_c)

The ratio of the instantaneous heat gain at normal incidence transmitted by a particular glass/blind combination to that transmitted by a reference glass, usually 3 mm or 4 mm thick clear glass, see 5.7.3

Shortwave radiation

Radiation from a high temperature source, e.g. the sun, electric lights. Shortwave radiation exchange does not depend upon the temperature of the receiving surface.

Simulation

Calculation of building behaviour; sometimes taken to refer to a complete simulation results set.

Sol-air temperature (θ_{eo})

The hypothetical temperature that determines the rate of heat flow into an external surface by convection from the surrounding air, shortwave solar radiation and radiative exchange to the surrounding (other buildings, ground and sky), see chapter 2: *External design data*.

Surface factor (F)

The ratio of the variation of radiant heat flow about its mean value readmitted to the space from the surface, to the variation of heat flow about its mean value incident upon the surface.

Thermal admittance (Y-value)

The rate of flow of heat between the internal surfaces of the structure and the environmental temperature in the space, for each degree of deviation of that temperature about its mean value.

Thermal transmittance (U-value)

The thermal transmission through unit area of a given structure, divided by the difference between the effective ambient temperature on either side of the structure under steady state conditions.

Total solar energy transmittance (G-value)

The sum of the direct solar transmittance and the heat transferred by radiation and convection into the space(5). It is generally applied to glazing combinations.

5.3 Quality assurance

A sound calculation procedure cannot by itself ensure a sound design process. To do achieve that it is necessary to have procedures in place to minimise the risk of error. This can be done by following an appropriate quality plan. This section introduces the application of quality assurance to design calculations; more details are given in Appendix 5.A1.

As defined in section 5.2, quality assurance (QA) is a management process designed to consistently achieve stated objectives. This requires the commitment of both the top management and staff. This commitment is called quality policy which is enforced by means of a set of quality procedures, thus forming the quality plan (QP) of the organisation. As such, a QP, in the context of design calculations, is a tool that helps achieve consistent results, every time a design decision or an assessment is made. Using the QP tool helps ensure the use of calculation methods that are fit for the purpose involved, avoid or reduce errors when using the method, provide an audit trail of calculations for future scrutiny and to implement best practice.

Introduction of QA in the use of calculation methods applies to the use of both simple hand calculation and software methods. However, due to the complexity of software tools, the number of options they offer, the larger number of input data they require and the fact that their results, in the majority of cases, will require interpretation, the use of well thought out procedures are a necessity when applying software tools. Most of what is discussed in this section and the related appendix is therefore biased towards the use of software tools. However, it should be emphasised that the decision of using a simple manual calculation method as opposed to using dynamic modelling software is in itself a QA issue that could be a critical decision and should be made and documented as part of a

QA process. Guidance for the selection of software is given in Appendix 5.A1, the subject is covered in more detail in CIBSE AM 11: *Building energy and environmental modelling*[6].

Once the selection of software has been made, the correct application of software is fundamental to obtaining reliable results. This is largely a user issue, and a matter of user training and skill in the use of the specific software selected. All modelling software contains simplifications and the engineer needs to know how to simplify the building description to the program in an appropriate manner.

For instance, have the floor and ceiling voids been modelled as the program requires or has the correct weather data set been selected for the intended design question? Ideally, the accuracy and precision of data entry would be achieved in every instance, but this is inevitably impossible. Blunders in data entry do occur, and at the design stage, not all building parameters are known. Reliance on default values, either chosen by the operator or automatically by the software, is a further evidence for the importance of training and knowledge of assumptions within the software.

Human error in entering data is difficult to guard against completely, particularly in larger building models. However, depending on the project size and value, the level of checking and the way checks are made may be different. For example, a firm may decide that for certain size projects, a second person should check the model developed and input data used. Lack of knowledge of the actual characteristics and performance of materials and the way a building is used is yet another source of uncertainty which has to be borne in mind when making a design decision. The design margins applied should take account of the sensitivity of the design to important uncertain parameters.

5.4 Selection of design parameters

This section describes the parameters and data required to carry out the design. Depending upon the building these are some, or all, of the following:

- location of the site
- internal design conditions (see also chapter 1)
- appropriate overheating criteria (see also chapter 1)
- external design conditions (see also chapter 2 and CIBSE Guide J)
- infiltration and ventilation requirements (see also chapter 4)
- internal gains and patterns of use (see also chapter 6)
- building fabric properties (see also chapter 3)
- building geometry.

It is essential that the design conditions are explained to the client and agreement obtained before proceeding with the design; this should include a risk assessment (for example the percentage of the year for which design conditions may not be achieved and the likely impact).

5.4.1 Internal design conditions

All heating and cooling load calculation methods assume a design temperature, see chapter 1: *Environmental criteria for design*. That temperature may be a comfort temperature such as the operative temperature. Comfort temperatures are a function of air temperature, radiant temperature and air speed. The last of these can often be assumed (and should be selected) to lie within a range where it has little effect upon the comfort index. In many cases the assumption that there is no variation in the spatial distribution of air temperature is reasonable. Radiant temperature depends upon surface temperatures and the distance between the room surfaces, the observer and the radiant component of the heat emitter. It is unrealistic to assume the radiant temperature to be constant within a space (see Figure 5.2). For practical purposes it is necessary to define a fixed location for the observer, which is implicit within the assumptions contained in the calculation method. It is also accepted that, for analytical purposes, calculations of comfort conditions throughout the space may need to be performed. A method has been suggested by Fanger[7] and a practical application has been demonstrated by Holmes and Connor[8].

For naturally or mechanically ventilated buildings, providing minimum fresh air requirements without humidification in cold dry winter conditions could mean that the relative humidity (RH) in the room may fall to a level that could have a detrimental effect on health[9], see chapter 1: *Environmental criteria for design*.

5.4.2 Overheating criteria

Where mechanical cooling is not provided, temperatures during hot periods of the year will usually exceed those deemed as comfortable for mechanically cooled spaces. While people will often accept these higher temperatures (see chapter 1), it is necessary to make an assessment of what conditions will be like on hot days. Recommended overheating criteria are considered in chapter 1, section 1.4.2, and CIBSE AM10: *Natural ventilation in non-domestic buildings*[10] provides some information on criteria used in other counties. Furthermore, what is acceptable will vary with the use of the space and employer. For example if working conditions are very flexible then it may be acceptable for people to vary their working hours to avoid times of high temperatures. For schools, the Department of Education and Skills requires that the space temperature does not exceed 28 °C more than 80 occupied hours in the year[11]. In a hospital ward where people are bed-bound conditions may be more critical. It is important that these issues are discussed with the client.

5.4.3 External design conditions

The parameters required will depend upon the application. For example only dry-bulb data will be required for the sizing of heat emitters whereas for an air conditioning system the following are necessary.

Some or all of the following external design criteria are required in order to determine the plant capacity:

- winter dry and wet bulb temperatures
- summer dry and wet bulb temperatures

— solar irradiation
— longwave radiation loss
— solair temperatures
— wind speed and direction.

Summary data for the UK and elsewhere are given in chapter 2: *External design data*. More extensive data are contained in CIBSE Guide J: *Weather, solar and illuminance data*[12].

While heat loss and cooling load calculations and peak summertime temperatures are based on specific design conditions, energy and overheating predictions, depending upon the selected calculation method, usually require either a complete year of hourly weather data, or a statistically reduced version (for example degree days).

Climate change may require that designers will need to 'future-proof' their buildings. Guide J provides some design data for future climates and chapter 2 of Guide A contains a subset of that information. At present, data sets for 'future weather years' are not available from CIBSE. Climate change issues are also addressed in CIBSE TM34: *Weather data with climate change scenarios*[13] and TM36: *Climate change and the internal environment*[14].

Various sets of climatic data are available which contain hourly values of the relevant parameters for a complete year. CIBSE Guide J[12] contains a detailed description of the types of data available. The CIBSE has approved the following data sets:

— *CIBSE Example Weather Years* (EWYS)[15,16]: are complete years selected to be representative of long-term averages. They were intended for the purpose of energy consumption calculations. They are not suitable for design or overheating risk assessment. They have now been superseded by the CIBSE Test Reference Years (TRYS).
— *CIBSE Test Reference Years* (TRYS): have been selected as good statistical representations of the past. Unlike the EWYS, the TRYS are built-up of months from different years. They are intended for general simulation purposes and the calculation of energy consumption. They are not suitable for design or overheating risk assessment.
— *CIBSE Design Summer Year* (DSYS): are a complete year for which the average temperature of the summer months is at the centre of the upper quartile of rankings obtained from about 20 individual years. This means that there is approximately a 1-in-8 chance of the temperatures on the DSY being exceeded in any one year. The DSY is intended to be used in the assessment of overheating risk.

5.4.3.1 Winter heating design criteria

External design dry bulb temperatures for the UK and sites worldwide are given in chapter 2. Where mechanical ventilation and/or air conditioning are used the fresh air plants should be sized using a lower temperature than that for the fabric heat loss calculation. This is because the air plant responds virtually instantaneously, whereas fabric storage has an averaging effect. The selection of a suitable design temperature will depend on the pattern of occupation for the building and the acceptable design risk, see chapter 2, section 2.3).

5.4.3.2 Summer design criteria

The maximum demand of an air conditioning system will depend on peak coincident zone loads. However, depending on the type of system, the peak external design dry and wet bulb temperatures will also have a significant influence on the installed plant capacity. External design dry bulb and wet bulb temperatures for the UK and sites worldwide are given in chapter 2. Design solar irradiances for a number of sites in the UK can also be found in chapter 2. Predicted solar irradiances for latitudes from 0–60° N/S can be found on the CD-ROM that accompanies this Guide.

5.4.4 Infiltration and ventilation

Design ventilation rates depend upon the system. For systems employing mechanical ventilation minimum outside air quantities are given in chapter 1. Empirical values for air infiltration in naturally ventilated buildings in winter are given in chapter 4, Tables 4.13 to 4.21.

For the calculation of peak temperatures in summer where natural ventilation is to be used a maximum of 10 air changes per hour is not unreasonable, although this figure will depend upon the ventilation system, see Table 5.2). It is the responsibility of the designer to ensure that there is adequate provision for natural ventilation. Natural ventilation is considered in chapter 4 and CIBSE AM10: *Natural ventilation on non-domestic buildings*[10].

5.4.5 Internal gains

These arise from the heat generated by the occupants, lighting and machines etc. used within the space. Depending upon the source sensible heat gains will be both convective and radiant in nature. Latent gains are mainly due to occupants; however there are some spaces such as swimming pools and kitchens where latent gains are more significant. In addition to being dependent upon the purpose of the building they may depend upon

Table 5.2 Effective mean ventilation rates for openable windows

Location of openable windows	Usage of windows		Effective mean ventilation rate	
	Day	Night	Air changes per hour / h^{-1}	Ventilation loss / $W \cdot m^{-2} \cdot K^{-1}$
One side of building only	Closed	Closed	1	0.3
	Open	Closed	3	1.0
	Open	Open	10	3.3
More than one side of building	Closed	Closed	2	0.6
	Open	Closed	10	3.3
	Open	Open	30	10.0

patterns of use. Diversity should therefore be considered, however this will need to be agreed with the client. Section A6 provides information on the magnitude and nature of many sources of internal gain.

5.4.6 Building fabric (properties of materials)

The properties of the materials used in the construction of a building are key in determining the way the building responds to the climate and systems within the building. Two types of properties are considered here; fundamental, that is parameters related to the material and derived, the parameters used by the designer.

The principal fundamental properties are as follows:

— density
— specific heat capacity
— thermal conductivity and resistance
— vapour resistivity
— absorptivity
— emissivity
— solar transmittance
— solar absorptance
— solar reflectance
— light transmittance.

The principal derived properties referred to in this section are:

— thermal transmittance (*U*-values)
— thermal admittance (*Y*-values)
— decrement factor
— surface factor
— solar gain factor for transparent materials.

Typical values of the above properties for a range of opaque fabrics and building structures are given in chapter 3: *Thermal properties of building structures*, along with a description of the calculation procedure for thermal transmittance and admittance.

The use of standard thermal properties, in particular the overall conductance of a material (*U*-value) may be related to a specific calculation method. The *U*-values given in chapter 3 are derived on the basis of an internal surface resistance to heat flow that is consistent with the theory for steady state heat transfer, see Appendix 5.A3 (standard external coefficients are also used). While such values are accepted as standard for the purposes of Building Regulations[17] in England and Wales, care should be taken when using other calculation methods to ensure that the surface heat transfer coefficients are appropriate to the method.

Solar gain factors for generic glass and blind combinations are given in 5.7.3, Table 5.7. The method used to calculate the tabulated solar gain factors is described in Appendix 5.A7. This appendix also gives the values of the transmission (*T*), absorption (*A*) and radiation (*R*) components (for thermal shortwave radiation) and emissivities (for thermal longwave radiation) used in calculating the solar gain factors.

5.5 Calculation methods

This section describes methods for calculating:

— steady state heat loss from spaces
— temperatures within spaces on hot days (summertime temperatures)
— design cooling loads.

In many cases the calculation method will depend upon the stage of the design. A simple 'rule of thumb'[18] is often sufficient to determine approximate plant room space requirements and sizes at the concept stage. Detailed design may range from simple steady state heat loss calculation to a coupled airflow and dynamic transient method. The selection of an appropriate calculation method should form part of the project quality plan (see Appendix 5.A1).

5.5.1 Overview of thermal models

For most applications the calculation of the size of space heating emitters makes use of a steady state analysis. The reasons for this are as follows.

The system is normally switched on when no loads other than those imposed by the external climate are active. Consequently the external design condition is simply a dry bulb temperature. Traditionally the design temperature is a 24- or 48-hour average. In most applications intermittent heating is used. If the building fabric did not absorb heat a steady state calculation would also be an accurate design calculation under the assumptions of constant dry-bulb and infiltration. Real buildings store heat; the amount of heat absorbed during pre-heat can be allowed for by means of a simple multiplying factor (see 5.10.3.3). An alternative approach would be to use dynamic simulation[6].

Peak space cooling loads usually occur during the occupied period. Therefore it is necessary to take account of both solar and internal gains. Radiant gains can only become a load upon the system by heating the building fabric. Fabric storage means that the gain is attenuated and shifted in time making it essential to use a calculation method that can take account of the dynamic response of the building.

Similarly space temperatures during periods of occupancy are the result of interactions between gains, building fabric and even occupant behaviour (effect on ventilation for example) are taken into consideration.

Summertime temperatures and cooling loads cannot be determined by steady state methods because it is necessary to take account of the time delays associated with the storage of heat within the building fabric. The most effective way to carry out these calculations is to make use of a computer program, although the CIBSE admittance method can be used in hand calculations and provides a way to check the output from computer programs.

The admittance procedure[19] is the simplest of the dynamic methods available. The procedure assumes that all internal and external load fluctuations can be represented by the sum of a steady state component and a sine wave with a period of 24 hours. It is therefore a 'cyclic model' that is implicit within this assumption is that steady cyclic conditions are achieved; i.e. a single day repeated for subsequent days until all long-term transients have died away. The method does not represent the effects of rapid load changes nor long-term storage. Therefore it is not suitable as a means of calculating the performance of buildings with a large thermal capacity or the effects of rapid changes in load. Nonetheless it is considered suitable for use at an early stage of design and as a means of predicting the limiting state. There is also some evidence that, for naturally ventilated buildings, the peak temperatures predicted by the admittance procedure are close to those, which actually occur[20]. For theses reasons the admittance method (or CIBSE cyclic model) is currently retained. However, it must be used with care because effects such as weekend shut downs cannot be taken into account. Alternative simple methods are available and Appendix 5.A2 provides some information.

Simulation, which uses a detailed numerical model of a building, offers the possibility of overcoming the deficiencies of cyclic models because it is possible to predict the response of a building to real sequences of weather data and loads. This type of model is called a 'transient model' and can only, be implemented on a computer.

Recognising that most engineering calculations (including steady state, cyclic and transient methods) are now carried out by means of computer programs, the CIBSE has produced a set of validation tests[2]. The initial set of tests is limited and will be extended in due course. It is also recognised that user error is a significant issue and ways to minimise such errors are presented in Appendix 5.A1.

5.6 Steady state models

The CIBSE recognises that different levels of detail may be required for different building types. A very simple approach will usually be sufficient where large temperature differences between surfaces are not anticipated, that is in most conventional buildings. Where accurate surface temperatures are required (e.g. comfort calculations, condensation risk and boundary conditions for computational fluid dynamics) then a more complex method will be necessary. Three methods are described in Appendix 5.A3. In addition to describing the application of the general purpose (simple) method, this section presents an 'approximate model' that is valid under some circumstances and provides a way of checking more detailed models. The accuracy of the simple method is compared with the more detailed approaches in Appendix 5.A4.

5.6.1 The approximate model

Prior to the publication of the 1999 edition of Guide A, the environmental temperature was used as the steady state design temperature for heating. This hypothetical temperature is not a comfort temperature but one that is a good approximation to the temperature that drives heat through the building fabric, see also Appendix 5.A3.5. For well insulated buildings, without large areas of glazing and low air change rates there is usually very little difference between the environmental temperature, operative temperature and air temperature.

In this model, the total heat loss is the sum of the fabric and ventilation losses, i.e:

$$\Phi_t = [\Sigma\,(A\,U) + C_v]\,(\theta_{ei} - \theta_{ao}) \tag{5.8}$$

where Φ_t is the total heat loss (W), $\Sigma\,(A\,U)$ is sum of the products of surface area and corresponding thermal transmittance over surfaces through which heat flow occurs ($W{\cdot}K^{-1}$), C_v is the ventilation conductance ($W{\cdot}K^{-1}$), θ_{ei} is the environmental temperature and θ_{ao} is the outside air temperature (°C).

The ventilation conductance is given by:

$$C_v = {}^1\!/_3\,N\,V \tag{5.9}$$

Where N is the number of room air changes for air entering the space at the outside air temperature (h^{-1}) and V is the room volume (m^3).

5.6.2 CIBSE Simple Model

This model enables the designer to size emitters to achieve a specified operative temperature.

The total heat loss is the sum of the fabric and ventilation losses, i.e.:

$$\Phi_t = [F_{1cu}\,\Sigma\,(A\,U) + F_{2cu}\,C_v]\,(\theta_c - \theta_{ao}) \tag{5.10}$$

where Φ_t is the total heat loss (W), F_{1cu} and F_{2cu} are factors related to characteristics of the heat source with respect to the operative temperature, $\Sigma\,(A\,U)$ is sum of the products of surface area and corresponding thermal transmittance over surfaces through which heat flow occurs ($W{\cdot}K^{-1}$), C_v is the ventilation conductance ($W{\cdot}K^{-1}$), θ_c is the operative temperature at centre of room (°C) and θ_{ao} is the outside air temperature (°C).

Where the fabric loss term contains heat loss through internal partitions, a modified U-value should be used:

$$U' = \frac{U\,(\theta_c - \theta_c')}{(\theta_c - \theta_{ao})} \tag{5.11}$$

where U' is the thermal transmittance modified for heat loss through internal partitions ($W{\cdot}m^{-2}{\cdot}K^{-1}$), and θ_c' is the operative temperature on the opposite side of partition through which heat flow occurs (°C).

F_{1cu} and F_{2cu} are calculated as follows (see Appendix 5.A5, equations 5.174 and 5.175):

$$F_{1cu} = \frac{3\,(C_v + 6\,\Sigma\,A)}{\Sigma\,(A\,U) + 18\,\Sigma\,A + 1.5\,R\,[3\,C_v - \Sigma\,(A\,U)]} \tag{5.12}$$

Table 5.4 Typical proportions of radiant (R) and convective heat from heat emitters

Emitter type	Proportion of emitted radiation	
	Convective	Radiative (R)
Forced warm air heaters	1.0	0
Natural convectors and convector radiators	0.9	0.1
Multi column radiators	0.8	0.2
Double and treble panel radiators, double column radiators	0.7	0.3
Single column radiators, floor warming systems, block storage heaters	0.5	0.5
Vertical and ceiling panel heaters	0.33	0.67
High temperature radiant systems	0.1	0.9

$$F_{2cu} = \frac{\Sigma\,(A\,U) + 18\,\Sigma\,A}{\Sigma\,(A\,U) + 18\,\Sigma\,A + 1.5\,R\,[3\,C_v - \Sigma\,(A\,U)]} \qquad (5.13)$$

where R is the radiant fraction of the heat source, A is the total area through which heat flow occurs (m^2) and $\Sigma\,(A\,U)$ is the sum of the products of surface area and corresponding thermal transmittance over surfaces through which heat flow occurs (W·m^{-2}·K^{-1}). Typical values for R are given in Table 5.4.

However, in many cases the values of F_{1cu} and F_{2cu} are very close to unity, and the calculations can be done assuming this with little loss of accuracy. In this case equation 5.10 reduces to:

$$\Phi_t = [\Sigma\,(A\,U) + C_v]\,(\theta_c - \theta_{ao}) \qquad (5.14)$$

Note this is the same as equation 5.8 except that θ_c is used for internal temperature instead of θ_{ei}

The corresponding air and mean surface temperatures are given by:

$$\theta_{ai} = \frac{\Phi_t\,(1 - 1.5\,R) + C_v\,\theta_{ao} + 6\,\Sigma\,A\,\theta_c}{C_v + 6\,\Sigma\,A} \qquad (5.15)$$

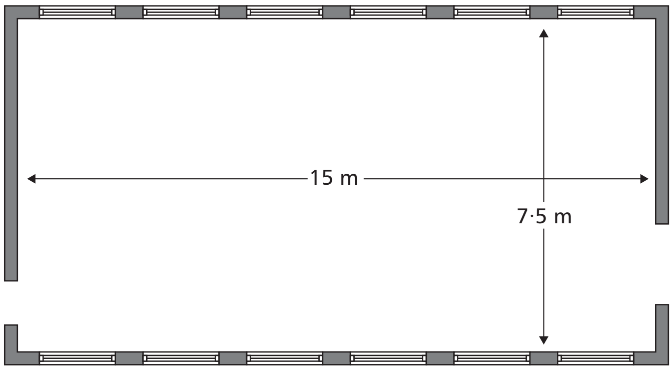
$$\theta_m = 2\,\theta_c - \theta_{ai} \qquad (5.16)$$

where θ_{ai} is the inside air temperature (°C), θ_m is the mean surface temperature (°C), θ_{ao} is the outside air temperature (°C) and θ_c is the operative temperature at the centre of the space (°C).

Note that the constants contained in the above equations assume standard heat transfer coefficients and emissivity values, i.e. h_c = 3 W·m^{-2}·K^{-1}, h_r = 5.7 W·m^{-2}·K^{-1}, ε = 0.9.

Example 5.1: Calculation of steady state design heat losses using the Simple Model

A small factory, see Figure 5.3, is to be heated to an operative temperature of 19 °C. The site is subject to 'normal' conditions of exposure. Surface areas and the corresponding U-values are given in Table 5.5. A ventilation rate of 0.5 air changes per hour is assumed. The external design temperature is –1 °C. It is required to determine the total heat loss and the inside air and mean radiant temperatures under steady state design conditions when heated by (a) forced circulation warm air heaters and (b) high temperature radiant strip heaters.



Height = 5 m; window area = 4 m² (each); door area = 6 m² (total)

Figure 5.3 Example 5.1: Small factory building

Table 5.5 Example 5.1: surface areas and U-values

Surface	Area, A / m^2	U-value / W·m^{-2}·K^{-1}	$(A \times U)$ / W·K^{-1}
Floor	112.5	0.45	50.6
Roof	112.5	0.3	33.8
External walls	171.0	0.5	85.5
Glazing	48.0	3.3	158.4
Doors	6.0	2.9	17.4
	$\Sigma\,A$ = 450.0		$\Sigma\,(A\,U)$ = 345.7

(a) *Heating by forced circulation warm air heaters*

Step 1: ventilation conductance (C_v)

Volume of building (V) = 562.5 m^3 and ventilation rate (N) = 0.5 h^{-1}, therefore using equation 5.9:

$$C_v = (0.5 \times 562.5) / 3 = 93.75 \text{ W·K}^{-1}$$

Step 2: factors related to characteristics of heat source (F_{1cu}, F_{2cu})

Assume F_{1cu}, F_{2cu} are both equal to 1.

Step 3: total heat loss (Φ_t)

From equation 5.10:

$$\Phi_t = [345.7 + 93.75]\,(19 + 1) = 8.789 \text{ kW}$$

Step 4: inside air temperature (θ_{ai})

From equation 5.15:

$$\theta_{ai} = \frac{(8.79 \times 10^3) + [93.75 \times (-1)] + (6 \times 450 \times 19)}{93.75 + (6 \times 450)}$$

$$= 21.48 \text{ °C}$$

Step 5: mean surface temperature (θ_m)

From equation 5.16:

$$\theta_m = (2 \times 19) - 21.48 = 16.52 \text{ °C}$$

(*b*) *Heating by radiant strips*

Step 1: ventilation conductance (C_v)

As for (*a*) above.

Step 2: factors related to characteristics of heat source (F_{1cu}, F_{2cu})
Assume F_{1cu}, F_{2cu} are both equal to 1.

Step 3: total heat loss (Φ_t)

As for (*a*) above, i.e:

$$\Phi_t = 8.789 \text{ kW}$$

Step 4: inside air temperature (θ_{ai})

From equation 5.15:

$$\theta_{ai} = \{(8.79 \times 10^3)\,[1 - (1.5 \times 0.9)] + [93.75 \times (-1)] + (6 \times 450 \times 19)\}\,/\,[93.75 + (6 \times 450)]$$
$$= 17.23\ °\text{C}$$

Step 5: mean surface temperature (θ_m)

From equation 5.16:

$$\theta_m = (2 \times 19) - 17.23 = 20.77\ °\text{C}$$

Example 5.1 illustrates how the method may be used to compare different types of heating system and to determine whether they can achieve the design comfort criteria. For example, a large difference between the air and radiant temperatures may lead to discomfort.

(Using equations 5.12 and 5.13 gives values of F_{1cu} = 0.996 and F_{2cu} = 0.984. The temperature values are then θ_{ai} = 20.35 °C and θ_m = 17.65 °C. Hence assuming values of 1 for F_{1cu} and F_{2cu} makes a difference of only 0.01 K in this example. The corresponding heat loss is 5.416 kW, a difference of only 0.046 kW or 0.8%).

Note that the errors resulting from assuming F_{1cu} and F_{2cu} are equal to 1 increase slowly as *U*-values increase, and increase rapidly as air change rate increases, and can become very large. In these cases, equations 5.12 and 5.13 should be used to calculate F_{1cu} and F_{2cu}.

5.7 Dynamic models

The model described here is intended for assessing the effects of sensible heat gains. In some climates moisture absorbed within the building fabric and its contents can significantly affect the loads on the system. Whilst there have been some attempts to derive simple models of the processes involved[21], this is a complex issue[22–25] and as yet there is no consensus on any particular analytical method.

The CIBSE recognises two levels of dynamic modelling:

— cyclic
— transient.

In a cyclic model the analysis is carried out using a sequence of identical days for which the external conditions vary on a 24-hour cyclical basis of which the CIBSE admittance method is a good example. In a transient model the state of the building and its components changes with time during the period of interest and do not go through a repeated cycle. Transient models can provide a realistic simulation of the performance of a building. Cyclic models are only suitable for applications that require an assessment of conditions after a long period of identical days, for example, the calculation of design sensible cooling load. Provided with appropriate climatic data a transient model can be used to carry out cyclic calculations.

The CIBSE does not recommend a particular transient model, however those models that meet the requirements of the CIBSE verification tests[2] may be used with more confidence than models that do not. Similarly, computer programs based upon the CIBSE admittance method require testing.

This section provides a description of the CIBSE admittance (cyclic) method. The selection of transient models is given in CIBSE Applications Manual AM11: *Building energy and environmental modelling*[6]. Appendix 5.A9 describes some of the features that are likely to be included in a detailed transient model.

This cyclic model can be used to make a rapid assessment of:

— peak summertime temperatures
— space cooling loads
— preheat requirement.

Because of the simplicity of the model, its application must be treated with care. This is particularly so when predicting summertime overheating. Steady cyclic analysis predicts a limit state, which under some circumstances may never be reached. For example many buildings take at least a week to achieve this condition. If the building is unoccupied at the weekend the predicted peak may not occur. This model cannot predict this type of event and so the true benefit of mass cannot be assessed.

The prediction of overheating risk may require the engineer to demonstrate that certain temperatures are not exceeded for more that a specified period of time. The only way the cyclic model can do this is by careful choice of climatic data. One approach is by the use of banded climatic data, such as those presented in chapter 2 of the 1986 edition of Guide A. Use of a fully transient model with an appropriate sequence of climatic data (i.e. a design summer year (DSY), see chapter 2 and CIBSE Guide J[12]) is a more satisfactory method to achieve this end.

This cyclic model is best suited to the calculation of space cooling loads. The CIBSE method has been compared with ASHRAE methods and shown to be comparable[26] Appendix 5.A6 describes in detail an algorithm for the calculation of design cooling loads using the CIBSE cyclic model and contains an example showing each stage of the calculation. Peak cooling loads are in good agreement with those given in CIBSE TM33[2]. There are, however, limitations as to the suitability of the method. In particular problems occur with:

— *high proportions of convective gain*: the cooling load is over estimated

— *chilled or heated surfaces*: these can only be represented by specifying a particular radiant convective split for the plant output

— *displacement ventilation*: only a well-mixed zone can be considered (not all transient models will be capable of handling this situation).

Accepting the limitations of the model it has value as a means of carrying out quick checks and the design of simple installations and because it can be used manually provides a better insight into thermal response than 'black box' computer methods. The Simple (cyclic) Model is based on the following assumptions:

— All parameters associated with thermal storage can be represented by the response to a sine wave with a period of 24 hours.

— Heat interchange between room surfaces follows the heat transfer assumptions given for the Simple (steady state) Model, see Appendix 5.A7.

— A uniform distribution of transmitted shortwave solar radiation over room surfaces.

The background to the admittance method is described in detail by Milbank and Harrington-Lynn(19) and the method is given further justification by Davies(27). The theoretical basis for extensions to the admittance method is covered elsewhere(28,29).

The parameters used by the model to characterise the performance of a space are as follows:

— thermal response

— admittance, decrement factor

— surface factor

— solar gain factor.

In the admittance method the response of a space is the sum of two components: a daily mean value and a cyclic value (the difference between the instantaneous value and the mean value; often called the 'swing'). The admittance and decrement factors relate to the cyclic response of building fabric, the mean response is characterised by the *U*-value. The surface factor is used to quantify the absorption and subsequent release (as longwave and convective gains) of the cyclic component of transmitted solar radiation. Solar gain factors combine the response of the space to shortwave radiation with the transmission and absorption characteristics of the glazing. They include the effect of the surface factor.

These parameters are considered in the following sections, along with the equations necessary for the calculation of space temperatures and cooling loads. The derivation of admittance and its related factors is given in chapter 3: *Thermal properties of building structures*, Appendix 3.A6.

This cyclic model may be implemented as either a manual calculation or a computer program. The manual calculation is a further simplification of the method involving the use of constants to represent the time-varying processes. The description of the application of the method given in the following sections is for the manual calculation method. Computer-based versions will follow the same general procedure but with adjustments, as follows:

— actual, rather than standardised, values for the admittance and surface factor of each surface, and their associated time delays

— solar transmission and absorption, and shading, calculated for each hour of the day (see Appendix 5.A7)

— internal gains divided into radiant and convective components

— calculation of air change rates.

Appendix 5.A6 contains an algorithm for the calculation of the design sensible cooling load using the admittance method.

5.7.1 Thermal response ('thermal weight')

Conventionally, buildings are classified as having either a slow or a fast response to heat transfer. The response of a space to thermal input depends upon:

— type of thermal input

— surface finishes

— thermal properties of the construction

— thickness of the construction

— furnishings within the space.

The heat input to the surfaces will be in the form of either shortwave radiation (solar radiation and energy from electric lights) or a combination of longwave radiation (from surfaces and other emitters) and convective exchange with the air. The way in which a surface responds to shortwave radiation depends on the shortwave absorption coefficient, the surface heat transfer coefficient and the thermal properties of the structure. The physical process involved is that shortwave radiation is absorbed at the surface and, after a delay due to thermal storage, causes the temperature of that surface to rise. Heat is then transferred between surfaces by longwave radiation and to the room air by convection. The effect is to raise the internal heat transfer (i.e. environmental) temperature.

The response of a space to changes in environmental temperature is characterised by the admittance of the surfaces, which is a function of the longwave emissivity, the surface heat transfer coefficient and the thermal properties of the structure.

There are two time delays associated with the thermal response of the space, one which applies to shortwave radiation and the other due to surface-to-surface and surface-to-air heat exchanges. Thus, it is possible for a space to be 'lightweight' in terms of its response to solar radiation but 'heavyweight' in terms of the change in temperature arising from other sources of heat input.

For the Simple (cyclic) Model, the definition of thermal response to shortwave radiation is as follows:

— *fast response*: surface factor = 0.8 with a delay of 1 hour

— *slow response*: surface factor = 0.5 with a delay of 2 hours

Response to the changes in the environmental temperature is characterised by the response factor, f_r, given by:

$$f_r = \frac{\Sigma\,(A\ Y) + C_v}{\Sigma\,(A\ U) + C_v} \tag{5.17}$$

where f_r is the response factor, $\Sigma\,(A\ Y)$ is the sum of the products of surface areas and their corresponding thermal admittances ($W \cdot K^{-1}$), $\Sigma\,(A\ U)$ is the sum of the products of surface area and corresponding thermal transmittance over surfaces through which heat flow occurs ($W \cdot K^{-1}$) and C_v is the ventilation conductance ($W \cdot K^{-1}$), see equation 5.9.

Structures with a high thermal response factor (> 4) are referred to as slow response buildings, i.e. 'heavyweight' structures, and those with a low thermal response factor (≤ 4) as fast response buildings, i.e. 'lightweight' structures. Note that these definitions are only approximations to the thermal behaviour of actual buildings. Since the thermal response factor partly depends on the ventilation rate, a building having a nominally slow thermal response may, if the rate of ventilation is high, have a low thermal response factor. Table 5.6 gives nominal building classifications and corresponding response factors. These classifications should be regarded as giving only a general indication of expected performance.

The response factor as defined in equation 5.17 should not be confused with that used (mainly in the USA) to describe the dynamic characteristics of walls.

5.7.2 Parameters associated with the response of surfaces

Values of the following parameters for a wide range of constructions are given in chapter 3: *Thermal properties of building structures*.

5.7.2.1 Thermal admittance (*Y*-value)

The most significant parameter is the admittance. This is the rate of flow of heat between the internal surfaces of the structure and the environmental temperature in the space, for each degree of deviation of that temperature about its mean value. The associated time dependency takes the form of a time lead denoted by ω.

For thin structures of low thermal capacity, the admittance is equal in amplitude to the *U*-value and has a time lead of zero. In the case of an exciting frequency with a period of 24 hours the amplitude tends towards a limiting value for thicknesses greater than about 100 mm.

For multi-layered structures, the admittance is primarily determined by the characteristics of the materials in the layers nearest to the internal surface. For example, the admittance of a heavy concrete slab construction lined internally with insulation will be close to the value for the insulation alone, whereas placing the insulation within the construction, or on the outside surface, will result in an admittance that differs little from that for the uninsulated slab.

5.7.2.2 Decrement factor (*f*)

The decrement factor is the ratio of the rate of heat flow through the structure, due to variations in the external heat transfer temperature from its mean value with the environmental temperature held constant, to the steady state conduction. The associated time dependency takes the form of a time lag denoted by ϕ.

For thin structures of low thermal capacity, the amplitude of the decrement factor is unity and the time lag zero. The amplitude decreases and the time lag increases with increasing thermal capacity.

5.7.2.3 Surface factor (*F*)

The surface factor is the ratio of the variation of radiant heat flow (from shortwave sources) about its mean value readmitted to the space from the surface, to the variation of heat flow about its mean value incident upon the surface. The associated time dependency takes the form of a time lag denoted by ψ.

The amplitude of the surface factor decreases and its time lag increases with increasing thermal conductivity but both are virtually constant with thickness.

For the Simple (cyclic) Model, the only shortwave gain considered is that from the sun; gains from other sources are treated as longwave gains.

5.7.3 Parameters associated with solar gain

The thermal load on a space due to solar irradiation depends upon the radiation transmitted by and absorbed within the glazing system. The amount of radiation transmitted and absorbed is a function of the intensity of the incident radiation and the angle of incidence between

Table 5.6 Thermal response

Thermal response	Typical features of construction	Response factor, f_r	Response to short-wave radiation		Time lead for admittance, ω / h
			Average surface factor, F	Time delay, ϕ / h	
Slow	Masonry external walls and Internal partitions, bare solid floors and ceilings	> 4	0.5	2	1
Fast	Lightweight external cladding, de-mountable partitions, suspended ceilings, solid floors with carpet or wood block finish or suspended floors	≤ 4	0.8	1	0

the solar beam and the glazing. Therefore, for an accurate determination of this load, it is necessary to calculate the gain occurring at various times of the day. A computer-based method is required to achieve this cost effectively. However, if the objective is to determine either the maximum space temperature or the maximum cooling load, the transmission and absorption characteristics calculated at the time of peak solar irradiation are often the most significant. The Simple (cyclic) Model is based on these assumptions.

The response of a space to solar radiation transmitted and absorbed in the glazing system is characterised by two parameters, see Appendix 5.A7:

— mean solar gain factor
— alternating solar gain factor.

These are further divided into factors relating the gain to the environmental node (causing an increase in environmental temperature) and to the room air. The latter is used only where internal blinds are fitted. This is because increased convection from blinds significantly changes the proportions of longwave and convective heat from the surface. The factors are defined as follows:

$$\bar{S}_a = \frac{\text{Mean solar gain at air node per m}^2\text{ of glazing}}{\text{Mean solar intensity incident on glazed façade}} \tag{5.18}$$

$$\bar{S}_e = \frac{\text{Mean solar gain at environmental node per m}^2\text{ of glazing}}{\text{Mean solar intensity incident on glazed façade}} \tag{5.19}$$

$$\tilde{S}_a = \frac{\text{Instantaneous cyclic solar gain at air node per m}^2\text{ of glazing}}{\text{Instantaneous cyclic solar intensity incident on glazed façade}} \tag{5.20}$$

$$\tilde{S}_e = \frac{\text{Instantaneous cyclic solar gain at environmental node per m}^2\text{ of glazing}}{\text{Instantaneous cyclic solar intensity incident on glazed façade}} \tag{5.21}$$

In the absence of shading devices, the alternating gain usually lags the solar intensity by between zero and two hours, the duration of the lag depending on the surface factors for the internal surfaces. High surface factors (e.g. 0.8) give rise to delays of about one hour, low surface factors (e.g. 0.5) give rise to delays of about two hours.

Typical values of the above factors for various glazing configurations are given in Table 5.7. The transmission (T), absorption (A) and radiation (I) components (for thermal shortwave radiation) and emissivities (for thermal longwave radiation) for generic glass and blind types used in calculating the solar gain factors are given in Appendix 5.A7, Table 5.51.

Glazing manufacturers do not usually provide values of these solar factors for their products. More commonly, one or more of the following are provided:

— properties at normal incidence (see Appendix 5.A7)
— shading coefficients
— total solar energy transmittance (‘G-value’).

Only the first of these can be used to calculate solar gain factors. However, shading coefficients and G-values can be derived from fundamental properties and so it is possible to obtain the corresponding solar gain factors by a process of iteration. Shading coefficients and G-values are defined in 5.2.2.

Shading coefficients are calculated as follows:

$$S_c = \frac{\text{Solar gain through subject glass and blind at direct normal incidence}}{\text{Solar gain through reference glass at direct normal incidence}} \tag{5.22}$$

where the solar gain through the reference glass at normal incidence is 0.87.

Solar gain in this case can be considered to mean the shortwave or the longwave component, or the total of both components. Therefore, the shading coefficient can take three forms, as follows:

— *shortwave shading coefficient*: the solar direct transmittance divided by 0.87
— *longwave shading coefficient*: the fraction of the solar absorptance that is re-radiated and contributes to the total transmittance, divided by 0.87
— *total shading coefficient*: the solar transmittance divided by 0.87.

Values of shading coefficients for shortwave and longwave radiation are given in Table 5.7 (page 5-16). (*Note*: these values have been calculated using ‘traditional CIBSE surface coefficients and may there differ slightly from calculations made using coefficients calculated according to the relevant BS EN ISO standard.) When values of shading coefficient are quoted it is important to know the basis on which they were calculated, see Appendix 5.A7.4.

5.8 CIBSE cyclic model

5.8.1 Summertime temperatures

This section describes the application of the Simple (cyclic) Model to the calculation of peak temperatures on hot days. The principles of the method and the relevant equations are given below, along with a numerical example.

The data required are as follows:

— climatic data: see chapter 2: *External design data* and CIBSE Guide J: *Weather, solar and illuminance data*[12]
— surface areas of internal and external structural elements
— construction details of internal and external structural elements

Table 5.7 Solar gain factors and shading coefficients for generic glazing/blind combinations

Description (inside to outside)	Solar gain factor at environmental node†			Solar gain factor at air node		Shading coefficient, S_c	
	$\bar{S}_e$	$\tilde{S}_{el}$	$\tilde{S}_{eh}$	$\bar{S}_a$	$\tilde{S}_a$	Shortwave	Longwave
Single glazing/blind combinations:							
— clear glass	0.76	0.66	0.50	—	—	0.91	0.05
— absorbing glass	0.61	0.54	0.44	—	—	0.53	0.19
— absorbing slats/clear	0.43	0.44	0.44	0.17	0.18	—	—
— reflecting slats/clear	0.35	0.32	0.31	0.12	0.12	—	—
— 'generic' blind/clear	0.34	0.33	0.29	0.11	0.11	—	—
Double glazing/blind combinations:							
— clear/clear	0.62	0.56	0.44	—	—	0.70	0.12
— clear/reflecting	0.36	0.32	0.26	—	—	0.37	0.08
— low emissivity/clear	0.62	0.57	0.46	—	—	0.62	0.18
— low emissivity/absorbing	0.43	0.38	0.32	—	—	0.36	0.15
— low emissivity/clear/'generic' blind	0.15	0.14	0.11	—	—	—	—
— absorbing slats/clear/clear	0.34	0.36	0.37	0.18	0.21	—	—
— absorbing slats/clear/reflecting	0.19	0.19	0.19	0.12	0.13	—	—
— absorbing slats/low emissivity/clear	0.33	0.35	0.35	0.21	0.23	—	—
— absorbing slats/low emissivity/absorbing	0.22	0.22	0.22	0.16	0.17	—	—
— reflecting slats/clear/clear	0.28	0.29	0.26	0.15	0.16	—	—
— reflecting slats/clear/reflecting	0.17	0.16	0.16	0.10	0.10	—	—
— reflecting slats/low emissivity/clear	0.28	0.27	0.26	0.18	0.20	—	—
— reflecting slats/low emissivity/absorbing	0.18	0.17	0.17	0.14	0.15	—	—
— 'generic' blind/low emissivity/clear	0.29	0.29	0.27	0.17	0.18	—	—
Triple glazing:							
— clear/clear/clear	0.52	0.49	0.40	—	—	0.55	0.17
— clear/clear/absorbing	0.37	0.35	0.29	—	—	0.33	0.15
— clear/clear/reflecting	0.30	0.28	0.23	—	—	0.30	0.09
— clear/low emissivity/clear	0.53	0.50	0.42	—	—	0.50	0.21

† For $\tilde{S}_e$, subscripts 'l' and 'h' denote thermally 'lightweight' and 'heavyweight' buildings, respectively

Note: shading coefficients for windows with slatted blind or windows with inner blind are not given since these not compatible with the properties of plain glass

— thermal transmittances (U-values) of internal and external structural elements: see chapter 3: *Thermal properties of building structures*

— thermal admittances (Y-values) of internal and external structural elements: see chapter 3: *Thermal properties of building structures*

— surface areas of glazing

— solar gain factors

— shading details

— internal heat gains due to occupants, electric lighting, it and other sources: see chapter 6: *Internal heat gains*

— ventilation rate and profile: empirical values for naturally ventilated buildings are given chapter 4, Tables 4.13 to 4.21.

Table 5.8 shows how the various sources of heat gain contribute to the internal temperature.

Using the Simple (cyclic) Model, the following need to be determined in order to calculate the peak internal space temperature:

— mean heat gains from all sources

— mean internal operative temperature

— swing (deviation), mean-to-peak, in heat gains from all sources

— swing (deviation), mean-to-peak, in operative temperature.

These calculations are described in the following sections and illustrated by means of a numerical example. For complex situations (e.g. the ventilation rate varies over 24 hours or there is shading) manual calculations are unlikely to be practicable. This is because it is necessary to determine, for example, the shaded area for each hour of the day so that the mean can be established.

5.8.1.1 Mean heat gains

Solar heat gains

Solar gains through glazing consist of solar radiation, which is absorbed in the glazing and transmitted to the environmental node and also the transmitted solar radiation, which is absorbed at the internal surfaces of the room and appears at the environmental node.

The mean solar heat gain to the internal environmental node is given by:

$$\bar{\Phi}_{se} = \bar{S}_e \, \bar{I}_T \, A_g \tag{5.23}$$

where $\bar{\Phi}_{se}$ is the mean solar heat gain to the environmental node (W), $\bar{S}_e$ is the mean solar gain factor at the environmental node, $\bar{I}_T$ is the mean total solar irradiance (W·m^{-2}) and A_g is the area of glazing (m^2).

For the case of internal shading (i.e. blinds), part of the solar gain will enter the air node and part will enter the environmental node.

Table 5.8 Sources of heat gain and their influence on internal temperature

Source of gain	Mechanism for transfer of heat gain	Means of converting gain at source to heat gain in the space			Node at which gain acts	Means of converting gain at source to temperature rise in the space		
		Modifier		Delay (h)		Modifier		Overall delay† (h)
		Mean	Swing			Mean	Swing	
Solar radiation	Direct transmission	Unmodified	Surface factor	1 or 2	Environmental	Unmodified	Admittance	0 or 1
	Absorption by glazing	Unmodified	Unmodified	0	Environmental	Unmodified	Admittance	0 or 1
	Absorption by internal shades	Unmodified	Unmodified	0	Air	Unmodified	Unmodified	0
	Absorption by opaque fabric	*U*-value	Decrement factor	0 – 24	Environmental	Unmodified	Admittance	0 – 24
Outside air	Conduction through opaque fabric	*U*-value	Decrement factor	0 – 24	Environmental	Unmodified	Admittance	0 – 24
	Infiltration/ ventilation	Ventilation conductance	Ventilation conductance	0	Air	Unmodified	Unmodified	0
Occupants, business machines, sundry equipment, heating/cooling emitters	Radiation‡	Unmodified	Unmodified	0	Environmental (1.5 × radiant component)	Unmodified	Admittance	0 or 1
	Convection‡	Unmodified	Unmodified	0	Air (convective component minus 0.5 × radiant component)	Unmodified	Unmodified	0

† Time between occurrence of source of gain and rise of temperature in the space
‡ Radiative and convective proportions depend on characteristics of the source
Note: 'Unmodified' means that the Simple (dynamic) Model uses the value of the gain at its source

Mean solar heat gain to the air node is given by:

$$\bar{\Phi}_{sa} = \bar{S}_a \, \bar{I}_T \, A_g \tag{5.24}$$

where $\bar{\Phi}_{sa}$ is the mean solar heat gain to the air node (W) and $\bar{S}_a$ is the mean solar gain factor at the air node.

Internal heat gains

The mean heat gain from internal sources such as occupants, lighting, computers etc. is calculated by multiplying each individual load by its duration, summing over all sources and averaging the total over 24 hours. It is assumed that all the internal gains are to the environmental node. Hence:

$$\bar{\Phi}_c = \frac{\Sigma\,(\Phi_{in}\, t_{in})}{24} \tag{5.25}$$

where $\bar{\Phi}_c$ is the mean internal heat gain (W), Φ_{in} is the instantaneous heat gain from internal heat source *n* (W) and t_{in} is the duration of internal heat source *n* (h).

Mean structural heat gain

The mean gain due to transmission through the fabric is calculated by summing the mean gains through the external opaque and glazed surfaces:

$$\bar{\Phi}_f = \Sigma\,(A\; U)\,\bar{\theta}_{eo} \tag{5.26}$$

where $\bar{\Phi}_f$ is the mean fabric heat gain (W), $\Sigma\,(A\; U)$ is the sum of the products of surface area and corresponding thermal transmittance over surfaces through which heat flow occurs ($W{\cdot}K^{-1}$) and $\bar{\theta}_{eo}$ is the mean sol-air temperature (°C) (see chapter 2: *External design data*).

Note that, for glazing, $\bar{\theta}_{ao}$ is used in equation 5.26 rather than $\bar{\theta}_{eo}$ because the effect of solar radiation is included in the solar heat gains, see above.

Total gain to the environmental node

The total gain to the environmental node is given by:

$$\bar{\Phi}_{te} = \bar{\Phi}_{se} + \bar{\Phi}_c + \bar{\Phi}_f \tag{5.27}$$

where $\bar{\Phi}_{te}$ is the mean total gain to the environmental node (W), $\bar{\Phi}_{se}$ is the mean solar heat gain to the environmental node (W), $\bar{\Phi}_c$ is the mean internal heat gain (W) and $\bar{\Phi}_f$ is the mean fabric heat gain (W).

Total gain to the air node

The total gain to the air node:

$$\bar{\Phi}_{ta} = \bar{\Phi}_{sa} + C_v\,\bar{\theta}_{ao} \tag{5.28}$$

where $\bar{\Phi}_{ta}$ is the mean total gain to the air node (W), $\bar{\Phi}_{sa}$ is the mean solar heat gain to the environmental node (W), C_v is the ventilation loss ($W{\cdot}K^{-1}$) and $\bar{\theta}_{ao}$ is the mean outside air temperature (°C).

5.8.1.2 Mean internal operative temperature

For a fixed ventilation rate the difference between the mean operative temperature and the outside air temperature is given by (see Appendix 5.A5, equation 5.196):

$$\bar{\theta}_c = \frac{\bar{\Phi}_{ta} + F_{cu}\,\bar{\Phi}_{te}}{C_v + F_{cu}\,\Sigma\,(A\,U)} \tag{5.29}$$

where $\bar{\theta}_c$ is the mean operative temperature at centre of room (°C), $\bar{\Phi}_{ta}$ is the 24-hour mean total gains at the air node (W), $\bar{\Phi}_{te}$ is the 24-hour mean total gains at the environmental node (W), F_{cu} is the room conduction correction factor with respect to operative temperature, C_v is the ventilation loss (W·K^{-1}) and $\Sigma\,(A\,U)$ is the sum of the products of surface area and corresponding thermal transmittance over surfaces through which heat flow occurs (W·K^{-1}).

Using standard heat transfer coefficients and emissivity values (i.e. h_c = 3 W·m^{-2}·K^{-1}, h_r = 5.7 W·m^{-2}·K^{-1}, ε = 0.9), F_{cu} is given by (see Appendix 5.A5, equation 5.167):

$$F_{cu} = \frac{3\,(C_v + 6\,\Sigma\,A)}{\Sigma\,(A\,U) + 18\,\Sigma\,A} \tag{5.30}$$

Table 5.2 suggests values of the daily mean ventilation rates that might be obtained under certain conditions. Information on ventilation rates for use in predicting the performance of naturally ventilated buildings is contained in chapter 4: *Infiltration and natural ventilation* and CIBSE AM10: *Natural ventilation in buildings*[10]. If the ventilation varies throughout the day then the solution is complex. One approach to the solution is given by Harrington-Lynn[28].

5.8.1.3 Swing (deviation), mean-to-peak, in heat gains

The variations in heat input due to solar radiation, outside air temperature and internal gains must be determined separately and summed to give the total swing in heat input.

Rooms with south or west facing external walls will usually experience a peak temperature in the early or late afternoon when high solar irradiance coincides with high outside temperatures. North facing rooms with little solar radiation will experience the peak indoor temperature in the afternoon due to the warmth of the ventilation air. East facing rooms will experience a peak temperature in the morning or afternoon dependant on the window size, amount of natural ventilation and the internal gains.

The thermal response of the building must be assessed, see 5.7.1, to determine whether the building has a fast or slow response since this determines the time at which heat stored in the structure will be re-transmitted into the space. The time of day at which the maximum indoor temperature is likely to occur must be established.

5.8.1.4 Swing in solar heat input

Tables of solar radiation data are contained in chapter 2: *External design data*.

The swing in solar gain to the environmental node is given by:

$$\tilde{\Phi}_{se} = \tilde{S}_e\,A_g\,(\hat{I}_t - \bar{I}_t) \tag{5.31}$$

and, where internal blinds are present, that to the air node by:

$$\tilde{\Phi}_{sa} = \tilde{S}_a\,A_g\,(\hat{I}_t - \bar{I}_t) \tag{5.32}$$

where $\tilde{\Phi}_{se}$ and $\tilde{\Phi}_{sa}$ are the swings in solar gain to environmental and air nodes respectively (W), $\tilde{S}_e$ and $\tilde{S}_a$ are the cyclic solar gain factors at the environmental and air nodes respectively, A_g is the area of glazing (m^2), is the peak total solar irradiance (W·m^{-2}) and $\bar{I}_t$ is the mean total solar irradiance (W·m^{-2}).

There will be a time delay between the occurrence of the gain and the consequent increase in space temperature due to the admittance of the room surfaces. This delay is one hour for spaces having a 'slow' response and zero for 'fast' response spaces. The time at which the peak space temperature occurs is called the 'peak hour'.

To demonstrate the calculation technique, it is assumed here that the peak internal temperature is the result of the incidence of solar radiation on the facade. The same techniques may be used to determine the internal temperature at other times.

5.8.1.5 Swing in structural heat gain

The peak temperature will generally be determined by the peak solar irradiance as this is often the largest heat input, however the outside sol-air temperature will contribute to the peak load. The swing in sol-air temperature is modified in amplitude and experiences a time delay. These factors are described in terms of the decrement factor (f) and an associated time lag (ϕ). Values of decrement factor and its associated time lag are given in chapter 3: *Thermal properties of building structures*.

The swing in the sol-air temperature is given by:

$$\tilde{\theta}_{eo} = (\theta_{eo} - \bar{\theta}_{eo}) \tag{5.33}$$

where $\tilde{\theta}_{eo}$ is the swing in sol-air temperature (K), θ_{eo} is the sol-air temperature (°C) at time $(t - \phi)$, t is the time of day at which the peak space temperature occurs (i.e. the 'peak hour'), ϕ is the time lag associated with decrement factor (h) and $\bar{\theta}_{eo}$ is the mean sol-air temperature (°C).

The swing in effective heat input due to fabric heat gain is given by:

$$\tilde{\Phi}_f = \sum_n f_n\,A_n\,U_n\,\tilde{\theta}_{eo} + \sum_n f_{gn}\,A_{gn}\,U_{gn}\,\tilde{\theta}_{ao} \tag{5.34}$$

where $\tilde{\Phi}_f$ is the swing in fabric heat gain (W), f_n is the decrement factor for (opaque) surface n, A_n is the area of (opaque) surface n (m^2), U_n is the thermal transmittance of (opaque) surface n (W·m^{-2}·K^{-1}), $\tilde{\theta}_{eo}$ is the swing in sol-air temperature (K), f_{gn} is the decrement factor for glazed surface n, A_{gn} the area of glazed surface n (m^2), U_{gn} is the thermal transmittance of glazed surface n (W·m^{-2}·K^{-1}), $\tilde{\theta}_{ao}$ is the swing in outside air temperature (K) at time t.

For glazing, the decrement factor (f) is unity and the time delay for θ_{ao} (ϕ) is zero; the outside air temperature and the sol-air temperature are calculated with the incident solar radiation set to zero.

Design values of sol-air temperature (θ_{eo}) for three locations within the UK are given in chapter 2: *External design data*.

5.8.1.6 Swing in internal heat gain

At the assumed time of peak load:

$$\tilde{\Phi}_c = \hat{\Phi}_c - \bar{\Phi}_c \tag{5.35}$$

where $\tilde{\Phi}_c$ is the swing in internal heat gain (W), $\hat{\Phi}_c$ is the peak internal heat gain (W) and $\bar{\Phi}_c$ is the mean internal heat gain (W). The peak internal heat gain is taken as the sum of all internal gains within the space.

5.8.1.7 Swing in heat gain from ventilation

The swing in heat gain is given by:

$$\tilde{\Phi}_{av} = C_v\,\tilde{\theta}_{ao} \tag{5.36}$$

where $\tilde{\Phi}_{av}$ is the swing in heat gain due to ventilation (W), C_v is the ventilation conductance (W·K^{-1}) and $\tilde{\theta}_{ao}$ is the swing in outside air temperature (K).

The swing in outside air temperature is given by the difference between the outdoor air temperature at the peak hour and the mean outdoor air temperature, i.e:

$$\tilde{\theta}_{ao} = \theta_{ao} - \bar{\theta}_{ao} \tag{5.37}$$

where θ_{ao} is the outside air temperature at the peak hour (°C) and $\bar{\theta}_{ao}$ is the mean outside air temperature (°C).

5.8.1.8 Total swing in heat gain

The total swing in heat gain to the environmental node is given by:

$$\tilde{\Phi}_{te} = \tilde{\Phi}_{se} + \tilde{\Phi}_f + \tilde{\Phi}_c \tag{5.38}$$

where $\tilde{\Phi}_{te}$ is the total swing in heat gain to the environmental node (W), $\tilde{\Phi}_{se}$ is the swing in solar heat gain to the environmental node (W), $\tilde{\Phi}_f$ is the swing in fabric heat gain (W) and $\tilde{\Phi}_c$ is the swing in internal heat gain (W).

The total swing in heat gain to the air node is given by:

$$\tilde{\Phi}_{ta} = \tilde{\Phi}_{sa} + \tilde{\Phi}_{av} \tag{5.39}$$

where $\tilde{\Phi}_{ta}$ is the total swing in heat gain to the air node (W), $\tilde{\Phi}_{sa}$ is the swing in solar heat gain to the air node (W) and $\tilde{\Phi}_{av}$ is the swing in heat gain due to ventilation (W).

5.8.1.9 Swing, mean-to-peak in internal operative temperature

The swing in operative temperature is determined from the following equation (see Appendix 5.A5, equation 5.197):

$$\tilde{\theta}_c = \frac{\tilde{\Phi}_{ta} + F_{cy}\,\tilde{\Phi}_{tet}}{C_v + F_{cy}\,\Sigma\,(A\;Y)} \tag{5.40}$$

where $\tilde{\theta}_c$ is the swing in operative temperature at the peak hour (K), $\tilde{\Phi}_{ta}$ is the total swing in heat gain to the air node at the peak hour (W), $\tilde{\Phi}_{tet}$ is the total swing in heat gain to the environmental node at time t (W), F_{cy} is the room admittance factor with respect to operative temperature and $\Sigma\,(A\;Y)$ is the sum of the products of surface areas and their corresponding thermal admittances (W·K^{-1}).

Using standard heat transfer coefficients and emissivity values (i.e. h_c = 3 W·m^{-2}·K^{-1}, h_r = 5.7 W·m^{-2}·K^{-1}, ε = 0.9), F_{cy} is given by (see Appendix 5.A5, equation 5.185):

$$F_{cy} = \frac{3\,(C_v + 6\,\Sigma\,A)}{\Sigma\,(A\;Y) + 18\,\Sigma\,A} \tag{5.41}$$

5.8.1.10 Peak operative temperature

The peak operative temperature is given by:

$$\hat{\theta}_c = \bar{\theta}_c + \tilde{\theta}_c \tag{5.42}$$

where $\hat{\theta}_c$ is the peak operative temperature (°C), $\bar{\theta}_c$ the mean operative temperature (°C) and $\tilde{\theta}_c$ is the swing in operative temperature (K).

Example 5.2: Determination of overheating risk using the Simple (cyclic) Model

This example uses the geometry of the room used in CEN prEN 15255[1] with typical design level internal gains).

Figure 5.4 shows a single office module. It is situated on an intermediate floor, facing south, in a building located in London. It is assumed that the peak operative temperature will occur during a sunny period in August. It is required to determine the peak operative temperature in order to assess the risk of overheating. Constructional and occupancy details are given in Table 5.9. Surface areas, thermal transmittances and admittances are given in Table 5.10.

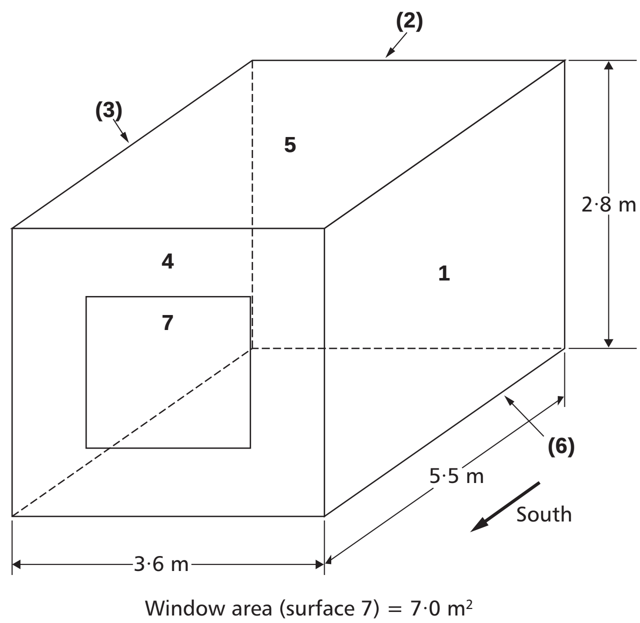


Figure 5.4 Example 5.2: south facing office module, volume 55.44 m^3

The calculation is based on the following assumptions:

— the operative temperature in adjoining rooms is equal to that for the module under consideration and hence heat flow occurs only through the outside window–wall

— the window will be open during the day and closed at night

— the thermal transmittance of the window frame is equal to that of the glass

— there are no internal blinds, therefore the solar gain to the air node is zero (i.e. $\bar{S}_a = 0$)

Step 1: solar gain through glazing ($\bar{Q}_{se}$):

From chapter 2: *External design data*, Table 2.30, the mean total solar irradiance (i.e. beam plus diffuse) for south facing surfaces in August is 177 W·m^{-2}. There are no internal blinds therefore all solar gains are to the environmental node. The window frame occupies 10% of window area, hence glazed area is (7 × 0.9) = 6.3 m^2. Hence, from equation 5.23 and Table 5.7, the mean solar gain is:

$$\bar{\Phi}_{se} = 0.62 \times 177 \times 6.3 = 691.36 \text{ W}$$

Step 2: internal gains ($\bar{\Phi}_c$)

From equation 5.25:

$$\bar{\Phi}_c = \frac{792 + 1280 + 1584}{24} = 152.33 \text{ W}$$

Step 3: fabric heat gains ($\bar{\Phi}_f$)

From chapter 2, Table 2.34(h), mean sol-air temperature (dark, south-facing surface) is 26.9 °C and mean air temperature is 19.8 °C. Therefore, using equation 5.26, for both opaque and glazed (including frame) areas of south-facing façade:

$$\bar{\Phi}_f = (3.08 \times 0.49 \times 26.9) + (7 \times 2.94 \times 19.8)$$
$$= 448.08 \text{ W}$$

Step 4: total gains to environmental node ($\bar{\Phi}_{te}$)

From equation 5.27:

$$\bar{\Phi}_{te} = 691.36 + 152.33 + 448.08 = 1291.8 \text{ W}$$

Step 5: total gains to air node ($\bar{\Phi}_{ta}$)

From equation 5.28:

$$\bar{\Phi}_{ta} = 0 + (55.44 \times 19.8) = 1097.7 \text{ W}$$

Step 6: mean operative temperature ($\bar{\theta}_c$)

For an air change rate of 3 ACH (see Table 5.2) and a room volume of 55.44 m^3, from equation 5.9:

$$C_v = {}^1/_3 \times 3 \times 55.44 = 55.44 \text{ W·K}^{-1}$$

From equation 5.17, the response factor for the room is:

Table 5.9 Example 5.2: constructional and occupancy details

Item	Details
External wall (opaque)	105 mm outer brickwork; 60 mm mineral fibre insulation; 175 mm inner blockwork; 15 mm dense plaster
Internal partition wall	12 mm lightweight plaster; 175 mm lightweight blockwork; 12 mm lightweight plaster
Internal floor/ceiling	(from floor) 4 mm floor covering; 60 mm cement screed; 40 mm insulation*; 180 mm concrete; 100 mm insulation*; 20 mm acoustic tile
Window	Double glazed, air-filled (air gap resistance: 0.16 m^2·K^{-1}·W^{-1}, ignoring frame effects)
Lighting	10 W·m^{-2} of floor area; in use 07:00–09:00 h and 17:00–19:00 h (total: 792 W·h)
Occupancy	Occupied 09:00–17:00 by 2 persons; 80 W sensible heat output per person (total: 1280 W·h)
Electrical equipment	IT equipment generating 10 W·m^{-2}; in use 09:00–17:00 (total: 1584 W·h)

Table 5.10 Example 5.2: surface areas, thermal transmittances and admittances

Surface†	Area / m^2	U value / W·m^2·K^{-1}	($A \times U$) / W·K^{-1}	Y value / W·m^2·K^{-1}	($A \times Y$) / W·K^{-1}	Decrement factor, f	Time lag, ϕ / h
External wall (opaque)	3.08	0.49	1.509	4.56	14.04	0.18	9.50
Internal wall	40.88	—	—	4.13	168.83	—	—
Internal floor	19.8	—	—	5.31	105.14	—	—
Ceiling (intermediate floor)	19.8	—	—	0.61	12.08	—	—
Glazing (inc. frame)	7.0‡	2.94	20.58	3.01	21.07	1.00	0.49
Summed values:	ΣA = 90.56		$\Sigma (A\,U)$ = 22.1		$\Sigma (A\,Y)$ = 321.2		

† Including internal partitions if present; $\Sigma (A\,U)$ is calculated over surfaces through which heat flow occurs

‡ Frame occupies 10% of window area; hence glazed area for assessment of solar gain = 6.3 m^2

$$f_r = \frac{321.2 + 55.44}{22.1 + 55.44} = 4.9$$

Hence from Table 5.6 the structure may be regarded as having a slow thermal response (i.e. thermally 'heavy-weight').

The room conductance correction factor is calculated using equation 5.30, hence:

$$F_{cu} = \frac{3\,[55.44 + (6 \times 90.56)]}{22.1 + (18 \times 90.56)} = 1.09$$

Therefore, from equation 5.29:

$$\bar{\theta}_c = \frac{1097.7 + (1.09 \times 1291.8)}{55.44 + (1.09 \times 22.1)} = 31.51 \text{ K}$$

Step 7: swing in solar gain ($\tilde{\Phi}_{se}$)

For a thermally heavyweight structure, Table 5.7 gives the cyclic solar gain factor ($\tilde{S}_{eh}$) as 0.44. Chapter 2, Table 2.30 indicates a mean total solar irradiance of 177 W·m^{-2} (see step 1) and peak solar irradiance (i.e. beam plus diffuse) of 603 W·m^{-2}, occurring at 11:30. Therefore, from equation 5.31, the swing in solar gain is:

$$\tilde{\Phi}_{se} = 0.44 \times 6.3 \times (603 - 177) = 1180.9 \text{ W}$$

Step 8: swing in structural gain ($\tilde{\Phi}_f$)

The swing in structural gain is obtained from equation 5.34. For a thermally heavyweight (i.e. slow response) building the time lag due to the thermal response of the building is one hour. Hence the time at which the peak space temperature occurs (i.e. the 'peak hour' for solar radiation) is 12:30. (Note that, for a building with a fast response, the peak hour would occur one hour earlier with zero lag, i.e. 11:30.

The swing in the sol-air temperature ($\tilde{\theta}_{eo}$) is determined by subtracting the mean sol-air temperature from the sol-air temperature at a time preceding the peak hour by the value of the time lag associated with the decrement factor of the structure. For this example, the peak hour is 11:30 and the time lag for the structure is 9.5 h; hence the sol-air temperature at 03:00 (i.e. 9.5 h previous) is required, thus the sol-air temperature for hour-ending 03:00 is used.

Chapter 2, Table 2.34, gives the mean sol-air temperature (for a dark, south-facing surface) as 26.9 °C and the sol-air temperature at hour-ending 03:00 as 11.7 °C. The outside air temperature, t_{ao}, at hour-ending 13:00 is 24.8 °C. Therefore the swing in structural gain is:

$$\tilde{\Phi}_f = (0.18 \times 3.08 \times 0.49)\,(11.7 - 26.9) + (1 \times 7 \times 2.94)\,(24.8 - 19.8) = 98.77 \text{ W}$$

Step 9: swing in internal gain ($\tilde{\Phi}_c$)

From equation 5.35:

$$\tilde{\Phi}_c = [(2 \times 80) + (10 \times 19.8)] - 152.33 = 205.67 \text{ W}$$

Step 10: swing in ventilation heat gain ($\tilde{\Phi}_{av}$)

From equation 5.36:

$$\tilde{\Phi}_{av} = 55.44 \times (24.8 - 19.8) = 277.2 \text{ W}$$

Step 11: total swing in heat gain to environmental node ($\tilde{\Phi}_{te}$)

From equation 5.38:

$$\tilde{\Phi}_{te} = 1180.9 + 98.77 + 205.67 = 1485.3 \text{ W}$$

Step 12: total swing in heat gain to air node ($\tilde{\Phi}_{ta}$)

From equation 5.39:

$$\tilde{\Phi}_{ta} = 0 + 227.30 = 227.30 \text{ W}$$

Step 13: mean-to-peak swing in operative temperature ($\tilde{\theta}_{c1200}$)

From equation 5.41:

$$F_{cy} = \frac{3\,[55.44 + (6 \times 90.56)]}{321.2 + (18 \times 90.56)} = 0.92$$

Therefore, using equation 5.40, the swing in operative temperature (at 12:30) is:

$$\tilde{\theta}_{c1230} = \frac{277.2 + (0.92 \times 1485.3)}{55.44 + (0.92 \times 321.2)} = 4.68 \text{ K}$$

Step 14: peak internal operative temperature ($\hat{\theta}_c$):

From equation 5.42:

$$\hat{\theta}_c = 31.51 + 4.68 = 36.2 \text{ °C}$$

Clearly this temperature is high for an office space and some form of shading and/or cooling may be required.

Table 5.11 shows the effect of different combinations of glazed area and thermal mass. The 'lightweight' building internal partition has the masonry block replaced by 100 mm insulation, density 30 kg·m^{-3}, specific heat capacity 850 J·kg^{-1}·K^{-1}, conductivity 0.04 W·m^{-1}·K^{-1} giving an admittance value of 0.75 W·m^{-2}·K^{-1}; the external wall, floor and ceiling are the same. 'Small' window means 3.5 m^2, large window means 7.0 m^2 as in the example.

Table 5.11 Variation in temperatures resulting from different internal thermal mass and glazed areas.

Room type	Mean operative temp., $\bar{\theta}_c$ / °C	Swing in operative temp., $\tilde{\theta}_c$ / K	Peak operative temp., $\hat{\theta}_c$ / °C
Large window, high thermal mass†	31.5	4.7	36.2
Large window, low thermal mass	31.5	8.5	40
Small window, high thermal mass	27.9	3	30.9
Small window, low thermal mass	27.9	5	32.9

† As worked example

5.8.2 Cooling load calculation

The cooling load is equal to the sum of the mean and alternating components as determined from Appendix 5.A5, equations 5.158 to 5.193, depending on the characteristics of the emitter and the control temperature. This section describes how the calculation is performed for a convective cooling system (i.e. radiant component is zero).

The total sensible cooling load (Φ_k) is given by:

$$\Phi_k = \bar{\Phi}_a + \tilde{\Phi}_a + \Phi_{sg} + \Phi_v \tag{5.43}$$

where Φ_k is total sensible cooling load to the air node (W), $\bar{\Phi}_a$ is the mean convective cooling load (W), $\tilde{\Phi}_a$ is the alternating component of the convective cooling load (W), Φ_{sg} is the cooling load due to windows and blinds (W) and Φ_v is the cooling load due to infiltration (W).

To calculate the mean ($\bar{\Phi}_a$) and alternating ($\tilde{\Phi}_a$) components for convective cooling and control on operative temperature, equations 5.168 and 5.184 apply (see Appendix 5.A5). For convenience, these may be written as follows:

$$\bar{\Phi}_a = \bar{\Phi}_{fa} + F_{cu}\, 1.5 \Sigma \bar{\Phi}_{rad} + \Sigma \bar{\Phi}_{con} - 0.5 \Sigma \bar{\Phi}_{rad} \tag{5.44}$$

$$\tilde{\Phi}_a = \tilde{\Phi}_{fa} + F_{cy}\, 1.5 \Sigma \tilde{\Phi}_{rad} + \Sigma \tilde{\Phi}_{con} - 0.5\, \tilde{\Phi}_{rad} \tag{5.45}$$

where $\bar{\Phi}_a$ is the mean convective cooling load (W), $\bar{\Phi}_{fa}$ is mean fabric gain to the air node (W), F_{cu} is the room conduction factor with respect to operative temperature, $\bar{\Phi}_{rad}$ is the daily mean radiant gain (W), $\bar{\Phi}_{con}$ is the daily mean convective gain (W), $\tilde{\Phi}_a$ is the alternating component of the convective cooling load (W), $\tilde{\Phi}_{fa}$ is alternating component of the fabric gain to the air node (W), F_{cy} is the room admittance factor with respect to operative temperature, $\tilde{\Phi}_{rad}$ is the alternating component of the radiant gain (W) and $\tilde{\Phi}_{con}$ is the alternating component of the convective gain (W).

The cooling load related to ventilation (Φ_v) is given by:

$$\Phi_v = C_v\,(\theta_{ao\,t} - \theta_c) \tag{5.46}$$

where Φ_v is the cooling load related to ventilation (W), C_v is the ventilation conductance (W·K^{-1}), $\theta_{ao\,t}$ is the outside air temperature (°C) at time t, and θ_c is the operative temperature (°C).

For control on air temperature, the above equations may be used by substituting for θ_c, F_{cu} and F_{cy}, as follows:

- θ_c is replaced by the inside air temperature, θ_{ai}
- F_{cu} is replaced by the room conduction factor with respect to air temperature, F_{au}
- F_{cy} is replaced by the room admittance factor with respect to air temperature, F_{ay}.

Corrections can be applied to deal with fluctuations in the control temperature, see 5.10.4.4. For 'comfort air conditioning', it is recommended that operative temperature be taken as the control temperature. It should be noted that most temperature detectors measure something other than the operative temperature. However, it may be assumed that the set point will be adjusted to provide comfortable working conditions equivalent to the design operative temperature.

Note that the procedure presented here assumes that, for intermittent operation, it is unnecessary to correct for gains other than solar gain. This is a simplification required to enable manual calculation. For a more accurate assessment of cooling loads, the correction given in 5.10.4.3 should be applied.

Thus assessment of the sensible cooling load falls into four stages, as follows:

- cooling load due to solar gain through windows and blinds, Φ_{sg}
- cooling load due to conduction through fabric (i.e. opaque surfaces), Φ_{fa}
- cooling loads due to internal gains, Φ_{rad} and Φ_{con}
- infiltration load, Φ_v.

The following sections give the principles and basis of these stages of the calculation, along with a numerical example. The use of psychrometric charts and the interaction of sensible and latent cooling loads which are required to size the plant are dealt with in chapter 2 of CIBSE Guide B: *Heating, ventilating, air conditioning and refrigeration*[30].

5.8.2.1 Cooling loads through windows and blinds (Φ_{sg})

A procedure for calculating the solar cooling load is described in Appendix 5.A6. It should be noted that this is not based upon the solar gain factors described under the calculation of summertime temperatures. It requires the calculation of transmitted and absorbed radiation for each hour of the day. In many cases however a good approximation to the peak solar load will be obtained by using the alternating and mean solar gain factors.

A further simplification is to make use of the tables of solar cooling loads (Tables 5.19 to 5.24) calculated for various generic glass types, see Table 5.7. The method used to determine the tabulated values of cooling load is given in Appendix 5.A6. The basis of the tables is as follows:

- constant internal temperature held by plant operating 10 hours per day (07:30–17:30 sun time)
- sunny spell of 4–5 days duration
- climatic data from chapter 2: *External design data*.

The data in the tables apply to 'fast response' (i.e. lightweight) buildings, see Table 5.6. The characteristics of such buildings are:

- average surface factor ≈ 0.8
- de-mountable partitioning
- suspended ceilings
- solid floor (with carpet or wood-block finish) or suspended floor.

Correction factors are given for buildings of heavyweight construction (i.e. 'slow response' buildings), the characteristics of which are:

— average surface factor ≈ 0.5
— solid internal walls and partitions
— solid floors and ceilings.

Surface factors for typical constructions are given in chapter 3: *Thermal properties of building structures*. It should be noted that the cooling load required to maintain constant air temperature will generally be about 10–15% less than that required to maintain constant operative temperature due to radiation from the surfaces. The data listed should be applied to the problem as follows.

In order to maintain constant operative temperature, for the UK, the cooling load may be read directly from Tables 5.19 to 5.24. (For worldwide locations, values may be obtained from the tables provided on the CD-ROM that accompanies this Guide.) For glasses other than 6 mm clear glass and for buildings of heavyweight construction, the tabulated values must be modified using the factors at the foot of each table. Interpolation is permitted where factors appropriate to the particular design situation are not given.

In order to maintain constant internal air temperature, the procedure follows that for constant operative temperature but the tabulated values must be modified by an additional factor, related to the thermal response of the building. Values are given at the foot of the tables of cooling loads (Tables 5.19 to 5.24).

5.8.2.2 Cooling load due to conduction (Φ_{fa})

If the operative temperature is held constant, the heat gain to the air node is given by the following equation (see Appendix 5.A5, equation 5.166):

$$\bar{\Phi}_{fa} = F_{cu} \Sigma (A\, U)\, (\bar{\theta}_{eo} - \bar{\theta}_{c}) \qquad (5.47)$$

where $\bar{\Phi}_{fa}$ is the mean fabric gain to the air node (W), F_{cu} is the room conduction factor with respect to operative temperature, see equation 5.30, A is the area of the surface through which heat flow occurs (m^2), U is the thermal transmittance of the surface (W·m^{-2}·K^{-1}), $\bar{\theta}_{eo}$ is the mean sol-air temperature (°C) and $\bar{\theta}_{c}$ is the mean operative temperature (°C).

The method for calculating sol-air temperatures, along with tabulated values for three UK locations, is given in chapter 2: *External design data*.

The cyclic variation about the mean is given by:

$$\tilde{\Phi}_{fa} = F_{cy} \Sigma (A\, U)\, f\, \tilde{\theta}_{eo(t-\phi)} \qquad (5.48)$$

where $\tilde{\Phi}_{fa}$ is the swing in fabric gain to the air node (W), F_{cy} is the room admittance factor with respect to operative temperature, see equation 5.41, and $\tilde{\theta}_{eo(t-\phi)}$ is the swing in sol-air temperature at time $(t - \phi)$ where ϕ is the time lag associated with decrement factor (h).

If the air temperature is held constant the factors F_{cu} and F_{cy} are replaced by the dimensionless factors F_{au} and F_{ay} respectively, and the operative temperature (θ_c) is replaced by the inside air temperature (θ_{ai}).

For standard values of heat transfer coefficients and emissivity (h_c = 3 W·m^{-2}·K^{-1}, h_r = 5.7 W·m^{-2}·K^{-1}, ε = 0.9), F_{au} and F_{ay} are given by (see Appendix 5.A5, equations 5.170 and 5.187):

$$F_{au} = \frac{4.5 \Sigma A}{4.5 \Sigma A + \Sigma (A\, U)} \qquad (5.49)$$

$$F_{ay} = \frac{4.5 \Sigma A}{4.5 \Sigma A + \Sigma (A\, Y)} \qquad (5.50)$$

where F_{au} is the room conduction factor with respect to the air node and F_{ay} is the room admittance factor with respect to the air node.

Where the building fabric is glass, the above calculation procedure may be used provided that the sol-air temperature, see chapter 2: *External design data*, is calculated with the incident radiation set to zero. This is to ensure that only longwave losses to the surroundings are considered. The decrement factor must be set to unity with time lag ϕ set to 0.

5.8.2.3 Cooling load due to internal gains (Q_{rad}, Q_{con})

Internal gains will usually comprise both radiant and convective components which must be converted to the equivalent loads at the air and environmental nodes, as follows:

— load at air node = $\Sigma\, \Phi_{con} - 0.5\, \Sigma\, \Phi_{rad}$
— load at environmental node = $1.5\, \Sigma\, \Phi_{rad}$

There is little information on the relative proportions of the radiant and convective components of internal heat gains but Table 5.12 provides some guidance for typical office equipment, adapted from published data by Wilkins et al.[31] and Hosni et al.[32] The proportions for heat gains from people may be taken to be 50% convective and 50% radiant.

To maintain constant operative temperature, the cooling load due to internal gains is the load at the environmental node multiplied by the factors F_{cu} and F_{cy} for the mean and cyclic components, respectively. The load at the air node is added directly to the cooling load. Where cooling to maintain a constant air temperature is required, F_{cu} and F_{cy} are replaced by F_{au} and F_{ay} respectively.

Table 5.12 Relative proportions of radiant and convective components of heat gains for some items of office equipment

Source of heat gains	Proportion of emitted radiation / %	
	Convective	Radiative
Personal computer and monitor	76	24
Photocopier	86	14
Laser printer	78	22

5.8.2.4 Cooling load due to air infiltration (Φ_v)

When the operative temperature is used as the control temperature, the load due to infiltration is given by equation 5.46.

When the air temperature is held constant, the heat gain at the air node due to air infiltration is given by:

$$\Phi_v = C_v\,(\theta_{ao\,t} - \theta_{ai}) \qquad (5.51)$$

where Φ_v is the cooling load related to ventilation (W), C_v is the ventilation conductance (W) (see equation 5.7), $\theta_{ao\,t}$ is the outside air temperature at time t (°C) and θ_{ai} is the inside air temperature (°C).

5.8.2.5 Cooling load due to outdoor air supply

The outdoor air supply is generally dealt with by a central air handling plant that regulates the temperature and humidity of the air supplied to the space. This air is used to control the temperature and humidity within the space. Hence account must be taken of the latent and sensible loads. These are direct loads at the air node.

Example 5.3: Calculation of air conditioning cooling load using the Simple (cyclic) Model

For this example, the office module used for example 5.2 is situated on the top floor of a south facing building located in SE England, see Figure 5.5. The constructional details are given in Tables 5.13 and 5.14. It is required to determine the sensible cooling load to be extracted at the air point at 12:30 in September for a convective system controlled on (*a*) operative temperature and (*b*) air temperature.

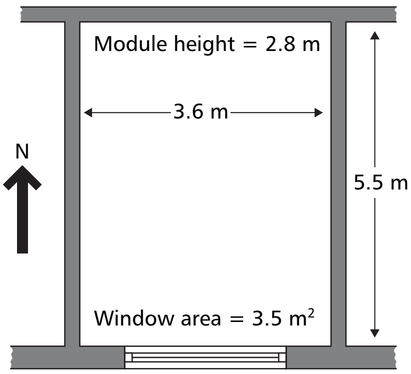


Figure 5.5 Example 5.3: constructional and occupancy details

The calculation is based on the following assumptions:

- building has slow thermal response
- external walls are light in colour; roof is dark in colour
- air conditioning operates for 10 hours per day
- air infiltration rate is equivalent to 1 air change per hour
- control temperature for room is 24 °C
- set point temperature for adjoining rooms is 24 °C.

(*a*) *Control on operative temperature*

Step 1: response factor (f_r)

From equation 5.7, ventilation conductance due to infiltration is:

$$C_v = (1/3) \times 1 \times 55.44 = 18.48\ \text{W·K}^{-1}$$

Hence, from equation 5.17:

$$f_r = \frac{459.6 + 18.48}{22.22 + 18.48} = 11.75$$

Step 2: solar gain through glazing (Φ_{sg})

From Table 5.20, the cooling load for a fast-response building in SE England with south facing glazing with intermittent blinds is 303 W·m^{-2} at 12:30 in September. Applying correction factor appropriate to a 'slow-response' building with double glazing and an external blind, and multiplying by window area:

$$\Phi_{sg} = 0.62 \times 303 \times 3.15 = 591.76\ \text{W}$$

Step 3: mean fabric gain at air node ($\bar{\Phi}_{fa}$)

Factors F_{cu}, and F_{cy} are determined from equations 5.30 and 5.41 respectively:

$$F_{cu} = \frac{3\,[18.48 + (6 \times 90.56)]}{22.22 + (18 \times 90.56)} = 1.020$$

$$F_{cy} = \frac{3\,[18.48 + (6 \times 90.56)]}{459.6 + (18 \times 90.56)} = 0.807$$

The mean sol-air and air temperatures are obtained from chapter 2, Table 2.34(*i*):

- opaque wall (south facing, light coloured): $\bar{\theta}_{eo} = 18.1$ °C
- roof (dark coloured): $\bar{\theta}_{eo} = 18.4$ °C
- window: $\bar{\theta}_{ao} = 14.2$ °C

Equation 5.47 is then used to determine the mean gains for a mean operative temperature of 24 °C.

For (opaque) wall:

$$(\bar{\Phi}_{fa})_{wall} = 1.020 \times 6.58 \times 0.49 \times (18.1 - 24.0)$$
$$= -19.40\ \text{W}$$

For roof:

$$(\bar{\Phi}_{fa})_{roof} = 1.020 \times 19.8 \times 0.44 \times (18.4 - 24.0)$$
$$= -49.76\ \text{W}$$

For window:

$$(\bar{\Phi}_{fa})_{win} = 1.020 \times 3.5 \times 2.94 \times (14.2 - 24.0)$$
$$= -102.86\ \text{W}$$

Hence total mean conduction gain at the air point is:

$$\bar{\Phi}_{fa} = -19.40 - 49.76 - 102.86 = -172.02\ \text{W}$$

Step 4: cyclic conduction gain at air node ($\tilde{\Phi}_{fa}$)

The sol-air and air temperatures are obtained from chapter 2, Table 2.34(*i*) for times of the day corresponding to 12:30 minus the time lag appropriate to the decrement

Table 5.13 Example 5.3: constructional and occupancy details

Item	Details
External wall (opaque)	115 mm outer brickwork; 60 mm mineral fibre insulation; 175 mm inner blockwork; 15 mm lightweight plaster
Internal partition wall	12 mm lightweight plaster; 175 mm lightweight blockwork; 12 mm lightweight plaster
Internal floor	(from floor) 4 mm floor covering; 60 mm cement screed; 40 mm insulation*; 180 mm concrete; 100 mm insulation*; 20 mm acoustic tiles
Roof/ceiling	(from outside) 4 mm roof tiles; 80 mm insulation, 200 mm concrete
Window	Double glazed, air-filled (air gap resistance: 0.16 $m^2 \cdot K \cdot W^{-1}$, ignoring frame effects)
Occupancy	09:00–17:00 by 1 person; 80 W sensible heat output (40 W convective, 40 W radiant)
Internal heat gains	90 W (50 W convective, 40 W radiant) during occupied hours

Table 5.14 Example 5.3: surface areas, thermal transmittances and admittances

Surface†	Area, A / m^2	U value / $W \cdot m^2 \cdot K^{-1}$	$(A \times U)$ / $W \cdot K^{-1}$	Y value / $W \cdot m^2 \cdot K^{-1}$	$(A \times Y)$ / $W \cdot K^{-1}$	Decrement factor, f	Time lag, ϕ / h
External wall (opaque)	6.58	0.49	3.22	4.56	30.00	0.18	9.5
Internal partition wall	40.88	—	—	4.13	168.83	—	—
Internal floor	19.8	—	—	5.31	105.14	—	—
Roof	19.8	0.44	8.71	7.33	145.13	0.25	4.98
Window (type 1)	3.5‡	2.94	10.29	3.01	10.54	1.00	0.49
Summed values:	ΣA = 90.56		$\Sigma (A\,U)$ = 22.22		$\Sigma (A\,Y)$ = 459.6		

† Including internal partitions if present; $\Sigma (A\,U)$ is calculated over surfaces through which heat flow occurs
‡ Frame occupies 10% of window area; hence glazed area for assessment of solar gain = 3.15 m^2

factor for the surface. Subtracting the mean temperatures from these values, see equations 5.33 and 5.37, gives the swing in the sol-air and air temperatures respectively.

For walls (i.e. θ_{eo} for hour-ending 03:00):

$$\tilde{\theta}_{eo} = 8.3 - 18.1 = -9.8\ °C$$

For roof (i.e. θ_{eo} for hour-ending 08:00):

$$\tilde{\theta}_{eo} = 18.9 - 18.4 = 0.5\ °C$$

For window (i.e. θ_{ao} for hour-ending 13:00):

$$\tilde{\theta}_{ao} = 18.7 - 14.2 = 4.5\ °C$$

Equation 5.48 is used to determine the cyclic conduction gains at the air node.

For (opaque) walls:

$$(\tilde{\Phi}_{fa})_{wall} = 0.807 \times 6.58 \times 0.49 \times 0.18 \times (-9.8)$$
$$= -4.59\ W$$

For roof:

$$(\tilde{\Phi}_{fa})_{roof} = 0.807 \times 19.8 \times 0.44 \times 0.25 \times 0.5$$
$$= 0.88\ W$$

For window:

$$(\tilde{\Phi}_{fa})_{win} = 0.807 \times 3.5 \times 2.94 \times 1.0 \times 4.5 = 37.37\ W$$

Hence total cyclic conduction gain at the air node:

$$\tilde{\Phi}_{fa} = -4.59 + 0.88 + 37.37 = 33.66\ W$$

Step 5: internal gains (Φ_{con}, Φ_{rad})

The convective and radiant components of the mean internal gain for 8-hour occupancy are as follows:

$$\bar{\Phi}_{con} = (50 + 40) \times (8 / 24) = 30.0\ W$$

$$\bar{\Phi}_{rad} = (40 + 40) \times (8 / 24) = 26.67\ W$$

Hence convective and radiant components of the swing in internal gain (i.e. instantaneous value minus mean value) are as follows:

$$\tilde{\Phi}_{con} = 90 - 30 = 60\ W$$

$$\tilde{\Phi}_{rad} = 80 - 26.67 = 53.33\ W$$

Step 6: infiltration gain (Φ_v)

From equation 5.46:

$$\Phi_v = 18.48\,(18.7 - 24.0) = -103.49\ W$$

Step 7: total sensible cooling load (Φ_k):

The components of the total sensible cooling load are $\bar{\Phi}_a$, $\tilde{\Phi}_a$, Φ_{sg} and Φ_v, see equation 5.43.

$\bar{\Phi}_a$ is determined using equation 5.44, i.e:

$$\bar{\Phi}_a = -172.02 + (1.020 \times 1.5 \times 26.67)$$
$$+ 30 - (0.5 \times 26.67) = -114.55\ W$$

$\tilde{\Phi}_a$ is determined using equation 5.45, i.e:

$$\tilde{\Phi}_a = 33.66 + (0.807 \times 1.5 \times 53.33)$$
$$+ 60 - (0.5 \times 53.33) = 131.55\ W$$

Φ_{sg} and Φ_v are calculated above, see steps 2 and 6, i.e:

$$\Phi_{sg} = 591.76 \text{ W}$$

$$\Phi_v = -103.49 \text{ W}$$

Hence, from equation 5.43:

$$\Phi_k = -114.55 + 131.55 + 591.76 - 103.49$$
$$= 505.3 \text{ W}$$

(*b*) *Control on air temperature*

Step 1: response factor (f_r)

As for (*a*) above.

Step 2: solar gain through glazing (Φ_{sg})

As for (*a*) above but an additional factor must be included for control at the air node. From Table 5.20, the value appropriate to a 'slow-response' building with double glazing and external blind is 0.85. Hence:

$$\Phi_{sg} = 0.83 \times 0.62 \times 303 \times 3.15 = 491.16 \text{ W}$$

Step 3: mean conduction gain at air node ($\bar{\Phi}_{fa}$):

Factors F_{au} and F_{ay} are determined from equations 5.49 and 5.50 respectively:

$$F_{au} = \frac{4.5 \times 90.56}{(4.5 \times 90.56) + 22.22} = 0.948$$

$$F_{ay} = \frac{4.5 \times 90.56}{(4.5 \times 90.56) + 459.6} = 0.470$$

The mean sol-air and air temperatures are as (*a*) above.

Substituting F_{au} for F_{cu} in equation 5.47, the mean gains to the air node may be determined for a mean inside air temperature of 24 °C.

For (opaque) wall:

$$(\bar{\Phi}_{fa})_{wall} = 0.948 \times 6.58 \times 0.49 \times (18.1 - 24.0)$$
$$= -18.03 \text{ W}$$

For roof:

$$(\bar{\Phi}_{fa})_{roof} = 0.948 \times 19.8 \times 0.44 \times (18.4 - 24.0)$$
$$= -46.25 \text{ W}$$

For window:

$$(\bar{\Phi}_{fa})_{win} = 0.948 \times 3.5 \times 2.94 \times (14.2 - 24.0)$$
$$= -95.59 \text{ W}$$

Hence total mean conduction gain at the air point is:

$$\bar{\Phi}_{fa} = -18.03 - 46.25 - 95.59 = -159.87 \text{ W}$$

Step 4: cyclic conduction gain at air node ($\tilde{\Phi}_{fa}$)

The swings in sol-air and air temperatures are as for (*a*) above.

Substituting F_{ay} for F_{cy} in equation 5.48, the cyclic gains to the air node may be determined for a mean inside air temperature of 24 °C.

For (opaque) walls:

$$(\tilde{\Phi}_{fa})_{wall} = 0.470 \times 6.58 \times 0.49 \times 0.18 \times (-9.8)$$
$$= -2.67 \text{ W}$$

For roof:

$$(\tilde{\Phi}_{fa})_{roof} = 0.470 \times 19.8 \times 0.44 \times 0.25 \times 0.5$$
$$= 0.51 \text{ W}$$

For window:

$$(\tilde{\Phi}_{fa})_{win} = 0.470 \times 3.5 \times 2.94 \times 1.0 \times 4.5 = 21.76 \text{ W}$$

Hence total cyclic conduction gain at the air node:

$$\tilde{\Phi}_{fa} = -2.67 + 0.51 + 21.76 = 19.6 \text{ W}$$

Step 5: internal gains at air and environmental nodes

As for (*a*) above, i.e:

$$\bar{\Phi}_{con} = 30.0 \text{ W}$$

$$\bar{\Phi}_{rad} = 26.67 \text{ W}$$

$$\tilde{\Phi}_{con} = 60.0 \text{ W}$$

$$\tilde{\Phi}_{rad} = 53.33 \text{ W}$$

Step 6: infiltration gain (Φ_v):

From equation 5.51:

$$\Phi_v = 18.48\,(18.7 - 24.0) = -103.49 \text{ W}$$

Step 7: total sensible cooling load (Φ_k)

The components of the total sensible cooling load are $\bar{\Phi}_a$, $\tilde{\Phi}_a$, Φ_{sg} and Φ_v, see equation 5.43.

$\bar{\Phi}_a$ is determined using equation 5.44, but substituting F_{au} for F_{cu}, i.e:

$$\bar{\Phi}_a = -159.87 + (0.948 \times 1.5 \times 26.67)$$
$$+ 30.0 - (0.5 \times 26.67) = -105.28 \text{ W}$$

$\tilde{\Phi}_a$ is determined using equation 5.45, but substituting F_{ay} for F_{cy}, i.e:

$$\tilde{\Phi}_a = 19.6 + (0.470 \times 1.5 \times 53.33)$$
$$+ 60.0 - (0.5 \times 53.33) = 90.53 \text{ W}$$

Φ_{sg} and Φ_v are calculated above, see steps 2 and 6, i.e:

$$\Phi_{sg} = 491.16 \text{ W}$$

$$\Phi_v = -103.49 \text{ W}$$

Hence, from equation 5.43:

$$\Phi_k = -105.28 + 90.53 + 491.16 - 103.49$$

$$= 372.9 \text{ W}$$

5.9 Airflow modelling

The performance of heating and cooling units is not determined solely by capacity load calculations. Some types of unit, e.g. downward projecting warm air diffusers and displacement airflow terminals, also depend on the airflow patterns within the space. The way in which the air moves through a building may also influence summertime overheating. For example, where an atrium is used to induce air flow through the surrounding offices, it is possible for warm air from the atrium to flow into the upper offices[33]. The prediction of airflow patterns cannot therefore always be separated from the calculation of the thermal performance of buildings.

Excluding laboratory mock ups there are four main approaches for quantifying airflow patterns:

— design guides

— zonal air flow models

— field airflow models using computational fluid dynamics (CFD)

— analogue models.

Detailed descriptions of air flow models are given by Awbi[34].

5.9.1 Design guides

Examples of the first of these approaches are CIBSE AM10: *Natural ventilation in non-domestic buildings*[10] and BSRIA Application Guide AG1/74: *Designing variable volume systems for room air movement*[35].

5.9.2 Zonal airflow models

The bulk movement of air within a building can be predicted using computer programs. Originally, such programs were popular for predicting infiltration rates but they are now often combined with dynamic thermal models to predict the performance of buildings where ventilation strategies are considered to be important. The analysis usually considers the movement of air as driven by forces due to both buoyancy and wind. Resistance to airflow is treated in a similar way to that for flow through orifices and ducts. The calculation of buoyancy driven flow through large openings is complicated because both inflow and outflow can occur at the same time. It is important to check on how this possible situation is handled by the software. CIBSE TM33[2] contains a verification test for zonal airflow models (although not the latter situation).

A significant difficulty in applying these models is in obtaining values for the pressure coefficients required to determine the wind pressures on the building. These can be obtained from wind tunnels tests (generally assumed to be the most accurate method), CFD simulations or tables[36]. The validity of computational methods is an area of ongoing research.

Examples of the application of zonal air flow models are given in CIBSE AM10: *Natural ventilation in non-domestic buildings*[10].

Control becomes a significant issue when a combined zonal and thermal model is used to simulate a building performance. One approach is to use a rule based system such as:

> 'If the space is occupied and internal temperature is above x °C and the external temperature below y °C start to open the window.'

Examples of the application of rule based natural ventilation control are contained in CIBSE TM36: *Climate change and the indoor environment*[14].

5.9.3 Computational fluid dynamics

An accurate assessment of the effect on the heat loss due to space height requires calculation of the airflow pattern within the space, for which computational fluid dynamics (CFD) may be required.

The objective of CFD is to provide a numerical solution to the equations governing the flow of fluids. The nature of fluid flow requires calculations to be made at a very large number of positions within the space under consideration. The result of the calculations is a detailed picture (both quantitative and qualitative) of the air movement and temperatures within the space.

Expert knowledge is required to make the many assumptions necessary in describing the problem, such as the choice of computational grid, boundary conditions and turbulence model, and to ensure that the results obtained are interpreted correctly. There are also difficulties associated in obtaining a converged solution (the solution involves iteration). Mathematical convergence can be a problem, particularly with buoyancy driven airflow.

The way air flows within a space is dependant upon the boundary conditions and unless the CFD code is integrated with a dynamic thermal model* these are usually determined separately. It is usual to take surface temperatures and heat gains directly from the predictions of a dynamic thermal model. When doing this it is important to ensure that:

— the same convective heat transfer coefficient is used in the CFD code as in the thermal model.

— only the convective component of internal heat sources is considered (the radiant component is absorbed at the surfaces).

— notwithstanding the difficulties and expense, CFD is a useful tool to obtain insight to areas where information is otherwise unavailable; it is not, however, an everyday design tool nor is it necessary for most conventional spaces.

* The integration of a CFD and dynamic thermal model is possible but only practicable for special cases. Integration with a steady state model is practicable.

5.9.4 Analogue airflow models

Excluding wind tunnels, the most common analogue model uses salt solutions and pure water to provide an inverted representation of the difference in density between the air inside and outside a building[37], thereby simulating buoyancy driven flow. The method requires significant laboratory facilities and expert application.

5.10 Application of CIBSE calculation methods

This section describes the general application of the calculation methods. Examples of the manual methods are given in 5.6.2 (heat loss), 5.8.1 (summertime overheating) and 5.8.2 (cooling load). Guidance on the application of software based methods is given in CIBSE AM11[6]. Under some circumstances it will be necessary to make use of an airflow model, e.g. to determine natural ventilation flows or the effect of displacement ventilation. The options available are discussed briefly more details are contained in CIBSE AM10[10] and AM11[6].

5.10.1 Building/room dimensions

The theory presented above assumes one-dimensional heat flow. In reality two- and three-dimensional flow occurs. It is common practice to account for the deviation from one-dimensional heat flow through walls by measuring surface dimensions between the mid-points of the walls. However, dimensions obtained in this way produce an incorrect value for room volume. Furthermore, they are not appropriate where the surface effects of longwave radiant and convective heat flows and solar absorption need to be considered. It is recommended that measurements be taken between internal wall surfaces (see chapter 3, section 3.3.2). Deviation from one-dimensional heat flow may be taken into account by modifying the thermal conductivity value for the wall. Consistency of surface dimensions is of particular importance when using software packages.

Ceiling and floor voids are also areas where care is needed. Computer programs may allow for these to be described as an additional space. For manual calculation unventilated voids can be represented by their effect on the thermal properties of the surface (U- and Y- values). For manual cooling load calculations where the void is used for supply or extract, unless heat can flow from the void to an adjacent space, the effect of the void may be ignored. If the spaces above and below have similar characteristics to that under consideration heat loss will be compensated by heat gain and again the effect of the void may be ignored. The void will however have an effect on the dynamic response of the space (generally increase the rate of response) and so must not be ignored as an item of building fabric.

While the void may not have a significant effect upon the space-cooling load the same may not be true for space temperatures. For example, heat pick-up in a floor void will increase the temperature of any air flowing through the void. Thus in the case of a floor supply system where perhaps air is supplied to the void a few degrees below the desired room temperature any increase in supply temperature will reduce the available cooling capacity.

The designer should ensure that these and similar deviations from the very simple cases represented by the models are understood and means taken to minimise the effect on the calculation.

5.10.2 Safety margins

Calculation of heating and cooling capacities in this Guide do not incorporate safety margins except where specifically stated (e.g. section 5.10.3.3, equation 5.54) and it is incumbent upon the designer to assess the plant margin required.

5.10.3 Heating system design

This section considers the calculation of the overall heating capacity for a building or zone on the assumption that heat will be supplied from a central plant. It is however recognised that there will be circumstances when local heat sources are more appropriate; in such cases each zone must be considered separately. The selection of heating plant is considered in chapter 1 of CIBSE Guide B[30].

The following calculation sequence is recommended:

— determine room heat emitter sizes (section 5.10.3.1)

— apply corrections for local effects for example screening of emitters by furnishings (section 5.10.3.2)

— apply corrections for intermittent operation (section 5.10.3.3)

— assess heat losses from the distribution system, where appropriate (section 5.10.3.4)

— assess diversity factor for central plant, where appropriate (section 5.10.3.5)

— select central plant, see CIBSE Guide B[30], chapter 1.

There will be circumstances in which a different approach will be required and it is often necessary for the designer to devise a suitable strategy.

Guidance is also provided for the following applications:

— storage systems (section 5.10.3.6)

— highly radiant systems (section 5.10.3.7)

— selective systems (section 5.10.3.8)

Air movement through and within spaces can have a significant effect on the performance of emitters. The calculation of air movement is considered in section 5.9.

5.10.3.1 Sizing room emitters

It is usual to exclude from the calculation heat gains from sources other than the emitter and its associated distribution system because, in most cases, plant will be operated at times when the building is unoccupied. However, where it is certain that heating will be operating continuously (over 24 hours) and heat sources such as electric lighting and occupants will be available at all times, the steady state heat requirement can be reduced by deducting the continuous loads from the calculated heat loss. In such cases, it must be made clear to the client that if the internal gains are removed or reduced, or the plant operated intermittently, then the design internal temperature will not be achieved.

The type of heating system can have a significant effect on the calculated design load therefore it is essential to use appropriate values for the heat transfer correction factors F_{1cu} and F_{2cu} when calculating the steady state heat requirements. An example of this calculation is given in section 5.6.1.

5.10.3.2 Local effects

The size of the emitter will be affected by its position within the space and the characteristics of the space. Four important effects are:

— back losses from radiators

— stratification

— presence of furniture

— direct radiation incident on occupants.

There will also be cases where the airflow pattern generated by the emitter, or due to features of the space, will be important. For example, a radiator placed at the back of a heated space, away from the window, will induce air movement whereby cool air close to the window will fall and be drawn across the room at floor level to replace warm air rising from the radiator. The temperature gradient produced and the increased air movement at floor level may exacerbate feelings of discomfort experienced by the occupants. The position of warm air inlets is also important, a high level inlet causing a larger temperature gradient than that produced by a low-level inlet. The calculation of air movement is discussed in section 5.9.

Back losses

For the theoretical models, it is assumed that emitters and sources of internal gains are free standing within a space and therefore their total energy output enters the space. For energy sources embedded in or fixed to the surfaces surrounding a space, it is necessary to introduce a correction for the additional heat losses which occur due to part of the surface being exposed to the temperature of the emitter rather than the internal environmental temperature in the room. This is referred to as the back loss.

For wall mounted radiators, the back loss is given by:

$$\Phi_b = A_h\, U\, (\theta_h - \theta_{ei}) \qquad (5.52)$$

where Φ_b is the back loss (W), A_h is the projected area of the emitter (m^2), U is the thermal transmittance of the exposed wall to which the emitter is fixed (W·m^{-2}·K^{-1}), θ_h is the surface temperature of the heat emitter (°C) and θ_{ei} is the environmental temperature in the space (°C).

Other sources of back loss can be corrected for in a similar way.

Stratification

For heat loss calculations a uniform temperature is assumed throughout the height of the heated space. Some heating systems cause vertical temperature gradients across the space, which can lead to increased heat losses, particularly through the roof, see Figure 5.6.

Temperature gradients also depend on the height of the space. An accurate assessment of the effect on the heat loss due to space height requires calculation of the airflow pattern within the space, for which computational fluid dynamics may be required. However, some allowance can be made for the increase in fabric losses using Table 5.15 (page 5-30). The table provides percentage increases in fabric loss for various systems and for heated spaces of various heights. This percentage increase applies to the fabric loss only, not to the total heat loss. The increase in air losses should be based directly on the estimated difference between the temperature of the bulk air in the space (as given by the heat loss calculation, see 5.6.2, equation 5.15) and that at the point where the air leaves the space, multiplied by the ventilation conductance, C_v.

Furnishings

If emitters are located behind furnishings much of the space will be denied radiant heat from the emitter. While the heat absorbed by the furnishings will eventually enter the space, often it will not be available during pre-heating,

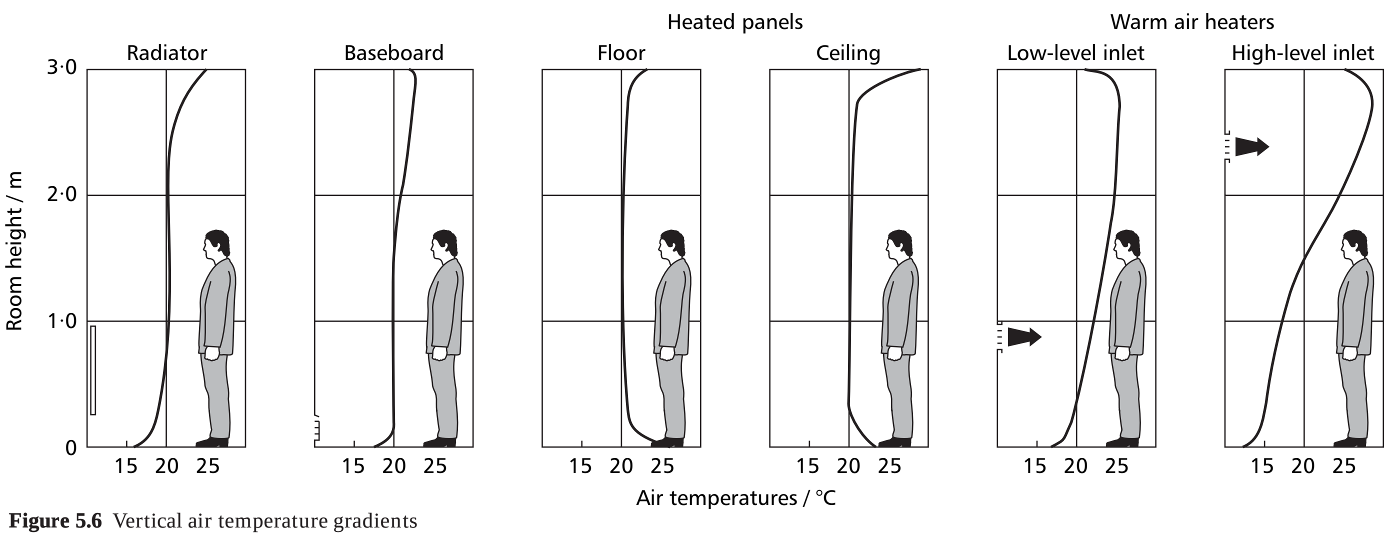


Figure 5.6 Vertical air temperature gradients

Table 5.15 Allowances for height of heated space

Type of heating system	Percentage increase in fabric loss for heated space of stated height / %		
	5 m	5–10 m	> 10 m
Mainly radiant:			
— warm floor	0	0	0
— warm ceiling	0	0–5	—†
— medium and high temperature downward radiation from high level	0	0	0–5
Mainly convective:			
— natural warm air convection	0	0–5	—†
— forced warm air; cross flow at low level	0–5	5–15	15–30
— forced warm air; downward from high level	0–5	5–10	15–20
— medium and high temperature cross radiation from intermediate level	0	0–5	5–10

† System not appropriate to this application

Note: most conventional 'radiators' may be regarded as mainly convective

prior to occupancy. In such cases the emitter should be sized as though the output were 100% convective. Unfortunately furnishings may also interfere with the free flow of air over the emitter and so change its characteristics. It is therefore difficult to provide simple correction factors. However if it is believed that the emitter behaves as though it were in free space then a suitable approach might be to ignore any radiant component of the output. For example, to meet a space heating demand of 1400 W using an emitter having an output which is nominally 70% convective and 30% radiant, if the emitter is positioned such that none of the radiant component will enter the space directly, its installed capacity should be increased to 2000 W (i.e. 1400/0.7).

Direct radiation

It may be possible to increase the dry bulb temperature to improve the comfort of occupants located in areas with lower radiant temperatures, for example those adjacent to windows. It may also be possible to lower the dry-bulb temperature to improve the comfort of occupants exposed to higher radiant temperatures, for example those near to radiant heating systems. It is however, necessary to know both the temperature of the emitter and the view factor between the occupant and the emitter. A method to determine this effect is given by Fanger[7]. Guidance is also given in chapter 1: *Environmental criteria for design*.

5.10.3.3 Intermittent operation

Intermittent heating is where the plant is switched off at the end of a period of building occupancy and turned on again at maximum output prior to the next period of occupancy in order to return the building to design conditions. Intermittent heating is the most common form of operation for heating plant in the UK. There are two main types of intermittent operation. 'Normal' intermittent operation is where the output of the heating system is reduced when the building is unoccupied or when the occupants are sleeping. 'Highly' intermittent operation is where a building is occupied for short periods only and therefore must be brought up to temperature rapidly, just prior to use.

With highly insulated buildings, there may be a case to minimise overall costs by using smaller plant operating continuously. This will require an estimate of the likely savings in capital costs and the increase in running costs over the life of the heating plant. Dynamic modelling will usually be required to perform such an estimate. If this is done it is essential that the designer ensures that the software implementation of an optimum start system is appropriate for the design.

Normal intermittent operation

The degree to which the plant output can be reduced will depend upon:

- building type and purpose
- level of insulation
- external temperature.

The building type will influence the level to which temperatures can be allowed to fall, e.g. the minimum temperature for bedrooms will differ from that for the storage of works of art.

The possibility of damage to the fabric due to condensation and structural movement caused by fluctuating temperatures must also be considered. In situations where the minimum temperature is critical, it must be determined by discussion with the client. Otherwise a temperature of 10 °C is suggested as a general minimum.

If it is required to raise the internal temperature from the overnight minimum temperature to the required level for occupancy within a reasonable period, it will be necessary to install plant with a capacity greater than the steady state design capacity. The excess capacity will depend upon:

- required heat up time
- dynamic characteristics of the boiler plant and distribution network
- thermal storage characteristics of the building.

Modern building control systems usually include software that determines the preheat time automatically from an assessment of previous building performance[38] (i.e. 'adaptive optimum start') and therefore an accurate determination of the design preheat time is not essential.

An accurate determination of excess capacity is not warranted and equation 5.53 below, based on admittance theory, is adequate for most applications. Plants with rapid response are most suitable for intermittent use in order to minimise the preheat time and energy losses due to the storage of thermal energy within the heating system. It has also been demonstrated that, provided means are taken properly to control multiple boiler installations, the energy consumption for an intermittently operated system is not very sensitive to excess capacity[39].

Exceptions are:

- highly intermittent operation (i.e. short occupation periods such as assembly halls, churches etc.)
- systems with a long time constant (e.g. underfloor heating).

Therefore, with the above exceptions, the intermittent or peak heating load can be calculated as follows:

$$\Phi_p = F_3\,\Phi_t \tag{5.53}$$

where Φ_p is the plant size for intermittent operation (W), F_3 is a correction factor for intermittent heating and Φ_t is the total heat loss (W).

F_3 is based on the thermal response factor and the total hours of plant operation, including preheat but neglecting the time for the system to reach its operating conditions, and is given by the following equation, see Appendix 5.A8:

$$F_3 = \frac{24\,f_r}{t_o\,f_r + (24 - t_o)} \tag{5.54}$$

where t_o is the duration of plant operation (including preheat) (h) and f_r is the thermal response factor (see equation 5.17).

If the calculated value of F_3 is less than 1.2, it is suggested that its value be taken as 1.2 to ensure that the customary safety margin of 20% is maintained. However, designers may choose to use other values. The value chosen may be justified by building simulation using the thermal Reference or Basic (dynamic) Models.

Large values for F_3 indicate that consideration should be given to extending the operating period or operating continuously, rather than greatly oversizing the plant capacity. Example 5.4 below illustrates the calculation of plant size for intermittent operation.

Equation 5.54 is based on heating to the internal environmental temperature with an emitter that has the same radiation/convection ratio (i.e. 2:1) as in the definition of the environmental node. It can be expected that the excess capacity will be underestimated for a system that is 100% radiant and overestimated for an all-air system. There is a further safety margin in that no account is taken of a minimum internal temperature in the derivation of the excess capacity.

For greater accuracy in assessing the excess capacity a dynamic simulation model should be used, which will enable a proper assessment of the system performance to be undertaken. The minimum requirements for such a model are that it must:

— be capable of representing transient heat flows

— represent the characteristics of the space heat emitter

— be capable of representing time delays associated with the boiler and distribution system (the latter may be particularly important where there are significant transportation delays)

— model the characteristics of an optimum start system.

Sizing of the emitters and the size of the central plant relative to the emitters must be considered. This may not require physically re-sizing the heating system. A degree of effective oversizing can be achieved during the pre-heat period by:

— operating the central plant to provide an elevated flow temperature

— reduction of the ventilation rate by window closure or control of the mechanical ventilation

— reduction of fabric transmittance by the use of window shutters or curtains

— increased emitter output due to initial low building temperatures.

The effective rating of the emitter can also be increased outside the occupied period by employing central plant sized to provide an elevated flow temperature.

Example 5.4: Correction of heating plant capacity for intermittent heating

For the small factory considered in Example 5.1, see Figure 5.3 (page 5-11) and Table 5.16, it is required to determine the effect of intermittent operation of the heating plant. The heating plant is assumed to operate for 8 hours with a preheat time of 3 hours.

Step 1: ventilation conductance (C_v)

From Example 5.1, C_v = 93.75 W·K^{-1}

Step 2: thermal response factor (f_r)

From equation 5.17:

$$f_r = \frac{1359.45 + 93.75}{179.3 + 93.75} = 5.32$$

Table 5.16 Example 5.4: surfaces areas, thermal transmittances and admittances

Surface*	Area, A / m^2	U-value / W·m^{-2}·K^{-1}	$(A \times U)$ / W·K^{-1}	Y-value / W·m^{-2}·K^{-1}	$(A \times Y)$ / W·K^{-1}
Floor	112.5	0.22	24.8	5.2	585.0
Roof	112.5	0.13	14.6	0.7	78.75
External walls	171.0	0.25	42.8	3.5	598.5
Glazing	48.0	1.8	86.4	1.8	86.4
Doors	6.0	1.8	10.8	1.8	10.8
	ΣA = 450.0		$\Sigma (A\,U)$ = 179.3		$\Sigma (A\,Y)$ = 1359.45

* Including internal partitions if present; $\Sigma (A\,U)$ is calculated over surfaces through which heat flow occurs

Step 3: correction factor for intermittent heating (F_3)

From equation 5.54:

$$F_3 = \frac{24 \times 5.32}{(11 \times 5.32) + (24 - 11)} = 1.79$$

Step 4: plant capacity for intermittent operation (Φ_p)

From Example 5.1, the total heat loss for the system types considered were, using the simple method, both as follows:

$$\Phi_t = 5.46 \text{ kW}$$

Hence, using equation 5.53, excluding distribution and back losses (see section 5.8.2.6) and ignoring internal heat gains, the calculated plant capacity for intermittent operation for either system is as follows.

$$\Phi_p = 1.79 \times 5.46 = 9.75 \text{ kW}$$

Highly intermittent operation

Where a building is used for very short periods (e.g. a meeting hall), the steady state heat loss is inappropriate for the purposes of plant sizing and it is necessary to consider the way in which the heat is absorbed by the fabric. In the terms of the Simple (dynamic) Model this is represented by the admittance of the structure, in which case the heat output required from the room appliance is given by:

$$\Phi_p = [F_{1cy} \Sigma (A\ Y) + C_v F_{2cy}] (\theta_c - \theta_{ao}) \quad (5.55)$$

where Φ_p is the plant size for intermittent operation (W), F_{1cy} and F_{2cy} are factors related to the characteristics of the heat source, $\Sigma (A\ Y)$ is the sum of the products of surface areas and the corresponding thermal admittances ($W \cdot K^{-1}$), C_v is ventilation conductance ($W \cdot K^{-1}$), θ_c is the operative temperature (°C) and θ_{ao} is the outside air temperature (°C).

C_v is given by equation 5.9. F_{1cy} and F_{2cy} are given by (see Appendix 5.A5, equations 5.189 and 5.190):

$$F_{1cy} = \frac{3 (C_v + 6 \Sigma A)}{\Sigma (A\ Y) + 18 \Sigma A + 1.5\ R\ [3\ C_v - \Sigma (A\ Y)]} \quad (5.56)$$

$$F_{2cy} = \frac{\Sigma (A\ Y) + 18 \Sigma A}{\Sigma (A\ Y) + 18 \Sigma A + 1.5\ R\ [3\ C_v - \Sigma (A\ Y)]} \quad (5.57)$$

where R is the radiant fraction of the heat source.

Dynamic thermal properties such as thermal admittance are functions of the frequency of the driving force for heat transfer. Therefore, for highly intermittent operation, the numerical value of the thermal admittance (Y) should be based on the actual hours of plant operation rather than the standard (24-hour) cycle. Table 5.17 shows the relationship between the thermal admittance and the hours of plant operation for surfaces with fast and slow thermal responses. The multiplying factors are the ratios of thermal admittance to 'standard' Y-values for an excitation period of 24 hours. The tabulated data are provided for general guidance and interpolation between the values is permitted.

Table 5.17 Multiplying factors to relate thermal admittance to daily hours of plant operation

Daily hours of plant operation	Multiplying factor for stated nominal thermal response*	
	Slow	Fast
12	1.0	1.0
6	1.1	2.0
4	1.2	2.8
3	1.2	3.5
2	1.2	4.7
1	1.3	6.5

* See Table 5.6

The derivation of thermal admittance given in chapter 3 (Appendix 3.A6) allows the use of variable cycle times.

5.10.3.4 Heat loss from the distribution system

Heat losses and gains to and from the distribution system may be determined from the data for heat loss from pipes contained in CIBSE Guide C(40), chapter 3: *Heat transfer*.

5.10.3.5 Central plant diversity

In practice, for intermittent heating, the preheat periods for all the rooms in a building will generally be coincident. Therefore, the central plant rating is the sum of the heat demands for the individual rooms, modified to account for the net infiltration.

The total net infiltration of outdoor air is usually about half the sum of the rates for the separate rooms. This is because at any one time, infiltration of outdoor air mainly takes place on the windward part of the building, the flow in the remainder being outwards, see chapter 4: *Infiltration and natural ventilation*. CIBSE TM33(2) contains a simple test for the calculation of central plant capacity.

For continuously heated buildings, some diversity can be expected between the room heating loads and Table 5.18 suggests some values. When mechanical ventilation is combined with heating, the heating plant and the ventilation plant may have different hours of use and the peak loads on the respective plants may often occur at different times.

Table 5.18 Diversity factors for central plant (continuous heating)

Space or buildings served by plant	Diversity factor
Single space	1.0
Single building or zone, central control	0.9
Single building, individual room control	0.8
Group of buildings, similar type and use	0.8
Group of buildings, dissimilar uses†	0.7

† Applies to group and district heating schemes where there is substantial heat storage in the distribution mains, whether heating is continuous or intermittent

The central plant may also be required to provide hot water for domestic and/or process purposes. These loads may have to be added to the net heating load to determine the necessary plant duty. However, careful design (for example using hot water storage systems) may avoid the occurrence of simultaneous peaks and, for large installations, boiler curves may indicate whether reductions in the boiler rating can be made[41]. In many cases, little or no extra capacity may be needed for hot water supply, the demand being met by diverting capacity from the heating circuits for short periods.

5.10.3.6 Storage systems

It may be financially advantageous to supply heat outside of normal hours and to store it within the fabric of the building or in water based storage systems. Such systems are generally used only when cheap off-peak electrical power is available. However any fuel can be used where cheap tariffs occur at certain times.

Two types of storage system are described below. In both cases energy will generally be supplied to the store at night, during the times of cheap fuel tariffs. Under design conditions, the storage and charge must be equal to the 24-hour demand.

The input rating for both systems can be calculated using the following equation, where the mean heat requirement and mean heat losses are expressed in joules:

$$\Phi_r = \frac{\text{mean heat requirement} + \text{mean heat losses}}{3600\, t_r} \qquad (5.58)$$

where Φ_r is input rating of plant (W) and t_r is the recharge time for the system (h).

The losses may be determined from the mean storage temperature or the mean space temperature and the level of insulation. Where the storage vessel is also the space heat emitter the losses may be considered as useful heat. Such systems differ widely and the designer must assess each individually. Often this will require the use of a dynamic model.

Uncontrolled storage systems (continuous heating)

In these cases, heat is dissipated continuously throughout 24 hours (e.g. uncontrolled floor or block heating) and no intermittent heating factors should be applied.

The daily storage and charge (in joules) is given by the mean heat required (in watts) multiplied by (24 × 3600). The mean heat requirement is equal to the steady state heat loss calculated at the estimated mean internal temperature, which will usually be the design internal temperature.

With these systems the highest room temperature is likely to be at the end of the re-charging period and the lowest will occur just prior to recharging, at the end of the usage period. This temperature swing will depend on the thermal properties of the building and those of the appliance but should be limited to 3 K or less for acceptability. The individual room units should be sized according to the method described for continuous heating.

With floor heating systems the floor surface temperature should conform to the comfort criteria set down in chapter 1: *Environmental criteria for design*.

Controlled storage systems

In cases where heat dissipation takes place only during the hours of use (e.g. controlled block heating or electrically heated water with thermal storage), the design rate of emission should be calculated as for conventional intermittent systems with the total storage capacity based on the mean internal temperature. This may be calculated as follows[42]:

$$\bar{\theta}_i = \frac{(t_o\, f_r)\,(\theta_r - \bar{\theta}_o)}{t_o\, f_r + (24 - t_o)} + \bar{\theta}_o \qquad (5.59)$$

where $\bar{\theta}_i$ is the 24-hour mean daily internal temperature (°C), t_o is the duration of plant operation including preheat (h), f_r is the thermal response factor (see equation 5.17), θ_r is the internal design temperature (°C) and $\bar{\theta}_o$ is the 24-hour mean daily outside temperature (°C).

The 24-hour mean daily outside temperature ($\bar{\theta}_o$) is assumed to equal the design outside temperature.

A review of the environmental benefits of thermal storage systems is given in CIBSE Research Report RR06: *Environmental benefits of thermal storage*[43].

5.10.3.7 Radiant heating systems

High temperature radiant systems are generally chosen for local heating or for situations where very intermittent heating is required. In either case the standard heat loss calculations are not appropriate for evaluating the total output for the purposes of equipment selection. For these systems it is essential to determine the distribution of radiant energy within the space. To do this it is necessary to obtain a radiant polar diagram for the emitter.

Medium and low temperature radiant systems can be sized using the usual heat loss calculation methods. For these cases, the total panel area is determined by dividing the net heat requirement by the emission per unit area of panel.

The distribution of hot surfaces needs to be considered to ensure a reasonable degree of uniformity over the working plane[44,45]. Generally, low and medium temperature panels should be placed near the exposed walls of the space. A single panel should not normally be placed in the centre of the ceiling as this could produce a temperature peak in the middle of the working zone. The panel area and temperature should be checked in relation to the mounting height in order to ensure comfort.

Limiting factors for the design of these systems are given in chapter 1: *Environmental criteria for design*, and Fanger[7] gives a method by which system performance can be assessed.

5.10.3.8 Selective systems

Systems of this type are used in situations where all the spaces in the building do not require heating at the same time. This is often the case in dwellings where demands for heat in the living spaces and the bedrooms do not coincide. The individual appliances must be sized as described above according to heat losses, gains and intermittency of operation. The central plant should be capable of meeting the peak simultaneous output of the units. The units can usually be operated intermittently.

5.10.4 Cooling system design

This section describes the calculation sequence to assess if a building needs cooling and, if so, to determine the room cooling load and the maximum system demand. As with heating systems there will be situations for which central plant is not appropriate and local cooling devices will be used. Although the calculation procedure described here references the simple, cyclic model it is equally applicable to any dynamic thermal model.

The recommended calculation sequence is as follows:

— assess risk of overheating (section 5.10.4.1)

— determine room cooling load (section 5.10.4.2)

— apply corrections for intermittent plant operation, if appropriate (section 5.10.4.3)

— apply corrections for fluctuations in control temperature, if appropriate (section 5.10.4.4)

— assess the diversity factor for the central plant, if appropriate (section 5.10.4.5)

— assess the heat gains/losses to the distribution system, if appropriate (section 5.10.4.7).

The calculations described here are for a fully convective cooling system. However, this is not always the case and guidance is given in 5.10.5.3 on passive cold (i.e. chilled) surfaces and multi-mode systems such as mechanical ventilation and chilled beams. The application of ice storage systems is covered in CIBSE TM18: *Ice storage*(46).

5.10.4.1 Assessment of overheating risk

This assessment should be carried out to determine if cooling is required. The method of assessment of overheating risk will differ according to the model chosen. If a transient model has been chosen then a period of hot days should be analysed. The exact overheating criteria and appropriate weather data must be agreed with the client. Guidance for overheating is given in chapter 1: *Environmental criteria for design*, section 1.4.2. Climatic data for the UK are given in chapter 2 and CIBSE Guide J: *Weather, solar and illuminance data*(12).

Where the criterion is the number of hours of overheating, i.e. the hours for which a specified temperature will be exceeded, that prediction is very sensitive to the method of assessment(6) and the climatic data used. For the UK, CIBSE recommends that the appropriate CIBSE Design Summer Year (DSY) be used, see chapter 1 and CIBSE Guide J: *Weather, solar and illuminance data*(12). At present DSYs are available for only 14 locations in the UK (see www.cibse.org/publications) and therefore some judgement is required when selecting the DSY appropriate to the site.

Section 5.8.1 gives equations for the calculation of overheating risk and a numerical example (example 5.2) using the Simple (dynamic) Model

If it is found that overheating is likely to occur an assessment should be made to establish whether it could be reduced to an acceptable level by changes to the building fabric, increasing the natural ventilation, or by other means.

The assessment of overheating risk for naturally ventilated buildings requires knowledge of the likely air change rates. Modern buildings are usually well sealed and infiltration rates may be as low as 0.25 air changes per hour. This is much lower than that required for acceptable air quality. It is expected that, in practice, a minimum ventilation rate of about one air change per hour will be achieved either by trickle ventilators or by windows ‘cracked’ open. Although ventilation rates greater than 10 air changes per hour can be achieved by cross ventilation etc., they can cause discomfort. Furthermore, the effectiveness of ventilation as a cooling medium is reduced as the air change rate increases. It is unusual to predict much of a reduction in space temperature when the air change rate exceeds 8 h^{-1}. Issues associated with the prediction of natural ventilation rates are discussed in section 5.9.2.

5.10.4.2 Room cooling load calculation

Once it has been established that air conditioning is necessary then the maximum room cooling loads need to be determined in order to obtain the plant size. Equations for calculating the cooling load and a numerical example using the Simple (cyclic) Model are given in section 5.8.2. Appendix 5.A6 contains an algorithm for the calculation of cooling loads.

Note that the cooling loads calculated using the methods described in section 5.6.3.5 do not include latent cooling, which depends on the type of system selected.

The Simple Model is based upon continuous operation and a constant space temperature. When plant is run intermittently and/or deviations from the control temperature are permitted it is necessary to apply corrections to the predicted loads. A transient model automatically takes account of these issues, however the calculations are a useful way to make a separate assessment of these effects.

5.10.4.3 Corrections for intermittent plant operation (Simple Model)

Generally the air conditioning plant is operated intermittently and the plant size can, in principle, be calculated directly(29). However, if the infiltration ventilation rate is constant or small, then the increase in cooling load $\Delta\Phi_k$ in going from continuous to intermittent operation can be calculated more simply(29) as follows:

$$\Delta\Phi_k = \frac{24\,[F_{cy}\,\Sigma\,(A\,Y) - F_{cu}\,\Sigma\,(A\,U)]\,\bar{\Phi}_k}{(24 - t_o)\,F_{cu}\,\Sigma\,(A\,U) + t_o\,F_{cy}\,\Sigma\,(A\,Y) + 24\,c_p\,\rho\,q_v} \tag{5.60}$$

where $\Delta\Phi_k$ is the increase in cooling load (W), F_{cy} is the room admittance factor with respect to operative temperature, F_{cu} is the room conduction factor with respect to operative temperature, $\bar{\Phi}_k$ is the 24-hour mean of the continuous cooling load which would otherwise have occurred during the off period (W), t_o is the duration of plant operation including preheat (h), c_p is the specific heat capacity of air (J·kg^{-1}·K^{-1}), ρ is the density of air (kg·m^{-3}), q_v is the natural ventilation rate (m^3·s^{-1}), $\Sigma\,(A\ U)$ is the sum of the products of surface area and corresponding thermal transmittance over surfaces through which heat flow occurs (W·K^{-1}) and $\Sigma\,(A\ Y)$ is the sum of the products of surface areas and their corresponding thermal admittances (W·K^{-1}).

This correction must be added to cooling loads calculated on the basis of continuous plant operation. Alternatively, when the load in the 'off' period is small:

$$\Delta\Phi_k = \bar{\Phi}_k \tag{5.61}$$

The cooling loads due to solar gain given in Tables 5.19 to 5.24 allow for intermittent operation. In temperate climates the effect of ignoring intermittency for fabric, internal and infiltration loads is not usually significant as their overnight values constitute only a small fraction of the peak load.

5.10.4.4 Corrections for fluctuations in control temperature (Simple Model)

If the controlled temperature in the space is allowed to rise at the time of peak load then the room sensible load can be reduced.

For a rise in the operative temperature ($\Delta\theta_c$), the reduction in cooling load is approximately given by the following equation (see Appendix 5.A5, equation 5.194):

$$\Delta\Phi_k = [c_p\ \rho\ q_{vt} + F_{cy}\,\Sigma\,(A\ Y)]\ \Delta\theta_c \tag{5.62}$$

where $\Delta\Phi_k$ is the change in plant output (W), c_p is the specific heat capacity of air (J·kg^{-1}·K^{-1}), ρ is the density of air (kg·m^{-3}), q_{vt} is the total ventilation (i.e. mechanical plus infiltration) rate (m^3·s^{-1}), F_{cy} is the room admittance factor with respect to operative temperature, $\Sigma\,(A\ Y)$ is the sum of the products of surface areas and their corresponding thermal admittances (W·K^{-1}) and $\Delta\theta_c$ is the increase in operative temperature (K). F_{cy} is given by equation 5.41.

For a rise in the internal air temperature ($\Delta\theta_{ai}$), the reduction in cooling load is approximately given by the following (see Appendix 5.A5, equation 5.195):

$$\Delta\Phi_k = [c_p\ \rho\ q_{vt} + F_{ay}\,\Sigma\,(A\ Y)]\ \Delta\theta_{ai} \tag{5.63}$$

where F_{ay} is the room admittance factor with respect to the air node, $\Sigma\,(A\ Y)$ is the sum of the products of surface areas and their corresponding thermal admittances (W·K^{-1}) and $\Delta\theta_{ai}$ is the increase in internal air temperature (K). F_{ay} is given by equation 5.50.

5.10.4.5 Diversity factor for central plant

In most cases the peak load for each space (i.e. room) will depend on the time of day. An east facing space will probably have a maximum heat gain at 10:00 while that for a west facing space may occur at 17:00. The maximum space cooling demand is therefore determined by summing the individual space demands for each hour of the day and selecting the peak hourly demand.

How this demand is satisfied will depend on the type of system, see CIBSE Guide B[30], chapter 2: *Ventilation and air conditioning*. It is also important to recognise that the maximum load on the central plant may arise from the fresh air load rather than the peak coincident space cooling loads.

The sizing of plant for thermal storage systems using ice is covered in CIBSE TM18: *Ice storage*[46] and a review of the environmental benefits of storage systems may be found in CIBSE Research Report RR06: *Environmental benefits of thermal storage*[43].

5.10.4.6 Latent loads

The cooling loads calculated using the methods described in section 5.8.2 do not include latent cooling loads. Even when humidity levels are not actively controlled, systems with very cold cooling surface temperatures will condense more moisture from the space than those with higher surface temperatures, which will result in greater cooling loads.

5.10.4.7 Heat gain/loss to the distribution system

Cooling will be distributed from a central plant directly in the form of air to room terminals or indirectly as water or refrigerant to devices such as fan coil units, induction units or chilled ceilings. Heat gains to air and water based systems can be determined by methods described in CIBSE Guide C[40], chapter 3: *Heat transfer*. The effect of heat gains to refrigeration distribution systems should be discussed with the equipment manufacturer.

5.10.5 Special applications

The majority of the design work carried out by building services engineers is associated with 'conventional' spaces, i.e. rectilinear (often approximately cubical) rooms with walls composed of brick, concrete and insulation about 200 mm in thickness and with glazed areas of the order of 20–60% of the facade area. Occupancy is usually at least 8 hours with levels of internal gains appropriate to office spaces (see chapter 6: *Internal heat gains*).

However, situations will arise which deviate significantly from the norm such as highly glazed atria and buildings with high thermal mass being operated intermittently. The designer may also be expected to cope with emitters having unusual characteristics. The following sections offer some guidance on calculation techniques appropriate in such cases, i.e:

— atria (section 5.10.5.1)

— intermittent operation of buildings with high thermal mass (section 5.10.5.2)

— surface cooling and heating systems (section 5.10.5.3)

— natural convection cooling systems (section 5.10.5.4).

5.10.5.1 Atria

The distinctive features of atria are their height, large areas of glazing, and the possible requirement for local temperature control. Heat transfer by radiation is particularly important together with the need to take temperature gradients into consideration. Steady state heat loss calculations should be made using the Reference Model. Dynamic models will generally require the capability to predict the distribution of shortwave radiation within the space together with the direct transmission of radiation from the space, either to outside or to adjacent spaces. Computational fluid dynamics (section 5.9.3) may be helpful in predicting temperature gradients; however it is often sufficient to assume a linear gradient within the space.

5.10.5.2 Spaces with high thermal mass

To predict the long-term effects of thermal mass a transient model, will be necessary. At present CIBSE does not have any recommended tests related to the performance of transient models where high thermal mass is involved.

5.10.5.3 Surface heating and cooling

These systems usually comprise either (*a*) a heated/cooled plate or matrix in direct contact with the space, e.g. a chilled ceiling, or (*b*) ducts or pipes embodied within the structure. In both cases the steady state performance will require the use of the Reference or Basic Model, adapted to take account of the constant temperature surfaces; the algorithms contained in Appendix 5.A3 demonstrate how this may be achieved. The Simple (cyclic) Model can be used to give an approximate duty for a chilled ceiling. Manufacturer's data will indicate the proportions of radiant and convective heating. The calculation method given in Appendix 5.A5 may then be used.

The performance of systems using ducts or pipes embodied within the fabric can generally be assessed using a transient model. However, provided care is taken to assess the effect of the fabric on surface temperatures, peak cooling duty may be estimated using the Simple (cyclic) Model.

In all cases it is essential to take account of losses from the heating or cooling medium to the surrounding structure.

The fresh air supply may be by means of a conventional ceiling or wall mounted diffuser, or a floor level displacement or swirl diffuser. When determining the peak cooling load it may be assumed that, for both cases, the air is well mixed within the space.

5.10.5.4 Natural convective cooling systems ('chilled beams')

These systems comprise a finned tube element mounted at ceiling level. They cause a downward flowing plume of cooled air. It is usual to supply fresh air at floor level using a displacement terminal. For the purposes of calculating peak loads, such systems may be regarded as 100% convective and the air in the space assumed to be fully mixed. Therefore the Simple (dynamic) Model may be applied.

5.11 Solar cooling load tables

The following tables provide solar cooling loads for three UK regions: SE England (Bracknell, 51° 33′N) (Tables 5.19 and 5.20), NW England (Manchester/Aughton: 53° 33′N) (Tables 5.21 and 5.22) and lowland Scotland (Edinburgh/Mylnefield, 56° 00′N) (Tables 5.23 and 5.24). These tables are based on measured irradiances that were not exceeded on more than 2½% of occasions during the period 1976–1995. Similar tables (Tables 5.25 to 5.41) for latitudes from 0–60° N/S, based on theoretical predictions of irradiance, are given on the CD-ROM that accompanies this Guide. The calculation method used to produce the tabulated values is described in Appendix 5.A6. These tables may be used in order to assess the plant loads at an early stage in the design.

The tabulated values apply to 'lightweight' buildings (i.e. those with a fast response to solar radiation) with single clear glass and assume that control of the cooling system will be based on the operative temperature. Multiplying factors are given at the foot of each table to adjust the tabulated cooling loads for application to buildings having a slow response to solar radiation and for other types of glazing. The glazing types should be regarded as generic, see Appendix 5.A7, Table 5.51. The term 'blind' refers to a generic shading device having a transmission coefficient of 0.2 and a reflectivity of 0.4. An additional factor is given to enable the values to be adjusted for situations where the cooling system will be controlled at the air node.

Table 5.19 Solar cooling loads for fast-response building with single clear glazing: SE England; unshaded

Date	Orien-tation	Solar cooling load at stated sun time / W·m^{-2}											Orien-tation
		07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	
Jan 29	N	6	12	20	29	36	40	40	37	29	19	11	N
	NE	9	29	26	32	37	41	41	38	29	20	12	NE
	E	28	103	201	229	171	77	61	54	45	35	28	E
	SE	61	148	292	408	452	406	316	193	96	69	60	SE
	S	77	128	233	366	476	514	507	422	299	190	111	S
	SW	51	57	70	109	218	327	404	402	328	228	117	SW
	W	20	26	34	43	50	63	86	167	191	157	78	W
	NW	7	13	20	30	37	41	41	38	33	26	27	NW
	Horiz.	29	43	76	133	182	206	203	168	115	67	40	Horiz.
Feb 26	N	16	24	36	49	57	62	60	55	46	34	22	E
	NE	49	78	60	58	60	65	63	59	49	37	25	NE
	E	100	203	277	281	211	107	90	79	70	57	45	E
	SE	126	236	361	446	472	427	325	203	106	89	75	SE
	S	110	163	269	388	484	533	522	473	389	267	155	S
	SW	74	83	99	127	229	350	434	476	468	380	241	SW
	W	45	53	65	78	86	101	125	230	305	298	209	W
	NW	22	30	42	55	63	68	69	86	62	68	83	NW
	Horiz.	61	93	155	226	282	313	306	274	221	150	86	Horiz.
Mar 29	N	38	50	64	74	83	86	86	81	72	60	47	N
	NE	160	181	135	83	97	98	98	93	84	72	58	NE
	E	252	355	400	361	261	147	133	120	111	99	84	E
	SE	231	353	456	498	483	426	319	196	142	121	105	SE
	S	109	176	283	384	457	505	497	454	377	270	161	S
	SW	104	118	135	162	224	347	443	501	514	467	358	SW
	W	87	101	115	126	135	150	175	293	389	420	370	W
	NW	55	69	83	93	102	105	106	104	97	154	193	NW
	Horiz.	134	206	292	365	417	449	443	413	358	282	195	Horiz.
Apr 28	N	70	87	95	103	110	115	116	110	103	93	86	N
	NE	322	308	236	143	146	143	143	137	131	120	106	NE
	E	439	501	510	430	314	192	179	167	160	149	136	E
	SE	331	433	511	515	487	413	301	186	169	149	135	SE
	S	115	157	255	346	418	457	455	409	339	236	142	S
	SW	133	149	162	184	212	330	437	503	535	505	434	SW
	W	138	154	167	176	184	199	226	351	465	518	513	W
	NW	104	120	133	142	149	154	155	159	167	258	325	NW
	Horiz.	251	342	437	499	548	572	570	540	496	417	329	Horiz.
May 29	N	118	121	126	131	137	140	139	135	130	123	120	N
	NE	382	362	290	190	174	166	165	161	155	146	134	NE
	E	475	519	518	450	329	208	194	184	179	169	157	E
	SE	321	408	471	485	448	367	258	167	166	152	140	SE
	S	110	129	201	289	355	388	384	345	274	182	116	S
	SW	142	156	168	183	193	290	393	465	497	477	420	SW
	W	170	184	196	204	210	223	247	371	484	544	554	W
	NW	142	156	168	176	182	185	185	197	221	321	394	NW
	Horiz.	326	417	507	578	615	625	621	605	564	492	407	Horiz.
Jun 21	N	144	139	142	147	151	154	154	151	147	144	143	N
	NE	440	413	320	211	190	180	181	178	173	165	154	NE
	E	533	579	550	469	343	220	207	199	195	186	176	E
	SE	345	437	482	487	446	363	253	168	171	159	148	SE
	S	111	123	191	276	342	376	374	331	261	174	116	S
	SW	147	159	170	182	190	282	389	460	492	476	424	SW
	W	185	197	207	215	220	232	257	382	497	561	578	W
	NW	160	172	182	190	195	198	199	214	244	347	424	NW
	Horiz.	365	463	545	609	642	650	648	629	590	521	439	Horiz.
Jul 4	N	126	126	130	136	142	146	146	141	136	130	127	N
	NE	394	372	294	196	179	171	171	166	161	152	139	NE
	E	480	525	509	437	323	210	196	187	182	172	160	E
	SE	314	400	450	458	422	346	244	162	162	149	136	SE
	S	102	118	184	265	328	359	360	320	253	168	108	S
	SW	135	149	160	174	184	272	372	438	467	453	401	SW
	W	168	182	192	202	209	221	245	360	467	529	542	W
	NW	144	157	168	177	183	187	188	200	226	322	394	NW
	Horiz.	329	420	503	565	599	606	608	590	551	487	407	Horiz.
Aug 4	N	81	92	99	107	113	117	117	115	106	97	91	N
	NE	316	314	244	153	148	143	144	141	132	123	109	NE
	E	414	486	487	420	309	189	176	167	159	149	135	E
	SE	302	406	471	484	459	378	270	173	160	144	130	SE
	S	107	137	223	311	382	410	405	367	298	203	122	S
	SW	125	139	152	170	189	294	393	460	490	452	372	SW
	W	134	148	161	170	176	189	215	334	440	480	454	W
	NW	105	120	132	141	147	150	152	161	171	257	303	NW
	Horiz.	261	354	442	509	557	567	560	539	498	417	325	Horiz.

Table continues

Table 5.19 Solar cooling loads for fast-response building with single clear glazing: SE England; unshaded — *continued*

Date	Orientation	Solar cooling load at stated sun time / W·m^{-2}											Orientation
		07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	
Sep 4	N	43	55	65	78	86	88	87	82	74	64	54	N
	NE	241	244	167	97	109	107	106	101	93	83	70	NE
	E	365	462	465	394	277	159	144	132	124	114	101	E
	SE	304	433	508	521	482	410	310	189	147	129	116	SE
	S	106	175	283	379	439	470	478	439	362	257	151	S
	SW	110	124	138	165	215	329	436	502	520	481	375	SW
	W	99	112	124	137	146	158	183	306	408	450	404	W
	NW	65	78	91	103	111	113	113	112	109	181	225	NW
	Horiz.	171	258	345	413	453	471	477	451	398	321	230	Horiz.
Oct 4	N	19	30	42	52	59	64	64	61	53	40	28	N
	NE	91	132	87	66	67	71	70	68	59	47	35	NE
	E	172	324	381	351	236	119	103	94	86	73	61	E
	SE	183	348	472	535	504	415	315	200	115	104	89	SE
	S	120	194	315	435	499	507	499	460	383	266	165	S
	SW	79	90	105	124	222	332	419	470	473	395	285	SW
	W	54	64	76	86	94	109	134	243	324	325	266	W
	NW	28	38	50	61	68	73	73	72	74	90	117	NW
	Horiz.	82	136	216	293	337	347	342	316	264	188	120	Horiz.
Nov 4	N	7	13	22	32	38	42	43	39	31	21	13	N
	NE	12	43	34	37	40	44	44	41	33	23	14	NE
	E	37	153	279	288	196	85	70	61	53	43	35	E
	SE	73	203	391	501	507	447	328	204	105	81	70	SE
	S	89	158	295	439	527	566	527	457	352	235	132	S
	SW	58	64	79	117	233	356	421	438	395	298	151	SW
	W	25	31	40	50	56	69	95	187	235	214	110	W
	NW	8	14	23	33	39	44	44	41	37	35	37	NW
	Horiz.	36	55	105	173	223	247	234	200	147	90	50	Horiz.
Dec 4	N	4	5	12	21	28	32	32	27	20	11	4	N
	NE	5	6	23	23	28	32	32	27	20	11	5	NE
	E	17	26	139	209	156	63	48	39	32	23	17	E
	SE	50	62	219	395	448	403	306	191	92	57	50	SE
	S	70	79	190	362	478	518	495	429	314	159	70	S
	SW	48	49	62	109	221	329	392	405	342	179	48	SW
	W	16	16	24	32	40	50	73	154	185	112	16	W
	NW	5	5	13	21	28	32	32	27	23	22	5	NW
	Horiz.	20	22	45	96	143	164	158	129	84	39	20	Horiz.

Glazing configuration (inside to outside)	*G*-value	Correction factor for stated building response		Glazing configuration (inside to outside)	*G*-value	Correction factor for stated building response	
		Fast	Slow			Fast	Slow
Clear	0.84	1.00	0.88	Low-E/clear	0.66	0.79	0.68
Absorbing	0.62	0.80	0.69	Low-E/absorbing	0.46	0.59	0.50
Clear/clear	0.72	0.83	0.72	Low-E/clear/clear	0.60	0.69	0.59
Clear/reflecting	0.41	0.50	0.43	Low-E/clear/absorbing	0.40	0.49	0.42
Clear/absorbing	0.49	0.60	0.52				
Clear/clear/clear	0.64	0.71	0.60				
Clear/clear/reflecting	0.35	0.43	0.36				
Clear/clear/absorbing	0.42	0.50	0.42				
Air node correction factor		0.86	0.83				

Table 5.20 Solar cooling loads for fast-response building with single clear glazing: SE England; intermittent shading

Date	Orientation	Solar cooling load at stated sun time / W·m^{-2}											Orientation
		07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	
Jan 29	N	6	11	18	26	33	37	37	35	27	18	11	N
	NE	7	27	24	30	34	38	38	36	28	19	12	NE
	E	11	164	149	131	41	62	47	40	33	24	16	E
	SE	22	210	238	287	276	225	74	155	62	36	27	SE
	S	29	71	305	281	324	331	291	214	74	142	67	S
	SW	18	22	31	140	183	245	263	228	165	54	83	SW
	W	16	21	28	36	43	52	67	234	50	148	74	W
	NW	7	11	18	27	34	38	38	36	31	24	27	NW
	Horiz.	20	31	60	113	248	142	55	153	104	59	32	Horiz.
Feb 26	N	15	22	33	45	52	57	56	52	44	33	21	N
	NE	42	72	56	53	55	60	59	55	47	35	24	NE
	E	73	287	191	161	55	87	71	61	53	42	31	E
	SE	77	321	271	304	291	234	80	162	68	53	39	SE
	S	48	180	210	283	328	335	310	261	186	67	98	S
	SW	33	41	54	71	301	262	301	308	267	181	64	SW
	W	25	33	43	55	63	74	181	189	204	161	56	W
	NW	20	28	39	51	58	63	62	82	55	61	79	NW
	Horiz.	34	60	117	297	200	204	188	157	59	122	61	Horiz.
Mar 29	N	34	46	58	68	76	79	80	76	68	57	45	N
	NE	144	167	129	76	90	90	91	86	78	68	55	NE
	E	216	262	256	202	77	118	105	93	85	74	61	E
	SE	297	271	315	318	290	228	83	147	94	75	60	SE
	S	47	201	214	271	311	318	297	253	187	68	106	S
	SW	58	71	85	105	282	267	316	337	319	258	99	SW
	W	45	58	71	81	89	100	222	230	268	252	188	W
	NW	51	64	77	87	95	98	99	98	83	138	180	NW
	Horiz.	76	240	211	252	280	285	270	238	191	74	142	Horiz.
Apr 28	N	64	80	87	95	101	106	107	102	96	86	80	N
	NE	223	189	85	121	124	121	122	117	111	101	90	NE
	E	325	347	312	242	104	150	139	127	121	111	99	E
	SE	261	322	341	331	292	224	93	138	122	103	91	SE
	S	59	177	194	248	284	293	271	230	166	62	92	S
	SW	68	83	95	112	238	249	304	335	329	291	216	SW
	W	74	89	101	109	115	127	258	269	320	332	285	W
	NW	72	87	99	107	114	118	119	122	209	206	214	NW
	Horiz.	264	244	294	332	355	362	347	321	274	215	94	Horiz.
May 29	N	109	111	116	121	126	129	129	125	120	114	109N	
	NE	263	227	107	163	148	140	139	135	130	122	111	NE
	E	344	355	325	255	116	163	150	140	136	127	116	E
	SE	251	301	323	310	265	197	88	122	122	109	98	SE
	S	70	80	253	217	248	254	236	195	74	143	79	S
	SW	80	93	104	117	205	222	280	312	310	280	223	SW
	W	96	109	120	128	133	141	269	280	334	354	330	W
	NW	92	105	116	123	129	132	132	139	260	243	266	NW
	Horiz.	349	293	347	382	396	398	390	368	324	269	124	Horiz.
Jun 21	N	134	128	131	135	140	142	143	140	136	133	129	N
	NE	303	255	124	182	161	152	152	150	146	138	129	NE
	E	388	385	343	268	128	173	160	153	149	142	132	E
	SE	272	314	328	311	264	195	90	123	127	116	107	SE
	S	74	77	239	209	240	248	229	187	73	137	79	S
	SW	85	97	106	117	198	220	277	309	309	283	229	SW
	W	107	119	128	136	140	148	276	289	345	369	350	W
	NW	106	117	127	134	139	141	142	152	284	264	291	NW
	Horiz.	251	311	360	391	404	406	397	375	334	280	218	Horiz.
Jul 4	N	117	116	120	125	131	135	135	131	126	121	115	N
	NE	272	233	113	169	153	144	145	140	136	128	116	NE
	E	350	354	319	251	119	165	153	144	139	131	120	E
	SE	250	294	310	296	253	107	204	122	123	111	100	SE
	S	66	74	231	201	230	238	220	181	69	132	73	S
	SW	80	92	102	115	114	328	267	296	297	271	219	SW
	W	96	109	119	127	133	142	265	271	325	346	327	W
	NW	94	107	117	125	131	135	135	142	265	245	269	NW
	Horiz.	355	294	342	373	385	389	382	360	321	269	126	Horiz.
Aug 4	N	74	85	91	99	104	108	108	106	99	91	84	N
	NE	225	193	88	130	126	121	122	120	112	104	91	NE
	E	314	332	303	238	104	148	135	127	120	111	99	E
	SE	244	298	320	313	270	203	86	127	116	101	88	SE
	S	64	84	281	233	262	267	249	210	79	160	83	S
	SW	67	81	92	108	212	224	278	309	299	253	187	SW
	W	75	88	99	108	113	122	248	257	300	298	254	W
	NW	75	88	100	108	114	117	118	125	216	197	201	NW
	Horiz.	278	250	301	340	356	357	346	323	276	214	96	Horiz.

Table continues

Table 5.20 Solar cooling loads for fast-response building with single clear glazing: SE England; intermittent shading — *continued*

Date	Orientation	Solar cooling load at stated sun time / $W \cdot m^{-2}$											Orientation
		07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	
Sep 4	N	39	51	60	71	79	81	81	76	69	60	51	N
	NE	180	77	158	87	100	97	97	93	86	77	66	NE
	E	294	318	288	219	88	127	113	102	95	86	75	E
	SE	252	317	341	326	285	224	88	140	101	83	71	SE
	S	46	203	213	263	291	303	288	244	179	65	99	S
	SW	51	63	75	96	250	248	304	328	315	257	180	SW
	W	52	64	75	87	95	103	226	240	284	272	220	W
	NW	45	57	68	80	88	90	90	89	77	236	148	NW
	Horiz.	105	305	244	278	296	305	295	264	217	87	171	Horiz.
Oct 4	N	18	27	38	47	54	59	59	57	50	38	27	N
	NE	78	122	82	60	61	65	65	63	56	44	33	NE
	E	261	240	242	183	60	90	75	67	60	48	37	E
	SE	248	273	333	334	286	221	76	151	67	57	44	SE
	S	55	227	244	302	319	318	299	256	184	67	107	S
	SW	38	47	60	68	285	251	295	310	274	207	75	SW
	W	32	41	52	62	68	79	190	201	220	194	70	W
	NW	26	35	46	56	63	68	68	67	66	80	111	NW
	Horiz.	44	92	279	209	223	224	211	182	68	150	87	Horiz.
Nov 4	N	6	12	20	29	35	39	40	37	30	21	13	N
	NE	8	40	32	34	36	40	41	38	31	22	14	NE
	E	14	241	194	155	48	67	53	45	38	28	20	E
	SE	25	296	302	330	306	238	78	160	65	42	32	SE
	S	27	183	238	312	352	342	302	238	161	58	76	S
	SW	20	26	35	147	198	257	281	265	212	70	114	SW
	W	12	18	26	35	41	49	147	149	152	51	98	W
	NW	8	13	21	30	36	40	41	38	34	31	37	NW
	Horiz.	22	37	82	238	163	163	61	178	129	76	37	Horiz.
Dec 4	N	4	4	11	19	25	29	30	25	19	11	4	N
	NE	5	5	22	21	26	29	30	25	19	11	5	NE
	E	10	10	211	122	38	53	39	30	24	16	10	E
	SE	16	92	213	279	271	217	142	44	57	23	16	SE
	S	24	24	262	279	325	324	290	223	74	114	24	S
	SW	18	18	26	145	187	241	263	240	78	149	18	SW
	W	12	12	18	26	33	40	55	222	46	108	12	W
	NW	5	5	11	19	26	29	30	25	21	22	5	NW
	Horiz.	20	20	39	86	131	153	149	123	82	39	20	Horiz.

Glazing configuration (inside to outside)	*G*-value	Correction factor for stated building response	
		Fast	Slow
Clear/blind	0.20	0.73	0.82
Absorbing/blind	0.17	0.58	0.62
Clear/clear/blind	0.16	0.57	0.62
Clear/blind/clear	0.30	0.70	0.73
Clear/blind/reflecting	0.21	0.46	0.47
Clear/blind/absorbing	0.24	0.54	0.56
Clear/reflecting/blind	0.11	0.35	0.38
Clear/absorbing/blind	0.13	0.41	0.45
Clear/clear/clear/blind	0.13	0.47	0.52
Clear/clear/blind/clear	0.24	0.56	0.58
Clear/clear/blind/reflecting	0.16	0.37	0.38
Clear/clear/blind/absorbing	0.19	0.42	0.43
Clear/clear/reflecting/blind	0.09	0.29	0.32
Clear/clear/absorbing/blind	0.10	0.34	0.37
Low-E/clear/blind	0.15	0.54	0.58
Low-E/absorbing/blind	0.12	0.40	0.44
Low-E/clear/clear/blind	0.13	0.46	0.51
Low-E/clear/blind/clear	0.23	0.55	0.57
Low-E/clear/blind/absorbing	0.18	0.42	0.44
Low-E/clear/absorbing/blind	0.10	0.36	0.38
Blind/clear	0.48	1.00	1.03
Blind/absorbing	0.40	0.81	0.83
Blind/clear/clear	0.49	0.95	0.95
Blind/clear/reflecting	0.31	0.61	0.61
Blind/clear/absorbing	0.35	0.71	0.71
Blind/low-E/clear	0.46	0.92	0.92
Blind/low-E/absorbing	0.32	0.67	0.68
Blind/clear/clear/clear	0.47	0.87	0.86
Blind/clear/clear/reflecting	0.28	0.55	0.54
Blind/clear/clear/absorbing	0.32	0.62	0.62
Blind/low-E/clear/clear	0.45	0.85	0.84
Blind/low-E/clear/absorbing	0.30	0.60	0.60
Air node correction factor:			
— internal blind		0.91	0.88
— mid-pane blind		0.87	0.83
— external blind		0.88	0.85

Table 5.21 Solar cooling loads for fast-response building with single clear glazing: NW England; unshaded

Date	Orien-tation	Solar cooling load at stated sun time / W·m^{-2}											Orien-tation
		07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	
Jan 29	N	6	11	18	28	35	39	40	35	27	16	10	N
	NE	9	27	37	31	36	40	41	36	28	18	11	NE
	E	24	80	152	189	156	71	58	48	40	30	24	E
	SE	53	119	224	338	406	370	298	184	88	60	53	SE
	S	70	110	188	310	430	471	477	407	284	172	100	S
	SW	48	53	65	101	204	304	383	389	313	203	105	SW
	W	19	24	31	41	48	60	84	160	181	140	68	W
	NW	7	12	19	29	36	40	41	36	33	39	24	NW
	Horiz.	25	36	61	108	156	178	179	148	101	57	34	Horiz
Feb 26	N	16	23	36	47	53	57	58	52	45	34	21	N
	NE	51	85	61	56	56	61	62	55	48	38	25	NE
	E	103	208	301	301	213	104	89	77	70	60	46	E
	SE	131	245	395	485	492	440	337	205	112	94	78	SE
	S	115	175	293	420	508	552	539	468	405	282	163	S
	SW	75	83	101	130	237	359	444	465	483	396	236	SW
	W	44	52	65	76	82	97	122	223	310	307	202	W
	NW	21	28	42	53	58	62	63	58	60	68	86	NW
	Horiz.	59	88	153	222	270	296	291	252	212	145	82	Horiz.
Mar 29	N	39	52	63	76	88	88	88	82	74	63	48	N
	NE	151	170	127	81	101	98	98	92	85	73	57	NE
	E	239	335	385	365	269	148	133	119	111	100	84	E
	SE	222	338	446	513	502	421	324	202	145	123	105	SE
	S	111	178	284	400	480	498	498	449	375	262	159	S
	SW	101	115	130	161	234	344	438	486	495	426	319	SW
	W	82	96	108	121	134	146	170	280	368	378	322	W
	NW	53	67	79	92	104	104	104	100	94	142	170	NW
	Horiz.	130	196	276	357	413	422	421	388	339	261	181	Horiz.
Apr 28	N	72	90	98	109	116	118	118	114	107	98	91	N
	NE	299	296	222	142	148	143	143	140	132	123	110	NE
	E	409	485	482	418	308	192	179	168	160	151	138	E
	SE	317	428	493	509	479	417	316	198	173	153	140	E
	S	121	167	265	357	423	466	477	425	351	247	154	SE
	SW	132	148	163	189	222	336	448	504	528	485	405	S
	W	133	149	163	173	181	195	219	342	448	486	466	SW
	NW	101	118	131	142	149	151	151	156	158	242	296	NW
	Horiz.	244	334	416	481	522	549	561	524	478	400	317	Horiz.
May 29	N	124	128	132	139	144	148	150	146	140	135	138	N
	NE	402	382	289	190	181	175	177	173	166	159	152	NE
	E	503	560	532	459	339	217	207	197	190	183	176	E
	SE	346	446	494	506	472	393	284	185	180	165	158	SE
	S	122	142	223	316	384	420	421	379	305	210	13	S9
	SW	148	161	173	190	204	310	418	490	519	495	418	SW
	W	170	183	194	203	209	222	249	374	488	547	530	W
	NW	142	155	167	175	181	184	187	199	215	315	378	NW
	Horiz.	342	436	514	580	619	633	635	614	573	502	413	Horiz.
Jun 21	N	136	139	144	149	155	158	157	151	145	141	144	N
	NE	416	391	307	206	195	186	185	179	172	165	156	NE
	E	508	554	535	462	343	225	212	200	193	186	177	E
	SE	336	428	480	490	455	379	268	175	172	159	151	SE
	S	115	134	210	296	365	400	395	351	281	192	125	S
	SW	147	162	175	190	201	300	401	467	503	496	434	SW
	W	180	194	207	214	221	234	257	376	492	568	576	W
	NW	153	168	181	188	194	198	198	208	229	336	415	NW
	Horiz.	356	450	531	593	631	644	638	615	583	525	441	Horiz.
Jul 4	N	135	137	141	148	155	158	155	150	141	135	137	N
	NE	412	386	298	202	192	183	180	176	167	158	150	NE
	E	506	553	527	457	338	223	208	197	188	180	171	E
	SE	337	429	476	488	448	377	270	174	167	154	146	SE
	S	116	134	210	298	361	398	402	359	285	191	122	S
	SW	150	163	175	193	205	303	409	478	512	496	431	SW
	W	186	199	211	220	227	240	261	385	500	568	573	W
	NW	159	172	184	193	199	202	200	211	228	333	410	NW
	Horiz.	351	442	520	584	614	631	638	617	580	513	427	Horiz.
Aug 4	N	81	95	103	112	116	121	121	116	109	99	96	N
	NE	312	319	231	151	149	146	147	142	135	124	114	NE
	E	413	505	464	403	305	192	180	168	160	150	140	E
	SE	306	426	457	471	459	391	286	180	164	145	135	SE
	S	112	146	233	318	393	430	425	389	313	214	134	S
	SW	130	145	160	181	204	313	411	479	497	456	392	SW
	W	140	155	169	178	183	198	223	341	440	477	472	W
	NW	111	126	139	149	153	157	159	164	171	251	313	NW
	Horiz.	261	359	425	484	539	559	554	535	486	407	331	Horiz.

Table continues

Table 5.21 Solar cooling loads for fast-response building with single clear glazing: NW England; unshaded — *continued*

Date	Orientation	07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	Orientation
		Solar cooling load at stated sun time / $W \cdot m^{-2}$											
Sep 4	N	46	58	71	83	89	93	92	87	77	68	58	N
	NE	216	217	154	96	109	109	108	103	93	84	72	NE
	E	327	409	415	384	282	160	144	131	122	112	100	E
	SE	280	393	462	515	506	434	318	200	150	131	117	SE
	S	112	177	279	389	471	505	487	457	383	273	161	S
	SW	114	127	144	174	230	353	442	514	538	492	370	SW
	W	101	114	128	140	147	163	186	309	415	455	391	W
	NW	67	80	95	106	113	117	116	114	108	178	218	NW
	Horiz.	167	242	320	399	454	475	461	442	393	318	227	Horiz.
Oct 4	N	20	30	42	55	63	66	67	61	52	40	28	N
	NE	78	110	77	67	68	71	72	66	57	45	34	NE
	E	143	261	311	293	210	114	100	88	79	67	55	E
	SE	155	284	388	444	435	391	306	189	107	94	80	SE
	S	110	169	272	373	439	479	481	420	357	243	152	S
	SW	74	85	100	122	212	320	404	429	437	355	254	SW
	W	50	60	72	85	94	107	131	223	298	290	234	W
	NW	27	37	49	62	70	74	74	70	71	82	105	NW
	Horiz.	74	118	183	248	291	315	317	281	237	168	109	Horiz.
Nov 4	N	7	14	23	33	41	44	45	41	31	21	13	N
	NE	11	39	33	38	42	45	46	42	32	22	14	NE
	E	30	116	210	226	159	80	66	57	47	37	29	E
	SE	60	157	295	388	390	387	308	190	93	69	59	SE
	S	76	131	233	348	412	490	493	424	316	205	115	S
	SW	53	60	74	108	200	317	398	410	357	260	132	SW
	W	23	30	39	49	57	69	93	177	212	185	95	W
	NW	8	15	24	34	42	45	46	42	36	32	34	NW
	Horiz.	31	49	87	139	176	209	209	178	127	77	44	Horiz.
Dec 4	N	4	4	10	18	25	28	27	24	17	10	4	N
	NE	4	6	21	20	25	28	27	24	17	10	4	NE
	E	14	22	110	170	134	56	40	34	27	20	14	E
	SE	43	52	177	324	386	356	277	168	80	50	43	SE
	S	60	68	159	300	414	458	448	373	263	134	60	S
	SW	42	42	53	94	194	291	355	351	284	147	42	SW
	W	13	14	20	27	34	44	64	132	153	90	13	W
	NW	4	5	11	18	25	28	27	24	19	20	4	NW
	Horiz.	16	18	36	72	112	132	128	103	65	32	16	Horiz.

Glazing configuration (inside to outside)	*G*-value	Correction factor for stated building response: Fast	Slow	Glazing configuration (inside to outside)	*G*-value	Correction factor for stated building response: Fast	Slow
Clear	0.84	1.00	0.90	Low-E/clear	0.66	0.79	0.68
Absorbing	0.62	0.80	0.69	Low-E/absorbing	0.46	0.59	0.51
Clear/clear	0.72	0.83	0.72	Low-E/clear/clear	0.60	0.69	0.59
Clear/reflecting	0.41	0.50	0.43	Low-E/clear/absorbing	0.40	0.49	0.42
Clear/absorbing	0.49	0.60	0.52				
Clear/clear/clear	0.64	0.71	0.60				
Clear/clear/reflecting	0.35	0.43	0.37				
Clear/clear/absorbing	0.42	0.50	0.43				
Air node correction factor		0.86	0.83				

Table 5.22 Solar cooling loads for fast-response building with single clear glazing: NW England; intermittent shading

Date	Orien-tation	Solar cooling load at stated sun time / W·m^{-2}											Orien-tation
		07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	
Jan 29	N	6	10	16	25	32	36	37	33	25	16	10	N
	NE	7	23	35	28	33	37	38	34	26	17	11	NE
	E	15	65	218	48	145	64	51	42	35	26	20	E
	SE	19	159	191	252	251	211	69	150	58	32	24	SE
	S	25	60	243	249	295	309	279	205	69	127	59	S
	SW	23	27	34	61	281	237	259	225	76	174	80	SW
	W	18	22	28	37	45	53	69	144	170	135	67	W
	NW	7	11	17	26	33	37	38	34	29	37	23	NW
	Horiz.	23	33	54	97	144	164	168	141	97	55	33	Horiz.
Feb 26	N	14	20	33	43	48	52	54	48	42	33	21	N
	NE	43	78	56	52	52	56	57	52	45	36	24	NE
	E	75	301	205	166	56	84	70	58	52	43	31	E
	SE	103	215	293	318	297	238	79	158	68	51	36	SE
	S	50	198	229	299	341	347	309	267	196	70	104	S
	SW	30	37	52	149	197	266	293	310	275	178	60	SW
	W	25	31	43	54	59	69	175	189	210	159	54	W
	NW	19	26	38	49	54	58	59	54	53	60	83	NW
	Horiz.	37	61	119	294	193	198	178	68	184	123	61	Horiz.
Mar 29	N	35	47	57	69	81	81	82	76	69	59	46	N
	NE	136	157	120	73	93	91	91	86	79	69	54	NE
	E	327	247	252	205	75	116	102	89	82	72	58	E
	SE	282	262	319	331	291	230	84	152	97	77	60	SE
	S	49	202	221	286	312	317	295	251	182	66	103	S
	SW	56	69	82	103	294	265	309	326	296	231	90	SW
	W	56	69	80	92	104	112	230	233	258	234	94	W
	NW	49	62	73	85	97	97	97	94	81	128	159	NW
	Horiz.	79	232	208	254	271	273	258	229	89	210	135	Horiz.
Apr 28	N	65	82	90	100	107	109	109	106	100	91	85	N
	NE	222	105	209	130	137	131	131	129	122	114	103	NE
	E	313	330	301	237	103	151	139	129	122	113	103	E
	SE	256	312	335	326	291	233	97	149	126	107	96	SE
	S	70	107	332	259	295	312	290	245	93	196	109	S
	SW	70	85	97	119	254	258	309	334	319	274	200	SW
	W	79	95	107	117	124	133	262	268	310	311	262	W
	NW	75	90	102	113	120	121	121	125	118	312	197	NW
	Horiz.	262	238	287	322	345	358	344	315	268	112	242	Horiz.
May 29	N	114	118	121	128	133	136	139	135	129	124	127	N
	NE	280	234	114	163	155	148	150	147	141	135	129	NE
	E	372	373	335	264	124	171	162	153	146	140	134	E
	SE	275	320	338	326	283	216	97	137	133	120	113	SE
	S	78	89	280	236	269	278	260	217	84	166	98	S
	SW	84	96	106	121	221	238	296	327	324	282	225	SW
	W	99	110	121	129	134	143	276	285	340	345	319	W
	NW	95	107	118	126	131	134	137	145	255	237	259	NW
	Horiz.	370	302	351	385	401	406	398	374	332	274	129	Horiz.
Jun 21	N	126	128	133	137	142	146	146	140	134	131	131	N
	NE	282	239	117	171	160	151	151	145	139	133	125	NE
	E	371	374	337	267	129	178	166	155	149	142	134	E
	SE	265	311	328	315	273	206	93	129	128	116	108	SE
	S	75	84	262	224	256	263	242	201	79	151	86	S
	SW	83	97	109	122	215	228	282	314	320	291	234	SW
	W	104	118	129	137	142	151	276	286	349	370	349	W
	NW	102	115	127	134	140	143	143	148	268	258	286	NW
	Horiz.	244	303	351	383	399	401	389	369	335	283	222	Horiz.
Jul 4	N	125	126	130	136	143	146	143	140	131	125	124	N
	NE	285	240	120	173	164	155	153	149	141	133	125	NE
	E	370	370	333	263	127	177	162	153	144	136	129	E
	SE	266	309	326	311	270	206	93	129	124	111	103	SE
	S	76	84	264	222	254	265	247	204	79	150	83	S
	SW	86	98	109	124	218	233	289	321	323	289	234	SW
	W	109	121	132	140	146	155	280	291	350	368	351	W
	NW	106	118	129	138	144	147	144	151	263	255	287	NW
	Horiz.	366	294	342	371	387	396	388	366	326	271	212	Horiz.
Aug 4	N	74	87	94	103	107	111	112	108	102	92	89	N
	NE	231	107	210	131	130	127	128	123	117	108	99	NE
	E	325	326	289	233	104	150	140	128	122	112	104	E
	SE	256	297	310	310	277	214	90	134	120	102	93	SE
	S	66	92	291	239	274	280	263	221	83	169	93	S
	SW	70	84	97	114	230	236	291	315	300	263	202	SW
	W	78	92	104	114	117	128	254	258	297	305	272	W
	NW	75	89	102	111	114	119	120	123	122	315	210	NW
	Horiz.	283	245	287	327	351	353	344	317	269	217	99	Horiz.

Table continues

Table 5.22 Solar cooling loads for fast-response building with single clear glazing: NW England; intermittent shading — *continued*

Date	Orientation	Solar cooling load at stated sun time / $W{\cdot}m^{-2}$											Orientation
		07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	
Sep 4	N	42	53	65	76	82	86	85	81	72	64	55	N
	NE	161	71	145	87	99	100	99	95	86	77	67	NE
	E	261	282	272	219	87	129	115	103	94	86	75	E
	SE	228	285	327	336	302	233	89	151	103	85	73	SE
	S	49	200	215	279	314	313	297	257	190	69	106	S
	SW	54	66	80	103	273	256	310	339	325	256	179	SW
	W	54	65	79	90	96	108	231	244	289	267	215	W
	NW	59	71	84	96	102	106	105	104	89	184	75	NW
	Horiz.	103	280	232	277	300	298	287	261	215	87	169	Horiz.
Oct 4	N	18	27	38	50	57	61	62	57	49	38	27	N
	NE	68	102	72	62	63	66	67	62	54	43	32	NE
	E	213	195	201	159	53	88	76	64	57	45	35	E
	SE	204	224	275	283	262	214	73	145	65	54	41	SE
	S	51	194	208	261	295	307	278	238	170	61	99	S
	SW	35	45	58	70	275	244	273	286	249	184	68	SW
	W	33	43	53	65	73	83	98	313	202	175	67	W
	NW	25	34	45	57	65	68	69	65	64	73	100	NW
	Horiz.	40	79	233	178	200	207	190	163	61	134	78	Horiz.
Nov 4	N	7	13	21	30	38	41	42	38	29	20	13	N
	NE	8	36	31	35	39	42	43	39	30	21	14	NE
	E	15	183	153	55	143	68	55	46	37	29	21	E
	SE	21	223	232	253	254	219	72	154	61	37	28	SE
	S	29	75	302	247	300	320	290	224	78	158	72	S
	SW	21	28	37	63	275	242	267	245	189	64	101	SW
	W	15	21	29	38	46	54	70	248	136	48	87	W
	NW	8	14	22	31	39	42	43	39	33	29	33	NW
	Horiz.	25	40	74	124	158	280	60	166	118	71	39	Horiz.
Dec 4	N	4	4	9	16	22	26	25	22	16	10	4	N
	NE	4	4	19	18	23	26	25	23	17	10	4	NE
	E	10	10	170	41	125	50	35	29	23	17	10	E
	SE	19	19	267	242	242	201	65	139	55	26	19	SE
	S	21	21	215	239	285	292	256	189	62	95	21	S
	SW	18	18	25	56	270	221	234	204	68	123	18	SW
	W	13	13	18	25	32	38	51	119	146	89	13	W
	NW	4	4	10	16	23	26	25	23	18	20	4	NW
	Horiz.	16	16	31	65	103	122	121	99	63	31	16	Horiz.

Glazing configuration (inside to outside)	*G*-value	Correction factor for stated building response		Glazing configuration (inside to outside)	*G*-value	Correction factor for stated building response	
		Fast	Slow			Fast	Slow
Clear/blind	0.20	0.78	0.86	Low-E/clear/clear/blind	0.13	0.48	0.58
Absorbing/blind	0.17	0.61	0.66	Low-E/clear/blind/clear	0.23	0.56	0.60
Clear/clear/blind	0.16	0.60	0.65	Low-E/clear/blind/absorbing	0.18	0.43	0.46
Clear/blind/clear	0.30	0.71	0.75	Low-E/clear/absorbing/blind	0.10	0.39	0.44
Clear/blind/reflecting	0.21	0.47	0.49	Blind/clear	0.48	1.00	1.04
Clear/blind/absorbing	0.24	0.54	0.58	Blind/absorbing	0.40	0.79	0.83
Clear/reflecting/blind	0.11	0.36	0.41	Blind/clear/clear	0.49	0.94	0.95
Clear/absorbing/blind	0.13	0.43	0.48	Blind/clear/reflecting	0.31	0.61	0.61
Clear/clear/clear/blind	0.13	0.49	0.56	Blind/clear/absorbing	0.35	0.70	0.71
Clear/clear/blind/clear	0.24	0.57	0.61	Blind/low-E/clear	0.46	0.91	0.92
Clear/clear/blind/reflecting	0.16	0.37	0.39	Blind/low-E/absorbing	0.32	0.67	0.68
Clear/clear/blind/absorbing	0.19	0.43	0.46	Blind/clear/clear/clear	0.47	0.86	0.86
Clear/clear/reflecting/blind	0.09	0.31	0.37	Blind/clear/clear/reflecting	0.28	0.54	0.54
Clear/clear/absorbing/blind	0.10	0.36	0.44	Blind/clear/clear/absorbing	0.32	0.61	0.62
Low-E/clear/blind	0.15	0.57	0.64	Blind/low-E/clear/clear	0.45	0.84	0.84
Low-E/absorbing/blind	0.12	0.43	0.52	Blind/low-E/clear/absorbing	0.30	0.59	0.59
Air node correction factor:							
— internal blind		0.91	0.88				
— mid-pane blind		0.87	0.83				
— external blind		0.88	0.85				

Table 5.23 Solar cooling loads for fast-response building with single clear glazing: NE Scotland (lowlands); unshaded

Date	Orien-tation	Solar cooling load at stated sun time / W·m^{-2}											Orien-tation
		07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	
Jan 29	N	4	8	12	21	27	34	34	26	18	11	7	N
	NE	6	14	25	23	28	34	35	27	18	11	8	NE
	E	19	38	147	216	160	69	52	40	31	24	20	E
	SE	53	74	225	399	451	415	325	206	89	59	54	SE
	S	74	91	193	363	481	533	528	480	341	171	81	S
	SW	52	55	65	108	222	340	421	459	381	202	62	SW
	W	18	21	26	35	41	56	83	176	210	131	27	W
	NW	5	8	13	22	28	34	35	27	21	22	12	NW
	Horiz.	21	28	46	94	137	163	162	134	84	40	26	Horiz.
Feb 26	N	12	18	28	38	43	47	47	43	37	27	16	N
	NE	43	72	51	46	47	50	50	46	40	30	19	NE
	E	96	193	302	308	216	95	77	69	62	52	42	E
	SE	128	235	406	513	533	463	349	221	110	90	77	SE
	S	115	171	301	444	550	585	568	526	405	265	151	S
	SW	73	80	96	127	246	373	462	519	476	360	204	SW
	W	39	45	55	65	71	83	112	228	296	271	167	W
	NW	16	22	33	42	48	51	51	48	48	54	68	NW
	Horiz.	50	73	132	200	251	271	265	241	184	119	67	Horiz.
Mar 29	N	41	53	64	75	83	87	87	83	73	63	51	N
	NE	117	134	120	95	92	95	95	90	81	70	57	NE
	E	182	255	362	344	239	139	125	113	103	93	80	E
	SE	179	268	426	490	453	412	317	205	124	118	102	SE
	S	110	166	283	393	444	494	490	461	360	261	160	S
	SW	100	113	128	143	228	344	431	492	463	413	305	SW
	W	81	94	107	118	126	142	166	277	340	361	303	W
	NW	54	67	80	91	98	103	103	101	106	133	162	NW
	Horiz.	123	175	259	329	360	390	387	369	306	244	174	Horiz.
Apr 28	N	70	85	92	101	106	109	109	105	99	91	83	N
	NE	257	268	205	124	133	131	130	126	120	112	100	NE
	E	356	451	486	414	303	180	166	155	149	140	128	E
	SE	289	411	512	526	507	436	324	201	167	148	136	SE
	S	126	175	283	380	457	498	498	457	372	256	160	S
	SW	135	148	160	186	229	352	461	531	543	480	411	SW
	W	133	146	157	166	172	186	212	341	446	470	465	W
	NW	100	113	124	133	139	142	141	143	143	222	285	NW
	Horiz.	233	316	408	467	516	538	539	516	463	379	306	Horiz.
May 29	N	120	125	129	136	141	143	141	140	139	164	126	N
	NE	418	381	276	176	175	169	167	166	162	153	142	NE
	E	533	580	540	462	336	214	198	192	187	179	168	E
	SE	370	471	514	525	489	410	298	189	178	161	150	SE
	S	125	151	244	339	410	446	449	402	320	224	138	S
	SW	150	162	173	191	213	326	437	505	521	507	445	SW
	W	175	187	197	204	210	223	245	373	480	550	561	W
	NW	146	158	168	175	181	183	182	191	203	302	383	NW
	Horiz.	350	440	511	572	608	622	629	602	549	491	412	Horiz.
Jun 21	N	140	143	144	151	154	158	157	154	151	144	144	N
	NE	416	391	293	190	182	178	178	175	171	163	152	NE
	E	518	576	548	469	343	219	207	198	194	186	176	E
	SE	353	456	507	519	485	401	288	184	178	163	152	SE
	S	125	145	231	327	399	431	429	389	308	212	133	S
	SW	160	171	181	198	214	324	430	505	524	511	470	SW
	W	203	214	224	231	235	249	272	399	508	581	620	W
	NW	175	187	196	203	207	210	211	220	237	340	437	NW
	Horiz.	370	463	539	600	639	645	642	626	578	519	454	Horiz.
Jul 4	N	132	137	139	147	152	154	153	151	145	138	138	N
	NE	411	376	284	191	188	182	180	178	173	164	154	NE
	E	512	550	521	446	337	222	209	200	195	186	176	E
	SE	351	439	484	490	467	394	289	187	178	162	152	SE
	S	126	148	231	318	389	424	427	385	310	216	136	S
	SW	153	165	176	195	213	317	421	489	513	500	436	SW
	W	181	193	203	211	217	230	252	373	482	552	557	W
	NW	154	166	176	184	189	192	191	201	214	313	390	NW
	Horiz.	356	438	508	560	603	618	622	602	560	503	423	Horiz.
Aug 4	N	88	101	107	116	122	126	124	121	115	107	100	N
	NE	306	299	227	150	155	153	150	147	141	133	120	NE
	E	406	474	470	408	309	198	184	173	167	159	146	E
	SE	310	414	475	489	474	396	307	198	175	156	143	SE
	S	126	164	257	344	417	436	451	408	323	232	150	S
	SW	128	141	154	178	208	314	417	474	466	426	367	SW
	W	125	138	150	159	165	181	202	316	393	424	418	W
	NW	101	113	125	135	140	144	143	147	151	222	272	NW
	Horiz.	265	347	420	476	522	526	541	515	455	388	321	Horiz.

Table continues

Table 5.23 Solar cooling loads for fast-response building with single clear glazing: NE Scotland (lowlands); unshaded — *continued*

Date	Orientation	Solar cooling load at stated sun time / W·m^{-2}											Orientation
		07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	
Sep 4	N	50	62	73	82	89	92	94	90	81	71	60	N
	NE	210	209	152	92	106	107	108	105	96	86	73	NE
	E	319	398	439	389	271	158	145	134	125	115	102	E
	SE	277	388	495	535	493	430	312	205	155	135	120	SE
	S	113	184	299	406	462	497	456	410	359	246	152	S
	SW	104	117	133	160	224	339	403	442	473	392	284	SW
	W	88	102	114	124	132	146	169	268	358	354	291	W
	NW	62	76	88	98	105	108	110	108	101	148	169	NW
	Horiz.	162	231	315	382	415	436	409	380	348	272	199	Horiz.
Oct 4	N	19	29	41	50	57	61	61	57	50	39	28	N
	NE	81	117	78	63	63	66	67	63	56	45	33	NE
	E	154	293	354	323	221	113	97	87	80	70	58	E
	SE	169	322	447	504	482	436	305	196	115	101	87	SE
	S	116	186	308	419	485	534	469	417	358	231	149	S
	SW	72	82	98	120	224	344	388	415	426	314	224	SW
	W	46	56	68	77	85	98	120	210	284	252	201	W
	NW	25	35	47	57	64	67	68	64	66	74	92	NW
	Horiz.	70	115	183	245	284	311	283	255	218	151	100	Horiz.
Nov 4	N	6	12	19	29	38	40	38	33	26	17	10	N
	NE	10	33	29	33	39	41	39	34	27	18	11	NE
	E	32	121	240	274	181	80	62	52	45	36	30	E
	SE	65	167	341	484	463	399	306	190	98	70	63	SE
	S	81	138	263	425	486	504	486	417	335	204	115	S
	SW	52	58	72	114	225	322	387	396	374	253	123	SW
	W	21	27	34	44	54	64	85	166	217	177	85	W
	NW	7	12	20	30	39	41	39	34	31	27	27	NW
	Horiz.	27	41	74	133	171	186	178	149	110	62	36	Horiz.
Dec 4	N	3	3	8	13	19	24	24	18	12	6	3	N
	NE	3	3	8	14	20	24	24	19	12	7	3	NE
	E	13	19	96	155	138	53	36	28	21	16	13	E
	SE	46	55	167	304	422	396	326	203	87	56	46	SE
	S	67	74	157	289	456	514	537	473	284	147	67	S
	SW	48	49	64	103	212	326	425	448	308	159	48	SW
	W	14	14	19	24	31	41	69	157	161	90	14	W
	NW	3	3	8	13	20	24	24	19	13	8	3	NW
	Horiz.	13	14	27	53	93	115	117	92	49	23	13	Horiz.

Glazing configuration (inside to outside)	*G*-value	Correction factor for stated building response		Glazing configuration (inside to outside)	*G*-value	Correction factor for stated building response	
		Fast	Slow			Fast	Slow
Clear	0.84	1.00	0.90	Low-E/clear	0.66	0.79	0.68
Absorbing	0.62	0.79	0.68	Low-E/absorbing	0.46	0.58	0.50
Clear/clear	0.72	0.83	0.72	Low-E/clear/clear	0.60	0.69	0.59
Clear/reflecting	0.41	0.50	0.43	Low-E/clear/absorbing	0.40	0.49	0.42
Clear/absorbing	0.49	0.60	0.52				
Clear/clear/clear	0.64	0.71	0.60				
Clear/clear/reflecting	0.35	0.43	0.37				
Clear/clear/absorbing	0.42	0.50	0.42				
Air node correction factor		0.86	0.83				

Table 5.24 Solar cooling loads for fast-response building with single clear glazing: NE Scotland (lowlands); intermittent shading

Date	Orien- tation	Solar cooling load at stated sun time / W·m^{-2}											Orien- tation
		07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	
Jan 29	N	4	7	11	19	25	31	32	25	17	10	7	N
	NE	5	12	24	21	25	31	33	26	18	11	7	NE
	E	10	20	221	125	39	58	43	31	23	17	13	E
	SE	16	106	216	280	277	228	154	48	53	23	19	SE
	S	26	34	262	279	331	341	321	246	81	123	34	S
	SW	17	19	24	137	187	253	292	266	163	48	26	SW
	W	11	13	17	25	31	42	138	140	49	123	19	W
	NW	5	7	11	20	25	31	33	26	19	22	11	NW
	Horiz.	20	26	40	85	126	151	153	128	82	39	25	Horiz.
Feb 26	N	11	16	26	35	40	43	43	40	35	26	16	N
	NE	36	66	47	42	43	46	46	43	38	28	19	NE
	E	68	287	208	169	55	75	59	50	45	36	26	E
	SE	73	328	307	342	315	248	164	57	65	46	34	SE
	S	48	191	240	323	364	363	343	278	185	66	89	S
	SW	31	36	48	152	208	278	327	321	258	87	159	SW
	W	24	29	38	47	52	61	173	187	193	68	149	W
	NW	15	20	30	39	44	47	48	45	43	48	65	NW
	Horiz.	30	48	102	268	177	179	168	63	160	99	48	Horiz.
Mar 29	N	38	48	59	69	76	81	81	77	69	59	48	N
	NE	106	122	113	88	84	88	88	84	75	66	55	NE
	E	241	220	238	185	67	111	98	86	78	69	57	E
	SE	127	374	311	306	281	229	85	163	83	78	64	SE
	S	50	191	220	266	303	312	301	246	179	66	106	S
	SW	57	68	83	87	290	261	312	312	283	221	86	SW
	W	57	68	80	90	97	110	229	219	244	222	89	W
	NW	50	62	74	84	91	96	96	94	94	120	151	NW
	Horiz.	79	125	327	227	250	255	248	214	85	200	134	Horiz.
Apr 28	N	64	78	84	92	98	101	101	98	92	85	78	N
	NE	198	95	193	113	122	120	119	116	111	104	93	NE
	E	289	330	305	239	104	147	134	123	117	110	100	E
	SE	239	316	345	341	305	239	97	151	119	100	90	SE
	S	65	199	214	272	309	319	301	254	180	68	105	S
	SW	72	84	94	113	265	266	324	347	320	276	200	SW
	W	80	93	103	111	116	127	255	268	302	308	257	W
	NW	75	87	97	105	111	114	113	115	106	289	189	NW
	Horiz.	153	368	280	317	339	346	336	307	255	202	94	Horiz.
May 29	N	111	116	118	125	130	133	131	129	126	155	116	N
	NE	285	229	113	150	149	144	142	141	137	130	120	NE
	E	389	382	338	264	126	168	154	147	144	136	126	E
	SE	291	334	350	337	294	225	100	140	130	115	105	SE
	S	78	94	306	252	285	296	276	228	88	177	95	S
	SW	85	96	105	120	234	250	308	330	328	298	231	SW
	W	102	114	123	130	135	144	268	281	339	362	328	W
	NW	99	110	119	126	131	134	132	138	235	238	259	NW
	Horiz.	253	306	351	382	398	406	395	364	326	276	134	Horiz.
Jun 21	N	129	132	133	139	142	145	145	143	140	134	131	N
	NE	283	233	114	155	148	144	144	141	138	131	121	NE
	E	384	385	344	269	131	172	161	152	149	142	133	E
	SE	281	329	346	335	289	219	99	136	131	117	107	SE
	S	81	91	290	245	277	284	266	220	86	167	91	S
	SW	92	102	111	125	228	246	306	332	330	310	257	SW
	W	113	123	132	139	142	152	278	291	349	386	378	W
	NW	110	120	129	136	139	143	143	148	254	262	303	NW
	Horiz.	255	311	357	390	403	404	397	370	332	289	234	Horiz.
Jul 4	N	122	126	128	136	140	143	142	140	135	127	126	N
	NE	275	225	113	157	154	148	147	145	140	132	123	NE
	E	372	368	327	262	130	176	164	155	151	143	134	E
	SE	274	315	329	321	283	219	99	141	133	117	109	SE
	S	82	94	287	239	272	282	264	221	87	172	94	S
	SW	88	100	109	125	231	242	297	324	325	293	231	SW
	W	108	119	128	136	141	150	273	283	341	360	333	W
	NW	105	116	126	134	138	141	140	146	247	244	269	NW
	Horiz.	255	306	346	379	397	403	395	371	336	286	140	Horiz.
Aug 4	N	80	93	98	107	112	117	115	112	107	99	93	N
	NE	220	105	207	130	135	133	131	129	124	116	105	NE
	E	308	323	294	236	107	157	144	133	128	121	109	E
	SE	250	303	323	322	282	225	97	151	129	111	100	SE
	S	77	107	322	257	281	294	279	229	88	183	106	S
	SW	83	95	106	125	253	252	303	312	291	259	107	SW
	W	94	106	117	125	131	143	264	255	287	296	129	W
	NW	90	101	112	121	126	131	129	132	128	303	99	NW
	Horiz.	289	246	288	324	336	346	338	304	259	114	248	Horiz.

Table continues

Table 5.24 Solar cooling loads for fast-response building with single clear glazing: NE Scotland (lowlands); intermittent shading — *continued*

Date	Orientation	Solar cooling load at stated sun time / W·m^{-2}											Orientation
		07:30	08:30	09:30	10:30	11:30	12:30	13:30	14:30	15:30	16:30	17:30	
Sep 4	N	46	57	66	75	82	85	87	84	76	67	57	N
	NE	191	193	145	83	98	99	100	97	89	80	69	NE
	E	252	292	280	215	85	128	116	105	97	88	76	E
	SE	224	300	345	334	298	229	88	156	108	89	75	SE
	S	55	220	233	283	314	302	271	240	89	190	102	S
	SW	62	74	88	107	287	250	282	309	278	208	83	SW
	W	67	80	91	100	107	118	132	369	250	219	93	W
	NW	58	70	82	91	98	101	102	102	89	134	157	NW
	Horiz.	107	280	231	261	280	272	252	232	95	219	150	Horiz.
Oct 4	N	18	26	37	46	53	56	57	53	47	37	27	N
	NE	70	109	73	58	58	61	62	58	52	42	32	NE
	E	233	222	223	170	56	86	71	62	56	46	35	E
	SE	225	257	313	316	293	221	74	148	69	56	43	SE
	S	57	224	239	294	336	314	275	240	88	174	96	S
	SW	38	46	61	71	304	245	267	282	230	82	18	SW
	W	34	43	54	63	70	79	94	301	184	71	184	W
	NW	24	32	43	53	59	62	63	60	60	66	87	NW
	Horiz.	42	81	238	178	201	194	175	67	186	123	75	Horiz.
Nov 4	N	6	10	17	26	35	37	36	31	25	16	10	N
	NE	7	30	27	30	36	38	37	32	25	17	11	NE
	E	12	194	180	144	44	64	47	38	31	23	17	E
	SE	23	244	285	308	274	218	72	150	61	35	27	SE
	S	27	160	231	295	318	315	279	227	79	151	66	S
	SW	18	23	31	146	182	235	254	249	187	59	89	SW
	W	17	21	28	37	46	53	66	243	55	167	80	W
	NW	7	11	18	27	35	38	36	32	28	24	27	NW
	Horiz.	25	36	65	121	158	173	167	141	106	61	36	Horiz.
Dec 4	N	3	3	7	11	18	22	22	18	11	6	3	N
	NE	3	3	7	13	18	22	22	18	12	6	3	NE
	E	10	10	102	37	129	48	31	24	18	13	10	E
	SE	17	17	240	250	265	229	157	49	57	27	17	SE
	S	23	23	198	253	317	343	322	217	67	102	23	S
	SW	18	18	28	135	182	256	293	231	71	128	18	SW
	W	8	8	12	16	22	29	120	112	37	84	8	W
	NW	3	3	7	12	18	22	22	18	13	7	3	NW
	Horiz.	13	13	24	47	85	106	110	89	48	23	13	Horiz.

Glazing configuration (inside to outside)	*G*-value	Correction factor for stated building response	
		Fast	Slow
Clear/blind	0.20	0.79	0.94
Absorbing/blind	0.17	0.59	0.61
Clear/clear/blind	0.16	0.61	0.73
Clear/blind/clear	0.30	0.71	0.73
Clear/blind/reflecting	0.21	0.47	0.47
Clear/blind/absorbing	0.24	0.55	0.57
Clear/reflecting/blind	0.11	0.36	0.39
Clear/absorbing/blind	0.13	0.43	0.46
Clear/clear/clear/blind	0.13	0.51	0.57
Clear/clear/blind/clear	0.24	0.58	0.59
Clear/clear/blind/reflecting	0.16	0.38	0.39
Clear/clear/blind/absorbing	0.19	0.43	0.44
Clear/clear/reflecting/blind	0.09	0.31	0.34
Clear/clear/absorbing/blind	0.10	0.36	0.39
Low-E/clear/blind	0.15	0.58	0.63
Low-E/absorbing/blind	0.12	0.42	0.48
Low-E/clear/clear/blind	0.13	0.49	0.59
Low-E/clear/blind/clear	0.23	0.56	0.59
Low-E/clear/blind/absorbing	0.18	0.44	0.44
Low-E/clear/absorbing/blind	0.10	0.37	0.42
Blind/clear	0.48	1.00	1.04
Blind/absorbing	0.40	0.79	0.82
Blind/clear/clear	0.49	0.94	0.95
Blind/clear/reflecting	0.31	0.61	0.61
Blind/clear/absorbing	0.35	0.70	0.71
Blind/low-E/clear	0.46	0.91	0.92
Blind/low-E/absorbing	0.32	0.67	0.69
Blind/clear/clear/clear	0.47	0.86	0.86
Blind/clear/clear/reflecting	0.28	0.54	0.54
Blind/clear/clear/absorbing	0.32	0.61	0.62
Blind/low-E/clear/clear	0.45	0.84	0.84
Blind/low-E/clear/absorbing	0.30	0.60	0.59
Air node correction factor:			
— internal blind		0.90	0.88
— mid-pane blind		0.87	0.83
— external blind		0.88	0.85

Tables 5.25 to 5.40 Solar cooling loads for fast-response building with single clear glazing: latitudes 00–60 deg N/S; unshaded/intermittent shading — refer to CD-ROM

References

1 prEN 15255: 2005: *Thermal performance of buildings. Cooling load calculation. General criteria and procedures* (London: British Standards Institution) (2005)

2 *CIBSE standard tests for the assessment of building services design software* CIBSE TM33 (London: Chartered Institution of Building Services Engineers) (2004)

3 Almadari F and Hammond G P Improved data correlations for buoyancy driven convection in rooms *Building Serv. Eng. Res. Technol.* **4** (3) 106–112 (1980)

4 Hatton A and Awbi H B Convective heat transfer in rooms *Proc. Building Simulation '95, August 1995, Wisconsin, USA* (1995)

5 BS EN 410: 1998: *Glass in buildings. Determination of luminous and solar characteristics of glazing* (London: British Standards Institution) (2005)

6 *Building energy and environmental modelling* CIBSE AM11 (London: Chartered Institution of Building Services Engineers) (1998)

7 Fanger P O *Thermal Comfort* (Copenhagen: Danish Technical Press) (1970)

8 Holmes M J and Connor P A ROOM: A method to predict thermal comfort at any point in a space *Proc. CIBSE Nat. Conf., Canterbury, 1991* (London: Chartered Institution of Building Services Engineers) (1991)

9 *Health and Safety (Display Screen Equipment) Regulations 1992* (London: Stationery Office) (1992)

10 *Natural ventilation in non-domestic buildings* CIBSE AM10 (London: Chartered Institution of Building Services Engineers) (2005)

11 *Guidelines for environmental design in schools* DfES Building Bulletin 87 2nd. ed. (London: The Stationery Office) (2003)

12 *Weather, solar and illuminance data* CIBSE Guide J (London: Chartered Institution of Building Services Engineers) (2002)

13 *Weather data with climate change scenarios* CIBSE TM34 (London: Chartered Institution of Building Services Engineers) (2004)

14 *Climate change and the built environment: impacts and adaptation* CIBSE TM36 (London: Chartered Institution of Building Services Engineers) (2005)

15 Holmes M J and Hitchen E R An Example Weather Year for the calculation of energy demand in buildings *Building Serv. Eng.* **45** 186–189 (1978)

16 *Passive solar design studies: Provision of weather data* ETSU Passive Solar Programme: UK Meteorological Data — Example Weather Years (Harwell: Energy Technology Support Unit) (1991)

17 *Conservation of fuel and power in buildings other then dwellings* The Building Regulations 2000 Approved Document L (London: The Stationery Office) (2002)

18 *Rules of thumb* BSRIA BG 14/2003 (Bracknell: Building Services Research and Information Association) (2003)

19 Milbank N O and Harrington-Lynn J Thermal response and the admittance procedure *Building Serv. Eng.* **42** 38–51 (May 1974)

20 *Passive and hybrid solar commercial buildings: Advanced Case Studies Seminar, April 1991* (Harwell: Energy Technology Support Unit) (1991)

21 Jones R Modelling water vapour conditions in buildings *Building Serv. Eng. Res. Technol.* **14** (3) 99–106 (1993)

22 Cunningham M J Modelling of moisture transfer in structures 1: A description of a finite difference model *Building and Environment* **25** (1) 55–61 (1990)

23 Huang C L D Multi-phase moisture transfer in porous media subjected to temperature gradient *Int. J. Heat Mass Transfer* **22** 1293–1307 (1973)

24 Luikov A V Systems of differential equations of heat and mass transfer in capillary porous bodies (review) *Int. J. Heat Mass Transfer* **18** 1–14 (1975)

25 Philip J R and DeVries D R Moisture movement in porous media under temperature gradients *Trans. Am. Geophysical Union* **38** (2) 222–232 (1957)

26 Rees S J, Spitler J D, Holmes M J and Haves P Comparison of peak load predictions and treatment of solar gains in the admittance and heat balance load calculation procedures *Building Serv. Eng. Res. Technol.* **21** (2) 125–138 (2000)

27 Davies M G The thermal response of an enclosure to periodic excitation: the CIBSE approach *Building and Environment* **29** (2) 217–235 (1994)

28 Harrington-Lynn J The admittance procedure: variable ventilation *Building Serv. Eng.* **42** 199–200 (November 1974)

29 Harrington-Lynn J The admittance procedure: intermittent plant operation *Building Serv. Eng.* **42** 219–221 (December 1974)

30 *Heating, ventilating, air conditioning and refrigeration* CIBSE Guide B (London: Chartered Institution of Building Services Engineers) (2005)

31 Wilkins C K, Kosonen R and Laine T An analysis of office equipment load factors *ASHRAE J.* **33** (9) 38–44 (1991)

32 Hosni M H, Jones B W, Sipes J M and Xu Y Total heat gain and the split between radiant and convective heat gain from office and laboratory equipment in buildings *ASHRAE Trans.* **104** (1A) 356–365 (1998)

33 Holmes M J Design for ventilation *Proc. 6th AIC Conf., Ventilation strategies and measurement techniques, Netherlands 16–19 September 1985* (1985)

34 Awbi H B *Ventilation of buildings* (London: Spon) (1991)

35 Holmes M J *Designing variable volume systems for a room air movement* BSRIA Application Guide AG1/74 (Bracknell: Building Services Research and Information Association) (1974)

36 Liddament M W *Air infiltration calculation techniques: an application guide* (Coventry: Air Infiltration and Ventilation Centre) (1986)

37 Linden P F, Lane Serff G F and Smeed D A Emptying filling boxes: the fluid mechanics of natural ventilation *J. Fluid Mech.* **212** 309–335 (1990)

38 Birtles A B and John R W A new optimum start algorithm *Building Serv. Eng. Res. Technol.* **6** (3) 117–122 (1985)

39 Adams S and Holmes M J *System simulation: A study of the effects of boiler oversize and heating controls on the energy consumption of a heating system* BSRIA Project Report 15/111 (Bracknell: Building Services Research and Information Association) (1978)

40 *Reference data* CIBSE Guide C (London: Chartered Institution of Building Services Engineers) (2001)

41 Jamieson H C The Mechanical Services at Shell Centre *J. Inst. Heating Ventilating Eng.* **31** 1–35 1963 (1963)

42 Harrington-Lynn J Derivation of equations for intermittent heating used in CIBSE Building Energy Code Part 2a *Building Serv. Eng. Res. Technol.* **19** (4) (1998)

43 *Thermal storage: environmental benefits* CIBSE Research Report RR06 (London: Chartered Institution of Building Services Engineers) (1997)

44 *Space heating by medium temperature radiant panels* HVRA Laboratory Report 36 (Bracknell: Building Services Research and Information Association) (1966)

45 *Siting of medium temperature radiant heating strip panels in long rooms* HVRA Laboratory Report 40 (Bracknell: Building Services Research and Information Association) (1967)

46 *Ice storage* CIBSE TM18 (London: Chartered Institution of Building Services Engineers) (1994)

Appendix 5.A1: Quality assurance in building services software

5.A1.1 Introduction

Quality assurance (QA), in the context of calculations for design, is a process which helps results to a consistently high standard by:

- using methods that fit for purpose
- avoiding or reducing errors, managing uncertainties, providing an audit trail of calculations for future scrutiny
- implementing best practice.

Quality assurance (QA) procedures are necessary to:

- instill confidence in clients that the work is undertaken to a consistently high standard
- ensure that the calculations address the needs of clients
- improve coordination between members of the design team
- enable new work to capitalise on previous projects
- provide evidence that may help in resolving disputes
- identify the need for training and recruitment.

Most large firms implement formal QA. For smaller firms the cost of establishing a quality assurance system and acquiring QA certification may be prohibitive. However, some QA principles may cost very little to implement, but are expected to improve efficiency, effectiveness and to save costs in the long run. Guidance provided here aims to help both the larger organisations to establish specific procedures for carrying out calculations when using computer software, and the smaller firms to encourage them to set up informal QA procedures to improve consistency and accuracy in their use of building services software or indeed simple hand calculations, see Parand and Bloomfield[A1.1] .

The main quality issues in this context can be summarised as follows:

- appropriate choice of the calculation method and/or the software tool (fitness for purpose)
- correct application of the method and/or the software tool
- accuracy and precision in data entry and operation
- checking of results
- documentation of the calculation process (including electronic files) and associated decisions.

The appropriate choice of the calculation method and/or the modelling software depends on the type of design being undertaken. Section 5.3 describes the different methodologies available and their uses. Note that greater detail is not necessarily synonymous with the improved results. Additional data entry can increase the risk of software input error (see below) or inappropriate reliance on default values. Sections 5.A1.3 and 5.A1.4 discuss software selection and use in detail.

Validation of software is normally undertaken by commercial software developers but this alone may not be sufficient. CIBSE has published its own independent test results for validation purposes[A1.2] and comparisons should either be sought from a vendor or carried out internally during an evaluation period, and upon subsequent release of software updates. Receipt of an updated version of a program is an instance when checks might be carried out. Keeping a small personal project model which is easily run after each version update and the results compared is a valuable guard against software errors.

Correct application of software is fundamental to obtaining reliable results. This is largely a user issue, and a matter of their training, their familiarity with specific programs and the application of QA procedures. Accuracy and precision of data entry would ideally be achieved in every instance but an element of human error should be expected and catered for (see Figure 5.7)*. This is best achieved with the aid of a second party to check overall values and sample individual data as necessary, preferably according to a documented procedure.

Document and model management is necessary for the efficient use of resources and to be able to retrieve specific project models with confidence. A detailed description is given in section 5.A1.4.5.

* Note that as the number of inputs increases so the likelihood of input errors also increases. However, since the number of inputs is related to the resolution with which heat transfer processes are modelled then model errors tend to reduce. At some point, it has been argued, an optimum is reached beyond which accuracy is impaired.

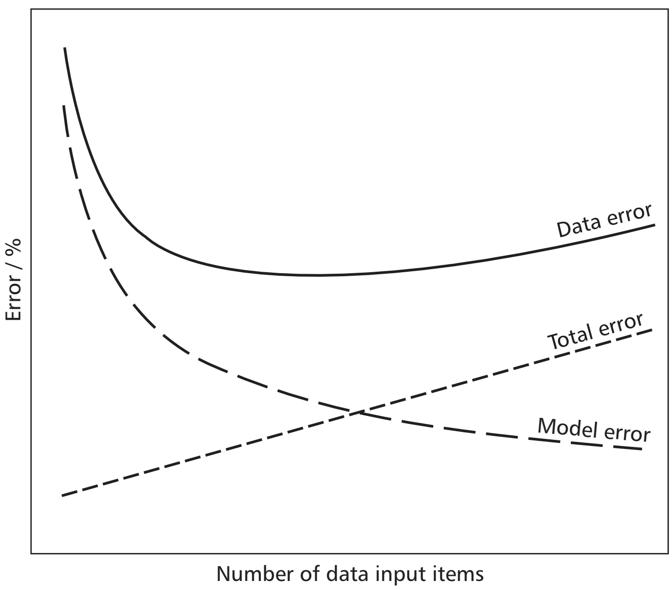


Figure 5.7 Relationship of model number of input data used, model errors and overall errors (after Chapman[A1.3])

QA procedures encompass how all the above issues are handled and controlled. Section 5.A1.4 gives detailed guidance on how a QA system may be implemented within the design process.

5.A1.2 Risk, uncertainty and sources of error

One of the main motivations in establishing QA is to reduce the risk of errors and to address uncertainties inherent in the process of design. However, a distinction should be made between avoidable errors in the use of calculation methods and the inherent uncertainties in using the calculation method, which irrespective of its level of detail is approximation of reality, be it a simple manual calculation or detailed simulation software model.

5.A1.2.1 Sources of avoidable errors

The errors that in theory could be avoided normally stem from the following given in an order of ease of tackling them:

— *Blunders in using the method and/or software*: QA procedures, such as routine checks of input data and careful examination of results can significantly reduce or even eliminate such errors.

— *Inappropriate use of a method and errors in abstracting a problem into a form suitable for calculation or simulation*: such errors are dependent on the user's knowledge of inherent assumptions in the calculation method, experience and training. The standardisation of procedures for carrying out routine calculations, for example using documented 'performance assessment methods'(A1.4), can help reduce such errors. Reviews of modelling methodology and assumptions by more experienced personnel is an alternative or indeed supplement in non-routine cases.

— *Errors in coding a software implementation of a method*: select software which has been through various validation tests, see AM11.

— *Approximations within the mathematical models being used*: this too may be addressed by appropriate software selection. Alternatively, if the nature of a physical simplification is understood then it is possible to test the sensitivity of predictions to this uncertainty and so account for it in design.

5.A1.2.2 Sources of uncertainty

Generally, calculation of the energy and environmental behaviour of buildings, is carried out deterministically, i.e. parameters used in modelling a building are treated as known values, which can either be fixed or time dependant. However, the values of most parameters are often uncertain or unknown at the time calculations are made. The most important sources of such uncertainties are as follows:

— Imperfect knowledge regarding detailed building design/operation characteristics and consequent assumptions (e.g. thermal/optical properties of materials, build quality and associated leakage, equipment used and their characteristics, etc.).

— The inherent unpredictability of the future (climate, occupants' use and operation of the building, etc.).

— Lack of knowledge about the underlying physical process and/or approximations within the mathematical models.

A systematic uncertainty analysis can help to identify the key sources of uncertainty which merit further attention as well as those that may be safely ignored. Furthermore, it can provide an insight into the level of confidence in estimates.

The purpose of quantitative uncertainty analysis is to use currently available information to quantify the degree of confidence in the existing data and models. The purpose is not to somehow 'reduce' uncertainty — reduction in uncertainty can only come from improved knowledge. Nevertheless it is important to be aware of the importance of uncertainties and ways of handling them by using appropriate design margins.

Several techniques have been developed, within an academic context, for studying predictive uncertainties. These include differential sensitivity analysis (DSA), Monte Carlo analysis (MCA) and stochastic sensitivity analysis (SSA)(A1.5). Of these the most commonly employed is DSA, in which individual parameters are varied between simulations, depending upon estimates of their uncertainty, and the results analysed. Such sensitivity analyses help to identify not only the overall uncertainty in model outputs but also the sensitivity of performance to input uncertainties. This may then prompt the user to investigate means for reducing input uncertainties (e.g. by acquiring better quality information) or by ensuring that the design is sufficiently robust so that performance is not contingent upon particular (uncertain) modelling assumptions. The issue of sensitivity analysis is discussed further in CIBSE AM11(A1.6).

5.A1.3 Fitness for purpose

As mentioned earlier, it is important to be aware of approximations in the mathematical models of the software being used, and the possible implications of these approximations. It is also important to be aware of how the software has performed in recent validation studies. This will help to define the range of applicability of the software or means for accommodating predictive limitations. This might be achieved by representing predictive uncertainty in some meaningful way (section 5.A1.2), or surmounting this uncertainty using additional support software, by identifying some means for emulating a physical process which is not being explicitly simulated*.

Guidance on the selection of suitable software is given in CIBSE AM11(A1.6). Furthermore, CIBSE has published some standard validation tests(A1.2) which can be used to

* As an example, many dynamic thermal models represent solar shading due to nearby buildings, but sky radiation is unchanged (with corresponding over-predictions) — although models are emerging to resolve this deficiency(A1.7). To account for this within the affected space(s), one might use predictions from a ray tracing program to calibrate the (temporary) scaling of diffuse solar radiation in the climate file.

determine the accuracy with which some of the fundamental heat transfer processes are being modelled whilst at the same time providing a basis for judging whether the software has been correctly deployed (this then acting as a tool to assist with staff training).

It is also important to be aware of the software's basis for environmental control. For example some programs will, by default, deliver all of the energy that is required to achieve a space set point within a single simulation time-step (typically one hour). Plant sizes may therefore be over predicted. One way of accounting for this is to bring forward the plant start time (say by one hour) and manually adjust the plant capacity (initially by say a third of that predicted) until the smallest size which reaches the set condition has been identified.

In addition to internal simplifications, it is generally desirable for the user to impose simplifications to the degree of detail with which a problem is represented within software (i.e., not to be seduced into describing problems in unnecessary detail by easy to use CAD tools). Failing to do so may render the models unmanageably complex (and lead to QA difficulties) and/or lead to unnecessarily high computational overheads, whilst over-simplification may cause predictive inaccuracies. It is important therefore to achieve a balance — keeping the model as simple as possible consistent with the avoidance of significant errors. To help with this, specific guidelines for abstracting problems into simulation metaphor are described in some detail in CIBSE AM11[A1.6]. Note that the correct application of these guidelines will tend to require both a sound knowledge of building physics fundamentals and a reasonable grounding in the theoretical basis and internal assumptions of the software being used.

5.A1.4 QA procedures

It has been suggested[A1.6] that three categories of individual are involved in the development and implementation of QA procedures in the context of using BEEM software: (*i*) the QA manager, (*ii*) the simulation team manager, and (*iii*) the program user. Whilst this may be true of relatively large organisations, the personnel associated with these roles and indeed the division of responsibilities may vary between organisations. The key issues are:

— that clear lines of responsibility are identified

— that senior personnel take responsibility for establishing and policing QA

— that a sound QA philosophy pervades all aspects of simulation work

— that QA procedures are established and continually refined for oft-repeated tasks.

It is perhaps useful to identify some key QA procedures with recourse to a typical simulation process* (see Table 5.41 for a summary).

* Note that this assumes that the simulation team manager and personnel designated with overall QA responsibility together develop a relevant QA statement and supporting procedures and thereafter ensure the proper implementation of these procedures.

5.A1.4.1 Problem definition

The simulation team manager should initially determine the need for environmental modelling and the types of software to be used. Following from this, perhaps in liaison with the program user, the modelling strategy for resolving the design questions of interest should be developed. This should involve defining the form of the base case model (including the approach for abstracting the design problem into simulation metaphor) and agreeing upon the need for and range of variations from this model. Sources of input data should be identified. Indeed a library of information resources should generally be developed and managed — ensuring that software time conventions are respected.

Table 5.41 Example of a QA checklist including personnel responsible

Person responsible	Task
Quality assurance (QA) manager	— Developing a quality statement — Developing and implementing QA procedures — Refining and updating of QA procedures
Simulation team manager	— Developing 'performance assessment method' (PAM) style documents for commonly occurring problems — Developing a documentation skeleton for generic simulation tasks and guidelines for adaptation to novel problems — Preparing skeleton documents for reporting results to clients (see CIBSE AM11[A1.6], section 5.5) — Procedures for archiving documentation on each job and the associated program input and output data — Devising project-specific simulation strategies — Checking users have followed QA check lists — Checking plausibility of output results — Identifying need for and arranging client meetings to review progress — Identifying the need for new staff or staff training — Recommending acquisition of new software or computing resources — Identifying need for internal development of productivity aids/supporting software and overseeing their development and testing
Program user	— Developing and maintaining standard databases — Developing and maintaining archives of simulation input and output — Adopting standard file and model attribute naming conventions — Maintaining a log book of elegant solutions as well as common mistakes and means for resolution — Checking plausibility of output results — Accounting for sensitivity to input uncertainties — Developing and applying checks to ensure correctness of the simulation results — Documenting procedures and databases used in a series of simulations — Making routine backups of modelling project folders — Routine testing of new programs against validation data sets — Recommending need for new programs and computer hardware — Developing supporting software tools and productivity aids

Important inputs should be agreed with the client and circulated to the design team (e.g. to inform the team manager of important departures from these), as should input assumptions. Input uncertainties and their effects should be quantified wherever possible, or techniques/third party software executed to minimise them.

The reference model should then be defined, according to internally agreed directory/file/attribute naming conventions. If appropriate, an internally developed performance assessment method (PAM)[(A1.8)] should be followed. This should be constructed to facilitate ‘painless’ changes to the model following from modelling or conclusions or other design changes and, in some circumstances, to track the design process later.

5.A1.4.2 Simulation

With the reference model defined, initial simulations should focus upon the program user understanding the behaviour of the building and systems being investigated; ensuring the model has been properly defined (checking input files as well as comparing the range of output variables with benchmarks or simplified calculations), that means for emulating system controls deliver the appropriate responses. Uncertainties to key inputs should also be studied as should, time permitting, performance sensitivity to key design variants. A review of the model and associated results with the team manager should act as a final test of the validity of the model.

The previously agreed list of model variants to be tested should be updated following any conclusions from initial ad-hoc simulations. Model variants should then be prepared and checked; modelling work should also be periodically backed-up. If supported, it is prudent to prepare scripts to conduct the simulations and extract the required result — again ensuring that no mistakes have been made, so as to avoid wasted simulation time. With large numbers of model variants, the need for preparing new or amending existing results analysis programs should then be internally reviewed. Such programs can save time and represent a source of consistency in analysis and presentation, but again checks should ensure that they are free from errors. Indeed, as with information sources, a library of such quality assured utility programs should be developed.

5.A1.4.3 Interpretation

In cooperation with the team manager, results should be reviewed in detail to relate cause to effect and consequently to prepare design advice. This may, particularly in the case of unintuitive or unforeseen trends in results, involve further checking of model input files and also further simulations and interrogation of results.

5.A1.4.4 Presentation

The team manager should arrange and attend periodic review meetings. Initially to check model inputs and agree design variants (both type and range) to be tested and later to review results. The need for and range of further simulations should then be agreed.

5.A1.4.5 Documentation

There are two strands to documentation: client reporting and project documentation. In the former, following the format of either a generic documentation skeleton or a task-specific reporting format as may be defined in a PAM, the modelling methodology, inputs (and any associated uncertainty analysis), reference and variant model results and conclusions should be logically documented. It is helpful to include in an appendix design drawings which associate tabulated construction build-ups with constructional elements, occupancy characteristics, internal heat gains etc.

Project documentation should include filing of hardcopy material and the thorough archiving of electronic material. Model documentation should be explained by a ‘readme’ file that identifies the project objectives, strategy and key assumptions, variants tested and associated file names/locations.

Finally, it is good practice to carry out project de-briefing. This is an opportunity to discuss the effectiveness of the modelling procedure and its influence on design in an open forum with a view to identifying scope for improvement in the efficacy and quality of the process as well as the need for staff training, recruitment or software/computing resources. At this stage the need for revising or creating new PAMs should also be identified and tasked and an ‘elegant solution’ logbook should be updated.

Note that the emphasis above is clearly placed upon the planning of the modelling study (note that sound planning generally pays dividends later), defining and checking the reference model. This is the most time-consuming aspect of the simulation process, but also the most critical. The preparation, checking and simulation of subsequent aspects of this model tend not to be excessively time consuming — particularly if automation opportunities are exploited. The temptation to move quickly onto other work without completing the project documentation process should be avoided. It is important that a project should be capable of being quickly resurrected in the future due to unforeseen circumstances or, indeed, because the staff member responsible is not available some time soon after the work was completed.

Regarding the software used, it is prudent to maintain a list of desired software features (and possible bugs) and liaise with software developers with a view to these being integrated (or resolved) in future releases. Alternatively, in the case of open source software, it may be worth investing the time required to understand the software structure so that such features may be embedded in-house, taking care subsequently to check the validity of the software.

5.A1.5 Summary

Software should be selected which is appropriate for the task in hand. The software should also be tested against standard validation data sets to understand the range of applicability of the software as well as to ensure that it is being correctly deployed.

Libraries of input data sets and productivity aids should be internally developed. Input data sets should respect software timing conventions and their basis should be corroborated. Productivity aids (including results analysis programs) should be documented and independently checked.

A quality assurance infrastructure should be established including personnel as well as documentation including QA checklists, PAMs, log book and documentation skeletons.

Quality assurance procedures should be defined and used to guide the modelling process.

The need for niche software as well as staff training, recruitment and computing resources should be periodically reviewed to ensure the best available resources are brought to bear on project work.

References

A1.1 Parand F and Bloomfield D Introducing quality assurance in practices using software for evaluation of building performance (Can small firms afford QA?) *Proc. CIBSE Conf., Computers in the Construction Industry, May 1993* (London: Chartered Institution of Building Services Engineers) (1993)

A1.2 *CIBSE standard tests for the assessment of building services design software* CIBSE TM33 (London: Chartered Institution of Building Services Engineers) (2004)

A1.3 Chapman J Data accuracy and model reliability *Proc. BEPAC Conf., Building Environmental Performance '91, Canterbury, 10-11 April 1991* (Reading: Building Environmental Performance Analysis Club) (1991)

A1.4 Parand F and Bloomfield D Integrated knowledge-based system for performance assessment methods *Proc. BEPAC Conf., University of York, April 1994* 263–272 (Reading: Building Environmental Performance Analysis Club) (1994)

A1.5 Lomas K J and Eppel H Sensitivity analysis techniques for building thermal simulation programs *Energy and Buildings* 19 21–44 (1992)

A1.6 CIBSE *Building energy and environmental modelling* CIBSE AM11 (London: Chartered Institution of Building Services Engineers) (1998)

A1.7 Robinson D and Stone A Solar radiation modelling in the urban context *Solar Energy* **77** (3) 295–309 (2004)

A1.8 Parand F and Bloomfield D Quality Assurance in Environmental Prediction *Proc. BEPAC Conf., Building Environmental Performance '91, Canterbury, 10–11 April 1991* 237–246 (Reading: Building Energy Performance Analysis Club) (1991)

Appendix 5.A2: Overview of calculation methods

5.A2.1 Brief history of CIBSE methods

In order to place this edition in context this appendix presents a brief history of previous CIBSE methods and a review of the various approaches to dynamic modelling in other countries.

5.A2.1.1 Steady state methods

These methods are only used for the sizing of heat emitters and a number of different approaches have been adopted by CIBSE in the past. The only significant difference between these approaches is the space temperature (index temperature) used to determine the fabric heat loss; that is, in the equation:

$$Q_f = U\,A\,(\theta_i - \theta_o) \qquad (5.64)$$

where Q_f is the fabric heat loss (W), U is the thermal transmittance of the surface ($W{\cdot}m^{-2}{\cdot}K^{-1}$), A is the surface area (m^2), θ_i is the index temperature (°C) and θ_o is the external temperature (°C).

Air temperature method

The rate of heat loss through a wall is balanced by the convective gain from the room air to that surface and longwave radiant interchange between room surfaces. Both processes are complex so any practical calculation technique needs to introduce approximations. The approximation made in the 1959 IHVE Guide(A2.1) was that all heat transfer occurred between the surface and the air temperature. This approximation is acceptable for well-insulated spaces but will generally lead to under-estimation for radiant heating systems and over-estimation for fully convective heating systems.

Environmental temperature method

The deficiencies of the 1959 approach were partly addressed in the 1970 IHVE Guide A(A2.2) by the introduction of the concept of environmental temperature. (Note that the only difference between the 1970 method and that given in the 1986 CIBSE Guide A(A2.3) is in the replacement of the environmental temperature by the operative temperature as the design temperature.) The intention was to combine the effects of radiant and convective heat transfer within a single air temperature index. While the method gives a fairly accurate representation of heat losses(A2.4) the presentation of the theory has been questioned(A2.5). The main problem occurs in the representation of long wave radiant heat transfer between room surfaces. A complete representation of this process is given in Appendix 5.A1 of the 1986 CIBSE Guide A(A2.3). Simplifications may take the form of approximations to the view factors, or, as chosen by the 1970 IHVE Guide A(A2.2), reduction of the radiant exchange to that between a single surface and an enclosure at some mean radiant temperature. An alternative proposal by Davies(A2.6) (the 'Two Star Method') uses exact view factor values for a room having six surfaces. In practice, there is little difference between heat losses calculated by that method and those calculated using the 1970 IHVE Guide A method. The Two Star Method suffered the disadvantage that it was impractical to do the calculation by hand but the widespread availability of computers now makes that method viable.

The 1999 edition of CIBSE Guide A[A2.7] recognised that different levels of accuracy were acceptable depending upon application and so the concept of the simple, basic and fundamental model was introduced. The present edition maintains that approach. The models are described in detail in Appendix 5.A3.

5.A2.1.2 Dynamic methods

The most common design applications for dynamic methods are the calculation of the cooling load on a space and peak temperatures within naturally ventilated buildings. Combined with appropriate HVAC system and plant models they can be powerful tools for the calculation of energy consumption.

The admittance procedure[A2.8] is the simplest of the dynamic methods available. It also offers transparency and the possibility to perform calculations by hand. The procedure assumes that all internal and external load fluctuations can be represented by the sum of a steady state component and a sine wave with a period of 24 hours. Implicit within this assumption is that steady cyclic conditions are achieved; i.e. a single day repeated for subsequent days until all long-term transients have died away. The method does not represent the effects of rapid load changes nor long-term storage. Therefore it is not suitable as a means of calculating the performance of buildings with a large thermal capacity or the effects of rapid changes in load. Nonetheless it is considered suitable for use at an early stage of design and as a means of predicting the limiting state. There is also some evidence that, for naturally ventilated buildings, the peak temperatures predicted by the admittance procedure are close to those which actually occur[A2.9]. This method may also be used as a manual check on the results from computer modelling. For these reasons the admittance method is currently retained.

Transient modelling can only be carried out by means of computer models. These models are capable of producing a realistic prediction of the performance of buildings and systems. However, the accuracy of all such models is open to question since, in common with all numerical representations of real systems, there are many approximations and uncertainties in the input data, including those made by the user. The effect of any particular approximation will depend upon circumstances. An idea of the potential performance of dynamic models may be obtained from the results of a validation exercise[A2.10]. Careful examination of the results of this exercise shows that:

— current models are quite good but no model gets everything right

— models that predict energy well may not be as good at predicting summertime temperatures.

The latter point applies equally to steady state models. Similarly, the considerations of heat flow within rooms for steady state models apply also to dynamic models. This Guide does not recommend any particular detailed dynamic model. The features that are required are discussed in Appendix 5.A9 and some ways in which the validity of dynamic models may be checked are given in CIBSE TM33[A2.11].

The main difference between transient methods and the admittance procedure is the way in which the fabric storage and solar gains are represented. Transient methods use particular numerical techniques to approximate and then solve the heat conduction equation whereas the admittance procedure uses an exact solution to the equation but approximates the boundary conditions to those represented by a cyclic process with a period of 24 hours.

5.A2.2 Review of alternatives to the admittance method

At the same time as the CIBSE admittance method[A2.12,A2.13] was developed in the UK other countries were developing their own methodologies, notably North America where engineers needed to size all-air plant for their growing market for air conditioning in offices. In contrast, the need in the UK was more towards sizing wet heating systems and predicting possible summertime overheating for offices with then fashionably large windows. Partly as a result of this difference of need, alternative approximations and approaches were taken in arriving at practical manual calculation techniques. For example, the early ASHRAE methods were based on internal air temperature whereas the UK adopted environmental temperature. Differences also arose in the mathematical treatment of heat flow through walls[A2.14]. The UK method assumed a sinusoidal input and response and the North Americans opted for a time series analysis combined with conduction transfer functions.

The admittance method, CIBSE's basic model, remains largely unchanged since it was first published in 1974 and its subsequent adoption by the precursor of CIBSE Guide A. However, the 1999 edition of CIBSE Guide A[A2.7] went on to identify and describe general algorithms for 'full' and 'reference' models. These specifications are not, as such, complete calculation recipes and consequently are neither likely to become regarded as generic methodologies in the same way as has the CIBSE admittance calculation. It is therefore not possible to compare directly the reference and full models with methodologies currently used outside the UK.

There are two cooling load calculation methodologies described in the ASHRAE *Fundamentals* Handbook[A2.15]. These are the heat balance (HB) method and the radiant time series (RTS) method. The latter is a simplification of the former but neither is a manual calculation procedure. The RTS method can, if required, be performed in a simplified manner on a spreadsheet. Both the HB and RTS methods are based on the fundamentals of building physics and are described as the most reliable means of estimating cooling loads. They do not, however, undermine the validity of earlier ASHRAE methods such as the transfer function method (TFM) and total equivalent temperature differential method with time averaging (TETD/TA) when used correctly.

The scientific basis for the HB and RTS methods is described in the ASHRAE publication *Cooling and Heating Load Calculation Principles*[A2.16]. Although this does not give detailed algorithms, a software implementation is provided on an accompanying disk.

Of the ASHRAE methods, the RTS method is the most similar in application to the CIBSE admittance method. It is suited for calculating the peak cooling load in a defined zone on a particular design day. Like the admittance method, it assumes the same weather occurs everyday and therefore tends to overestimate extreme cooling loads. A shortcoming is its inability to model the re-radiation of long wave radiation from inside back to the environment as would be important for a room with extensive glazing. All energy that enters a zone is assumed to remain stored in the fabric until later dissipated as part of a convective cooling load.

The HB method is the most accurate but the added complexity of calculation makes it unusable for interacting zones in a large building. It addresses energy re-radiation by solving simultaneously for the internal and external surface temperatures from hour to hour. Like the RTS method, it assumes the same weather pattern is repeated each day. It is therefore unable to calculate annual energy or overheating frequencies.

Both the RTS and HB methods are essentially 'design day' calculations predominantly geared towards modular offices with mixed air cooling strategies. They are not well adapted for performing cooling load calculations on rooms with displacement-type air conditioning systems or chilled ceilings. Other countries have also published procedures for sizing cooling equipment and calculating energy use. Some of these are similar in terms of modelling the physics from first principles but others use an adaptive technique based on a simple steady state calculation.

In Germany, VDI 2078: *Computation of Cooling Load for Air-conditioned Areas*[A2.17], uses a times series solution of the underlying physics but it differs significantly from the HB method. Rather than calculate the necessary intermediate data from first principles, the VDI 2078 procedure adapts data from standardised solutions of 'lightweight' and 'heavyweight' offices. In application, it is similar to the HB and RTS methods and is suited to modular offices. An example of a very different approach is the Dutch standard NEN 2919: *Energy Performance of Non-Residential Buildings — Determination Method*[A2.18]. This is based on the hypothesis that it is possible to calculate the cooling loads by considering only steady state conditions and then modifying results with coefficients to account for thermal capacity. A procedure in a similar vein is the Italian UNI 10375: *Method for calculating the summer internal temperature of environments*[A2.19]. Research has shown that simplified adaptive methods are less precise and flexible than those based on hourly simulation[A2.20].

In the UK and abroad, government sponsored research has also led to a number of university developed building energy models. Whilst these are principally academic research tools, they can be used for cooling and heating plant sizing. Some, indeed, are available as commercial packages although the underlying methodology may not necessarily be in the public domain or in an accessible format. Establishing suitability for commercial use and lack of staff familiarity with their operation is a significant factor. Of particular concern is whether the algorithms and user interfaces cater for relevant types of cooling equipment and cooling strategies.

A different approach has recently been taken by the US Government, which is sponsoring the development of a widely applicable and freely available open source code called EnergyPlus[A2.21]. This is, in fact, a merger of two previous building energy models that were developed independently at American university laboratories. EnergyPlus uses response functions to model conduction rather than the finite difference method[A2.22] that is predominant in UK dynamic thermal modelling codes. This is essentially only an issue of computational efficiency and does not affect validity. Of some debate is whether EnergyPlus can be regarded as a calculation method or is better described as an example of an application that may conform to a performance specification such as that developed by the CIBSE.

References

A2.1 *A Guide to Current Practice* (London: Institution of Heating and Ventilating Engineers) (1959) (out of print)

A2.2 *Design Data* IHVE Guide A (London: Institution of Heating and Ventilating Engineers) (1970) (out of print)

A2.3 *Design Data* CIBSE Guide A (London: Chartered Institution of Building Services Engineers) (1986) (out of print)

A2.4 Holmes M J Heat loss from rooms: comparison of determination methods *Building Serv. Eng. Res. Technol.* **9**(2) 69–78 (1988)

A2.5 Davies M G Flaws in the environmental temperature model *Building Serv. Eng. Res. Technol.* **13**(4) 209–215 (1992)

A2.6 Davies M G The basis for a room global temperature *Phil. Trans. Roy. Soc. A.* **339** 153–191 (1992)

A2.7 *Environmental design* CIBSE Guide A (London: Chartered Institution of Building Services Engineers) (1999) (out of print)

A2.8 Milbank N O and Harrington-Lynn J Thermal response and the admittance procedure *Building Serv. Eng.* **42** 38–51 (May 1974)

A2.9 *Passive and hybrid solar commercial buildings: Advanced Case Studies Seminar, April 1991* (Harwell: Energy Technology Support Unit) (1991)

A2.10 Lomas K J et. al. Empirical validation of thermal building simulation programs using test room data *IEA Energy conservation in buildings and community systems programme Annex 21* (Leicester: De Montford University/International Energy Agency) (September 1994)

A2.11 *CIBSE standard tests for the assessment of building services design software* CIBSE TM33 (London: Chartered Institution of Building Services Engineers) (2004)

A2.12 Loudon A G Summertime Temperatures in Buildings Without Air Conditioning *J. Heating and Ventilating Engineers* **32** 280–292 (1970)

A2.13 Milbank N O and Harrington-Lynn J The Admittance Procedure: Variable Ventilation *Building Serv. Eng.* **42** 38–51 (1974)

A2.14 Spitler J D, Rees S J and Haves P Quantitative comparison of North American and UK cooling load calculation procedures- Part 1: methodology, Part 2: results ASHRAE Trans. **104**(2) 36–44, 47–61 (1998)

A2.15 *Fundamentals* ASHRAE Handbook (Atlanta GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers Inc.) (2005)

A2.16 Penderson C O, Fisher D E, Spitler J D and Liesen R J *Cooling and heating load calculation principles* (Atlanta GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers Inc.) (1998)

A2.17 VDI 2078: *Berechnung der Kühllast klimatisierter Räume* (Düsseldorf, Germany: Verein Deutscher Ingenieure e.V.)

A2.18 NEN 2916: 1994: *Energy Performance of Non-Residential Buildings — Determination Method* (Delft, The Netherlands: Nederlands Normalisatie Instituut) (1994)

A2.19 UNI 10375: *Method for calculating the summer internal temperature of environments* (Milano, Italy: Ente Nazionale Italiano di Unificazione) (1995)

A2.20 Beccali M, Mazzarella L and Motta M Simplified Models for Building Cooling Energy Requirement *Proc. IBPSA Conf., Building Simulation, Rio de Janeiro, Brazil, August 2001* (College Station, TX: International Building Performance Simulation Association) (2001)

A2.21 Penderson C O, Fisher D E, Liesen R J, Strand R K, Taylor R D, Buhl W F, Winkelmann F C, Lawrie L K, Crawley D B, EnergyPlus, The Merger of BLAST and DOE-2 *Proc. IBPSA Conf., Building Simulation '97, Prague, Czech Republic, September 1997* (College Station, TX: International Building Performance Simulation Association) (1997)

A2.22 Clarke J A *Energy simulation in building design* 2nd edn. (Oxford: Butterworth-Heinemann) (2001)

Appendix 5.A3: Derivation of thermal steady state models

5.A3.1 Notation

Symbols used in this appendix are as follows.

A_n Area of surface n (m^2)
a, b Linearising constants
b_n Radiant heat transfer coefficient (W·m^{-2}·K^{-1})
c_p Specific heat capacity of air (J·kg^{-1}·K^{-1})
C_v Ventilation conductance (W·K^{-1})
E_{bn} Black body radiation from surface n (W·m^{-2})
F_a fraction of air temperature detected by sensor (0.5 for a sensor detecting operative temperature)
$F_{m,n}$ View factor from surface m to surface n
h_a Heat transfer coefficient between air and environmental nodes (W·m^{-2}·K^{-1})
H_c Thermal transmittance due to convection (W·K^{-1})
h_c Convective heat transfer coefficient (W·m^{-2}·K^{-1})
h_{cn} Convective heat transfer coefficient for surface n (W·m^{-2}·K^{-1})
H_r Thermal transmittance due to radiation (W·K^{-1})
h_r Radiative heat transfer coefficient (W·m^{-2}·K^{-1})
J_n Radiosity of surface n (W·m^{-2})
L_n Longwave radiant heat flux incident on surface n (W·m^{-2})
m Integer denoting particular surface
$\dot{m}_a$ Mass flow rate of air (kg·s^{-1})
N Total number of surfaces
N_v Number of room air changes (h^{-1})
n Integer denoting particular surface
R Radiant fraction of source from source
R_{sin} Thermal resistance between inner face of surface n and environmental temperature (m^2·K·W^{-1})
R_{sn} Thermal resistance of surface n (m^2·K·W^{-1})
U_n Thermal transmittance for material of which surface n is composed (W·m^{-2}·K^{-1})
U_n' Thermal transmittance between inner face of surface n and heat transfer temperature on outer face of surface n (W·m^{-2}·K^{-1})
U_p Thermal transmittance modified for heat flow through internal partition (W·m^{-2}·K^{-1})
V Room volume (m^3)
α Surface absorption coefficient
ε_n Emissivity of surface n
Φ_{con} Convective energy from emitter (W)
Φ_f Fabric heat gain (W)
Φ_{ln} Longwave energy incident on surface n from sources other than room surfaces (W)
Φ_{rad} Radiant energy from emitter (W)
Φ_t Total heat loss (W)
ϕ_n Radiant heat flow from surface n (W·m^{-2})
θ_{ai} Inside air temperature (°C)
θ_{ain} Air temperature for convective heat exchange with surface n (°C)
θ_c Operative temperature at centre of room (°C)
θ_c' Operative temperature on far side of internal partition through which heat flow occurs (°C)
θ_{ei} Environmental temperature (°C)
θ_o External heat transfer temperature (°C)
θ_{on} External heat transfer temperature for surface n (°C)
θ_r Mean radiant temperature (°C)
θ_s Surface temperature (°C)
θ_{sn} Surface temperature of surface n (°C)
θ^* Radiant-star temperature (°C)

5.A3.2 Full Model

The rate of loss of heat from a space through the building fabric can be expressed as:

$$\Phi_f = \sum_{n=1}^{N} A_n\,(\theta_{sn} - \theta_{on}) \,/\, R_{sn} \tag{5.65}$$

The rate of heat flow through a wall is equal to that into the wall, thus the fabric heat loss can also be expressed as:

$$\Phi_f = \sum_{n=1}^{N} [A_n\,(\theta_{ain} - \theta_{sn})\,h_{cn} - \phi_n] \tag{5.66}$$

Note that ϕ_n is positive for heat flows leaving the surface.

The first term inside the square brackets represents the rate of convection of heat from the room air to the surface and the second term is the rate of radiant heat flow into the surface. This radiant term represents the exchange of longwave radiation between the surface and all other surfaces within the room. (The calculation of steady state loss ignores shortwave radiation.) The exchange of long-wave radiation can be seen as analogous to the reflection of light from a diffuse source, i.e. there are an infinite number of reflections of radiation between the surfaces.

The rate of radiant heat flow into the surface is the difference between that incident (L_n) upon the surface and that leaving the surface (J_n) that is:

$$\phi_n = A_n (J_n - L_n) \tag{5.67}$$

Now the rate at which radiant energy leaves a surface may be expressed as:

$$J_n = (1 - \varepsilon_n) L_n + \varepsilon_n E_{bn} \tag{5.68}$$

Thus:

$$\phi_n = A_n (E_{bn} - J_n) \varepsilon_n / (1 - \varepsilon_n) \tag{5.69}$$

The radiation incident upon the surface is the sum of that received from other surfaces and that from radiant heating sources. Radiation from another surface depends upon the view factor between that surface and the subject surface (n) and the rate at which radiation leaves that surface (i.e. the radiosity).

Thus the radiation incident upon surface n is:

$$A_n L_n = \sum_{m=1}^{N} (J_m A_m F_{m,n}) - \Phi_{1n} \tag{5.70}$$

However:

$$A_n F_{n,m} = A_m F_{m,n} \tag{5.71}$$

Therefore:

$$A_n L_n = \sum_{m=1}^{N} (J_m A_n F_{n,m}) - \Phi_{1n} \tag{5.72}$$

This represents a set of simultaneous equations (one for each surface), that when solved give the amount of radiation leaving each surface (J_n). Simultaneous solution means that the infinite number of reflections of radiation is accounted for automatically.

In order to solve equation 5.72 it is first necessary to substitute for L_n by combining equations 5.69 and 5.72. Assuming that $F_{n,m}$ is zero (i.e. that is all surfaces are planar), this results in the following equation set:

$$J_n / (1 - \varepsilon_n) - \sum_{1}^{N} F_{n,m} J_m \ldots\ldots$$
$$= \varepsilon_n E_{bn} / (1 - \varepsilon_n) + Q_{1n} / A_n \tag{5.73}$$

This relationship is converted into a heat loss model by linearising the black body emissive power and introducing the steady state surface heat balance. Thus:

$$E_{bn} = a + b\,\theta_{sn} \tag{5.74}$$

where a and b are constants.

From equation 5.65, for a single surface (n):

$$\Phi_f = A_n (\theta_{sn} - \theta_{on}) U_n' \tag{5.75}$$

Therefore, equating 5.75 and 5.66 to eliminate Φ_f gives:

$$-\phi_n + \theta_{ai\,n} h_{cn} + U_n' \theta_{on} = \theta_{sn} (h_{cn} + U_n') \tag{5.76}$$

U_n' is the transmittance between the surface temperature θ_{sn} and the outside temperature θ_{on}, i.e. the heat transfer temperature on the other side surface n, given by:

$$U_n' = U_n / (1 - U_n R_{si\,n}) \tag{5.77}$$

$R_{si\,n}$ is the standard value of the inner surface resistance used to calculate the standard U-value (see chapter 3: *Thermal properties of building structures*) for surface n, U_n. Since U_n' is also dependent on the external surface heat transfer coefficient, i.e. the surface coefficient appropriate to the 'other' side of surface, it may be necessary to include a correction for exposure.

Substitution of equations 5.74 and 5.76 into equation 5.73 gives the set of equations 5.78 (see below), which represent both radiant interchange between surfaces and the conduction of heat through room surfaces.

Equation 5.78 places no restrictions on the air temperature distribution within the space. A means of obtaining air temperatures would be to combine the solution of the above with computational fluid dynamics. Alternatively, some rules could be assigned to the distribution of air temperature throughout the space(A3.1).

5.A3.3 Reference Model

The Reference Model is developed by adding convective heat transfer and control sensor models to the Full Model and making some assumptions about the distribution of the radiant component of heat from the emitter.

The Full Model contains an arbitrary model of the convective heat transfer process. The Reference Model assumes a fully mixed space, i.e. the dry bulb temperature of

$$(h_{cn} + U_n' + \varepsilon_n h_{rn}) (\theta_{sn} / \varepsilon_n) - \sum_{m=1}^{N} (F_{n,m} / \varepsilon_m) [(h_{cm} + U_m') (1 - \varepsilon_m) + h_r \varepsilon_m] \theta_{sm} - (h_{cn} \theta_{ai\,n} / \varepsilon_n) + \sum_{m=1}^{N} (F_{n,m} / \varepsilon_m) [h_{cm} (1 - \varepsilon_m) \theta_{ai\,m}]$$
$$= (\theta_{on} U_n' / \varepsilon_n) - \sum_{m=1}^{N} (F_{n,m} / \varepsilon_m) [U_m' (1 - \varepsilon_m) \theta_{om}] + \Phi_{1n} / A_n \tag{5.78}$$

where:

$$h_{rn} = \varepsilon_n b_n \tag{5.79}$$

the air does not vary from point to point within the space. Thus in equation 5.78 all values of $\theta_{\mathrm{ai}n}$ are equal to the inside air temperature θ_{ai} and the convective heat balance is then given by:

$$-\sum_{n=1}^{N} h_{\mathrm{c}n} A_n \theta_{\mathrm{s}n} + \theta_{\mathrm{ai}} (\dot{m}_{\mathrm{a}} c_{\mathrm{p}} + \sum_{n=1}^{N} h_{\mathrm{c}n} A_n) = \Phi_{\mathrm{t}} (1 - R) + \theta_{\mathrm{ao}} \dot{m}_{\mathrm{a}} c_{\mathrm{p}} \tag{5.80}$$

The model is completed by the introduction of the control temperature, θ_{c}, for example the operative temperature which at low air speeds is the average of the air and mean radiant temperatures. The mean radiant temperature 'seen' by a sensor may be considered to be the equivalent temperature for radiant heat exchange between the sensor and its surroundings. It therefore depends upon:

— surface temperature
— surface emissivity
— emissivity of the sensor
— view factor between the surfaces and the sensor
— radiation from a heat emitter incident on the sensor.

Thus, the mean radiant temperature varies throughout the space. It is possible to model the sensor as an additional room surface. However, for the purposes of design calculations, the sensor is deemed to be located at a position where the proportion of longwave radiation received from each surface is directly proportional to the ratio of the area of the surface to the total room area. Furthermore, the sensor is also assumed to have an emissivity of unity (i.e. a black body). Thus the design mean radiant temperature is:

$$\theta_{\mathrm{r}} = \frac{\sum \varepsilon_n \theta_{\mathrm{s}n} A_n}{\sum A_n \ \varepsilon_n} + \frac{R \ \Phi_{\mathrm{t}}}{h_{\mathrm{r}} \sum A_n} \tag{5.81}$$

Note that h_{r} is calculated for an emissivity of unity.

The control temperature is given by:

$$\theta_{\mathrm{c}} = F_{\mathrm{a}} \theta_{\mathrm{ai}} + (1 - F_{\mathrm{a}}) \theta_{\mathrm{r}} \tag{5.82}$$

where F_{a} = 0.5 if the sensed parameter is the operative temperature.

Assuming that any radiant heat input is uniformly distributed over each surface, and is equal to ($Q_{\mathrm{t}}\ R / \sum A$), the Reference Model may be represented by the equation set:

$$A\ X = C \tag{5.83}$$

where A, X and C are matrices, defined as follows.

Matrix A:

(a) Surface heat balance equations

Terms $A(n,n)$ for n = 1 to n = total number of room surfaces:

$$A(n,n) = (h_{\mathrm{c}n} + U_n' + h_{\mathrm{r}} \varepsilon_n) / \varepsilon_{\mathrm{n}} \tag{5.84}$$

Terms $A(n,m)$ where $n \neq m$, for n = 1 to n = total number of room surfaces and for m = 1 to m = total number of room surfaces:

$$A(n,m) = -F_{n,m} [(h_{\mathrm{c}m} + U_m')(1 - \varepsilon_m) + h_{\mathrm{r}} \varepsilon_m)] / \varepsilon_m \tag{5.85}$$

Terms $A(n,m)$ for n = 1 to n = total number of room surfaces and for m = total number of room surfaces + 1:

$$\mathbf{A}(n,m) = (-h_{\mathrm{c}n} / \varepsilon_n) + \sum_{i=1}^{N} h_{\mathrm{c}i} F_{n,i} (1 - \varepsilon_n) / \varepsilon_i \tag{5.86}$$

Terms $A(n,m)$ for n = 1 to n = total number of room surfaces and for m = total number of room surfaces + 2:

$$A(n,m) = -R / \sum A \tag{5.87}$$

(b) Control sensor heat balance equations

Terms $A(n,n)$ for n = total number of room surfaces + 1:

$$A(n,n) = F_{\mathrm{a}} \tag{5.88}$$

where F_{a} is the fraction of the air temperature detected by the sensor. (F_{a} = 0.5 for a sensor detecting operative temperature.)

Terms $A(n,m)$ for n = total number of room surfaces + 1 and for m = 1 to m = total number of room surfaces:

$$A(n,m) = \varepsilon_n (1 - F_{\mathrm{a}}) A_n / \sum (A_n \varepsilon_n) \tag{5.89}$$

Terms $A(n,m)$ for n = total number of room surfaces + 1 and for m = total number of room surfaces + 2:

$$A(n,m) = R (1 - F_{\mathrm{a}}) / (h_{\mathrm{r}} \sum A) \tag{5.90}$$

(c) Convection heat balance

Terms $A(n,n)$ for n = total number of room surfaces + 2:

$$A(n,n) = (R - 1) / \sum A \tag{5.91}$$

Terms $A(n,m)$ for n = total number of room surfaces + 2 and for m = 1 to m = total number of room surfaces:

$$A(n,m) = -h_{\mathrm{c}n} A_n / \sum A \tag{5.92}$$

Terms $A(n,m)$ for n = total number of room surfaces + 2 and for m = total number of room surfaces + 1:

$$A(n,m) = [C_{\mathrm{v}} + \sum (A_n h_{\mathrm{c}n})] / \sum \mathrm{A} \tag{5.93}$$

Vector C:

Terms $C(n)$ for n = 1 to n = the total number of room surfaces:

$$C(n) = (\theta_{\mathrm{o}n} U_n' / \varepsilon_n) - \sum_{i=1}^{N} [F_{n,i} U_n' \theta_{\mathrm{o}i} (1 - \varepsilon_i) / \varepsilon_i \tag{5.94}$$

Terms $C(n)$ for n = total number of room surfaces + 1:

$$C(n) = \theta_{\mathrm{c}} \tag{5.95}$$

Terms $C(n)$ for n = total number of room surfaces + 2:

$$C(n) = \theta_{ao}\, C_v / \Sigma A \tag{5.96}$$

Solution vector X:

Terms $X(n)$ for n = 1 to n = total number of room surfaces provide the temperatures for each surface.

Term $X(n)$ for n = total number of room surfaces + 1 provides the room air temperature.

Term $X(n)$ for n = total number of room surfaces + 2 provides the emitter output (i.e. sum of convective and radiant outputs).

The ventilation transmittance is represented by the conventional term C_v, see equation 5.9. If it is necessary to take account of air flows from a number of sources, that term in matrix A should be replaced by the summation $\Sigma\,(\dot{m}_a\, c_p)_i$ where the summation covers all sources i.

In vector C, the term $(\theta_{ao}\ C_v\ /\ \Sigma\ A)$ is then replaced by $[\Sigma\ (\theta_i\, \dot{m}_{ai}\, c_{pi})\ /\ \Sigma\ A]$ where θ_i is the temperature of air from source i.

View factors are not easy to calculate and while some standard relationships are given in chapter 3 of CIBSE Guide C(A3.2), these will not cover many applications. Figure 5.8 and the following algorithm enables view factors to be determined for rectangular rooms(A3.3).

(a) Two parallel room surfaces

Radiation shape factor ($F_{1\text{-}2}$) between parallel surfaces 1 and 2 separated by a distance G, see Figure 5.8(a), is given by:

$$\begin{aligned} &2\pi\,(b_1 - a_1)\,(d_1 - c_1)\,F_{1\text{-}2} = \\ &\{[P(b_2 - b_1) + P(a_2 - a_1)] \times [Q(c_2 - c_1) + Q(d_2 - d_1) \\ &- Q(c_2 - d_1) - Q(d_2 - c_1)]\} \\ &+ \{[P(b_2 - a_1) + P\,(a_2 - b_1)] \times [Q(c_2 - d_1) \\ &+ Q(d_2 - c_1) - Q(c_2 - c_1) - Q(d_2 - d_1)]\} \end{aligned} \tag{5.97}$$

P and Q are functions; expanding equation 5.97 gives products of the form $P(b_2 - b_1)\, Q(c_2 - c_1)$, given by:

$$\begin{aligned} P(Z_1)\, Q(Z_2) &= Z_1\, W \tan^{-1}(Z_1 / W) \\ &+ Z_2\, V \tan^{-1}(Z_2 / V) - (G^2/2) \ln\,[(W^2 + Z_1^2) / W^2)] \end{aligned} \tag{5.98}$$

where Z_1 and Z_2 are generalised variables, e.g. $Z_1 = (b_2 - b_1)$ and $Z_2 = (c_2 - c_1)$, and:

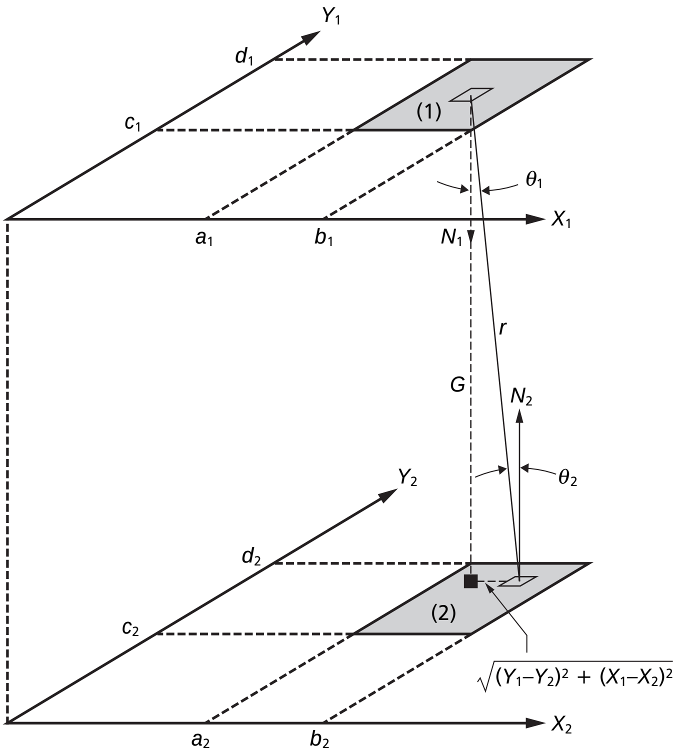
$$V^2 = G^2 + Z_1^2 \tag{5.99}$$

$$W^2 = G^2 + Z_2^2 \tag{5.100}$$

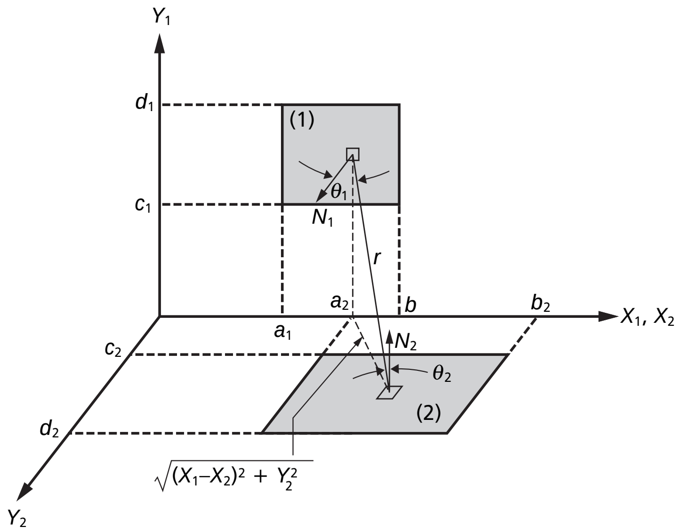
(b) Two perpendicular room surfaces

Radiation shape factor ($F_{1\text{-}2}$) between perpendicular surfaces 1 and 2, see Figure 5.8(a), is given by:

$$\begin{aligned} &2\pi\,(b_1 - a_1)\,(d_1 - c_1)\,F_{1\text{-}2} = \{[R(b_2 - b_1) + R(a_2 - a_1)] \\ &\times [S(c_2 - c_1) + S(d_2 - d_1) - S(c_2 - d_1) - S(d_2 - c_1)]\} \\ &+ \{[R(b_2 - a_1) + R(a_2 - b_1)] \times [S(c_2 - d_1) \\ &+ S(d_2 - c_1) - S(c_2 - c_1) - S(d_2 - d_1)]\} \end{aligned} \tag{5.101}$$



(a)



(b)

Figure 5.8 View factors for radiation heat exchange; (a) between two parallel room surfaces, (b) between two perpendicular room surfaces

R and S are functions; expanding equation 5.101 gives products of the form $R(b_2 - b_1)\, S(c_2 - c_1)$, given by:

$$\begin{aligned} R(Z_1)\, S(Y_2 - Y_1) &= T\, Z_1 \tan^{-1}(Z_1 / T) \\ &+ {}^1/_4\, (Z_1^2 - T^2) \ln\,(T^2 + Z_1^2) \end{aligned} \tag{5.102}$$

where Z_1 and $(Y_2 - Y_1)$ are generalised variables, as above, and:

$$T^2 = Y_2^2 + Y_1^2 \tag{5.103}$$

These equations when combined with view factor algebra will satisfy the majority of needs. The relevant view factor algebra is as follows.

For conservation of energy:

$$\sum_{m=1}^{M} F_{n,m} = 1.0 \tag{5.104}$$

where the summation is over all surfaces comprising the enclosure.

For reciprocity:

$$A_n F_{n,m} = A_m F_{m,n} \tag{5.105}$$

If surface m is constructed from a number of sub-surfaces, e.g. windows, doors, wall, then:

$$A_n F_{n,m} = A_n F_{n,m1} + A_n F_{n,m2} + \ldots\ldots \tag{5.106}$$

where surface m is made up from sub-surfaces $m1$, $m2$ etc.

For cases where non-rectangular or concealed surfaces are involved or where rooms are not orthogonal, numerical techniques will be necessary for calculating view factors. These methods usually make use of contour integration[(A3.4)] although statistically based methods have also been used[(A3.5)]. The application of these methods is outside of the scope of this Guide.

5.A3.4 Basic Model

The Reference Model above considered the heat transfer process within a room from the view of direct surface-to-surface radiant heat flows, surface-to-air convection and surface-to-outside conduction. The surface-to-surface radiant flow is the most difficult of these processes to model. An alternative approach is to assume that just as all convective heat input must first increase the air temperature, i.e. enters the 'air temperature node' so all radiant heat enters at the 'radiant temperature node' (θ*). Heat then flows into each room surface by means of a heat transfer coefficient that is adjusted to take account of the multiple reflections of radiation between surfaces. Davies[(A3.6)] has shown that it is possible to make a very close approximation to the exact equivalent of the radiosity matrix used in the Reference Model for a six-sided enclosure.

In such a case the radiant heat transfer coefficient is equal to the product ($E^*_n h_r$) where:

$$E^*_n = \varepsilon_n / (1 - \varepsilon_n + ß_n \varepsilon_n) \tag{5.107}$$

where $ß_n$ is given by the regression equation:

$$ß_n = 1 - f_n [1 + 3.53 (f_n - 0.5) - 5.04 (f_n^2 - 0.25)] \tag{5.108}$$

where:

$$f_n = A_n / \Sigma A \tag{5.109}$$

The standard error for the regression is 0.0068 and the exact value of $ß_n$ for a cube is $^5/_6$.

While the terms $ß_n$ are specific to a six-sided enclosure, they can often be used in most design applications. Introducing this close approximation to the radiant exchange process, greatly simplifies matrix A at the minor expense of the introduction of a new temperature θ*, as follows:

$$\left(\sum_{n=1}^{N} h_{cn} A_n + \dot{m}_a c_p\right) \theta_{ai} - \sum_{n=1}^{N} h_{cn} A_n \theta_{sn} = \Phi_{con} + \theta_{ao} \dot{m}_a c_p \tag{5.110}$$

$$\sum_{n=1}^{N} h^*_r E^*_n A_n \theta^* - \sum_{n=1}^{N} h^*_r E^*_n A_n \theta_{sn} = \Phi_n + \Phi_{rad} \tag{5.111}$$

where Φ_{con} is the convective output from an emitter (W) and Φ_{rad} is the radiant output (W).

For each surface n:

$$-A_n h_{cn} \theta_{ai} - A_n E^*_n h_r \theta^* + (h_{cn} + E^*_n h_r + U_n') A_n \theta_{sn} = Q_n + t_{on} A_n U_n' \tag{5.112}$$

where Φ_n is a heat input to surface n (W), e.g. the absorbed solar radiation incident upon the surface. For the purposes of a heat loss model, all Φ_n are set to zero.

The Basic Model is given by the equation set represented by the matrix equation:

$$A^* X^* = C^* \tag{5.113}$$

where A*, X* and C* are matrices, defined as follows.

Matrix A*:

(*a*) Surface heat balance

Terms $A^*(n,n)$ for n = 1 to n = total number of room surfaces:

$$A^*(n,n) = (h_{cn} + U_n' + E^*_n h_r) \tag{5.114}$$

Terms $A^*(n,m)$ where $n \neq$ m, for n = 1 to n = total number of room surfaces and for m = 1 to m = total number of room surfaces:

$$A^*(n,m) = 0 \tag{5.115}$$

Terms $A^*(n,m)$ for n = 1 to n = total number of room surfaces and for m = total number of room surfaces + 1:

$$A^*(n,m) = -h_{cn} \tag{5.116}$$

Terms $A^*(n,m)$ for n = 1 to n = total number of room surfaces and for m = total number of room surfaces + 2:

$$A^*(n,m) = 0 \tag{5.117}$$

Terms $A^*(n,m)$ for n = 1 to n = total number of room surfaces and for m = total number of room surfaces + 3:

$$A^*(n,m) = -E^*_n h_r \tag{5.118}$$

(*b*) Control sensor heat balance

Terms $A^*(n,n)$ for n = total number of room surfaces + 1:

$$A^*(n,n) = F_a \quad (5.119)$$

Terms $A^*(n,m)$ for n = total number of room surfaces + 1 and for m = 1 to m = total number of room surfaces:

$$A^*(n,m) = \varepsilon_m (1 - F_a) A_m / \Sigma (A\,\varepsilon) \quad (5.120)$$

Terms $A^*(n,m)$ for n = total number of room surfaces + 1 and for m = total number of room surfaces + 2:

$$A^*(n,m) = R\,(1 - F_a) / (h_r \Sigma A) \quad (5.121)$$

Terms $A^*(n,m)$ for n = total number of room surfaces + 1 and for m = total number of room surfaces + 3:

$$A^*(n,m) = 0 \quad (5.122)$$

(*c*) Convection heat balance

Terms $A^*(n,m)$ for n = total number of room surfaces + 2 and for m = 1 to m = total number of room surfaces:

$$A^*(n,m) = -h_{cn} A_m / \Sigma (A\,\varepsilon) \quad (5.123)$$

Terms $A^*(n,m)$ for n = total number of room surfaces + 2 and for m = total number of room surfaces + 1:

$$A^*(n,m) = [C_v + \Sigma (A_m h_{cm})] / \Sigma (A\,\varepsilon) \quad (5.124)$$

Terms $A^*(n,m)$ for n = total number of room surfaces + 2 and for m = total number of room surfaces + 2:

$$A^*(n,m) = (R - 1) / \Sigma A \quad (5.125)$$

Terms $A^*(n,m)$ for n = total number of room surfaces + 2 and for m = total number of room surfaces + 3:

$$A^*(n,m) = 0 \quad (5.126)$$

(*d*) Radiant heat balance

Terms $A^*(n,n)$ for n = total number of room surfaces + 3:

$$A^*(n,n) = \Sigma E^*_n h_r A_n / \Sigma A \quad (5.127)$$

Terms $A^*(n,m)$ for n = total number of room surfaces + 3 and for m = 1 to m = total number of room surfaces:

$$A^*(n,m) = -E^*_m h_r A_m / \Sigma A \quad (5.128)$$

Terms $A^*(n,m)$ for n = total number of room surfaces + 3 and for m = total number of room surfaces + 1:

$$A^*(n,m) = 0 \quad (5.129)$$

Terms $A^*(n,m)$ for n = total number of room surfaces + 3 and for m = total number of room surfaces + 2:

$$A^*(n,m) = -R / \Sigma A \quad (5.130)$$

Vector C^*:

Terms $C^*(n)$ for n = 1 to n = the total number of room surfaces:

$$C^*(n) = t_{on} U_n' \quad (5.131)$$

Terms $C^*(n)$ for n = total number of room surfaces + 1:

$$C^*(n) = t_c \quad (5.132)$$

Terms $C^*(n)$ for n = total number of room surfaces + 2:

$$C^*(n) = \theta_{ao} C_v / \Sigma A \quad (5.133)$$

Terms $C^*(n)$ for n = total number of room surfaces + 3:

$$C^*(n) = 0 \quad (5.134)$$

Solution vector X^*:

Terms $X^*(n)$ for n = 1 to n = total number of room surfaces provide the temperature of each surface.

Term $X^*(n)$ for n = total number of room surfaces + 1 provides the room air temperature.

Term $X^*(n)$ for n = total number of room surfaces + 2 provides the total heat input.

Term $X^*(n)$ for n = total number of room surfaces + 3 provides the radiant-star temperature (t^*), which is *not* the radiant temperature.

The ventilation transmittance is represented by the conventional term C_v, see equation 5.9. If it is necessary to take account of air flows from a number of sources, that term in matrix A^* should be replaced by the summation $\Sigma (\dot{m}_a c_p)_i$ where the summation covers all sources i.

In vector C^*, the term (t_{ao} C_v / Σ A) is then replaced by $[\Sigma (\theta_i \dot{m}_{ai} c_{pi}) / \Sigma A]$ where θ_i is the temperature of air from source i.

5.A3.5 Simple Model

If the radiant exchange between surfaces can be treated separately, the surface heat balance equations are decoupled and the need for matrix manipulation is removed. This leads to a manual calculation procedure.

One means of achieving this approximation is to assume that, with the exception of the subject surface, all surface temperatures are known. In this case, the heat balance on the subject surface is described by the surface heat balance equations given for the Basic Model, see equation 5.112. Hence:

$$\theta_s (h_c + U' + E^* h_r) - h_c \theta_{ai} - E^* h_r \theta^* = \theta_o U' \quad (5.135)$$

Rearranging equation 5.135 gives the fabric heat loss:

$$U' (\theta_s - \theta_o) = h_c (\theta_{ai} - \theta_s) + h_r E^* (\theta^* - \theta_s) \quad (5.136)$$

It then remains to determine a value for E^*. A simple method should use parameters that are independent of the shape of the enclosure. The simplest assumption is that the subject surface has an area equivalent to one sixth of that of the enclosure of which it forms a part. Therefore, from equation 5.107, with f_n = $^1/_6$ (see equation 5.109) and $ß_n = {^5/_6}$ (see equation 5.108):

$$E^* = \frac{\varepsilon}{(1 - \varepsilon + {^5/_6}\,\varepsilon)} \quad (5.137)$$

Now for $\varepsilon = 1$, $E^* = {}^6/_5\,\varepsilon$, hence:

$$\phi_f = h_c\,(\theta_{ai} - \theta_s) + {}^6/_5\,\varepsilon\,h_r\,(\theta^* - \theta_s) \tag{5.138}$$

Equation 5.138 may be summed for all surfaces to give the total fabric loss, that is:

$$\Phi_f = h_c \sum A\,(\theta_{ai} - \theta_m) + {}^6/_5\,\varepsilon\,h_r \sum A\,(\theta^* - \theta_m) \tag{5.139}$$

where it is assumed that h_c and h_r are constants and that:

$$\theta_m = \sum A\,\theta_s / \sum A \tag{5.140}$$

The heat input to a space comprises a radiant and convective component. From equation 5.111, the radiant component is:

$$\Phi_{rad} = {}^6/_5\,\varepsilon\,h_r \sum A\,(\theta^* - \theta_m) \tag{5.141}$$

The convective components associated with the fabric heat loss is:

$$\Phi_{con} = h_c \sum A\,(\theta_{ai} - \theta_m) \tag{5.142}$$

Equations 5.139, 5.141 and 5.142 can be expressed in analogue form by the network shown in Figure 5.9, where:

$$H_c = h_c \sum A \tag{5.143}$$

and:

$$H_r = {}^6/_5\,\varepsilon\,h_r \sum A \tag{5.144}$$

Figure 5.9 shows a radiant input Φ_{rad} acting at the radiant star node θ^*, being lost by conduction Φ_f from θ_s and by ventilation Φ_v from θ_{ai}. This network may be transformed exactly into that shown in Figure 5.10 where the rad-air node t_{ra} is located on the convective transmittance H_c, dividing it into two components: $X = H_c\,(H_c + H_r) / H_r$ and $Y = (H_c + H_r)$. An augmented flow, $\Phi_{rad}\,(1 + H_c / H_r)$ acts at θ_{ra} and the excess, $\Phi_{rad}\,(H_c / H_r)$ is withdrawn from θ_{ai}. Components X and Y can be considered, in effect, in parallel[A3.6]. The physically significant quantities, i.e. the observable temperatures θ_s and θ_{ai}, and the heat flows from them, Q_f and Q_v, can be considered the same in both cases.

There is a further transmittance, $[(H_c + H_r)\,H_c / H_r]$, between θ_{ra} and θ_{ai}. The rad-air temperature, θ_{ra}, is related to the two generating temperatures by the following equation:

$$\theta_{ra} = \frac{H_c\,\theta_{ai}}{H_c + H_r} + \frac{H_r\,\theta^*}{H_c + H_r} \tag{5.145}$$

If the mean surface temperature, t_m, is taken as an approximation for t^*, then $t_{ra} \approx t_{ei}$. Hence:

$$\theta_{ei} = \frac{H_c\,\theta_{ai}}{H_c + H_r} + \frac{H_r\,\theta_m}{H_c + H_r} \tag{5.146}$$

It is appropriate to standardise the heat transfer coefficients as follows:

$h_c = 3.0\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$ (an average figure)

$h_r = 5.7\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$ (for temperatures ≈ 20 °C)

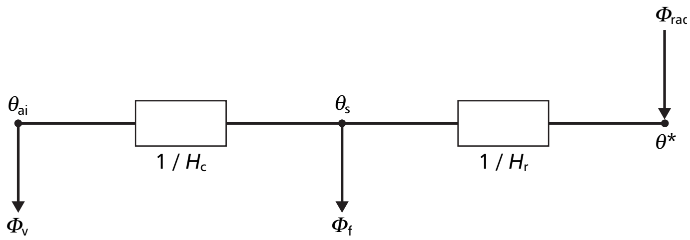


Figure 5.9 Simplified heat flow network

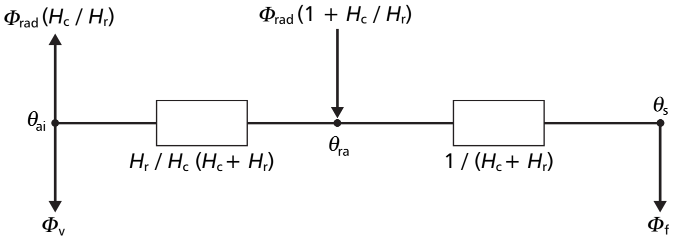


Figure 5.10 Equivalent heat flow network

$H_r / \sum A = 6.0\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$ (for $\varepsilon = 0.9$)

$H_c / \sum \mathrm{A} = 3.0\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$

Also,

$$H_a / \sum \mathrm{A} = (H_r + H_c)\,H_c / H_r = 4.5\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$$

So,

$$h_a = 4.5\ \mathrm{W\,{\cdot}m^{-2}{\cdot}K^{-1}}$$

Therefore, it follows from equation 5.146 that:

$$\theta_{ei} = {}^1/_3\,\theta_{ai} + {}^2/_3\,\theta_m \tag{5.147}$$

That is, the effective radiant heat input is 1.5 times the actual input, with the excess (50%) of radiant input subtracted from the convective component of the heat input. A further implication is that a heat source that is effectively directly linked to the environmental temperature has the characteristics of ${}^2/_3$ radiation and ${}^1/_3$ convection.

It is accepted that a number of approximations are embodied in this relationship. However empirical testing over a number of years has not revealed any serious deficiencies in practice and, as such, it is therefore accepted as the basis of the CIBSE simple heat loss model, which is developed as follows.

The heat loss due to the fabric is defined as:

$$\Phi_f = \sum_{n=1}^{N} A_n\,U_n\,(\theta_{ei} - \theta_{on}) \tag{5.148}$$

Where the fabric term contains heat loss through internal partitions, a modified U-value should be used:

$$U_p = \frac{U\,(\theta_c - \theta_c')}{(\theta_c - \theta_{ao})} \tag{5.149}$$

This correction is based on the internal design operative temperature (θ_c) and therefore is not exact. However, the operative is usually very close to the heat loss temperature

(θ_{ei}) which means that any error is small in what is already a second order correction. This approximation makes it unnecessary to determine the value of the environmental temperature in adjacent spaces.

The heat loss due to infiltration and/or ventilation by outdoor air is:

$$\Phi_v = \frac{c_p\,\rho\,N_v\,V}{3600}(\theta_{ai} - \theta_{ao}) \qquad (5.150)$$

For practical purposes ($c_p\ \rho$ / 3600) = $^1/_3$, therefore $C_v = N_v\,V/3$.

Hence:

$$\Phi_v = C_v(\theta_{ai} - \theta_{ao}) \qquad (5.151)$$

Ventilation rates must include infiltration, natural ventilation due to open windows and, where appropriate, mechanical ventilation. Guidance on ventilation requirements and design allowances for infiltration are given in chapter 1: *Environmental criteria for design* and chapter 4: *Infiltration and natural ventilation*, respectively.

The total heat loss is the sum of the fabric and infiltration losses:

$$\Phi_t = \sum_{n=1}^{N} A_n\,U_n(\theta_{ei} - \theta_{on}) + C_v(\theta_{ai} - \theta_{ao}) \qquad (5.152)$$

For winter heating design conditions it is conventional to assume that the outside heat transfer temperature (θ_{on}) equals the outside air temperature (θ_{ao}), therefore:

$$\Phi_t = \sum_{n=1}^{N} A_n\,U_n(\theta_{ei} - \theta_{ao}) + C_v(\theta_{ai} - \theta_{ao}) \qquad (5.153)$$

In order to relate the heat loss to the design operative temperature, it is necessary to eliminate θ_{ai} and θ_{ei}. This is achieved by introducing factors F_{1cu} and F_{2cu}, as follows (see Appendix 5.A5, equations 5.174 and 5.175):

$$F_{1cu} = \frac{3.0\,(C_v + 6.0\,\Sigma A)}{\Sigma(A\,U) + 18.0\,\Sigma A + 1.5\,R\,[3.0\,C_v - \Sigma(A\,U)]} \qquad (5.154)$$

$$F_{2cu} = \frac{\Sigma(A\,U) + 18.0\,\Sigma A}{\Sigma(A\,U) + 18.0\,\Sigma A + 1.5\,R\,[3.0\,C_v - \Sigma(A\,U)]} \qquad (5.155)$$

Therefore the Simple Model is:

$$\Phi_t = (F_{1cu}\sum_{n=1}^{N} A_n\,U_n + F_{2cu}\,C_v)(\theta_c - \theta_{ao}) \qquad (5.156)$$

and the corresponding air temperature is calculated using the following equation (see Appendix 5.A5.3, equation 5.164):

$$\bar{\theta}_{ai} = \frac{\bar{\Phi}_t\,(1 - 1.5\,R) + C_v\,\bar{\theta}_{ao} + 6.0\,\Sigma A\,\bar{\theta}_c}{C_v + 6.0\,\Sigma A} \qquad (5.157)$$

References for Appendix 5.A3

A3.1 Gagneau S, Nataf J M and Wurtz E An illustration of automatic generation of zonal models *Proc. Int. Building Performance Simulation Association.Conf., Prague, Czech Republic* **2** 437–444 (1997)

A3.2 *Heat transfer* CIBSE Guide C3 (London: Chartered Institution of Building Services Engineers) (2001)

A3.3 *Energy calculation procedures to determine heating and cooling loads for computer analysis* (Atlanta GA: American Society of Heating Refrigerating and Air-conditioning Engineers) (1976)

A3.4 Walton G N *Algorithms for calculating radiation view factors between plane convex polygons with obstructions* NBSIR 86-3463 (Washington DC: US Department of Commerce) (October 1986)

A3.5 Malalasekera W M G and James E H Thermal radiation in a room: numerical evaluation *Building Serv. Eng. Res. Technol.* **14** (4) 159-168 (1993)

A3.6 Davies M G An idealised model for room radiant exchange *Building and Environment* **25** (4) 375–378 (1990)

Appendix 5.A4: Comparison of thermal steady state models

This appendix contains a number of example calculations intended to demonstrate differences between the three methods described for the calculation of design heat losses. These examples will assist building services engineers in assessing the suitability of the methods for a particular application and enable software designers to demonstrate that the results obtained from computer programs are consistent with the methods described in this Guide.

The example calculations are as follows:

— a cubic enclosure with uniform *U*-value but variable internal emissivity.

— a cubic enclosure with variable *U*-value but constant emissivity (i.e. 0.9)

— a typical room, with typical *U*-values and surface emissivity

— a non-typical application; in this case an atrium

— an example of applying the calculation methods to multiple surfaces.

In each case, the following are calculated:

— heat losses for various generic system types

— surface temperatures

— radiation view factors.

During the development of software validation tests, it was felt that the definition of mean radiant temperature used in the Reference and Basic models should be revised. Therefore the values calculated for these models differ from those given in the 1999 edition of this Guide. No changes have been made to the Simple Model.

Example A5.1: Cubic enclosure; uniform U-values, varying emissivities

See Figure 5.11 and Tables 5.42 to 5.45. This example demonstrates a weakness of the Simple Model when emissivities differ from 0.9. It offers a means of checking that a computer program can account for variable emissivity.

Operative temperature: 21 °C
Outside air temperature: –1 °C
Infiltration rate: 0.5 h^{-1}

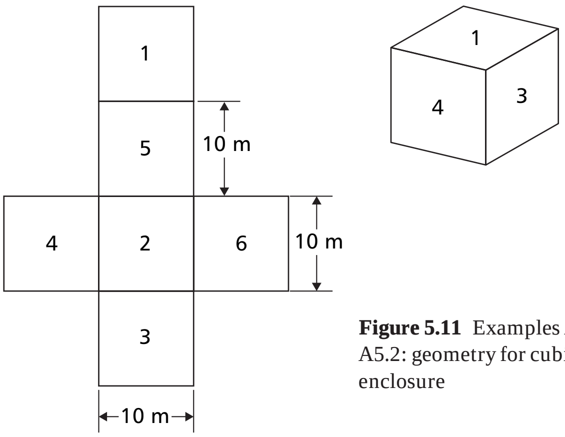


Figure 5.11 Examples A5.1 and A5.2: geometry for cubic enclosure

Table 5.42 Example A5.1: surface data

Surface number	Area / m^2	*U*-value / W·m^{-2}·K^{-1}	Emissivity of surface, ε_n	Convective heat transfer coefficient, h_c	Inside surface resistance, R_{si} / m^2·K·W^{-1}	Temp. on outer side of surface / °C
1	100	1.00	1.00	3.00	0.12	–1.0
2	100	1.00	1.00	3.00	0.12	–1.0
3	100	1.00	0.8	3.00	0.12	–1.0
4	100	1.00	0.6	3.00	0.12	–1.0
5	100	1.00	0.4	3.00	0.12	–1.0
6	100	1.00	0.2	3.00	0.12	–1.0

Table 5.43 Example A5.1: heat loss

Emitter characteristics (% convective)	Component of heat loss	Heat loss using Reference Model / W	Percentage difference using stated model	
			Basic Model	Simple Model
100	Fabric	12912	0	–6.4
	Ventilation	4258	0	–0.6
	Total	17170	0	–4.8
70	Fabric	13209	0	–4.6
	Ventilation	3886	0	0.7
	Total	17095	0	–3.2
30	Fabric	13600	0	–2.4
	Ventilation	3396	0	2.1
	Total	16996	0	–1

Table 5.44 Example A5.1: surface temperatures

Emitter characteristics (% convective)	Surface number	Surface temperature using Reference Model / °C	Difference in calculated surface temperature using stated model / K	
			Basic Model	Simple Model
100	1	17.52	0	–0.2
	2	17.52	0	–0.2
	3	17.49	0	0.11
	4	17.43	0	0.52
	5	17.29	0	1.09
	6	16.97	0	1.9
70	1	18.6	0	0.28
	2	18.6	0	0.28
	3	18.29	0	0.32
	4	17.88	0	0.4
	5	17.28	0	0.52
	6	16.33	0	0.75
30	1	20.01	0	0.89
	2	20.01	0	0.89
	3	19.34	0	0.59
	4	18.47	0	0.22
	5	17.26	0	–0.24
	6	15.48	0	–0.79

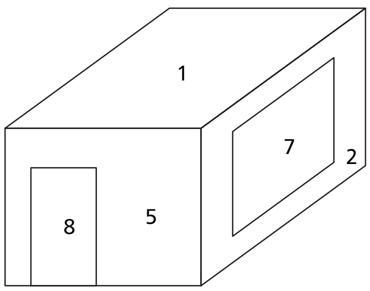
Table 5.45 Example A5.1 radiation view factors

Surface number	View factor for stated surface as viewed from surface indicated in first column					
	1	2	3	4	5	6
1	0	0.2000	0.2000	0.2000	0.2000	0.2000
2	0.2000	0	0.2000	0.2000	0.2000	0.2000
3	0.2000	0.2000	0	0.2000	0.2000	0.2000
4	0.2000	0.2000	0.2000	0	0.2000	0.2000
5	0.2000	0.2000	0.2000	0.2000	0	0.2000
6	0.2000	0.2000	0.2000	0.2000	0.2000	0

Example A5.2: Typical room; varying U-values, uniform emissivities

See Figure 5.12 and Tables 5.46 to 5.49. This example indicates the uncertainties likely to occur in a typical design situation.

Operative temperature: 21 °C
Outside air temperature: –1 °C
Infiltration rate: 1.0 h^{-1}



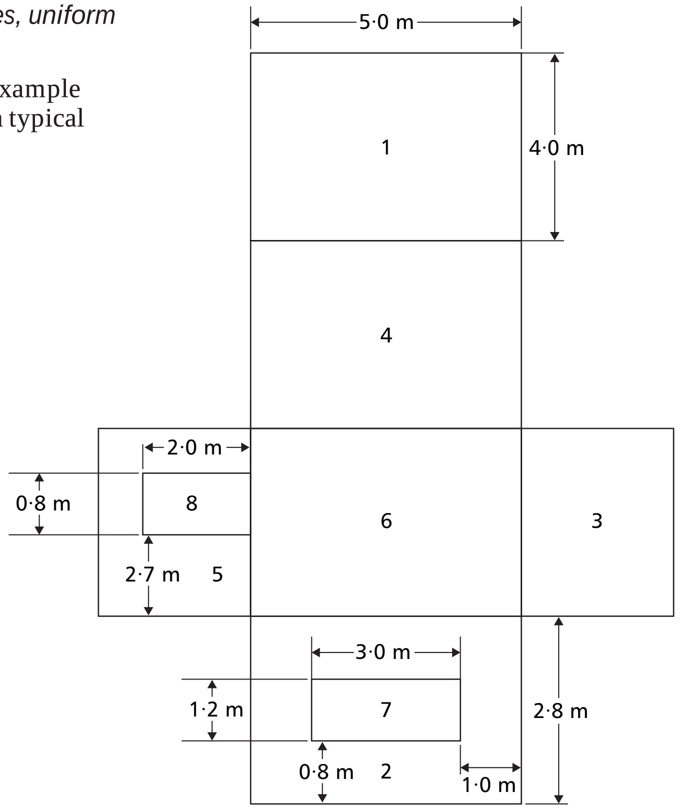


Figure 5.12 Example A5.2: geometry for typical room

Table 5.46 Example A5.2: surface data

Surface number	Area / m^2	*U*-value / W·m^{-2}·K^{-1}	Emissivity of surface, ε_n	Convective heat transfer coefficient, h_c	Inside surface resistance, R_{si} / m^2·K·W^{-1}	Temperature on outer side of surface / °C
1	20.0	0.4	0.9	4.3	0.1	–1.0
2	10.4	0.6	0.9	3.0	0.12	–1.0
3	11.2	2.5	0.9	3.0	0.12	18.0
4	14.0	2.5	0.9	3.0	0.12	18.0
5	9.6	0.6	0.9	3.0	0.12	–1.0
6	20.0	0.2	0.9	1.5	0.14	–1.0
7	3.6	3.0	0.9	3.0	0.12	–1.0
8	1.6	2.0	0.9	3.0	0.12	–1.0

Table 5.47 Example A5.2: heat loss

Emitter characteristics (% convective)	Component of heat loss	Heat loss using Reference Model / W	Percentage difference using stated model	
			Basic Model	Simple Model
100	Fabric	959	0.1	3.9
	Ventilation	444	0	0.2
	Total	1403	0.1	2.7
50	Fabric	1018	–0.2	0.8
	Ventilation	408	0	0.3
	Total	1426	–0.1	0.7

Table 5.48 Example A5.2: surface temperatures

Emitter characteristics (% convective)	Surface number	Surface temperature using Reference Model / °C	Difference in calculated surface temperature using stated model / K	
			Basic Model	Simple Model
100	1	19.85	–0.01	0.28
	2	19.12	0.12	0.14
	3	19.77	–0.01	–0.35
	4	19.72	–0.05	–0.4
	5	19.01	0.01	0.02
	6	19.37	–0.02	–0.45
	7	12.5	0.04	–0.31
	8	15.15	0.05	0.19
50	1	21.15	–0.06	–0.04
	2	19.7	0.18	0.11
	3	20.21	0.03	0.03
	4	20.16	–0.04	–0.02
	5	19.58	0.07	–0.01
	6	20.44	–0.1	–0.02
	7	12.82	0.11	–0.41
	8	15.53	0.15	–0.3

Table 5.49 Example A5.2: radiation view factors

Surface number	View factor for stated surface as viewed from surface indicated in first column							
	1	2	3	4	5	6	7	8
1	0	0.1358	0.1458	0.1849	0.1289	0.3387	0.0491	0.0169
2	0.2611	0	0.1544	0.1690	0.1449	0.2611	0	0.0095
3	0.2604	0.1434	0	0.1845	0.0943	0.2604	0.0411	0.0159
4	0.2641	0.1255	0.1476	0	0.1158	0.2641	0.0511	0.0318
5	0.2686	0.1570	0.1101	0.1689	0	0.2525	0.0429	0
6	0.3387	0.1358	0.1458	0.1849	0.1212	0	0.0491	0.0246
7	0.2727	0	0.1279	0.1986	0.1143	0.2727	0	0.0136
8	0.2107	0.0616	0.1116	0.2782	0	0.3073	0.0306	0

Example A5.3: Atrium; varying U-values, varying emissivities

See Figure 5.13 and Tables 5.50 to 5.53. This example illustrates a non-typical application, in this case an atrium. It is intended mainly to assist in comparing computer programs.

Operative temperature: 21 °C
Outside air temperature: –1 °C
Infiltration rate: 0.1 h^{-1}

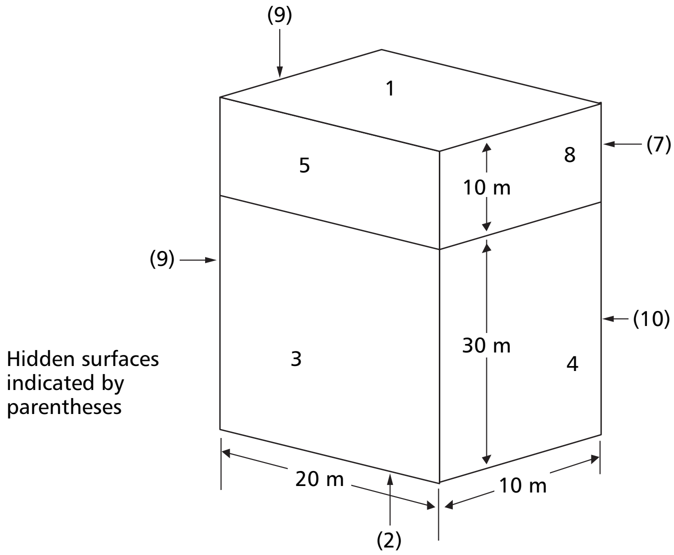


Figure 5.13 Example A5.3: geometry for atrium

Table 5.50 Example A5.3: surface data

Surface number	Area / m^2	U-value / $W \cdot m^{-2} \cdot K^{-1}$	Emissivity of surface, ε_n	Convective heat transfer coefficient, h_c	Inside surface resistance, R_{si} / $m^2 \cdot K \cdot W^{-1}$	Temperature on outer side of surface / °C
1	200.0	3.0	0.8	4.3	0.1	–4.0
2	200.0	0.7	0.9	1.5	0.14	–1.0
3	600.0	2.0	0.9	3.0	0.12	21.0
4	300.0	2.0	0.9	3.0	0.12	21.0
5	200.0	2.0	0.8	3.0	0.12	21.0
6	100.0	3.0	0.8	3.0	0.12	–1.0
7	200.0	3.0	0.8	3.0	0.12	–1.0
8	100.0	3.0	0.8	3.0	0.12	–1.0
9	300.0	0.45	0.9	3.0	0.12	–1.0
10	600.0	0.45	0.9	3.0	0.12	–1.0

Table 5.51 Example A5.3: heat loss

Emitter characteristics (% convective)	Component of heat loss	Heat loss using Reference Model / W	Percentage difference using stated model	
			Basic Model	Simple Model
100	Fabric	46923	4.7	8.6
	Ventilation	6976	0.8	1.7
	Total	53489	4.2	7.7
50	Fabric	49903	4.5	6.1
	Ventilation	5928	0.4	1.3
	Total	55831	4.1	5.6

Table 5.52 Example A5.3: surface temperatures

Emitter characteristics (% convective)	Surface number	Surface temperature using Reference Model / °C	Difference in calculated surface temperature using stated model / K	
			Basic Model	Simple Model
100	1	11.84	–1.02	–0.69
	2	18.44	0.99	0.37
	3	20.83	0.56	–0.17
	4	20.93	0.51	–0.07
	5	18.71	–1.88	–2.29
	6	11.22	–1.14	–1.34
	7	11.35	–1.06	–1.2
	8	11.19	–1.17	–1.36
	9	19.79	0.65	0.83
	10	19.96	0.9	1.01
50	1	11.97	–0.99	–1.22
	2	19.78	1.05	0.95
	3	21.46	0.47	0.46
	4	21.55	0.54	0.55
	5	19.14	–1.88	–1.86
	6	11.58	–1.12	–1.51
	7	11.71	–1.06	–1.38
	8	11.56	–1.14	–1.54
	9	20.55	0.69	0.8
	10	20.72	0.78	0.97

Table 5.53 Example A5.3: radiation view factors

Surface	View factor for stated surface as viewed from surface indicated in first column									
	1	2	3	4	5	6	7	8	9	10
1	0	0.0362	0.0740	0.0509	0.2406	0.1164	0.2406	0.1164	0.0509	0.0740
2	0.0362	0	0.3081	0.1617	0.0065	0.0056	0.0065	0.0056	0.1617	0.3081
3	0.0247	0.1027	0	0.1595	0	0.0121	0.0539	0.0121	0.1595	0.4756
4	0.0339	0.1078	0.3190	0	0.0242	0.0255	0.0242	0	0.1464	0.3190
5	0.2406	0.0065	0	0.0362	0	0.1164	0.2859	0.1164	0.0362	0.1617
6	0.2329	0.0112	0.0725	0.0766	0.2329	0	0.2329	0.0686	0	0.0725
7	0.2406	0.0065	0.1617	0.0362	0.2859	0.1164	0	0.1164	0.0362	0
8	0.2329	0.0112	0.0725	0	0.2329	0.0686	0.2329	0	0.0766	0.0725
9	0.0339	0.1078	0.3190	0.1464	0.0242	0	0.0242	0.0255	0	0.3190
10	0.0247	0.1027	0.4756	0.1595	0.0539	0.0121	0	0.0121	0.1595	0

Example A5.4: Enclosure with multiple surfaces; varying U-values, uniform emissivities

See Figure 5.14 and Tables 5.54 to 5.57. In this example, the calculation methods are applied to multiple surfaces. It is intended to assist in checking computer programs to determine the accuracy with which surface temperatures are calculated.

Operative temperature: 21 °C
Outside air temperature: –1 °C
Infiltration rate: 1.0 h^{-1}

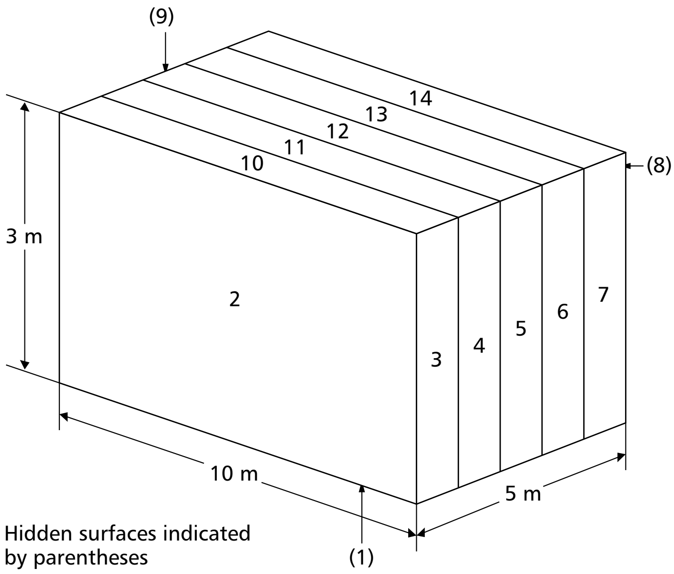


Figure 5.14 Example A5.4: geometry for enclosure with multiple surfaces

Table 5.54 Example A5.4: surface data

Surface number	Area / m^2	*U*-value / W·m^{-2}·K^{-1}	Emissivity of surface, ε_n	Convective heat transfer coefficient, h_c	Inside surface resistance, R_{si} / m^2·K·W^{-1}	Temperature on outer side of surface / °C
1	50.0	1.0	0.8	1.5	0.14	–1.0
2	30.0	5.6	0.8	3.0	0.12	–1.0
3	3.0	1.0	0.8	3.0	0.12	–1.0
4	3.0	1.0	0.8	3.0	0.12	–1.0
5	3.0	1.0	0.8	3.0	0.12	–1.0
6	3.0	1.0	0.8	3.0	0.12	–1.0
7	3.0	1.0	0.8	3.0	0.12	–1.0
8	30.0	1.0	0.8	3.0	0.12	–1.0
9	15.0	1.0	0.8	3.0	0.12	–1.0
10	10.0	1.0	0.8	4.3	0.10	–1.0
11	10.0	1.0	0.8	4.3	0.10	–1.0
12	10.0	1.0	0.8	4.3	0.10	–1.0
13	10.0	1.0	0.8	4.3	0.10	–1.0
14	10.0	1.0	0.8	4.3	0.10	–1.0

Table 5.55 Example A5.4: heat loss

Emitter characteristics (% convective)	Component of heat loss	Heat loss using Reference Model / W	Percentage difference using stated model	
			Basic Model	Simple Model
100	Fabric	6560	–0.2	0.4
	Ventilation	1402	0.1	–1.8
	Total	7962	–0.2	0.1
50	Fabric	7116	0.1	0.2
	Ventilation	1141	–0.1	0.0
	Total	8256	0.0	0.1

Table 5.56 Example A5.4: surface temperatures

Emitter characteristics (% convective)	Surface number	Surface temperature using Reference Model / °C	Difference in calculated surface temperature using stated model / K	
			Basic Model	Simple Model
100	1	15.07	0.25	–1.13
	2	5.74	0.02	–0.06
	3	16.06	–0.91	–0.64
	4	16.71	–0.26	0.01
	5	17.07	0.1	0.37
	6	17.3	0.33	0.6
	7	17.46	0.48	0.75
	8	16.94	0.09	0.23
	9	16.92	–0.01	0.22
	10	17.26	–1.03	0.24
	11	17.81	–0.47	0.79
	12	18.18	–0.11	1.16
	13	18.42	0.14	1.4
	14	18.61	0.32	1.59
50	1	17.66	–0.19	0.05
	2	6.28	0	–0.08
	3	17.42	–0.77	–0.75
	4	18.14	–0.06	–0.03
	5	18.53	0.34	0.36
	6	18.77	0.58	0.61
	7	18.92	0.73	0.76
	8	18.37	0.04	0.21
	9	18.35	0.11	0.19
	10	17.87	–0.85	–0.64
	11	18.51	–0.21	0
	12	18.91	0.2	0.41
	13	19.17	0.45	0.66
	14	19.33	0.61	0.82

Table 5.57 Example A5.4: radiation view factors

Surface number	View factor for stated surface as viewed from surface indicated in first column													
	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	0	0.1867	0.0155	0.0188	0.0196	0.0188	0.0155	0.1867	0.0083	0.0783	0.0957	0.1019	0.0957	0.0783
2	0.3112	0	0.0359	0.0220	0.0151	0.0109	0.0082	0.1934	0.0921	0.1311	0.0803	0.0490	0.0307	0.0211
3	0.2585	0.3593	0	0	0	0	0	0.0821	0.0416	0.1361	0.0652	0.0304	0.0168	0.0101
4	0.3136	0.2198	0	0	0	0	0	0.1092	0.0438	0.0652	0.1361	0.0652	0.0304	0.0168
5	0.3272	0.1505	0	0	0	0	0	0.1505	0.0445	0.0304	0.0652	0.1361	0.0652	0.0304
6	0.3136	0.1092	0	0	0	0	0	0.2198	0.0438	0.0168	0.0304	0.0652	0.1361	0.0652
7	0.2585	0.0821	0	0	0	0	0	0.3593	0.0416	0.0101	0.0168	0.0304	0.0652	0.1361
8	0.3112	0.1934	0.0082	0.0109	0.0151	0.0220	0.0359	0	0.0921	0.0200	0.0307	0.0409	0.0803	0.1311
9	0.2943	0.1842	0.0083	0.0088	0.0089	0.0088	0.0083	0.1842	0	0.0517	0.0627	0.0654	0.0627	0.0517
10	0.3916	0.3934	0.0408	0.0196	0.0091	0.0050	0.0030	0.0600	0.0776	0	0	0	0	0
11	0.4786	0.2409	0.0196	0.0408	0.0196	0.0091	0.0050	0.0921	0.0941	0	0	0	0	0
12	0.5094	0.1471	0.0091	0.0196	0.0408	0.0196	0.0091	0.1471	0.0982	0	0	0	0	0
13	0.4786	0.0921	0.0050	0.0091	0.0196	0.0408	0.0196	0.2409	0.0941	0	0	0	0	0
14	0.3916	0.0600	0.0030	0.0050	0.0091	0.0196	0.0408	0.3934	0.0776	0	0	0	0	0

Appendix 5.A5: Equations for determination of sensible heating and cooling loads

5.A5.1 Introduction

The Simple (dynamic) Model is based on the assumption that all loads and heat flows comprise a daily mean (or steady state) value and an alternating component (i.e. swing, or deviation from the mean). The total load (or temperature) is the sum of these two components. Thus the calculation of the space load requires the evaluation of each. The space load comprises the following elements:

- solar gain through glazing
- internally generated heat
- conduction through walls and glazing
- air infiltration.

The first of these is treated separately, see Appendix 5.A7.

Internally generated heat gains are calculated from design conditions. For the purposes of the Simple Model a single design level is assumed constant during the occupied hours. Thus the mean level is the sum of the gain over the occupied hours divided by 24, the number of hours in a day. The swing is the difference between the design load during the occupied hours and the daily mean value. These two elements must be divided into convective and radiant components. The model also assumes that heat is added (or removed) at two nodes, the air node and the environmental node. In the case of radiant gains 150% of that gain is realised at the environmental node (a hypothetical heat transfer node) with the excess 50% removed at the air node, see Appendix 5.A3, section 5.A3.5. The whole of the convective gain is realised at the air node.

Conduction and infiltration gains are discussed in the following sections where the mean and alternating components are handled separately. Furthermore, it is recognised that the design intent may be to cool to either a specific operative or air temperature therefore calculations for both cases are given.

For the purposes of this appendix, emitter output is taken as positive for heating and negative for cooling. When applying the equations, they are presented such that both heating and cooling loads are expressed as positive values. This also necessitates the use of a single symbol (Φ_p) for the emitter output. When applied, cooling is distinguished from heating by replacing Φ_p by Φ_k in the case of cooling load.

5.A5.2 Notation

Symbols used in this appendix are as follows. Some of the quantities occur in three forms: the instantaneous value, which is denoted by the appropriate letter (e.g. X). The 24-hour mean or steady state value, denoted by $\bar{X}$; and the instantaneous variation about the mean value, denoted by $\tilde{X}$. Where appropriate, the variation symbol is given a subscript to indicate the time at which it occurs, e.g. $\tilde{X}_t$ is the value of $\tilde{X}$ at time t.

A	Surface area (m^2)
C_v	Ventilation conductance (W·K^{-1})
c_p	Specific heat capacity of air (J·kg^{-1}·K^{-1})
f	Decrement factor
F_{au}	Room conduction factor with respect to air node
F_{ay}	Room admittance factor with respect to air node
F_{cu}	Room conduction factor with respect to operative temperature
F_{cy}	Room admittance factor with respect to operative temperature
F_{1au}, F_{2au}	Conduction factors related to characteristics of heat source with respect to air temperature
F_{1cu}, F_{2cu}	Conduction factors related to characteristics of heat source with respect to operative temperature
F_{1ay}, F_{2ay}	Admittance factors related to characteristics of heat source with respect to air temperature
F_{1cy}, F_{2cy}	Admittance factors related to characteristics of heat source with respect to operative temperature
h_a	Heat transfer coefficient between air and environmental nodes (W·m^{-2}·K^{-1})
h_c	Convective heat transfer coefficient (W·m^{-2}·K^{-1})
h_r	Radiative heat transfer coefficient of a black body (W·m^{-2}·K^{-1})
q_v	Total ventilation (mechanical plus infiltration) rate (m^3·s^{-1})
R	Radiant fraction of the heat source
t	Time (h)
U	Thermal transmittance (W·m^{-2}·K^{-1})
Y	Thermal admittance (W·m^{-2}·K^{-1})
θ_{ai}	Inside air temperature (°C)
θ_{ao}	Outside air temperature (°C)
θ_c	Operative temperature at centre of room (°C)
θ_{ei}	Environmental temperature (°C)
θ_{eo}	Sol-air temperature (°C)
θ_m	Surface temperature (°C)
θ_r	Mean radiant temperature (°C)
Φ_a	Heat flow to the air node (W)
Φ_k	Cooling load (W)
Φ_e	Heat flow to the environmental node (W)
Φ_p	Emitter output (W)
Φ_{pa}	Emitter output supplied to the air node (W)
Φ_{pe}	Emitter output supplied to the environmental node (W)
Φ_{sg}	Solar gain (W)
ρ	Density of air (kg·m^{-3})
ΣA	Sum of room surface areas, unless otherwise indicated (m^2)
$\Sigma (A\,U)$	Sum of the products of surface area and corresponding thermal transmittance over surfaces through which heat flow occurs (W·K^{-1})
$\Sigma (A\,Y)$	Sum of the products of surface area and corresponding thermal admittance over all surfaces (W·K^{-1})
$\Sigma \bar{\Phi}_{con}$	Sum of daily mean convective heat gains (W)
$\Sigma \bar{\Phi}_{rad}$	Sum of daily mean radiant heat gains (W)

5.A5.3 Calculation of the steady state load

The Simple (steady state) Model assumes that heat enters the space at two points: the air node and a hypothetical node called the environmental temperature node. For a convective heat input, the heat flow into these nodes may be written as (see also Appendix 5.A3, section 5.A3.5):

Heat flow to air node:

$$\bar{\Phi}_a = C_v (\bar{\theta}_{ai} - \bar{\theta}_{ao}) - h_c \Sigma A (\bar{\theta}_m - \bar{\theta}_{ai}) \quad (5.158)$$

Heat flow to environmental node:

$$\bar{\Phi}_e = \Sigma (A\,U) (\bar{\theta}_{ei} - \bar{\theta}_{eo}) + h_a \Sigma A (\bar{\theta}_{ei} - \bar{\theta}_{ai}) \quad (5.159)$$

The method also defines the following relationships:

(*a*) Environmental temperature:

$$\bar{\theta}_{ei} = {}^1/_3\, \bar{\theta}_{ai} + {}^2/_3\, \bar{\theta}_r \quad (5.160)$$

(*b*) Operative temperature

$$\bar{\theta}_c = {}^1/_2\, \bar{\theta}_{ai} + {}^1/_2\, \bar{\theta}_r \quad (5.161)$$

The definition of environmental temperature follows from the introduction of standard heat transfer coefficients (see Appendix 5.A3, section 5.A3.5). It is therefore rational to use the same coefficients in the following derivation. That is:

$$h_c = 3.0\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$$

$${}^6/_5\, \varepsilon\, h_r = 6.0\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$$

$$h_a = 4.5\ \mathrm{W{\cdot}m^{-2}{\cdot}K^{-1}}$$

Now, all gains to the space have been expressed in terms of gain to the air and environmental temperature nodes. Therefore to calculate the heating or cooling load to maintain to a specific operative temperature it is necessary to eliminate $\bar{\theta}_{ei}$ and $\bar{\theta}_{ai}$ from equations 5.158 and 5.159, while for a specific air temperature, $\bar{\theta}_{ei}$ and $\bar{\theta}_c$ are not required. Rearranging equation 5.161 and substituting into equation 5.160 provides the following relationships that can be used to eliminate the unwanted variables:

$$\bar{\theta}_m = 2\,\bar{\theta}_c - \bar{\theta}_{ai} \quad (5.162)$$

$$\bar{\theta}_{ai} = 4\,\bar{\theta}_c - 3\,\bar{\theta}_{ei} \quad (5.163)$$

Substituting for $\bar{\theta}_m$ and replacing the heat transfer coefficients with the standard numerical values given above, equation 5.158 gives:

$$\bar{\theta}_{ai} = \frac{\bar{\Phi}_a + C_v\,\bar{\theta}_{ao} + 6.0\,\Sigma A\,\bar{\theta}_c}{C_v + 6.0\,\Sigma A} \quad (5.164)$$

Similarly, equation 5.159 gives:

$$\bar{\theta}_{ei} = \frac{\bar{\Phi}_e + \Sigma (A\,U)\,\bar{\theta}_{eo} + 18.0\,\Sigma A \bar{\theta}_c}{\Sigma (A\,U) + 18.0\,\Sigma A} \quad (5.165)$$

5.A5.3.1 Convective heating/cooling source for control on operative temperature

Equations 5.164 and 5.165 together with equation 5.163 can be manipulated to give the convective heat input ($\bar{\Phi}_a$) for control at a specific operative temperature as:

$$\bar{\Phi}_a = C_v (\bar{\theta}_c - \bar{\theta}_{ao}) + F_{cu} [\Sigma (A\,U) (\bar{\theta}_c - \bar{\theta}_{eo}) - \bar{\Phi}_e] \quad (5.166)$$

where:

$$F_{cu} = \frac{3.0\,(C_v + 6.0\,\Sigma A)}{\Sigma (A\,U) + 18.0\,\Sigma A} \quad (5.167)$$

The steady state, or mean, load for control of operative temperature is calculated by including solar gains, occupancy gains and equipment and lighting gains in equation 5.161. These gains comprise both radiant and convective components. In the case of radiant gains, 150% is released as a gain to the environmental node and 50% as a gain to the convective node, see Appendix 5.A3, section 5.A3.5. Thus the mean convective cooling load is:

$$\bar{\Phi}_a = C_v (\bar{\theta}_c - \bar{\theta}_{ao}) + F_{cu} [\Sigma (A\,U) (\bar{\theta}_c - \bar{\theta}_{eo}) - \bar{\Phi}_e - 1.5\,\Sigma\,\bar{\Phi}_{rad}] - \Sigma\,\bar{\Phi}_{con} + 0.5\,\Sigma\,\bar{\Phi}_{rad} \quad (5.168)$$

where $\Sigma\,\bar{\Phi}_{rad}$ and $\Sigma\,\bar{\Phi}_{con}$ are the sums of the daily mean radiant and convective gains, respectively (cooling loads are negative).

The extension of this calculation to cover a combination of convective and radiant heating or cooling sources is given below, see section 5.A5.3.3.

Note that in equation 5.168 and elsewhere, the shortwave radiant component of lighting, i.e. the visible light, is assumed to form part of the longwave radiant component.

5.A5.3.2 Convective heating/cooling source for control on air temperature

In this case, equation 5.163 is used to provide a value for $\bar{\theta}_c$ in equations 5.164 and 5.165, thus:

$$\bar{\Phi}_a = C_v (\bar{\theta}_{ai} - \bar{\theta}_{ao}) + F_{au} [\Sigma (A\,U) (\bar{\theta}_{ai} - \bar{\theta}_{eo}) - \bar{\Phi}_e - 1.5\,\Sigma\,\bar{\Phi}_{rad}] - \Sigma\,\bar{\Phi}_{con} + 0.5\,\Sigma\,\bar{\Phi}_{rad} \quad (5.169)$$

where:

$$F_{au} = \frac{4.5\,\Sigma A}{\Sigma (A\,U) + 4.5\,\Sigma A} \quad (5.170)$$

The calculation then follows that for control on the operative temperature, see section 5.A5.3.1.

5.A5.3.3 Combined convective and radiant heating/cooling sources for control on operative temperature

If the emitter output is Φ_p with a radiant fraction of R (where R = 1.0 for a 100% radiant load), then the heat supplied to the air node (Φ_{pa}) is:

$$\Phi_{pa} = \Phi_p (1 - R) - 0.5\,\Phi_p\,R = \Phi_p (1 - 1.5\,R) \quad (5.171)$$

and the heat supplied to the environmental node (Φ_{pe}) is given by:

$$\Phi_{pe} = 1.5\,Q_p\,R - \Phi_{sg} \quad (5.172)$$

where Φ_{sg} is the solar gain (W).

From equation 5.168, by replacing $\bar{\Phi}_a$ and $\bar{\Phi}_e$ by $\bar{\Phi}_{pa}$ and $\bar{\Phi}_{pe}$, respectively, and substituting from equations 5.171 and 5.172, the daily mean, or steady state, load is:

$$\bar{\Phi}_p = F_{1cu}\,[\Sigma\,(A\,U)\,(\bar{\theta}_c - \bar{\theta}_{eo}) - \bar{\Phi}_{sg} - 1.5\,\Sigma\,\bar{\Phi}_{rad}] + F_{2cu}\,[C_v\,(\bar{\theta}_c - \bar{\theta}_{ao}) - \Sigma\,\bar{\Phi}_{con} + 0.5\,\Sigma\,\bar{\Phi}_{rad}] \qquad (5.173)$$

where:

$$F_{1cu} = \frac{3.0\,(C_v + 6.0\,\Sigma\,A)}{\Sigma\,(A\,U) + 18.0\,\Sigma\,A + 1.5\,R\,[3.0\,C_v - \Sigma\,(A\,U)]} \qquad (5.174)$$

$$F_{2cu} = \frac{\Sigma\,(A\,U) + 18.0\,\Sigma\,A}{\Sigma\,(A\,U) + 18.0\,\Sigma\,A + 1.5\,R\,[3.0\,C_v - \Sigma\,(A\,U)]} \qquad (5.175)$$

For the purposes of calculation of the steady state design heat loss (where all internal gains are ignored), equation 5.173 reduces to:

$$\bar{\Phi}_p = [F_{1cu}\,\Sigma\,(A\,U) + F_{2cu}\,C_v]\,(\bar{\theta}_c - \bar{\theta}_{ao}) \qquad (5.176)$$

where it is assumed that $\bar{\theta}_{eo}$ and $\bar{\theta}_{ao}$ are equal.

The corresponding air temperature, θ_{ai}, is obtained by substituting Φ_{pa} for Φ_a in equation 5.164 and then replacing Φ_{pa} by Φ_p using equation 5.169, hence:

$$\theta_{ai} = \frac{\Phi_p\,(1 - 1.5\,R) + C_v\,\theta_{ao} + 6.0\,\Sigma\,A\,\theta_c}{C_v + 6.0\,\Sigma\,A} \qquad (5.177)$$

5.A5.3.4 Combined convective and radiant heating/cooling sources for control on air temperature

In this case, the emitter load relationships given by equations 5.171 and 5.172 are substituted into equation 5.169 to give:

$$\bar{\Phi}_p = F_{1au}\,[\Sigma\,(A\,U)\,(\bar{\theta}_{ai} - \bar{\theta}_{eo}) - \bar{\Phi}_{sg} - 1.5\,\Sigma\,\bar{\Phi}_{rad}] + F_{2au}\,[C_v\,(\bar{\theta}_{ai} - \bar{\theta}_{ao}) - \Sigma\,\bar{\Phi}_{con} + 0.5\,\Sigma\,\bar{\Phi}_{rad}] \qquad (5.178)$$

where:

$$F_{1au} = \frac{4.5\,\Sigma\,A}{(1 - 1.5\,R)\,\Sigma\,(A\,U) + 4.5\,\Sigma\,A} \qquad (5.179)$$

$$F_{2au} = \frac{\Sigma\,(A\,U) + 4.5\,\Sigma\,A}{(1 - 1.5\,R)\,\Sigma\,(A\,U) + 4.5\,\Sigma\,A} \qquad (5.180)$$

For the purposes of calculation of the steady state design heat loss, equation 5.178 reduces to:

$$\bar{\Phi}_p = [F_{1au}\,\Sigma\,(A\,U) + F_{2au}\,C_v]\,(\bar{\theta}_c - \bar{\theta}_{ao}) \qquad (5.181)$$

5.A5.4 Alternating component of cooling load

5.A5.4.1 Convective cooling for control on operative temperature

This may be derived in a similar way to that for the mean cooling loads. However, in this case the fabric heat load is dependent on the thermal admittance of the surfaces. Thus the heat flow to the air node is:

$$\tilde{\Phi}_{at} = C_v\,\tilde{\theta}_{ai\,t} - h_c\,\Sigma\,A\,(\tilde{\theta}_{mt} - \tilde{\theta}_{ai\,t}) \qquad (5.182)$$

Note: equation 5.182 assumes that changes in the ventilation load due to fluctuations in external temperature are taken into account separately.

Heat flow to the environmental node is:

$$\tilde{\Phi}_{et} = \Sigma\,(A\,Y)\,\tilde{\theta}_{ei\,t} + h_a\,\Sigma\,A\,(\tilde{\theta}_{ei\,t} - \tilde{\theta}_{ai\,t}) + \Sigma\,(A\,f\,U\,\tilde{\theta}_{eo}) + \tilde{\Phi}_{sg} \qquad (5.183)$$

where $\tilde{\Phi}_{sg}$ is the solar gain at the environmental node (W).

Note: it is assumed that phase differences are not significant.

Thus the alternating component of the cooling load for control to the operative temperature is:

$$\tilde{\Phi}_{at} = C_v\,\tilde{\theta}_{ct} + F_{cy}\,[\Sigma\,(A\,Y)\,\tilde{\theta}_{ct} + \Sigma\,(A\,f\,U\,\tilde{\theta}_{eo}) - \tilde{\Phi}_{sg} - 1.5\,\tilde{\Phi}_{rad}] - \Sigma\,\tilde{\Phi}_{con} + 0.5\,\tilde{\Phi}_{rad} \qquad (5.184)$$

where:

$$F_{cy} = \frac{3.0\,(C_v + 6.0\,\Sigma\,A)}{\Sigma\,(A\,Y) + 18.0\,\Sigma\,A} \qquad (5.185)$$

Note: $\tilde{\theta}_{ct} = 0$ for 24-hour plant operation.

5.A5.4.2 Convective cooling for control on air temperature

The alternating component of the cooling load for control to the air temperature is:

$$\tilde{\Phi}_{at} = C_v\,\tilde{\theta}_{at} + F_{ay}\,[\Sigma\,(A\,Y)\,\tilde{\theta}_{at} - \tilde{\Phi}_{et} - 1.5\,\Sigma\tilde{\Phi}_{rad} - \tilde{\Phi}_{sg}] - \Sigma\,\tilde{\Phi}_{con} + 0.5\,\tilde{\Phi}_{rad} \qquad (5.186)$$

where:

$$F_{ay} = \frac{4.5\,\Sigma\,A}{\Sigma\,(A\,Y) + 4.5\,\Sigma\,A} \qquad (5.187)$$

Note: $\tilde{\theta}_{at} = 0$ for 24-hour plant operation.

5.A5.4.3 Combined convective and radiant cooling for control on operative temperature

The emitter load relationships given in section 5.A5.3.3 can also be applied to the alternating component of the emitter load. In this case substitution of equations 5.171 and 5.172 into equation 5.176 gives:

$$\tilde{\Phi}_{\mathrm{p}t} = F_{1\mathrm{cy}}\,[\Sigma\,(A\,Y)\,\tilde{\theta}_{\mathrm{c}t} + \Sigma\,A\,f\,U\,\tilde{\theta}_{\mathrm{eo}} - 1.5\,\Sigma\,\tilde{\Phi}_{\mathrm{rad}} - \tilde{\Phi}_{\mathrm{sg}}] + F_{2\mathrm{cy}}\,[C_{\mathrm{v}}\,\tilde{\theta}_{\mathrm{c}t} - \Sigma\,\tilde{\Phi}_{\mathrm{con}} + 0.5\,\Sigma\,\tilde{\Phi}_{\mathrm{rad}}] \tag{5.188}$$

where:

$$F_{1\mathrm{cy}} = \frac{3.0\,(C_{\mathrm{v}} + 6.0\,\Sigma\,\mathrm{A})}{\Sigma\,(A\,Y) + 18.0\,\Sigma\,A + 1.5\,R\,[3.0\,C_{\mathrm{v}} - \Sigma\,(A\,Y)]} \tag{5.189}$$

$$F_{2\mathrm{cy}} = \frac{\Sigma\,(A\,Y) + 18.0\,\Sigma\,A}{\Sigma\,(A\,Y) + 18.0\,\Sigma\,A + 1.5\,R\,[3.0\,C_{\mathrm{v}} - \Sigma\,(A\,Y)]} \tag{5.190}$$

Note: $\tilde{\theta}_{\mathrm{c}t} = 0$ for 24-hour plant operation.

5.A5.4.4 Combined convective and radiant cooling for control on air temperature

Substitution of equations 5.171 and 5.172 into equation 5.186 gives:

$$\tilde{\Phi}_{\mathrm{p}t} = F_{1\mathrm{ay}}\,[\Sigma\,(A\,Y)\,\tilde{\theta}_{\mathrm{a}t} + \Sigma\,A\,f\,U\,\tilde{\theta}_{\mathrm{eo}} - 1.5\,\Sigma\,\tilde{\Phi}_{\mathrm{rad}} - \tilde{\Phi}_{\mathrm{sg}}] + F_{2\mathrm{ay}}\,[C_{\mathrm{v}}\,\tilde{\theta}_{\mathrm{a}t} - \Sigma\,\tilde{\Phi}_{\mathrm{con}} + 0.5\,\Sigma\,\tilde{\Phi}_{\mathrm{rad}}] \tag{5.191}$$

where:

$$F_{1\mathrm{ay}} = \frac{4.5\,\Sigma\,A}{(1 - 1.5\,R)\,\Sigma\,(A\,Y) + 4.5\,\Sigma\,A} \tag{5.192}$$

$$F_{2\mathrm{ay}} = \frac{\Sigma\,(A\,Y) + 4.5\,\Sigma\,A}{(1 - 1.5\,R)\,\Sigma\,(A\,Y) + 4.5\,\Sigma\,A} \tag{5.193}$$

Note: $\tilde{\theta}_{\mathrm{a}t} = 0$ for 24-hour plant operation.

5.A5.5 Effect of allowing room temperature to rise above set point

The peak cooling capacity can be reduced if the room temperature is allowed to rise above the set point for a period sufficiently short that the effect on the mean temperature is small (e.g. two hours at the time of peak load). Because the mean is not changed, the reduction may be calculated from the alternating component of the gain, i.e. from either equation 5.184 or equation 5.186. Thus for control on the operative temperature, the change in load becomes:

$$\Delta\Phi_{\mathrm{k}} = [c_{\mathrm{p}}\,\rho\,q_{\mathrm{v}} + F_{\mathrm{cy}}\,\Sigma\,(A\,Y)]\,\Delta\theta_{\mathrm{c}} \tag{5.194}$$

For control on the air temperature:

$$\Delta\Phi_{\mathrm{k}} = [c_{\mathrm{p}}\,\rho\,q_{\mathrm{v}} + F_{\mathrm{ay}}\,\Sigma\,(A\,Y)]\,\Delta\theta_{\mathrm{ai}} \tag{5.195}$$

where $\Delta\Phi_{\mathrm{k}}$ is the change in cooling load resulting from a small change in temperature ($\Delta\theta$), c_{p} is the specific heat capacity of air ($\mathrm{J{\cdot}kg^{-1}{\cdot}K^{-1}}$), ρ is the density of air ($\mathrm{kg{\cdot}m^{-3}}$), q_{v} is the total ventilation (mechanical plus infiltration) rate ($\mathrm{m^3{\cdot}s^{-1}}$), F_{cy} is the room admittance factor with respect to operative temperatures, F_{ay} is the room admittance factor with respect to the air node, $\Sigma\,(A\,Y)$ is the sum of the products of surface areas and their corresponding thermal admittances ($\mathrm{W{\cdot}K^{-1}}$), $\Delta\theta_{\mathrm{c}}$ is the rise in operative temperature (K) and $\Delta\theta_{\mathrm{ai}}$ is the rise in internal air temperature (K).

5.A5.6 Summertime temperatures

The Simple (dynamic) Model may be used to assess peak temperatures when there is no heating or cooling. The method used is essentially the inverse of the cooling load calculation. However, further simplifications are introduced to enable rapid hand checks on designs.

The intent of the calculation is to obtain the peak operative temperature, which is achieved using a trans-position of equations 5.166 and 5.167. Thus from equation 5.166 the daily mean operative temperature is:

$$\bar{\theta}_{\mathrm{c}} = \frac{C_{\mathrm{v}}\,\bar{\theta}_{\mathrm{ao}} + F_{\mathrm{cu}}\,\Sigma\,(A\,U)\,\bar{\theta}_{\mathrm{eo}} + F_{\mathrm{cu}}\,\bar{\Phi}_{\mathrm{e}} + \bar{\Phi}_{\mathrm{a}}}{C_{\mathrm{v}} + F_{\mathrm{cu}}\,\Sigma\,(A\,U)} \tag{5.196}$$

where:

$$\bar{\Phi}_{\mathrm{a}} = \bar{\Phi}_{\mathrm{con}}$$

Note that for glazed surfaces, $\bar{\theta}_{\mathrm{eo}} = \bar{\theta}_{\mathrm{ao}}$.

The alternating operative temperature follows from equation 5.188:

$$\tilde{\theta}_{\mathrm{c}t} = \frac{\bar{\Phi}_{\mathrm{a}t} + F_{\mathrm{cy}}\,\bar{\Phi}_{\mathrm{e}t}}{C_{\mathrm{v}} + F_{\mathrm{cy}}\,\Sigma\,(A\,Y)} \tag{5.197}$$

where:

$$\bar{\Phi}_{\mathrm{a}t} = \Sigma\,\bar{\Phi}_{\mathrm{con}}$$

Equation 5.197 is based on the ventilation rate remaining constant throughout the day. An assessment of the effect of a varying ventilation rate is given by Harrington-Lynn(A5.1).

Reference for Appendix 5.A5

A5.1 Harrington-Lynn J The admittance procedure: variable ventilation *Building Serv. Engineer* **42** 99–200 (November 1974)

Appendix 5.A6: Algorithm for the calculation of cooling loads by means of the admittance method

5.A6.1 Introduction

This appendix describes the way room cooling loads are calculated using the CIBSE admittance method. The solar position and transmission algorithms are those used to produce the cooling load tables in the 1999 and earlier editions of Guide A. Other calculations follow the equations presented in Appendix 5.A5 which, although different in appearance, are identical to those in editions of Guide A preceding the 1999 edition.

The solar cooling load tables are based upon a particular space and rules related to the use of blinds. Details are given in 5.A6.17.

5.A6.2 Input data

The input data required are:

— *Latitude of the building*: the calculations here are carried out for Local Apparent Time (solar time) and so longitude is not required. CIBSE Guide J(A6.1) provides information on how to correct to clock time.

— *Internal design temperature*: operative temperature, see chapter 1, Table 1.5.

— *Hourly dry bulb temperatures*: design values for three UK locations can be found in chapter 2, Tables 2.34 to 2.36.

— *Hourly values of direct and diffuse solar radiation*: see CIBSE Guide J(A6.1), sections 2.6.2.2 and 2.6.2.3. Design values can be found in chapter 2 of this Guide, Tables 2.30 to 2.32 for UK locations and Table 2.33 for worldwide latitudes. (Note that editions of Guide A prior to the 1999 edition used theoretical solar data whereas the 1999 edition and the present edition use measured data.)

— *Dimensions of the space*.

— *Material properties*: these are the dimensions and thermal properties of the fabric elements bounding the space. For glazing the data must be sufficient to determine the transmission, absorption and reflection for each pane of glass as a function of the solar angle of incidence.

— *Internal heat gains*: the hourly profile of use and the radiant/convective split are required.

— *Infiltration rate/ventilation rate*: this is for outside air only and it is assumed here to be constant throughout the day. Harrington-Lynn(A6.2) shows how to allow for a variable ventilation rate.

— *Boundary conditions for internal surfaces*: this algorithm assumes that internal surfaces are adiabatic. If the temperature in adjacent spaces is known then they can be treated in the same way as external spaces. If not an iterative procedure is required.

— *Time plant is switched on and off*.

5.A6.3 Overview

The basic process is as follows, for each hour of the day. Note that if measured climatic data are used the solar radiation at any given hour is, usually, the average over the preceding hour and so calculations should be made on the half hour. Measured temperatures are usually reported on the hour and so interpolation may be required in order to obtain the half hour value.

(1) Calculate the *U*-value, thermal admittance, decrement and surface factor for all fabric elements (see chapter 4).

(2) Calculate the factors required by the method.

The following preliminary calculations are carried out for each hour of the day:

(3) Calculate the position of the sun.

(4) Generate the direct and diffuse radiation normal to the sun (or obtain from tabulated or measured data).

(5) Obtain appropriate hourly dry bulb temperatures.

(6) Calculate the sol-air temperature for all external surfaces.

(7) Calculate the radiation transmitted through and absorbed within the glazing. If necessary allowing for external shading devices and the raising or lowering of blinds. It is assumed here that blinds are lowered because of external conditions (level of solar radiation) and not internal space temperature. Iteration will be necessary if internal temperature control is required.

This completes the preliminary calculations.

(8) The following loads at the environmental and air node are needed (see 5.A5.3 and 5.A5.4). Note that in the case of the solar cooling load tables only the solar and infiltration loads are required.

  — solar

  — infiltration/ventilation

  — fabric

  — internal gains.

(9) Sum the gains and determine the cooling load for 24-hour plant operation.

(10) Apply correction for intermittent plant operation.

The method of calculation is given in the following sections.

5.A6.4 Correction factors

The calculation requires the following input data for each surface:

VOL	Room volume (m^3)
AWALL	Opaque area (m^2)

AGLASS	Glazed area (m^2)
U	*U*-value
Y	Thermal admittance
YL	Time lead associated with thermal admittance
D	Decrement factor
DL	Time delay associated with the decrement factor
SF	The surface factor
SFD	The time delay associated with the surface factor
AIRCH	The air change rate
PLNTON	Time plant switched on
PLNTOFF	Time plant switched off

The following summations are necessary, the derived *U*-values etc. are those appropriate to an individual surface.

SIGA	Sum of all AWALL and AGLASS
SIGAU	Sum of all AWALL*U and AGLASS*U
SIGAY	Sum of all AWALL*Y and AGLASS*Y
SIGAYL	Sum of all AWALL*YL and AGLASS*YL
SIGASF	Sum of all AWALL*SF and AGLASS*SF
SIGASFD	Sum of all AWALL*SFD and AGLASS*SFD

Ventilation conductance:

CV = AIRCH*VOL / 3

Response factor:

RFACT = (SIGAY + CV) / (SIGAU + CV)

Non-dimensional factors:

FU = 18.*SIGA/(18.*SIGA + SIGAU)

FY = 18.*SIGA/(18.*SIGA + SIGAY)

FV = 6.*SIGA/(6.*SIGA + CV)

F1A = 4.5*SIGA/((1.-1.5R)*SIGAU + 4.5*SIGA)

F2A = (SIGAU + 4.5*SIGA)/((1.-1.5R)*SIGAU + 4.5*SIGA)

F1AY = 4.5*SIGA/((1.-1.5R)*SIGAY + 4.5*SIGA)

F2AY = (SIGAY + 4.5*SIGA)/((1.-1.5R)*SIGAY + 4.5*SIGA)

F1C = 3.0*(CV + 6.SIGA)/((SIGAU + 18.0*SIGA + 1.5*R*(3.0*CV-SIGAU))

F2C = (SIGAU + 18.0*SIGA)/(SIGAU + 18.0*SIGA + 1.5*R*(3.0*CV-SIGAU))

F1CY = 3.0*(CV + 6.0*SIGA)/(SIGAY + 18.0*SIGA + 1.5*R*(3.0*CV-SIGAY))

F2CY = (SIGAY + 18.0*SIGA)/(SIGAY + 18.0*SIGA + 1.5*R*(3.0*CV-SIGAY))

Factor for correction for intermittent operation:

PRUN = PLTOFF-PLTON + 1

DOUTPT = (FY*SIGAY-FU*SIGAU)/(24-PRUN) *FU*SIGAU + PRUN*FY*SIGAY + 24*CV*FV)

Admittance and the associated factors are vector quantities and so all delays and leads should be handled separately. This simple method assumes that the overall response to solar radiation can be represented by a mean value for the surface factor as follows.

Mean surface factor:

SFBAR = SIGASF/SIGA

Mean surface factor delay:

SFDEL = SIGASFD/SIGA

The delay is rounded to the nearest hour, but if zero set to 1 hour.

5.A6.5 Calculation of solar position.

The calculation requires the following input data:

RLAT	Latitude (radians)
NUMDAY	Day of year (January 1st = 1, December 31st = 365)
HOUR	Sun time (sun will be overhead at 12.00)

The calculated data are:

DECANG	Declination angle (radians)
SUNALT	Solar altitude (radians)
SUNAZI	Solar azimuth (radians)
SUNRIS	Time sun rises (decimal hours)
SUNSET	Time sun sets (decimal hours)

5.A6.5.1 Declination angle

This the latitude at which the sun is overhead at solar noon.

DAY = NUMDAY/365.25

SINDEC = (0.398*Sine (0.01721*DAY + 0.03347 *Sine(0.01721*DAY)-1.4096))

DECANG = Arcsine (SINDEC)

COSDEC = Cosine (DECANG)

TANDEC = Tangent (DECANG)

5.A6.5.2 Solar altitude and azimuth

COSLAT = Cosine (RLAT)

SINLAT = Sine (RLAT)

TANLAT = Tangent (RLAT)

Check if TANLAT is not equal to zero then:

TANRAT = TANDEC/TANLAT

Otherwise TANRAT is equal to a large number (10E32) and given the sign of TANDEC.

Hour angle:

HANG = Absolute value (sign ignored so taken as positive) of ((π*15/180)*(12.-HOUR))

COSHAG = Cosine (HANG)

SINALT = COSLAT*COSDEC*COSHAG + SINLAT*SINDEC

SUNALT = Arcsine (SINALT)

COSALT = Cosine (SUNALT)

TANALT = Tangent (SUNALT)

If the solar altitude (SUNALT) is negative the sun is below the horizon, otherwise it is necessary to carry out some checks.

TV0 = COSDEC*Sine (HANG)/COSALT

If TV0 is greater than 1 then it is set to 1, if it is less than –1 then it is set to –1.

C = Arcsine (TV0)

TV1 = COSHAG

TV2 = TANRAT

Northern hemisphere

If the sine of the latitude (SINLAT) is greater than zero (Northern hemisphere) then if the hour is before 12 the following conditional checks are necessary.

Morning:

If TV1 is greater than TV2 then SUNAZI = π-C

If TV1 is equal to TV2 then SUNAZI = π/2

If TV1 is less than TV2 then SUNAZI = C

Southern hemisphere

Switch values:

TV3 = TV1

TV1 = TV2

TV2 = TV3

The checks are now made for the afternoon:

If TV1 is greater than TV2 then SUNAZI = π + C

If TV1 is equal to TV2 then SUNAZI = 1.5*π

If TV1 is less than TV2 then SUNAZI = 2π-C

5.A6.5.3 Sunrise and sunset times

COSANG = TANDEC*TANLAT

Normal situation

Some checks:

TV4 = the absolute value of (COSANG-1)

If TV4 is negative then the time of sunrise = 12Arccosine (COSANG)/π; the time of sunset = 24 – the time of sunrise.

Other cases

If COSANG is less than unity the sun never rises.

If COSANG is equal to or greater than unity the sun never sets.

5.A6.6 Calculation of the solar radiation incident upon a surface and the angle of incidence

This calculation requires the following input data where it is assumed that any corrections for sky clarity and altitude have been applied if theoretical solar data is used.

SUNAZI	Solar azimuth (radians)
SUNALT	Solar altitude (radians)
ORIEN	Surface orientation (radians, North 0 or 2π)
SLOPE	Angle of surface to horizontal (radians, flat roof 0, vertical wall π/2)
DIRAD	Direct radiation normal to the sun ($W \cdot m^{-2}$)
DIFRAD	Diffuse radiation on the horizontal ($W \cdot m^{-2}$)
GREF	Solar albedo (Ground reflectance)

Calculated values:

DIRECT	Direct radiation incident upon an exposed surface ($W \cdot m^{-2}$)
SKYDIF	Sky diffuse radiation incident on a surface ($W \cdot m^{-2}$)
GRDREF	Ground reflected radiation incident on a surface ($W \cdot m^{-2}$)
ANGINC	Solar angle of incidence (radians)

5.A6.6.1 Solar angle of incidence

Default this to π/2 (i.e. the sun's rays are parallel to the surface).

The solar azimuth relative to the surface is:

WALSUN = SUNAZI-ORIEN

COSSLO = Cosine (SLOPE)

SINSLO = Sine (SLOPE)

COSINC = Cosine (SUNALT)*SINSLO*Cosine (WALSUN) + Sine (SUNALT)*COSSLO

If COSINC is positive the solar angle of incidence (ANGINC) is equal to Arccosine (COSINC).

5.A6.6.2 Incident radiation

Direct radiation incident on surface, first check if surface is facing the sun, for this the angle of incidence must be less than 90 degrees; that is the cosine of the angle of incidence (COSINC) is greater than zero. The intensity of the direct radiation (beam) is:

DIRECT = DIRAD*COSINC

The diffuse radiation falling on the surface depends on the orientation of the surface. The simple correction given here was used in the original calculation of the cooling load tables.

AZCOR = 0.9-0.1*Cosine (ORIEN)

AZCOR = 1.-SINSLO*(1.-AZCOR)

SKYDIF = DIFRAD*AZCOR*(1. + COSSLO)/2.0

GRDREF = GREF*((BASRAD*Sine (SUNALT) + DIFBAS)/2.)*(1.-COSSLO)

5.A6.7 Dry bulb temperature

Where measured values are used combined with measured solar radiation data, it is necessary to ensure consistent timing. For example the data contained in the CIBSE Guide J(A6.1) provides:

- solar data as the average over the preceding hour and at the Local Apparent Time (LAT), using:
- dry-bulb data at the hour and at Greenwich Mean Time.

In this case calculations should be made on the half hour. In theory, the dry-bulb time should be corrected to LAT using the 'equation of time' (see CIBSE Guide J(A6.1), section 5.2.6). Bearing in mind the approximations involved in the admittance method, this is considered unnecessary for use of the method in the UK. The same may not be true where time zones span greater distances. It is therefore considered sufficient to use the average of the dry-bulb over the hour.

It is important to be consistent in timing and the convention of 'hour 1' being representative of the period midnight to 01:00 is recommended.

5.A6.8 Sol-air temperature

The following input data are required:

TDRY	External dry bulb temperature
HSO	External surface heat transfer coefficient (see chapter 3, section 3.3.9)
ALPHA	External surface absorption coefficient
EMISS	External surface emissivity
SLOPE	Angle of surface to horizontal (radians: flat roof 0, vertical wall $\pi/2$)
RAD	Incident solar radiation = sum of direct, ground reflected and sky diffuse solar radiation, as appropriate.
RRLM	Longwave radiation loss; standard value for an emissivity of *I* is 100 $W \cdot m^{-2}$.

Calculated value:

TSOL	Sol-air temperature (°C)

The following allows for a reduction in longwave loss dependant on the angle between the surface and the sky. For the surface to see the sky the slope must be less than π. In which case the reduction factor is zero (COR = 0), otherwise:

Let X = SURANG/π

The correction factor is:

COR = 1.–X*(2.–X)

The longwave loss is then:

RLONG = COR*RRLM

and the sol-air temperature is:

TSOL = TDRY + (ALPHA*RAD–EMISS*RLONG)/HSO

5.A6.9 Solar load imposed by the glazing

Appendix 5.A7 describes the way the admittance method calculates the transmission and absorption of solar radiation within a glazing system. That appendix includes the calculation of the mean and alternating solar gain factors. These factors are only intended to be used in 'hand' calculations; the cooling load calculation makes use of the appropriate value for each hour of the day. That is, the glazing system properties are determined as a function of the solar angle of incidence (5.A6.6.2). Section 5.A7.3.3 describes how to determine the transmitted and absorbed radiation and calculate the gain to the environmental node and, where internal blinds are used, the air node. The admittance method requires the following to be done at each hour of the day (only between sunrise and sunset, 5.A6.5.3):

The following input data are required:

Glass and blind properties including cavity, internal and external thermal resistances (see chapter 3, section 3.6).

Dimensions of the glazed surface (window).

Dimensions and position of any shading device relative to the window.

ORIEN	Window orientation (radians: North 0 or 2π)
SLOPE	Angle of window to horizontal (radians: flat roof 0, vertical wall $\pi/2$)
DIRECT	Direct radiation incident upon an exposed surface ($W \cdot m^{-2}$)
SKYDIF	Sky diffuse radiation incident on a surface ($W \cdot m^{-2}$)
GRDREF	Ground reflected radiation incident on a surface ($W \cdot m^{-2}$)
ANGINC	Solar angle of incidence (radians)

Calculated values (hourly):

QTRANS	Total transmitted solar radiation
QGE	Solar gain to the environmental node
QGA	Solar gain to the air node

For each glazed surface:

— If there are external shading devices other than blinds, determine the amount of shadow created by the device. The calculation of shade is not given here. The calculation of the effect of shade on the performance of a window can be treated at many levels ranging from the simplistic (and probably conservative) approach described here to taking full account of the relationship between the shade, window and sky vault, the reflections of solar radiation within the shading system and the temperature of the shades. The approach here is to assume that the whole window is exposed to diffuse radiation with the direct intensity reduced by the shade fraction.

— Where blinds are fitted determine if they are lowered. This may be a simple schedule or at a particular solar intensity. The cooling load tables assume the blinds are lowered if the intensity of direct radiation on the façade was greater than 200 $W \cdot m^{-2}$.

— From the solar angle of incidence and the properties of the glazing calculate the transmission coefficient for direct (TAUD) and diffuse radiation (TAUd) and the absorption coefficient for each element in the glazing system.

AGLASS = total window area

AS = area of the window that is in shade

QTRANS = DIRECT*TAUD*(AGLASS–AS) + AGLASS*(SKYDIF + GRDREF)*TAUd

— If QGED and QGEd are the loads at the environmental node for direct and diffuse radiation respectively (determined following 5.A7.3.3), and similarly for the air point loads QGAD and QGAd, the load at the environmental node is:

QGE = (AGLASS-AS)*QGED + AGLASS*QGEd

QGA = (AGLASS-AS)*QGAD + AGLASS*QGAd

5.A6.10 Calculation of the solar component of the gain

The following data are required.

Surface areas:

SFBAR	Mean surface factor
SFDEL	Mean surface factor delay

The solar loads imposed by the glazing (5.A6.9):

QTRANS	Total transmitted solar radiation for each hour of the day
QGE	Solar gain to the environmental node for each hour of the day.
QGA	Solar gain to the air node for each hour of the day.

Calculated values:

QSESW	The swing in solar cooling load at the environmental node at each hour of the day
QSEBAR	The daily mean solar cooling load at the environmental node
QSASW	The swing in solar cooling load at the air node at each hour of the day
QSABAR	The daily mean solar cooling load at the air node

Carry out the following summations over the day.

QTBAR = Σ QTRANS/24

QSEBAR = Σ QGE/24

QSABAR = Σ QSA/24

The direct solar gain must be absorbed by the room surfaces before it can contribute to the heat load in the room. Due to thermal storage within the surfaces of the room there will be a delay and 'smoothing out' of the direct gain. In the admittance method this is quantified by the 'surface factor' and the associated delay.

The swing in the transmitted load at hour H is that due to the radiation transmitted at:

Hdel = H–SFDEL

The swing in the solar gain at the environmental node at hour, H is:

QSESW(H) = (QGE(H)-QSEBAR) + SFBAR *(QTRANS (Hdel)–QTBAR)

The swing in the load at the air node at hour H is:

QSASW(H) = QGA(H)-QSABAR

5.A6.11 Calculation of the ventilation component of the gain

The following data are required:

TDRY	External dry bulb temperature
TDES	Internal design temperature
CV	Ventilation conductance (5.A6.4)

Calculated values:

QVENTSW	the swing in ventilation gain at the air node.
QVENTB	the mean ventilation load.

The ventilation load at hour H is:

QVENT(H) = CV*(TDRY(H)–TDES)

QVENTB = Σ QVENT/24

QVENTSW(H) = QVENT(H)-QVENTB

5.A6.12 Calculation of the conduction component of the gain

The following data are required.

For each surface:

A	Area (opaque and glazed)
U	*U*-value
D	Decrement factor
DL	Time delay associated with decrement factor
TSOL	Hourly sol-air temperature for each external surface
TSBAR	Daily mean sol-air temperature for each external surface
TDES	Internal design temperature

Calculated values:

QCSW	The swing in conduction gain at the environmental node
QCSB	The mean conduction gain at the environmental node

Mean conduction gain

For each hour (H) of the day, calculate for each external surface (N) and sum:

QC (H) = U(N)*A(N)*(TSOL(H,N)–TDES)

QCSB = Σ QC/24

The swing in gain is due to the gain that occurred DL hours before the current hour, that is:

HDEL = H–DL

QCSW(H) = U(N)*A(N)*D(N)*(TSOL(HDEL,N) -TSBAR(N))

5.A6.13 Calculation of the internal gain

The following data are required.

QGI	Hourly value of the internal gain (W)
FRG	Radiant fraction (0 = 100% convective)

Calculated values:

QGASW	Swing in internal gain at the air node
QGAB	Mean internal gain at the air node
QGESW	Swing in internal gain at the environmental node
QGEB	Mean internal gain at the environmental node

For each hour (H):

convective load: QC = QGA(H)*(1.-FRG)

radiant load: QR = QGA(H)*FRG

The load at the air node:

QGA(H) = QC–0.5*QR

The load at the environmental node is:

QGE(H) = 1.5*QR

The mean loads are:

QGAB = Σ QGA/24

QGEB = Σ QGE/24

The swing in load is:

QGASW(H) = QGA(H)–QGABAR

QGESW(H) = QGE(H)–QGEBAR

5.A6.14 Calculation of the total gain and the solar cooling load for 24-hour plant operation

The calculation here is for control by air temperature or operative temperature. In the case of the cooling load tables only the solar load is considered. The air change rate is used in the calculation of the non-dimensional parameters only.

The following data are required.

Daily mean values of loads at the environmental node:

QSEBAR	Daily mean solar cooling load (5.A6.10)
QCSB	Mean conduction gain (5.A6.12)
QGEB	Mean internal gain (5.A6.13)

Daily mean values at the air node:

QSABAR	Daily mean solar cooling load (5.A6.10)
QVENTB	Mean ventilation load (5.A6.11)
QGAB	Mean internal gain (5.A6.13)

Hourly swing in load at the environmental node:

QSESW	Swing in solar cooling load (5.A6.10)
QCSW	Swing in conduction gain (5.A6.12).
QGESW	Swing in internal gain (5.A6.13)

Hourly swing in load at the air node:

QSASW	Swing in solar cooling load (5.A6.10).
QVENTSW	Swing in ventilation gain (5.A6.11).
QGASW	The swing in internal gain (5.A6.13).

Non-dimensional parameters (5.A6.4):

FU	FY
F1C	F1AY
F2C	F2AY
F1A	F1CY
F2A	F2CY

Calculated values:

QPBAR	Daily mean plant load
QPSWG	Hourly swing in the plant load
QPLANT	Hourly cooling load

The following sums are required:

QGENVB	Sum of all mean gains to the environmental node
QGAIRB	Sum of all mean gains to the air node
QGENVS	Sum of all swings in gain at the environmental node for each hour (H) of the day
QGAIRS	Sum of all swings in gain at air node for each hour (H) of the day.

Control by operative temperature

QPBAR = F1C*QGENVB + F2C*QGAIRB

QPSWG(H) = F1CY*QGENVS(H) + F2CY*QGAIRS(H)

Control by air temperature

QPBAR = F1A*QGENVB + F2A*QGAIRB

QPSWG(H) = F1AY*QGENVS(H) + F2AY*QGAIRS (H)

Hourly cooling load

QPLANT(H) = QPBAR + QPSWG(H)

5.A6.15 Calculation of total gain for intermittent plant operation

The following data are required:

QPLANT	Hourly cooling load for 24-hour plant operation (5.A6.14)
PLNTON	Time plant switched on
PLNTOFF	Time plant switched off
DOUTPT	Intermittency correction factor (5.A6.4)

Sum the cooling load for all hours (H) for which the plant is off (QB). This is the sum of QPLANT when H is less than PLNTON or H is greater than PLNTOFF.

The cooling load for each hour for which the plant is switched on is:

QPLANTI(H) = QPLANT(H) + QB*DOUTPT

Otherwise:

QPLANTI(H) = 0.0

5.A6.16 Example calculation

This example is based upon the room used for Example 5.2 (see page 5-19). Table 5.46 Provides space dimensions and properties. The non-dimensional parameters are given in Table 5.47 and the calculation in Tables 5.48 and 5.49.

The design-cooling load (peak) is highlighted as 678 W occurring between hours 13 and 14.

Table 5.47 Example calculation: dimensionless parameters

Name	Value	Name	Value	Name	Value
FU	0.992	FCU	1	F1A	0.968
FY	0.835	F1AU	0.968	F2A	1
FV	0.992	F2AU	1	F1C	1
FAU	0.968	F1AY	0.559	F2C	1
FAY	0.559	F2AY	1	F1CY	0.842
				F2CY	1

Table 5.46 Example calculation: dimensions and properties

Surface no.	Area / m^2	*U*-value / $W \cdot m^{-2} \cdot K^{-1}$	*Y*-value / $W \cdot m^{-2} \cdot K^{-1}$	*Y*-value lead / h	Dec. factor time lag / h	Dec. factor	Surf. factor	Surf. factor time lag / h	τ	α
1	10.8	0.49	4.56	1.3	0.18	11.05	0.49	1.65	0	0
2	15.4	—	0.75	5.54	—	—	0.99	0.38	0	0
3	10.8	—	0.75	5.54	—	—	0.99	0.38	0	0
4	15.4	—	0.75	5.54	—	—	0.99	0.38	0	0
5	19.8	—	5.31	2.7	—	—	0.69	2.01	0	0
6	19.8	—	7.33	1.1	—	—	0.37	2.36	0	0
Window in 1	3.5	2.94	2.94	0	1	0	0.62	0.34	0.835	0.017

Notes:
(1) The window is double-glazed, the transmission coefficient (τ) and absorption coefficient (α) are for a single pane.
(2) The space volume is 55.44 m^2.
(3) The glazed surface (surface no. 1) faces south and is the only external surface.

5.A6.17 Solar cooling load tables

The cooling load tables (Tables 5.19 to 5.24 (UK) and Tables 5.25 to 5.40 (worldwide)) were calculated using the algorithm described in this appendix with no internal, conduction and ventilation gains.

Table 5.48 Example calculation: calculated hourly values

HOUR	TDRY	DIRAD	DIFRAD	QSNORM*	TSOL	QTRANS
00–01	14	0	0	0	11.74	0
01–02	13.3	0	0	0	11.69	0
02–03	12.2	0	0	0	11.11	0
03–04	11	0	0	0	10.22	0
04–05	11.5	0	13	13	10.89	9
05–06	12.1	0	43	43	13.06	28
06–07	13.2	0	99	99	17.46	66
07–08	15.1	28	91	119	20.08	64
08–09	16.9	164	117	281	32.43	156
09–10	17.8	53	173	226	29.95	146
10–11	17.5	4	139	144	24.94	95
11–12	18.3	11	218	229	31.22	152
12–13	19.2	112	266	378	41.93	254
13–14	19.1	1	201	202	30.53	134
14–15	19.4	4	168	172	28.86	114
15–16	18.9	1	130	130	26.01	87
16–17	18.8	0	141	142	26.66	94
17–18	18.8	0	63	63	21.38	42
18–19	18	0	21	21	18.16	14
19–20	17	0	1	1	16.09	0
20–21	13.4	0	0	0	13.89	0
21–22	13	0	0	0	12.61	0
22–23	12.9	0	0	0	11.92	0
23–01	12.8	0	0	0	11.53	0
Mean	15.58	16	78	94	20	61

* QSNORM = DIRECT + SKYDIF + GRDREF

Table 5.49 Example calculation: calculated swings and hourly values

HOUR	Swings								Hourly values	
	QSESW	QSASW	QCSW	QVENT	QGESW	QGENVS	QGENVS	QGAIRS	QPLANT	QPLANTI
00–01	–162	0	–4	–10	–58	–39	–223	–49	7	0
01–02	–162	0	–5	–10	–58	–39	–225	–49	6	0
02–03	–162	0	–10	–13	–58	–39	–230	–52	1	0
03–04	–162	0	–15	–17	–58	–39	–234	–56	–6	0
04–05	–162	0	–19	–18	–58	–39	–239	–57	–9	0
05–06	–137	0	–21	–17	–58	–39	–216	–56	5	0
06–07	–83	0	–19	–14	–58	–39	–160	–53	39	0
07–08	13	0	–14	–8	–46	–31	–47	–39	116	0
08–09	13	0	–7	–1	–46	–31	–40	–32	127	246
09–10	258	0	–1	5	–46	–31	211	–26	273	392
10–11	224	0	1	7	–46	–31	179	–24	257	377
11–12	98	0	5	10	61	41	163	50	323	442
12–13	254	0	9	13	61	41	324	54	416	535
13–14	506	0	10	15	61	41	577	55	559	678
14–15	194	0	11	16	61	41	266	57	387	506
15–16	140	0	11	16	–46	–31	105	–15	225	344
16–17	70	0	12	15	–46	–31	36	–15	186	305
17–18	84	0	15	15	–46	–31	54	–16	195	315
18–19	–54	0	15	13	120	80	82	93	320	0
19–20	–126	0	20	10	120	80	15	90	279	0
20–21	–161	0	7	0	120	80	–33	80	243	0
21–22	–162	0	–3	–6	120	80	–45	74	230	0
22–23	–162	0	–2	–9	61	41	–103	31	155	0
23–01	–162	0	3	–11	–58	–39	–216	–50	11	0
Mean:	217	0	–78	–48	58	39	197	–10	181	172.6
								Total:	4344	4142.2

(*a*) For the tables relating to unshaded situations, values of parameters used for calculating the non–dimensional factors were as follows:

- module location: intermediate floor with one exposed surface
- module dimensions: (4.8 × 4.8 × 2.7) m
- glazed percentage: 40% of external wall
- properties of surfaces of module: see Table 5.50 (page 5-85).

For both cases (i.e. fast and slow thermal response), a relatively well-sealed facade was assumed, with an infiltration rate of 0.25 h^{-1}.

(*b*) Glazing properties were as given in Appendix 5.A7, Table 5.51.

(*c*) Shading: the tables relating to shaded situations, a generic shading device having 20% transmission and 40% reflection was assumed, see Appendix 5.A7, Table 5.51. The shading device was assumed to operate when direct radiation on the façade was greater than 200 $W{\cdot}m^{-2}$.

References for Appendix 5.A6

A6.1 *Weather, solar and illuminance data* CIBSE Guide J (London: Chartered Institution of Building Services Engineers) (2002)

A6.2 Harrington-Lynn J The admittance procedure: variable ventilation *Building Serv. Eng.* **42** 199–200 (November 1974)

Table 5.50 Properties of surfaces for module used for determination of cooling load tables

Surface	Slow thermal response				Fast thermal response			
	U-value / $W \cdot m^{-2} \cdot K^{-1}$	Y-value / $W \cdot m^{-2} \cdot K^{-1}$	Surface factor, F	Time lag Ψ / h	U-value / $W \cdot m^{-2} \cdot K^{-1}$	Y-value / $W \cdot m^{-2} \cdot K^{-1}$	Surface factor, F	Time lag Ψ / h
Glass	3.0	3.0	—	—	3.0	3.0	—	—
Wall	0.45	5.5	0.5	2	0.45	2	0.8	1

Appendix 5.A7: Derivation of solar gain factors

5.A7.1 Introduction

This appendix describes the method used to determine the mean and alternating solar gain factors given in Table 5.7. While these factors are applicable only to the CIBSE Simple (dynamic) Model (i.e. the admittance procedure), much of what follows is general in nature and therefore also appropriate to more complex thermal models.

The cooling load due to solar radiation has three components:

(a) *Direct gain:* radiation transmitted through the glazing system falls upon the room surfaces and contents where it is both reflected and absorbed. The majority of the reflected radiation is absorbed by the surfaces but some passes out through the windows. For normal rooms this retransmitted radiation is small and can be ignored for the purposes of design calculations. Absorbed radiation increases the temperature of the surfaces and so becomes a room load through both convection and radiation. There is a time delay between the incidence of the direct solar gain and the corresponding room gain because thermal storage occurs within the room fabric. In terms of the Simple (dynamic) Model, this process is represented by the surface factor with the load being realised at the environmental node.

(b) *Indirect gain:* radiation is absorbed within the elements of the glazing system resulting in an increase in the temperature of those elements. Therefore there is a heat gain to the room due to the difference between the inner surface temperature of the glazing, the room surfaces and the room air. For the Simple (dynamic) Model, in the absence of an internal blind, this gain is realised at the environmental node.

(c) Where internal blinds are fitted, the possibility for air to circulate around the blind results in an increase in the rate of convective heat transfer from the blind surface. In terms of the Simple (dynamic) Model, this is expressed as an additional gain to the air node.

The calculation of transmitted and absorbed radiation provides the basis for the calculation of room cooling loads. However, this is too laborious for manual calculation and some simplification is necessary. This is achieved by the introduction of a 'solar gain factor' which is the ratio of the gain to the external radiation producing the gain.

The room load at a given time is due to the combined effect of the three components described above. That is, the room load at time t consists of:

(a) load due to direct gain at time (t – delay), plus

(b) load due to indirect gain at time t, plus

(c) load due to additional gain at air node at time t.

The appropriate solar gain factor could be obtained by normalising the room load by the external irradiance at time t. However, the direct gain is due to external radiation incident at some time (t_o) before t and should therefore be normalised by the external radiation at time t_o. Therefore, it is suggested that three solar gain factors are required:

— a 'shortwave' factor based on the ratio between the direct room load at time t and the external irradiance at time t_o

— a 'longwave' factor based on the ratio between the indirect room load at time t and the external irradiance at time t

— an 'air node' factor based on the ratio between the additional air gain load at time t and the external irradiance at time t.

The use of a separate air node factor is justified by the theoretical approach used in the Simple Model where a distinction is made between the different sources of heat gains. While theoretically justifiable, the use of three factors (two of which are based on external conditions at a different time to the third) cannot be considered simple. Furthermore, solar gain factors defined in this way are not constants but vary hour-by-hour and thus can only be used to predict conditions at a particular time. (More strictly, at a single angle of incidence between the solar beam and the glazing.) However, for most types of glass the transmission and absorption properties are almost independent of the angle of incidence up to an angle of about 45 degrees. It is unlikely that peak loads will occur at high angles of incidence and so that effect can be ignored.

Figure 5.15 shows the transmitted shortwave and indirect gain to a space for a particular glazing type, (without an internal blind) together with the associated external irradiance. The Simple Model is based on the response of a space to the 24-hour mean (see Figure 5.16) and the deviation from that mean (i.e. the swing), see Figure 5.17. In the Simple (dynamic) Model, the solar loads that act at the environmental node within the space are equal to the sum of the mean and alternating gains. In the case of direct (shortwave) transmission, the alternating gain is the swing multiplied by the appropriate surface factor and delayed by the corresponding time delay. The indirect gain acts directly at the environmental node. Figure 5.18 shows these components of the gain to the environmental node for a space with an average surface factor of 0.5 and a time delay of 2 hours. The total solar gain to the environmental node (i.e. sum of mean and swings) is given in Figure 5.19.

The Simple (dynamic) Model is based on the response to a mean and the deviation from that mean; the solar gain factor must be consistent with that model.

The mean solar gain factor is the sum of the daily average values of the transmitted gain, divided by the daily average level of the incident irradiation.

In theory, the alternating component should comprise the three factors described above. However, for the practical purpose of the calculation of peak gain, the shortwave and longwave components are combined. Thus, two alternating factors are defined:

— *alternating solar gain factor:* the swing in solar load due to the sum of direct transmission and indirect gain from the surface of the glazing, divided by the swing in the external incident irradiation

— *alternating air node factor:* the swing in the additional air node load, divided by the swing in the external incident irradiation.

In the case of the air node factor, the swing in irradiation is calculated at the same time as that used to determine the swing in the corresponding gain. In the case of the alternating solar gain factor, there is a time delay associated with the directly transmitted component. Modern glazing systems are usually designed to minimise transmission and so a pragmatic decision must be taken to normalise the

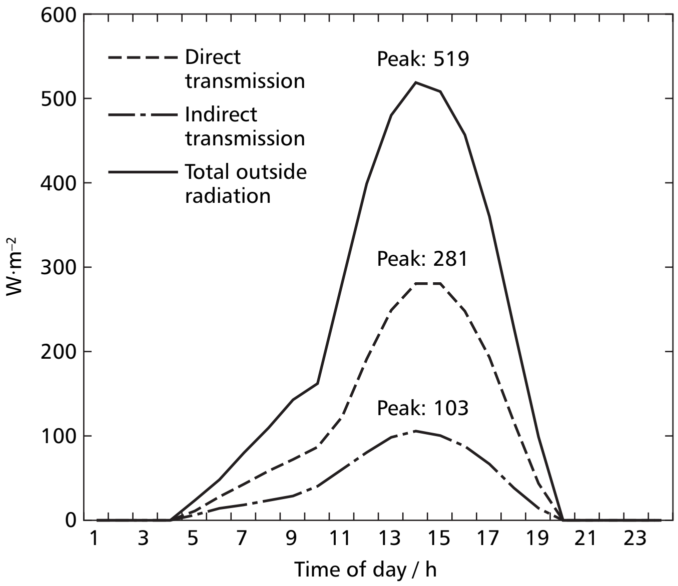


Figure 5.15 Solar transmission and external irradiance

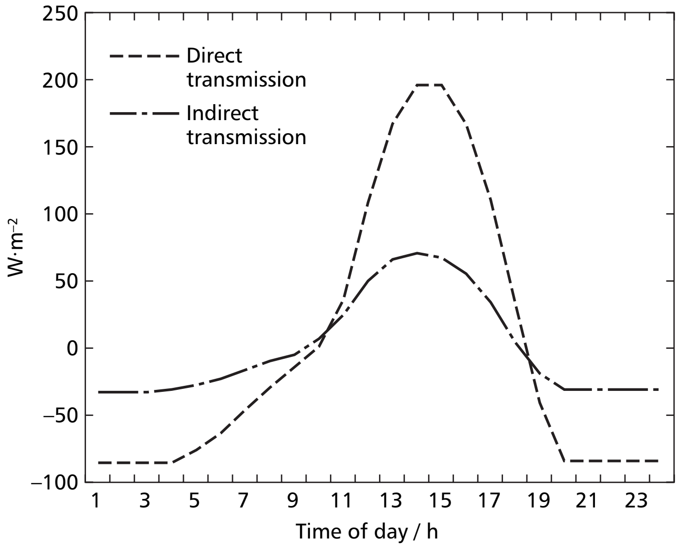


Figure 5.17 Swing in solar transmission

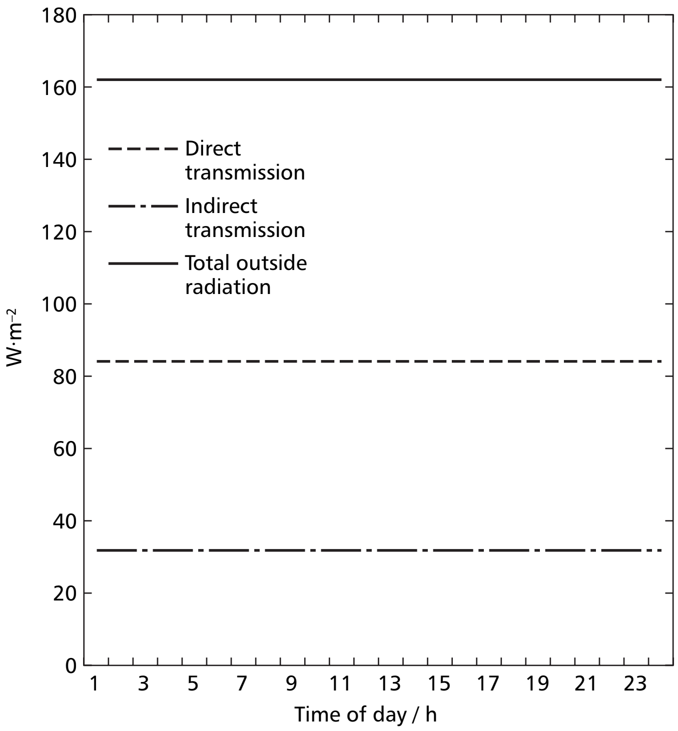


Figure 5.16 Mean solar transmission and external irradiance

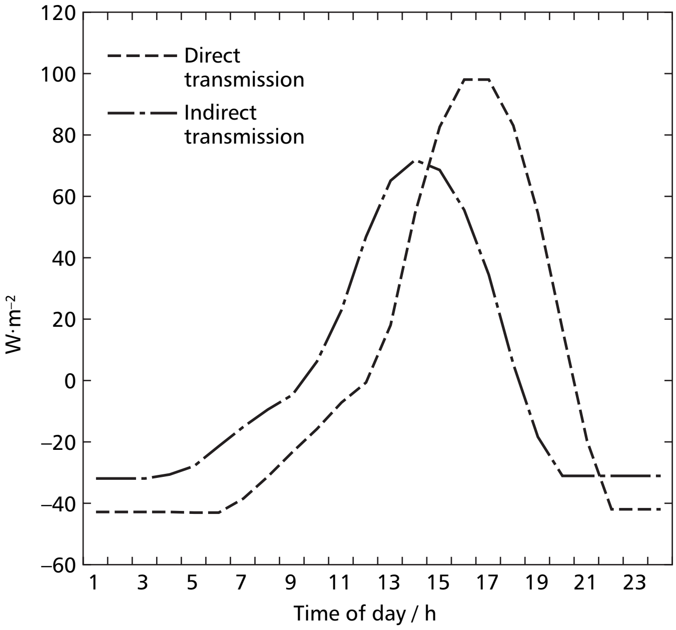


Figure 5.18 Solar gains; swing in gains

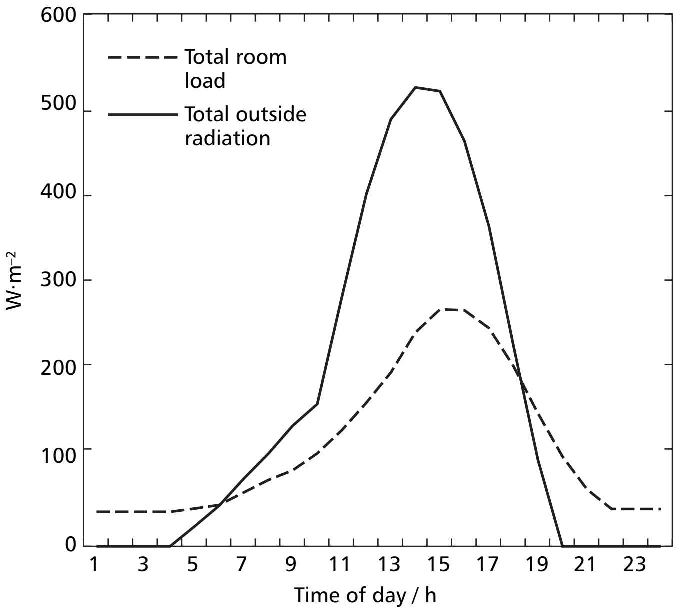


Figure 5.19 Solar gains; total gains

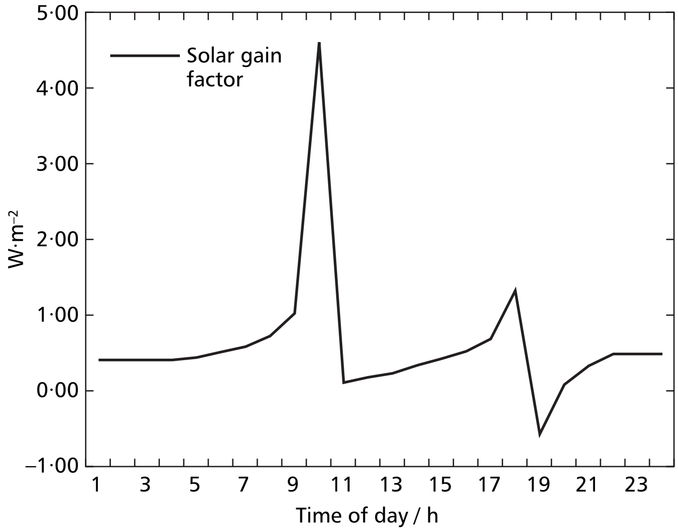


Figure 5.20 Alternating solar gain factor

alternating solar gain factor by the swing at the time of the gain to the space. Figure 5.20 shows the variation of alternating solar gain factor throughout the day. Therefore, there is no single value which is representative of all hours of the day. CIBSE solar gain factors are calculated to be representative of conditions around the time of peak gain.

5.A7.2 Notation

Symbols used in this appendix are as follows.

a	Fraction of incident energy absorbed by thickness L (mm) of glass
A	Absorption coefficient
A'	Absorption coefficient for double glazing
A''	Absorption coefficient for triple glazing
A_D	Absorption coefficient for direct radiation
A_d	Absorption coefficient for diffuse radiation
A_{dg}	Absorption coefficient for ground reflected radiation
A_{ds}	Absorption coefficient for sky diffuse radiation
C_1, C_2	Configuration factors for slatted blinds
D	Slat thickness (mm)
F	Surface factor
h	Solar altitude (degree)
H	Transmittance factor
I	Incident solar irradiance (W·m^{-2})
j	Number of surface
k	Glass extinction coefficient
L	Glass thickness (mm)
M	Width of slat illuminated (mm)
n	Total number of surfaces
R	Reflection coefficient
R'	Reflection coefficient for double glazing
R''	Reflection coefficient for triple glazing
r	Ratio of incident beam to reflected beam at air/glass interface
$r_{//}$	Ratio of incident beam to reflected beam at air/glass interface for radiation polarised parallel to the plane of incidence
$r_{\perp}$	Ratio of incident beam to reflected beam at air/glass interface for radiation polarised perpendicular to the plane of incidence
R_D	Reflection coefficient for direct radiation
R_d	Reflection coefficient for diffuse radiation
R_{dg}	Reflection coefficient for ground reflected radiation
R_{ds}	Reflection coefficient for sky diffuse radiation
R_{se}	External surface resistance (W·m^{-2}·K^{-1})
R_{si}	Internal surface resistance (W·m^{-2}·K^{-1})
T	Transmission coefficient
T'	Transmission coefficient for double glazing
T''	Transmission coefficient for triple glazing
T_D	Transmission coefficient for direct radiation
T_d	Transmission coefficient for diffuse radiation
T_{dg}	Transmission coefficient for ground reflected radiation
T_{ds}	Transmission coefficient for sky diffuse radiation
T_n	Transmission coefficient at normal incidence
t	Time (h)
W	Slat width (m)
α	Absorptivity (thermal shortwave radiation)
β	Profile angle (degree)
γ	Wall azimuth (degree)
γ_s	Wall–solar azimuth (degree)
ζ_i	Angle of incidence (degree)
ζ_r	Angle of refraction (degree)
θ	Temperature (°C)
μ	Refractive index of glass (= 1.52)
σ_v	Vertical shadow angle (degree)
Φ	Room gain (W·m^{-2})
Φ_a	Room gain to air node (W·m^{-2})
$\Phi_{a\,t}$	Room gain to air node at time t (W·m^{-2})
Φ_e	Room gain to environmental node (W·m^{-2})
$\Phi_{e\,t}$	Room gain to environmental node at time t (W·m^{-2})
Φ_t	Room gain at time t (W·m^{-2})
ϕ	Solar azimuth (degree)
ψ	Slat angle (degree)
ω	Time lag associated with surface factor (h)

Where required additional subscripts 'A', 'R' and 'T' indicate gains due to absorbed, reflected and transmitted components of radiation, respectively.

5.A7.3 Response of room to solar radiation

Shortwave solar radiation incident upon a window will be reflected, absorbed in the glazing elements or directly transmitted into the space beyond the window. The absorbed radiation will increase the temperature of the glazing and is therefore both a longwave radiant heat gain and a convective gain to the space. In terms of the Simple (dynamic) Model these gains are considered to enter the model at the environmental node. If internal blinds are present, there will be an increase in the convective portion of the gain which enters the model at the air node. Transmitted radiation must be absorbed at the room surfaces before it can become a heat gain to the space. With the exception of any shortwave radiation that passes directly out of the space by transmission through glazed surfaces, all the radiation entering the space is absorbed at the room surfaces or within the furnishings.

Once absorbed, the radiation warms the surfaces and, after a time delay, enters the space at the environmental node by means of convection and radiation.

For the purposes of the Simple (dynamic) Model, the room gain is divided into a 24-hour mean component and an hourly cyclic component.

For any given source:

$$\bar{\Phi} = (1/24) \sum_{t=1}^{t=24} \Phi_t \tag{5.198}$$

and:

$$\tilde{\Phi}_t = \Phi_t - \bar{\Phi} \tag{5.199}$$

In the case of the Simple (dynamic) Model the gain will be either to the environmental node only or to both the environmental and air nodes. The gain to the environmental node from transmitted radiation is:

$$\Phi_{eTt} = \bar{\Phi}_{eT} + F\,\tilde{\Phi}_{teT\,(t-\omega)} \tag{5.200}$$

$$\tilde{\Phi}_{eTt} = \Phi_{eTt} - \bar{\Phi}_{eT} \tag{5.201}$$

$$\bar{\Phi}_{eT} = (1/24) \sum_{t=1}^{t=24} \Phi_{eTt} \tag{5.202}$$

$$\Phi_{eTt} = T\,I_t \tag{5.203}$$

where Φ_{eTt} is the overall gain to the environmental node from transmitted radiation at time t. However, in practice, direct and diffuse transmitted radiation must be treated separately.

The gain to the environmental node due to conduction and radiation from the inner surface of the glazing is:

$$\Phi_{eAt} = \sum_{j=1}^{j=n} (H_{ej}\,A_j\,I_t) \tag{5.204}$$

where A is the component of the radiation absorbed by the glass, subscript j denotes the number of the glazing element and n is the total number of glazing elements within the window system.

Thus the total gain to the environmental node is:

$$\Phi_{et} = \Phi_{eTt} + \Phi_{eAt} \tag{5.205}$$

Additionally, if there is an internal blind, the gain to the air node is:

$$\Phi_{aAt} = \sum_{j=1}^{j=n} (H_{aj}\,A_j\,I_t) \tag{5.206}$$

Thus the total gain to the air node is:

$$\Phi_{at} = \Phi_{aAt} \tag{5.207}$$

To simplify the calculation of these gains, solar gain factors are used. These are the ratios of the components of the gain to the incident solar radiation. The room load has both steady state and cyclic components and the space gains are to the environmental and, possibly, the air nodes. Additionally, the surface factor depends on the response time of the space. To calculate the solar gain factors, typical values are taken, as follows:

— for slow response space: $F = 0.5$; time delay = 2 h

— for a fast response space: $F = 0.8$; time delay = 1 h

The solar gain factors are as follows:

$$\bar{S}_e = \bar{\Phi}_e / \bar{I} \tag{5.208}$$

$$\tilde{S}_{et} = \tilde{\Phi}_{et} / \tilde{I}_t \tag{5.209}$$

$$\bar{S}_a = \bar{\Phi}_a / \bar{I} \tag{5.210}$$

$$\tilde{S}_{at} = \tilde{\Phi}_{at} / \tilde{I}_t \tag{5.211}$$

Solar gain factors for generic glass and blind combinations are given in Table 5.7. These have been calculated using banded solar radiation data for Kew (1959–1968)[A7.1] incident on a south-west facing vertical window (see chapter 2: *External design data*). The transmission (T), absorption (A) and reflection (R) components (for thermal shortwave radiation) and emissivities (for thermal longwave radiation) for the generic glass and blind types used in calculating the solar gain factors are given in Table 5.51.

Effectively, solar gain factors can only be calculated by means of a computer program. The following sections describe the basis of the calculation procedure.

5.A7.3.1 Transmission, absorption and reflection for direct solar radiation

Clear glass

For clear glass the transmission, absorption and reflection (TAR) coefficients can be derived theoretically[A7.2].

The angle of refraction is obtained from the angle of incidence using Snell's Law:

$$\zeta_r = \arcsin(\sin \zeta_i / \mu) \tag{5.212}$$

The reflected beams for radiation polarised parallel to and perpendicular to the plane of incidence are determined using Fresnel's formula:

$$r_{//} = \frac{\tan^2(\zeta_i - \zeta_r)}{\tan^2(\zeta_i + \zeta_r)} \tag{5.213}$$

Table 5.51 Transmission, absorption and reflection components and emissivities for generic glass and blind combinations

Description	Shortwave radiation (proportions of total)			Longwave emissivity	
	Transmitted	Reflected 1	Reflected 2	Surface 1	Surface 2
Glass:					
— clear	0.789	0.072	0.072	0.837	0.837
— low emissivity*	0.678	0.091	0.108	0.837	0.17
— absorbing	0.46	0.053	0.053	0.837	0.837
— reflecting (high performance)*	0.39	0.31	0.45	0.837	0.025
Slatted blind†:					
— reflecting	0.0	0.60	0.40	0.80	0.80
— absorbing	0.0	0.80	0.20	0.80	0.80
'Generic' blind	0.20	0.40	0.40	0.80	0.80

* Asymmetric glass properties

$$r_{\perp} = \frac{\sin^2(\zeta_i - \zeta_r)}{\sin^2(\zeta_i + \zeta_r)} \tag{5.214}$$

As the angle of incidence approaches 0 (i.e. normal incidence):

$$\tan \zeta_i > \sin \zeta_i > \zeta_i \tag{5.215}$$

hence:

$$r_{//} > r_{\perp} > \frac{(\mu - 1)^2}{(\mu + 1)^2} \tag{5.216}$$

This is a useful result as it enables the calculation of the extinction coefficient (k) if the transmission at normal incidence (T_n) is known. The extinction coefficient is a non-linear function of the glass thickness (L) and is related to the transmission coefficient by:

$$T_n = \frac{(1 - r)^2 \exp(-k\,L)}{1 - r^2 \exp(-2\,k\,L)} \tag{5.217}$$

For the beam polarised parallel to the plane of incidence, the fraction of incident energy absorbed for each beam is calculated as follows:

$$a_{//} = 1 - \exp(-k\,L / \cos \zeta_r) \tag{5.218}$$

and similarly for the perpendicularly polarised beam.

The transmitted, absorbed and reflected coefficients are calculated separately for each beam (i.e. parallel and perpendicularly polarised) and the average taken to give the overall coefficients. For the beam polarised parallel to the plane of incidence:

$$T_{D//} = \frac{(1 - r)^2 (1 - a_{//})}{1 - r^2 (1 - a_{//})^2} \tag{5.219}$$

$$A_{D//} = \frac{a_{//} (1 - r)\,[1 + r (1 - a_{//})]}{1 - r^2 (1 - a_{//})^2} \tag{5.220}$$

$$R_{D//} = \frac{r (1 - r)^2 (1 - a_{//})}{1 - r^2 (1 - a_{//})} + r \tag{5.221}$$

and similarly for the perpendicularly polarised beam.

Therefore:

$$T_D = {}^1/_2\,(T_{D//} + T_{D\perp}) \tag{5.222}$$

and similarly for the absorption and reflection coefficients.

Note that since the transmitted, absorbed and reflected components add up to unity, only two need be calculated, the third being obtained by subtraction.

Reflecting and other glasses

The characteristics of such glasses differ from those for plain glass and therefore must be obtained from the manufacturers. If the characteristics are supplied as a graph of TAR coefficients against angle of incidence, the appropriate values can be read-off directly or by curve-fitting techniques.

Slatted blinds

The analysis is the same for both horizontal and vertical slatted blinds. Radiation may be transmitted into a room by the following paths(A7.3).

— *direct:* i.e. passes through the blind without touching any surface; may be zero

— *reflected* (1): i.e. passes through the blind after one reflection from the slat surface which is directly irradiated by the sun

— *reflected* (2): i.e. passes through the blind after undergoing any number of reflections, the final reflection being from the slat surface opposite the one directly illuminated by the sun

— *reflected* (3): i.e. passes through the blind after undergoing any number of reflections, the final reflection being from the one directly illuminated by the sun.

In order to calculate these components, up to five configuration factors are required, each of which depends on the blind geometry. The number of factors needed depends on whether all or only part of the slat is illuminated.

The amount of a slat that is illuminated (i.e. not shaded by the slat above it) depends on the geometry of the blind and the 'profile angle'.

The profile angle (β) is the angle that the direct radiation beam makes with the blind in a vertical plane perpendicular to the plane of the window. For horizontal slatted blinds on a vertical window, the profile angle is the vertical shadow angle:

$$\beta = \sigma_v = \arctan (\tan h \sec \gamma_s) \tag{5.223}$$

For vertical slatted blinds on a vertical window, the profile angle is the wall–solar azimuth:

$$\beta = \gamma_s = \phi - \gamma \tag{5.224}$$

In the following analysis, it is assumed that the radiation is incident on the upper surface of the slat. The width of slat that is illuminated is calculated from:

$$M = \min\left(W, \frac{D \cos \beta}{\sin (\beta + \psi)}\right) \tag{5.225}$$

The configuration factors are calculated as follows.

Radiation that is reflected by the lower slat and passes into the room when the whole width is illuminated (C_1):

$$C_1 = {}^1/_2 \{1 + (D / W) - [1 + (D^2 / W^2) + (2 D \sin \psi / W)]^{1/2}\} \tag{5.226}$$

Radiation that is reflected by the lower slat and intercepted by the upper slat when the whole width is illuminated (C_2):

$$C_2 = {}^1/_2 \{[1 + (D^2 / W^2) + (2 D \sin \psi / W)]^{1/2} + [1 + (D^2 / W^2) - (2 D \sin \psi / W)]^{1/2} - (2 D / W)\} \tag{5.227}$$

Radiation reflected by the upper slat which passes into the room (C_3):

$$C_3 = {}^1/_2 \{[1 + (D / W) - [1 + (D^2 / W^2) - (2 D \sin \psi / W)]^{1/2}\} \tag{5.228}$$

Radiation reflected by the lower slat, which passes into the room when the lower slat is partially shaded (C_4):

$$C_4 = {}^1/_2 (1 + \{[(W - M)^2 / M^2] + (D^2 / M^2) + [2 (W - M) D \sin \psi / M^2]\}^{1/2} - [(W^2 / M^2) + (D^2 / M^2) + (2 W D \sin \psi / M^2)]^{1/2}) \tag{5.229}$$

Radiation reflected by the lower slat, which is intercepted by the upper slat when the lower slat is partially shaded (C_5):

$$C_5 = {}^1/_2 ([(W^2 / M^2) + (D^2 / M^2) + (2 D W \sin \psi / M^2)]^{1/2} - (D / M) + [1 + (D^2 / M^2) - (2 D \sin \psi / M)]^{1/2} - \{[(W - M)^2 / M^2] + (D^2 / M^2) + [2 (W - M) D \sin \psi / M^2]\}^{1/2}) \tag{5.230}$$

If the whole of the lower slat is illuminated and some radiation may pass directly into the room, the TAR coefficients for the blind are calculated from:

$$T_D = 1 - \left(\frac{W \sin (\phi + \psi)}{D \cos \phi}\right) \times \left(1 - C_1 (1 - a) - \frac{C_2 (1 - a)^2 - [C_3 + C_1 C_2 (1 - a)]}{1 - C_2^2 (1 - a)^2}\right) \tag{5.231}$$

$$A_D = a\, W \sin (\phi + \psi) / D \cos \phi\, [1 - C_2 (1 - a)] \tag{5.232}$$

$$R_D = 1 - A_D - T_D \tag{5.233}$$

Where part of the lower slat is shaded by the slat above:

$$T_D = C_4 (1 - a) + \{C_5 (1 - a)^2 \times [C_3 + C_1 C_2 (1 - a)] / [1 - C_2^2 (1 - a)^2]\} \tag{5.234}$$

$$A_D = a (1 + \{[C_5 (1 - a)] / [1 - C_2 (1 - a)]\}) \tag{5.235}$$

$$R_D = 1 - A_D - T_D \tag{5.236}$$

Roller blinds

The properties for roller blinds are not well defined. It is generally sufficient to assume that the TAR coefficients are independent of the angle of incidence and take the values at normal incidence supplied by the manufacturers.

5.A7.3.2 Transmission, absorption and reflection for sky diffuse and ground reflected radiation

Transmission, absorption and reflection coefficients for glasses and blinds are calculated by considering the direct properties over a range of angles appropriate to the radiation. For glass, the TAR values for sky diffuse and ground reflected radiation are the same since glass has symmetrical properties. The characteristics for roller blinds can be assumed to be the same for direct and diffuse radiation. However, slatted blinds are highly asymmetrical so the two sources of diffuse radiation must be calculated separately.

Glasses

The standard properties are calculated on the assumption that the glass is exposed to a hemispherical source of uniform radiance therefore the transmission and absorption angles are from 0° to 90°. Mathematically, the expressions for TAR could be integrated over this range, i.e:

$$T_d = \int_0^{90} T_D (\zeta_i) \sin (2 \zeta_i)\, d\zeta_i \tag{5.237}$$

In practice the direct properties are summed for angles of incidence from 2.5° to 87.5° at intervals of 5°, i.e:

$$T_d = \sum_{\zeta = 2.5}^{\zeta = 87.5} \{T_{D\zeta} [\sin^2 (\zeta_i + 2.5) - \sin^2 (\zeta_i - 2.5)]\} \tag{5.238}$$

A_d is calculated similarly and R_d is obtained by subtraction from unity, see equation 5.233.

Slatted blinds

The direct properties are summed for profile angles from 5° to 85° at intervals of 10° for sky diffuse radiation. For ground reflected radiation, they are summed from –85° to –5° at intervals of 10° taking into account the configuration factor of the hemispherical radiating source bounded by profile angles of $(\beta + 5)°$ and $(\beta - 5)°$[A7.4]. Thus, for sky diffuse radiation:

$$T_{ds} = \sum_{\beta=5}^{\beta=85} \{T_{D\beta}[\sin(\beta+5) - \sin(\beta-5)]\} \qquad (5.234)$$

For ground reflected radiation:

$$T_{dg} = \sum_{\beta=-5}^{\beta=-85} \{T_{D\beta}[\sin(\beta+5) - \sin(\beta-5)]\} \qquad (5.235)$$

A_{ds} and A_{dg} are calculated similarly and R_{ds} and R_{dg} are obtained by subtraction from unity, see equation 5.233.

5.A7.3.3 Properties of glass and blind combinations

The properties of multiple layer windows can be calculated from the properties of the individual components. There are many glass types and many permutations; the method of calculation is demonstrated in the following for double and triple glazing using generic glass and blind types.

In the same way that the properties of a single sheet of glass are calculated from the fundamental properties of the glass and an infinite number of inter-reflections at both glass/air interfaces, the properties of multiple glazing are calculated by considering the inter-reflections between the component layers[A7.2,A7.5]. These calculations are performed for both direct and diffuse radiation. However, if the window incorporates a blind, the radiation reflected by or transmitted through it is assumed to be diffuse whatever the nature of the source. This is because the slat surfaces are assumed to be diffusing rather than specular reflectors[A7.3].

The following equations are derived from Figure 5.21 where all layers are symmetrical, i.e. both surfaces of the layer have the same reflection and the specularity of the radiation is not changed by the layer. If any of the layers are asymmetrical, the equations become more complicated since they have to include the reflection of both surfaces of the layer. If any of the layers is a diffusing slatted blind, then the direct radiation equations need to include the diffuse properties of the elements for radiation that has been reflected by the blind(s). Examples for some of these situations are given elsewhere[A7.2].

Double glazing

TAR coefficients for double glazing, denoted by prime (′), are as follows:

$$T' = (T_o\,T_i)/(1 - R_o\,R_i) \qquad (5.241)$$

$$A_o' = A_o + [(T_o\,A_o\,R_i)/(1 - R_o\,R_i)] \qquad (5.242)$$

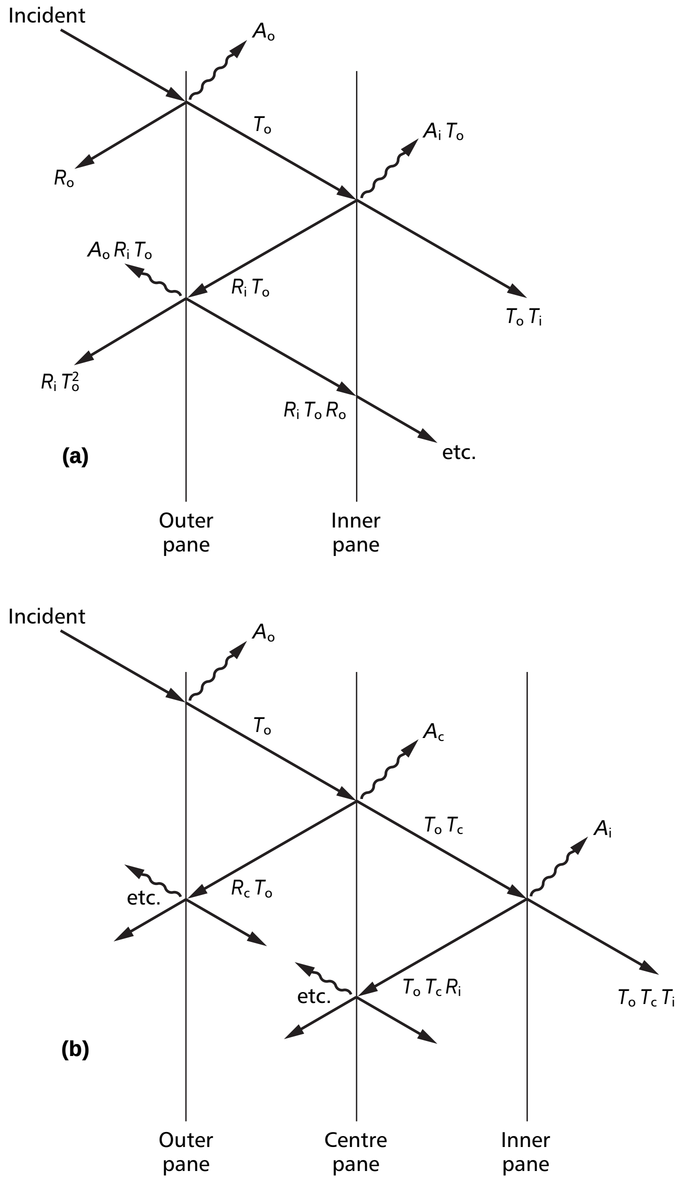


Figure 5.21 Transmitted, absorbed and reflected radiation; (*a*) double glazing, (*b*) triple glazing

$$A_i' = (T_o\,A_i)/(1 - R_o\,R_i) \qquad (5.243)$$

$$R' = 1 - T' - A_o' - A_i' \qquad (5.244)$$

where subscript 'o' denotes the outer glazing element and subscript 'i' denotes the inner glazing element.

Triple glazing

TAR coefficients for triple glazing, denoted by double prime (″), are as follows:

$$T'' = \frac{T_o\,T_c\,T_i}{(1 - R_o\,R_c)(1 - R_c\,R_i) - T_c^2\,R_o\,R_i} \qquad (5.245)$$

$$A_o'' = A_o + \frac{T_o\,A_o\,R_c}{1 - R_o\,R_c} + \frac{T_o\,T_c^2\,A_o\,R_i}{(1 - R_o\,R_c)(1 - R_c\,R_i) - T_c^2\,R_o\,R_i} \qquad (5.246)$$

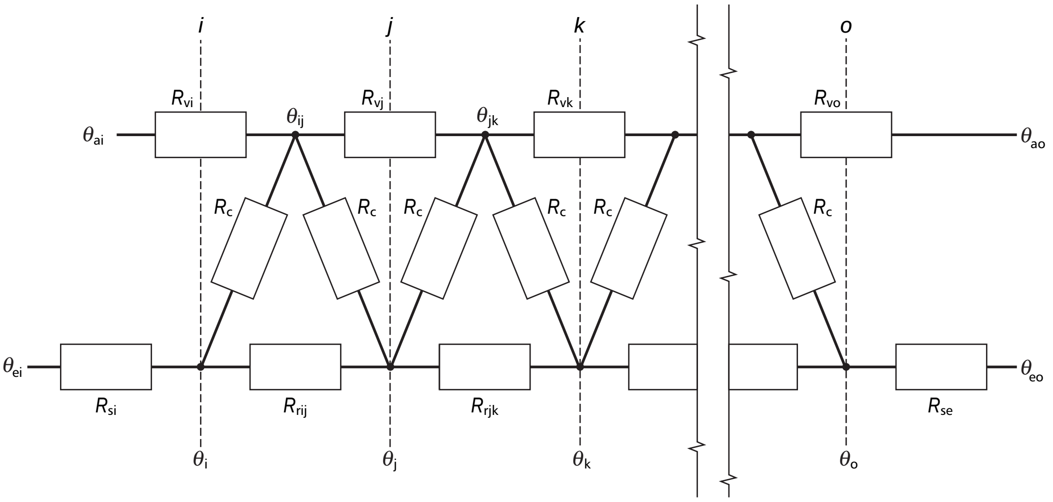


Figure 5.21 General thermal resistance network for a multiple-layer window

$$A_c'' = \frac{T_o\,A_c\,(1 - R_c\,R_i + T_c\,R_i)}{(1 - R_o\,R_c)\,(1 - R_c\,R_i) - T_c^2\,R_o\,R_i} \tag{5.247}$$

$$A_i'' = \frac{T_o\,T_c\,A_i}{(1 - R_o\,R_c)\,(1 - R_c\,R_i) - T_c^2 R_o\,R_i} \tag{5.248}$$

$$R'' = 1 - T'' - A_o'' - A_c'' - A_i'' \tag{5.249}$$

where subscript 'o' denotes the outer glazing element, subscript 'c' denotes the central glazing element and subscript 'i' denotes the inner glazing element.

The heat gain to the environmental node due to conduction and radiation from the inner surface of the glazing is given by equation 5.204. If there is an internal blind, the additional heat gain to the air node is given by equation 5.206. In these equations, the transmittance factors (H) depend on the values taken for the thermal resistances (i.e. the radiant and convective heat transfer coefficients) of the layers of the window. They are calculated by considering the thermal resistance network for the window. Figure 5.22 shows the general thermal resistance network for a multiple-layer window.

The properties of the glazing systems are calculated using the following standard thermal resistances and heat transfer coefficients:

— thermal resistance between inner surface of window and environmental point (i.e. inside thermal resistance): R_{si} = 0.12 m^2·K·W^{-1}

— thermal resistance between outer surface of window and sol-air temperature (i.e. outside thermal resistance): R_{se} = 0.06 m^2·K·W^{-1}

— convective resistance between a window layer and the air: R_c = 0.33 m^2·K·W^{-1} (for vertical window, corresponding to h_c = 3 W·m^2·K^{-1})

— radiative resistance between two layers (j, k) of window:

$$(R_r)_{j,k} = (\varepsilon_j + \varepsilon_k - \varepsilon_j\,\varepsilon_k) / (h_r\,\varepsilon_j\,\varepsilon_k) \tag{5.251}$$

(if both layers have an emissivity of 0.84 and h_r = 5.7 W·m^{-2}·K^{-1}; $(R_r)_{j,k}$ = 0.24 m^2·K·W^{-1})

— ventilation resistance across window layer between adjacent air spaces: R_v = 0 m^2 K·W^{-1} if the layer is a blind; $R_v = \infty$ if the layer is glass.

Example A7.1: Triple glazing without blinds

Figure 5.23 shows the network for triple glazing and Figure 5.24 shows the simplified network resulting from evaluation of the parallel resistances.

The total resistance of the network is:

$$\begin{aligned}\Sigma\,(R) &= R_{si} + R_{ic} + R_{co} + R_{se} = \\ &= 0.12 + 0.18 + 0.18 + 0.06 \\ &= 0.54\ \text{m}^2\cdot\text{K}\cdot\text{W}^{-1}\end{aligned}$$

where R_{ic} is the thermal resistance between inner and central elements of the glazing (m^2·K·W^{-1}) and R_{co} is the thermal resistance between central and outer elements of the glazing (m^2·K·W^{-1}).

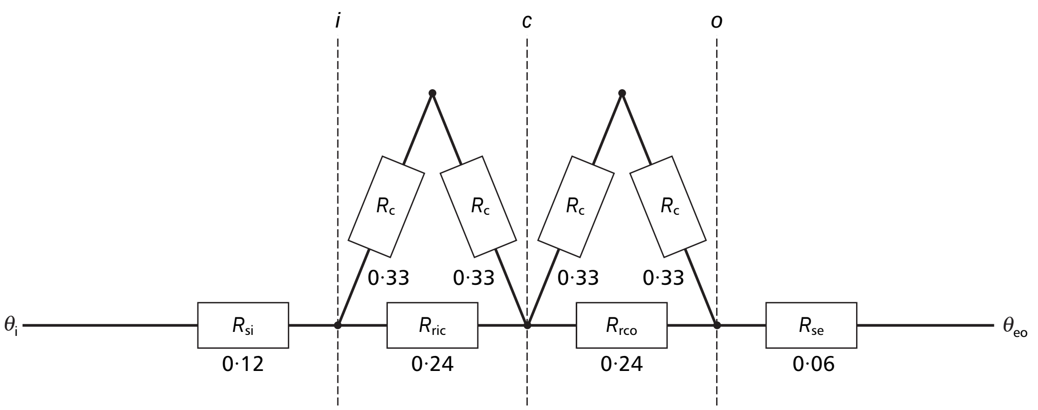


Figure 5.23 Thermal resistance network for triple glazing

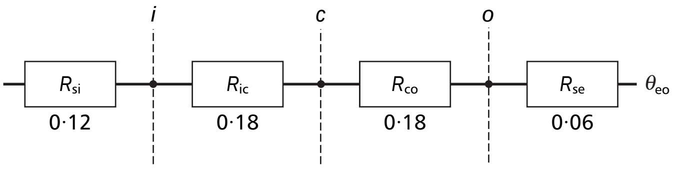


Figure 5.24 Simplified thermal resistance network for triple glazing

The transmittance factors for the inner, central and outer elements of the glazing can be shown to be:

$$H_{ei} = (R_{ic} + R_{co} + R_{se}) / \Sigma R$$
$$= (0.18 + 0.18 + 0.06) / 0.54 = 0.78$$

$$H_{ec} = (R_{co} + R_{se}) / \Sigma R = (0.18 + 0.06) / 0.54 = 0.44$$

$$H_{eo} = R_{se} / \Sigma R = 0.06 / 0.54 = 0.11$$

From equation 5.204, the cyclic component of the convective and longwave radiant gain from the glazing to the environmental node is calculated as follows:

$$\tilde{H}_e A = H_{ei} \tilde{A}_i + H_{ec} \tilde{A}_c + H_{eo} \tilde{A}_o$$

Table 5.52 summarises the steps in the calculation of the solar gain to the space by means of an example. The calculation was carried out as follows.

For 12:00 h:

$$\tilde{H}_e A = 0.78\,(35 - 15) + 0.44\,(58 - 24)$$
$$+ 0.11\,(89 - 34) = 36.6$$

The gain at other times is calculated similarly. The mean gain is calculated using the mean, rather than the cyclic, absorption values.

The cyclic component of the directly transmitted short-wave radiation is attenuated by the surface factor (F) which is appropriate to the thermal weight of the building and corresponding time delay, see Table 5.6.

For a lightweight building, $F = 0.8$ and the time delay is 1 hour, i.e:

$$\tilde{T}_L = 0.8 \times (T_{t+1} - \bar{T})$$

and for a heavyweight building, $F = 0.5$ and the time delay is 2 hours, i.e:

$$\tilde{T}_H = 0.5 \times (T_{t+2} - \bar{T})$$

where subscript 'L' denotes thermally lightweight building and subscript 'H' denotes thermally heavyweight building.

The mean solar gain factor is given by:

$$\bar{S}_e = \frac{\text{mean transmitted radiation plus mean absorbed radiation}}{\text{daily mean incident radiation}}$$

Hence:

$$\bar{S}_e = (65 + 26) / 179 = 0.51$$

The cyclic solar gain factors are calculated using the gains appropriate to a time one or two hours after the time of peak radiation, depending on the thermal weight of the structure, i.e:

$$\tilde{S}_e = \frac{\text{total swing in gain to space}}{\text{swing in external gain}}$$

Peak solar irradiance occurs at 14:00 h; hence, for a thermally lightweight structure (i.e. 1 hour delay):

$$\tilde{S}_{eL} = (126 + 59) / (563 - 179) = 0.48$$

and for a thermally heavyweight structure (i.e. 2 hour delay):

$$\tilde{S}_{eH} = (79 + 50) / (504 - 179) = 0.4$$

Example A7.2: Single glazing with internal absorbing blind

Figure 5.25 shows the network for single glazing with an internal blind and Figure 5.26 shows the simplified network resulting from evaluation of the parallel resistances.

In this case, there are transmittance factors to both the air and environmental nodes, which are calculated as follows:

$$R_{ix} = R_{rio} + [(R_c\, R_{se}) / (R_c + R_{se})]$$
$$= 0.23 + [(0.33 \times 0.06) / (0.33 + 0.06)] = 0.28$$

$$H_{ei} = \frac{R_c\, R_{ix} / (R_c + R_{ix})}{R_{si} + [(R_c\, R_{ix}) / (R_c + R_{ix})]}$$

$$= \frac{(0.33 \times 0.28) / (0.33 + 0.28)}{0.12 + [(0.33 \times 0.28) / (0.33 + 0.28)]} = 0.56$$

Table 5.52 Example A7.1: components of radiation

Time / h	Solar irradiance / W·m^{-2}	Radiation absorbed/transmitted by glazing system / W·m^{-2}				Gains to space / W·m^{-2}		
		Radiation absorbed by inner, central and outer glazing elements			Directly transmitted radiation, T	Cyclic component of absorbed radiation, ($H_e A$)	Cyclic component of transmitted radiation for lightweight (L) and heavyweight (H) buildings	
		A_i	A_c	A_o			$\tilde{T}_L$	$\tilde{T}_H$
1200	442	35	58	89	133	37	—	—
1300	531	46	71	104	189	53	54	—
1400	572	52	77	108	223	60	99	34
1500	563	52	75	105	229	59	126	62
1600	504	47	67	94	205	50	131	79
Mean:	179	15	24	34	65	26	—	—

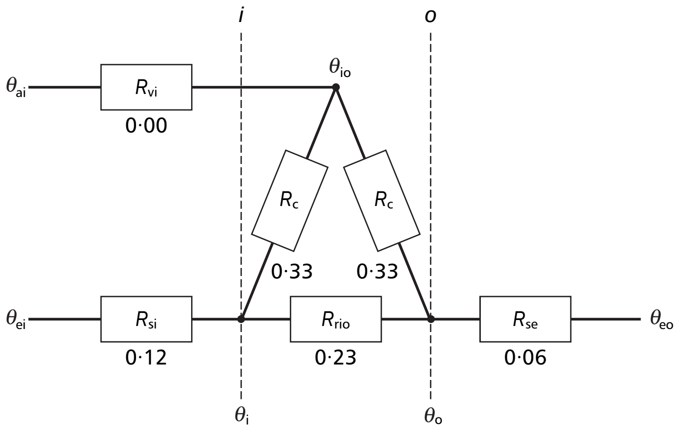


Figure 5.25 Thermal resistance network for single glazing with internal blind

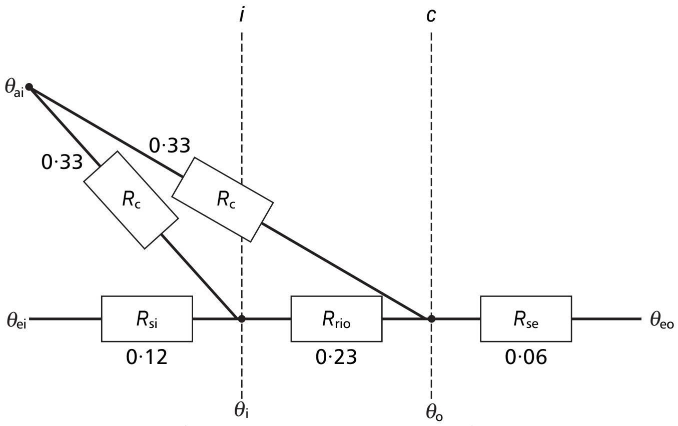


Figure 5.26 Simplified thermal resistance network for single glazing with internal blind

$$H_{ai} = \frac{R_{si}\,R_{ix} / (R_{si} + R_{ix})}{R_c + [(R_{si}\,R_{ix}) / (R_{si} + R_{ix})]} + \left(\frac{R_{si}\,R_c / (R_{si} + R_c)}{R_{ix} + [(R_{si}\,R_c) / (R_{si} + R_c)]}\right)\left(\frac{R_{se}}{R_{se} + R_c}\right)$$

$$= 0.24$$

$$R_{ox} = R_{rio} + [(R_c\,R_{si}) / (R_c + R_{si})] = 0.32$$

$$H_{eo} = \frac{R_c\,R_{se} / (R_c + R_{se})}{R_{ox} + (R_c\,R_{se}) / (R_c + R_{se})} \times \frac{R_c}{(R_c + R_{si})} = 0.10$$

$$H_{ao} = \frac{R_{se}\,R_{ox} / (\mathrm{R}_{se} + R_{ox})}{R_c + [(\mathrm{R}_{se}\,R_{ox}) / (\mathrm{R}_{se} + R_{ox})]} + \left(\frac{R_{se}\,R_c / (\mathrm{R}_{se} + R_c)}{R_{ox} + [(R_{se}\,R_c) / (R_{se} + R_c)]}\right)\left(\frac{R_{si}}{R_c + R_{si}}\right)$$

$$= 0.17$$

Table 5.53 summarises the steps in the calculation of the solar gain to the space.

The cyclic component of the convective and longwave radiant gain from the glazing to the environmental node is calculated as for triple glazing, see example A7.1, i.e:

$$\tilde{H}_e A = H_{ei}\,\tilde{A}_i + H_{ec}\,\tilde{A}_c + H_{eo}\,\tilde{A}_o$$

The instantaneous component of the convective and longwave radiant gain from the glazing to the air node is calculated as follows:

$$H_a A = H_{ai}\,A_i + H_{ao}\,A_o$$

Hence, for 12:00:

$$H_a A = (0.24 \times 116) + (0.17 \times 146) = 52.66$$

The gain at other times is calculated similarly, see Table 5.53. The mean gain is calculated using the mean, rather than the cyclic, absorption values.

The cyclic component of the directly transmitted shortwave radiation and attenuated by the surface factor (F) appropriate to the thermal weight of the building and delayed by a time corresponding to the thermal weight, see Table 5.6.

For a lightweight building, F = 0.8 and the time delay is 1 hour, i.e:

$$\tilde{T}_L = 0.8 \times (T_{t+1} - \bar{T})$$

and for a heavyweight building, F = 0.5 and the time delay is 2 hours, i.e:

$$\tilde{T}_H = 0.5 \times (T_{t+2} - \bar{T})$$

Table 5.53 Example A7.1: components of radiation

Time (h)	Solar irradiance / W·m^{-2}	Radiation absorbed/transmitted by glazing system / W·m^{-2}			Gains to space / W·m^{-2}			
		Radiation absorbed by inner, and outer glazing elements		Directly transmitted radiation, T	Cyclic components of absorbed radiation		Cyclic components of transmitted radiation for lightweight (L) and heavyweight (H) buildings	
		A_i	A_o		$(H_e A)$	$(H_a A)$	$\tilde{T}_L$	$\tilde{T}_H$
1200	442	116	146	47	45	53	—	—
1300	531	151	178	52	68	67	22	—
1400	572	171	191	54	81	74	26	14
1500	563	173	186	51	81	73	28	17
1600	504	157	167	45	70	66	26	18
Mean:	179	51	58	19	34	22	—	—

where subscript 'L' denotes thermally lightweight building and subscript 'H' denotes thermally heavyweight building.

As for example A7.1, the mean solar gain factor at the environmental node is given by:

$$\bar{S}_e = \frac{\text{mean transmitted radiation plus mean absorbed radiation}}{\text{daily mean incident radiation}}$$

Hence:

$$\bar{S}_e = (19 + 34) / 179 = 0.3$$

Again, as for example A7.1, the cyclic solar gain factors at the environmental node are calculated using the gains appropriate to a time depending on the thermal weight of the structure, i.e:

$$\tilde{S}_e = \frac{\text{total swing in gain to space}}{\text{swing in external gain}}$$

Peak solar irradiance occurs at 14:00; hence, for a thermally lightweight structure (i.e. 1 hour delay):

$$\tilde{S}_{eL} = (28 + 81) / (563 - 179) = 0.28$$

and for a thermally heavyweight structure (i.e. 2 hour delay):

$$\tilde{S}_{eH} = (18 + 70) / (504 - 179) = 0.27$$

The mean solar gain factor at the air node is given by:

$$\bar{S}_a = \frac{\text{mean gain to air node}}{\text{mean incident radiation}}$$

Hence:

$$\bar{S}_a = 22 / 179 = 0.12$$

The cyclic solar gain factor at the air node is given by:

$$\tilde{S}_a = \frac{\text{total swing in gain to air node}}{\text{swing in external gain}}$$

There is no time delay associated with the air node, hence:

$$\tilde{S}_a = 74 / (572 - 179) = 0.19$$

5.A7.4 Shading coefficients

In addition to solar gain factors, Table 5.7 also gives the shortwave and longwave shading coefficients (S_c). These correspond to the direct and indirect transmission to the space for direct radiation at normal incidence, divided by 0.87 (i.e. the transmission coefficient of nominal 4 mm plain glass at normal incidence). However, it should be noted that the room gains, and hence the solar gain factors, depend on the direct and diffuse components of the incident radiation and the angle of incidence of the direct radiation.

References for Appendix 5.A7

A7.1 *Weather and solar data* CIBSE Guide A2 (London: Chartered Institution of Building Services Engineers) (1986)

A7.2 Jones R H L Solar radiation through windows —theory and equations *Building Serv. Eng. Res. Technol.* **1**(2) 83–91 (1980)

A7.3 Parmelee G V and Vild D J Design data for slat-type sun shades for use in load estimating *ASHVE Trans.* **59** 1403–1434 (1953)

A7.4 Nicol J F Radiation transmission characteristics of louver systems *Building Science* **1** 167–182 (1966)

A7.5 Mitalas G P and Stephenson D G *Absorption and transmission of thermal radiation by single and double glazed windows* Research Paper No. 173 (National Research Council of Canada, Division of Building Research) (1962)

A7.6 Parmelee G V and Aubele W W The shading of sunlit glass *ASHVE Trans.* **58** 377–395 (1952)

Appendix 5.A8: Derivation of factor for intermittent heating

The symbols used in this appendix are defined in section 5.2.1.

Equation 5.53 defines the factor for intermittent heating, F_3, as follows:

$$F_3 = \Phi_p / \Phi_t \qquad (5.251)$$

where Φ_p is the installed capacity for intermittent operation (W) and Φ_t is the total heat loss (W).

Assuming that the installed capacity is that required to raise the space temperature from the daily mean space temperature ($\bar{\theta}_i$) to the internal design temperature (θ_i) then, for a design day where the mean outside temperature ($\bar{\theta}_o$) is equal to the design outside temperature:

$$F_3 = \frac{[\Sigma (A\,U) + C_v]\,(\bar{\theta}_i - \bar{\theta}_o) + [\Sigma (A\,Y) + C_v]\,(\theta_i - \bar{\theta}_i)}{[\Sigma (A\,U) + C_v]\,(\theta_i - \bar{\theta}_o)} \qquad (5.252)$$

Assuming the ventilation rate is constant and equal to the design value:

$$F_3 = \frac{\bar{\theta}_i - \bar{\theta}_o}{\theta_i - \bar{\theta}_o} + \frac{[\Sigma (A\,Y) + C_v]\,(\theta_i - \bar{\theta}_i)}{[\Sigma (A\,U) + C_v]\,(\theta_i - \bar{\theta}_o)} \qquad (5.253)$$

$$= \frac{\bar{\theta}_i - \bar{\theta}_o}{\theta_i - \bar{\theta}_o} + f_r \frac{(\theta_i - \bar{\theta}_i)}{(\theta_i - \bar{\theta}_o)} \qquad (5.254)$$

where f_r is the thermal response factor (see equation 5.17).

It has been shown that[(A8.1)]:

$$\frac{\bar{\theta}_i - \bar{\theta}_o}{\theta_i - \bar{\theta}_o} = \frac{t_o\, f_r}{t_o\, f_r + (24 - t_o)} \tag{5.255}$$

where t_o is hours of plant operation including preheat (h).

Therefore, subtracting both sides from θ_i and rearranging gives:

$$\frac{\theta_i - \bar{\theta}_i}{\theta_i - \bar{\theta}_o} = 1 - \frac{t_o\, f_r}{t_o\, f_r + (24 - t_o)} \tag{5.256}$$

Substituting equations 5.256 and 5.255 into equation 5.254 gives:

$$F_3 = \frac{24\, f_r}{t_o\, f_r + (24 - t_o)} \tag{5.257}$$

Reference for Appendix 5.A8

A8.1 Harrington-Lynn J Derivation of equations for intermittent heating used in CIBSE Building Energy Code Part 2a *Building Serv. Eng. Res. Technol.* **19**(4) (1998)

Appendix 5.A9: Specification for Reference (dynamic) Model

In order to satisfy the requirements of the Reference (dynamic) Model, the features indicated below should be incorporated. This specification is not exhaustive but gives sufficient detail to provide a basis for assessing computer models.

5.A9.1 Analytical method

Calculations should be carried out for time increments not exceeding one hour using appropriate time sequences of climatic data, internal load patterns and required control set points. These may be hourly average values or, if the calculation requires a time increment of less than one hour, measured data corresponding to the time increment should be used or values may be interpolated from hourly data.

5.A9.2 Climatic data

Data for the following parameters are required at time increments not exceeding one hour:

— dry bulb temperature

— moisture content (or equivalent)

— solar irradiation, comprising direct, sky diffuse, ground reflected (taking account of site factors), sky temperature (or other parameter appropriate to the determination of longwave radiation from external surfaces)

— wind speed

— wind direction.

The effect on the convective heat transfer coefficient of wind speed and direction should be taken into account.

The solar component should include longwave radiation transfer to the sky and surroundings.

The conversion of solar irradiance data measured at a particular orientation and slope into values for other orientations and slopes should be achieved using the methods described in CIBSE Guide J: *Weather and solar data*[(A9.1)].

Solar altitude and azimuth should be determined using the methods contained in CIBSE Guide J: *Weather and solar data*[(A9.1)].

The conversion of measured climatic data into the form required by the calculation procedure should be achieved using the relationships given in chapter 1 of CIBSE Guide C: *Reference data*[(A9.2)].

5.A9.3 Properties of opaque fabric

The following properties should be represented:

— thermal resistance

— thermal capacitance

— surface emissivity (at boundaries and internal cavities)

— surface absorption coefficient for shortwave radiation

— convective and radiant heat transfer characteristics within cavities.

The dynamic response of opaque components may be determined using finite difference techniques or by response factors; other methods may be used provided that it may be demonstrated that they can achieve equal precision[(A9.3)].

5.A9.4 Glazing

The following properties should be represented:

— thermal resistance

— solar absorption

— solar transmission

— surface emissivity

— convective and radiant heat transfer characteristics within internal cavities.

The performance of glazing systems should be based on the values of solar altitude and azimuth calculated at the solar time corresponding to the time for which the calculation is being performed. This may differ due to longitude and/or the effect of local adjustments for daylight saving.

The performance of glazing systems must take account of reflections between the elements comprising the system.

Separate calculations must be made for shaded and unshaded areas of glazed surfaces.

5.A9.5 Shading

Shading devices may consist of purpose built overhangs, side fins adjacent to or part of a window or moveable devices such as blinds, shutters or curtains.

The shading effect should be calculated for time increments not exceeding one hour using values of solar altitude and azimuth at the appropriate solar time. Where shading devices may be adjusted or controlled the effect of such features should be represented.

The model should take account of the effect of shading on glazing performance, as follows:

— in the case of purpose built shades, the determination of the amount and location of shade falling on the glazing; reflected radiation from the shades should also be considered

— for blinds and curtains, the absorbed and transmitted radiation to be calculated, if appropriate, as a function of slat angle; the interaction between glazing elements and blinds due to reflection of radiation from blinds must be represented.

Other obstacles to radiation such as shading by adjacent buildings and other site features should also be included, as should self-shading by the building under analysis.

5.A9.6 Internal longwave radiation

Longwave radiant heat transfer between surfaces and convective heat transfer between room air and room surfaces should be modelled using the fundamental heat balance described in Appendix 5.A3.

Longwave interchange between sources of internal heat gain and room surfaces must be modelled. The location of heat emitters should be taken into account.

5.A9.7 Internal shortwave radiation (direct solar gain)

The distribution of shortwave energy should be determined by calculation of the amount of direct and diffuse transmitted solar radiation incident upon each room surface. If a surface transmits shortwave radiation the quantity transmitted must be calculated using the same methods as for the transmission of solar radiation into the building. Reflections of shortwave radiation should be modelled.

The solar distribution must be calculated at the same frequency as that for the climatic data.

5.A9.8 Room air model

Convective heat gains may be assumed to enter directly into the air. The convective heat balance should include a representation of the thermal capacity of the room air. Under some circumstances it may be appropriate to increase the heat storage capacity of the air artificially to take account of furnishings. However, there is little guidance available on when this is necessary.

The convective heat transfer coefficient at room surfaces should be calculated as a function of surface and air temperatures; suitable correlations are given by Alamdari[(A9.4)] and Hatton[(A9.5)]. It is not considered practicable at present to include the influence of room air movement patterns.

5.A9.9 Infiltration and ventilation

The needs of design models and simulation models differ in that, for design purposes, it is usual to specify the value of infiltration whereas simulation techniques require this parameter to be calculated. Furthermore, ventilation to remove excess heat gain is an essential factor in the calculation of overheating risk. One way to determine ventilation rates is by means of a zonal airflow model. See chapter 4: *Infiltration and natural ventilation* for guidance on the calculation of natural ventilation rates. The program supplier should provide details of the method used and be able to justify the assumptions made in the model.

References for Appendix 5.A9

A9.1 *Weather and solar data* CIBSE Guide J (London: Chartered Institution of Building Services Engineers) (2002)

A9.2 *Properties of humid air* ch. 1 in *Reference data* CIBSE Guide C (London: Chartered Institution of Building Services Engineers) (2000)

A9.3 *Building energy and environmental modelling* CIBSE Applications Manual AM11 (London: Chartered Institution of Building Services Engineers) (1998)

A9.4 Almadari F and Hammond G P Improved data correlations for buoyancy driven convection in rooms *Building Serv. Eng. Res. Technol.* **4**(3) 106–112 (1980)

A9.5 Hatton A and Awbi H B Convective heat transfer in rooms *Proc. Building Simulation '95, August 1995, Wisconsin, USA* (1995)

6 Internal heat gains

6.1 Introduction

Internal heat gain is the sensible and latent heat emitted within an internal space from any source that is to be removed by air conditioning or ventilation, and/or results in an increase in the temperature and humidity within the space. It includes the following sources:

— bodies (human and animal)

— lighting

— computers and office equipment

— electric motors

— cooking appliances and other domestic equipment.

This chapter provides information on heat emission from various sources to enable designers to estimate internal heat gains. Designers can choose to estimate either the rate of internal heat gain, where sufficient is known about the use of the building, or base it on 'benchmark' values typical for the building and intended use and normally used by the industry. The choice will depend on the known or predicted use and likely change of use during the life of the building and building environmental services. If, for example, the building and services were to be designed speculatively in anticipation of a generic type of user, it could be appropriate to base the estimates of internal heat gains based on current practice or benchmark values. However, benchmark values are only available for common buildings.

If the building use is known it may be more appropriate to estimate the level of internal heat gains by comparison with measurements from similar buildings or calculate a value using measured heat gains from individual heat emitting devices and first principles. Such estimates should allow for the probability that all devices do not emit heat concurrently and at a constant rate. This is particularly the case for office equipment manufactured to comply with the US 'Energy Star' program*. Diversity factors should be allowed for the values of heat gain for the design of centralised air conditioning systems. The diversity factor increases, i.e. the load reduces, as the sum of areas served by the cooling distribution system approaches to the central cooling source.

This chapter also provides information on the proportions of radiant and convective heat from various sources of heat gain and the time delay caused by thermal storage of heat gains in building fabric.

* www.energystar.gov

6.2 Benchmark values for internal heat gains

Benchmark values for internal heat gains are based on either surveys of measured internal heat gains from a number of buildings of particular types and usage, or empirical values found appropriate from experience and considered good practice in the industry.

6.2.1 Office buildings

Most of the published surveys of internal heat gains have been carried out in office buildings. The main sources of internal heat gains in offices are the occupants, artificial lighting and office equipment connected to the small power electrical distribution. Surveys have identified a relationship between internal heat gains from small power office equipment and the density of occupation.

Stanhope[(1)] commissioned surveys of a number of different office buildings with different densities of occupation in 1993 and 2000. There was no significant difference in the results of measured internal heat gains from small power office equipment between the two surveys. The results of the surveys showed that an allowance of 15 $W{\cdot}m^{-2}$ was adequate for practically all types of offices and that occupation densities of 12 m^2/person and 16 m^2/person were appropriate design occupation densities for city centre offices and business parks respectively. The only exception might be offices used for financial activities such as dealers' rooms. The typical values were adopted by the British Council for Offices and recommended in the BCO's *Guide to best practice in the specification for offices*[(2)]. The results of these surveys were similar to surveys in the USA. Measurements on 44 office buildings were made in 1995 by Komor[(3)], who concluded that an allowance of not more than 13.4 $W{\cdot}m^{-2}$ for office equipment heat gains should not be exceeded unless the circumstances were exceptional.

Knight and Dunn[(4)] calculated the internal heat gains of 30 air conditioned office buildings in the UK, based on surveys. The occupant densities in the surveys ranged from 4 to 24 m^2 per person. The results show that total internal heat gains are proportional to occupant density. The relationship can be seen in Figure 6.1.

The internal heat gains include lighting and occupant gains in addition to equipment. Benchmark values for lighting heat gain are given in Energy Consumption Guide ECG019: *Energy use in offices*[(5)]. The maximum value for offices is 12 $W{\cdot}m^{-2}$. The results of the above surveys of benchmark values for lighting energy use are combined in Table 6.1 to give benchmark values for total internal heat gains for typical offices at various occupant densities.

Table 6.1 Benchmark values for internal heat gains for offices (at 24 °C, 50% RH)

Building type	Use	Density of occupation / person·m^{-2}	Sensible heat gain / W·m^{-2}			Latent heat gain / W·m^{-2}	
			People	Lighting	Equip't	People	Other
Office	General	4	20	12	25	15	—
		8	10	12	20	7.5	—
		12	6.7	12	15	5	—
		16	5	12	12	4	—
		20	4	12	10	3	—

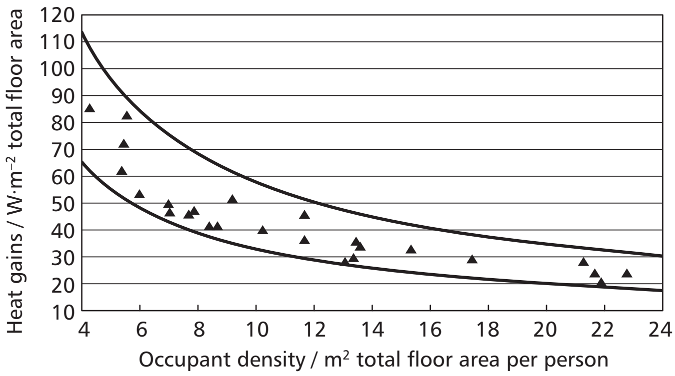


Figure 6.1 Variation of calculated total heat gains with occupation density[4]

6.2.2 Other building types

There are few published surveys of measured internal heat gains for other types of buildings. Table 6.2 (opposite) provides typical internal heat gains for some common buildings and uses.

6.3 Occupants

All active animal bodies including humans lose heat to their surroundings. This section deals with human beings but a method for estimating the heat gains from animals is included as Appendix 6.A1.

The emission of heat from a human body in relation to the surrounding indoor climate is discussed in chapter 1 section 1.3.1.5. Table 6.3 provides representative heat emissions (sensible and latent) from an average adult male in different states of activity. The figures for a mixture of males and females assume typical percentages of men, women and children for the stated building type.

The latent heat gain from a human body results in an instantaneous addition to the moisture content of the air, whereas the part of the sensible heat gain is absorbed by the surrounding surfaces and stored in the material. Between 20 and 60%[6] of the sensible heat emission, can be radiant depending on type of clothing, activity, mean radiant temperature and air velocity. Indicative values for high and low rates of air movement are shown in two columns on the right hand side of the table.

Table 6.3 Typical rates at which heat is given off by human beings in different states of activity.

Degree of activity	Typical building	Total rate of heat emission for adult male / W	Rate of heat emission for mixture of males and females / W			Percentage of sensible heat that is radiant heat for stated air movement / %	
			Total	Sensible	Latent	High	Low
Seated at theatre	Theatre, cinema (matinee)	115	95	65	30	—	—
Seated at theatre, night	Theatre, cinema (night)	115	105	70	35	60	27
Seated, very light work	Offices, hotels, apartments	130	115	70	45	—	—
Moderate office work	Offices, hotels, apartments	140	130	75	55	—	—
Standing, light work; walking	Department store, retail store	160	130	75	55	58	38
Walking; standing	Bank	160	145	75	70	—	—
Sedentary work	Restaurant	145	160	80	80	—	—
Light bench work	Factory	235	220	80	140	—	—
Moderate dancing	Dance hall	265	250	90	160	49	35
Walking; light machine work	Factory	295	295	110	185	—	—
Bowling	Bowling alley	440	425	170	255	—	—
Heavy work	Factory	440	425	170	255	54	19
Heavy machine work; lifting	Factory	470	470	185	285	—	—
Athletics	Gymnasium	585	525	210	315	—	—

Source: ASHRAE Handbook: *Fundamentals* (2001)[6]

Table 6.2 Benchmark allowances for internal heat gains in typical buildings

Building type	Use	Density of occupation / person·m^{-2}	Sensible heat gain / W·m^{-2}			Latent heat gain / W·m^{-2}	
			People	Lighting*	Equip't†	People	Other
Offices	General	12	6.7	8–12	15	5	—
		16	5	8–12	12	4	—
	City centre	6	13.5	8–12	25	10	—
		10	8	8–12	18	6	—
	Trading/dealing	5	16	12–15	40 +	12	—
	Call centre floor	5	16	8–12	60	12	—
	Meeting/conference	3	27	10–20	5	20	—
	IT rack rooms	0	0	8–12	200	0	—
Airports/stations‡	Airport concourse	0.83	75	12	5	4	—
	Check-in	0.83	75	12	5	50	—
	Gate lounge	0.83	75	15	5	50	—
	Customs /immigration	0.83	75	12	5	50	—
	Circulation spaces	10	9	12	5	6	—
Retail	Shopping malls	2–5	16–40	6	0	12–30	—
	Retail stores	5	16	25	5	12	—
	Food court	3	27	10	†	20	§
	Supermarkets	5	16	12	†	12	§
	Department stores:						
	— jewellery	10	8	55	5	6	—
	— fashion	10	8	25	5	6	—
	— lighting	10	8	200	5	6	—
	— china/glass	10	8	32	5	6	—
	— perfumery	10	8	45	5	6	—
	— other	10	8	22	5	6	—
Education	Lecture theatres	1.2	67	12	2	50	—
	Teaching spaces	1.5	53	12	10	40	—
	Seminar rooms	3	27	12	5	20	—
Hospitals	Wards	14	57	9	3	4.3	—
	Treatment rooms	10	8	15	3	6	—
	Operating theatres	5	16	25	60	12	—
Leisure	Hotel reception	4	20	10–20	5	15	—
	Banquet/conference	1.2	67	10–20	3	50	—
	Restaurant/dining	3	27	10–20	5	20	—
	Bars/lounges	3	27	10–20	5	20	—

* The internal heat gain allowance should allow for diversity of use of electric lighting coincident with peak heat gain and maximum temperatures. Lighting should be switched off in perimeter/window areas (up to say 4.5 m) and no allowance account for any dimming or other controls.

† Equipment gains do not allow for large duty local equipment such as heavy-duty photocopiers and vending machines.

‡ The exact density will depend upon airport and airplane capacity, the type of gate configuration (open or closed) and passenger throughput. Absolute passenger numbers if available would be a more appropriate design basis. Appropriate building scale diversities need to be derived based on airport passenger throughput.

§ Latent gains are likely but there are no benchmark allowances and heat gains need to be calculated from the sources, e.g. for meals, 15 W per meal(6) served, of which 75% is sensible and 25% latent heat; see also Appendix 6.A2

6.4 Lighting

6.4.1 General

All the electrical energy used by a lamp is ultimately released as heat. The energy is emitted by means of conduction, convection or radiation. When the light is switched on the luminaire itself absorbs some of the heat emitted by the lamp. Some of this heat may then be transmitted to the building structure, depending on the manner in which the luminaire is mounted. The radiant energy emitted (both visible and invisible) from a lamp will result in a heat gain to the space only after it has been absorbed by the room surfaces. This storage effect results in a time lag before the heat appears as a part of the cooling load.

In determining the internal heat gains due to artificial lighting the following must be known:

- total electrical input power
- fraction of heat emitted which enters the space
- radiant, convective and conductive components.

Both the total electrical input power and the distribution of the heat output will vary with manufacturer. In particular, the optical properties of the luminaire can affect greatly the radiant/convective proportion emitted by the lamp. All figures quoted in the following section are typical. Manufacturers' data should be used where possible.

6.4.2 Total electrical power input

The total electrical power input to the lighting installation must be known. For lamps with associated control gear, it is important to add the power dissipated by the control gear to that dissipated by the lamp. The control gear power loss is likely to be about 10% of the lamp rating for electronic ballast and about 20% for conventional ballast[7].

Case studies carried out on a number of offices built or refurbished between 1977 and 1983 found that the lighting loads were between 10 and 32 $W \cdot m^{-2}$ for a maintained illuminance levels of 150–800 lux[8]. Surveys carried out on newer buildings found that the lighting loads were in the range 8–18 $W \cdot m^{-2}$ for a maintained illuminance levels of 350–500 lux[9].

Where the actual installed power is not known reference should be made to Table 6.4[10], which provides target installed power densities for various task illuminances.

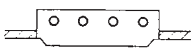
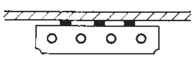
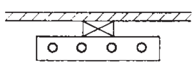
Table 6.4 Lighting energy targets

Application	Lamp type	Task illuminance / lux	Average installed power density / $W \cdot m^{-2}$
Commercial and similar applications (e.g. offices, shops*, schools)	Fluorescent-triphosphor	300	7
		500	11
		750	17
	Compact fluorescent	300	8
		500	14
		750	21
	Metal halide	300	11
		500	18
		750	27
Industrial and manufacturing	Fluorescent-triphosphor	300	6
		500	10
		750	14
		1000	19
	Metal halide	300	7
		500	12
		750	17
		1000	23
	High pressure sodium	300	6
		500	11
		750	16
		1000	21

* Excluding display lighting

Source: *Code for Lighting* (2002)[10]

Table 6.5 Measured energy distribution for fluorescent fittings having four 70 W lamps[7]

Type of fitting			Energy distribution / %	
Mounting	Schematic	Description	Upwards	Downwards
Recessed		Open	38	62
		Louvre	45	55
		Prismatic or opal diffuser	53	47
Surface		Open	12	88
		Enclosed prismatic or opal	22	78
		Enclosed prismatic on metal spine	6	94

6.4.3 Fraction of emitted heat entering the space

The proportion of heat entering the space depends upon the type and location of the light fittings.

Where the lamp or luminaire is suspended from the ceiling or wall-mounted or where uplighters or desk lamps are used, all the heat input will appear as an internal heat gain.

Where recessed or surface-mounted luminaires are installed below a false ceiling, some of the total input power will result in a heat gain to the ceiling void. An accurate assessment of the distribution of energy from particular types of luminaire should be obtained from the manufacturer. In the absence of manufacturer's data, Table 6.5 provides an indication of the energy distribution for various arrangements of fluorescent lamp luminaires, based on laboratory measurements(7).

For air handling luminaires, up to 80% of the total input power can be removed by the air stream, leaving only 20% to enter the space as heat gain. The specific manufacturer should be contacted for actual test data. Heat taken away from a luminaire through a ceiling plenum, or directly from the luminaire itself, will not form part of the room sensible heat gain but may still constitute part of the total refrigeration load.

6.4.4 Radiant, convective and conductive components

Little information exists on the proportions of radiant, convective and conducted heat gain from lighting. Lamps radiate in both the visible and invisible wavebands and there will be a net gain of infrared radiation from the lamp and luminaire due to their radiant temperature being above the room mean radiant temperature. Table 6.6 provides approximate data for different lamp types and shows that a substantial proportion of the energy dissipated by all sources is emitted as radiant heat. Radiant heat can cause discomfort to the occupants. It is mainly detected by the occupants on the forehead and the backs of the hands as these parts are more sensitive to radiant heat than other parts of the body. The optics and body design of the luminaire can reduce substantially the radiant component and, for the purposes of determining room cooling load, it may be sufficient to assume that the heat is purely convective.

Table 6.6 Energy dissipation in lamps(10)

Lamp type	Heat output / %		
	Radiant	Conducted/ convected*	Total
Fluorescent	30	70	100
Filament (tungsten)	85	15	100
High pressure mercury/ sodium, metal halide	50	50	100
Low pressure sodium	43	57	100

* The power loss of ballasts should be added to the conducted/convected heat.

6.5 Personal computers and office equipment

6.5.1 General

Personal computers (PCs) and associated office equipment result in heat gains to the room equal to the total power input. The internal heat gains for this equipment is normally allocated as an allowance in watts per square metre ($W \cdot m^{-2}$) of net usable floor area. Typical values are given in section 6.2 above.

The internal heat gains can be estimated from basic data but care must be taken to allow for diversity of use, idle operation and the effects of energy saving features of the equipment

6.5.2 Individual machine loads

It is well documented that nameplate power overstates the actual power and consequent heat gain. Hosni et al.(11) found with nameplate consumption of less than 1000 W the ratio of heat gain to nameplate power ranged from 25% to 50% and concluded the most accurate ratio for determining heat gain was 25%.

The heat gain from PCs will fall significantly when they are equipped with the Energy Star(12) feature. The Energy Star features apply to all office equipment including PCs, CRT and flat screen monitors, printers, fax machines, photocopiers and scanners. The Energy Star qualification started in the US and has been adopted by the European Union(13). To qualify as Energy Star compliant equipment has three levels of power consumption: normal, standby and sleep. The specifications set out the maximum levels of power consumption in the sleep mode and the default time for the equipment to enter sleep mode. Wilkins and Hosni(14) measured the power consumption of various PCs and other office equipment. Tables 6.7 and 6.8 show their results as typical heat gains from PCs and monitors in the continuous and energy saver modes. (Similar data for flat screen monitors could not be located.)

Table 6.7 Typical heat gains from PCs(14)

Nature of value	Value for stated mode / W	
	Continuous	Energy saving
Average	55	20
Conservative	65	25
Highly conservative	75	30

Table 6.8 Typical heat gains from PC monitors(14)

Monitor size	Value for stated mode / W	
	Continuous	Energy saving
Small (13–15 inch)	55	0
Medium (16–18 inch)	70	0
Large (19–20 inch)	80	0

Wilkins and Hosni[14] also reported measurements of the power consumption of other types of office equipment. They found the power varied with the level of throughput for laser printers, see Table 6.9.

Office photocopiers were divided into desktop and freestanding office grade copiers. Freestanding copiers are more likely to be operated continuously and are often located outside the main office area. Typical heat gains from photocopiers are shown in Table 6.10.

Table 6.11 gives heat outputs from some other items of office equipment.

Table 6.9 Typical heat gains from laser printers[14]

Printer size	Value for stated mode / W		
	Continuous	1-page/min.	Idle
Small desktop	130	75	10
Desktop	215	100	35
Small office	320	160	70
Large office	550	275	125

Table 6.10 Typical heat gains from photocopiers[14]

Copier size	Value for stated mode / W		
	Continuous	1-page/min.	Idle
Desktop copier	400	85	20
Office copier	1100	400	300

Table 6.11 Typical heat gains from office equipment[14]

Device	Value for stated mode / W	
	Continuous	Energy saving
Fax machine	30	15
Scanner	25	15
Dot matrix printer	50	25

6.5.3 Diversity

The actual peak internal heat gain for all office equipment in a single common area is less than the sum of the individual continuous gains due to diversity of use. Diversity is not the difference between the nameplate power and heat gain. It is the factor that accounts for a percentage of equipment being idle or turned off. There have been a number of surveys of office equipment in use that include the effect of diversity[1,2]. Wilkins and McGaffin[15] surveyed 23 areas in five buildings and found diversity varied between 37 and 78%. The results are shown in Figure 6.2, which compares nameplate power with no diversity and the actual heat gain including diversity.

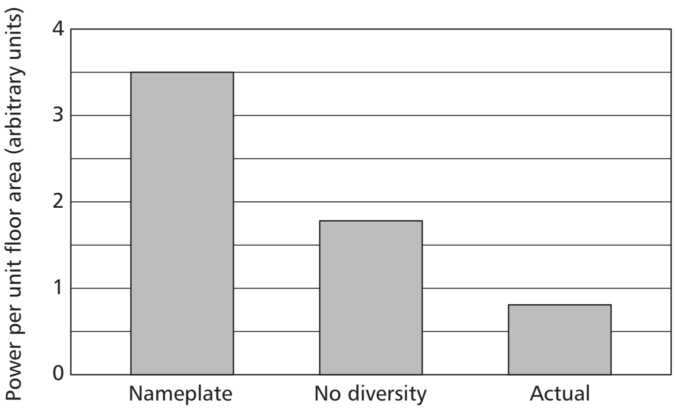


Figure 6.2 Load factor comparison

6.5.4 Radiation, convection and conduction components

The electrical consumption of office equipment gives the total heat dissipated but it is also useful to know how that heat is transferred to the space. Heat that is convected or conducted is an instantaneous gain whereas that which is radiated is absorbed by the building mass and dissipated over time. Little information is available on the radiant, conductive and convective components of heat gain from office equipment. Table 6.12 provides an indication of the likely breakdown of heat gains for a computer, monitor and laser printer[16]. These figures should be used with caution since they were obtained from tests of a single computer, two monitors and one laser printer.

Table 6.12 Components of heat gain dissipated from office equipment[16]

Device	Heat gain component / %	
	Convective/conductive	Radiant
Desktop computer	86	14
Monitor	65	35
Laser printer	67	33

6.6 Electric motors

6.6.1 General

For situations where the motor and the motor driven equipment are both situated within the space (e.g. machinery in a workshop), the heat output is given by:

$$\Phi_g = P_a / \eta_t \qquad (6.1)$$

where Φ_g is the rate of heat gain to the space (W), P_a is the power at the equipment shaft (W) and η_t is the overall efficiency of transmission. The overall efficiency of transmission (η_t) is the product of the motor efficiency (η_m) and the drive efficiency (η_d).

For situations where the motor is situated within the space but the driven equipment is situated elsewhere:

$$\Phi_g = P_a\,[(1 / \eta_t) - 1] \qquad (6.2)$$

For motor driven equipment situated within, or related to the space (e.g. fans) but with the motor situated outside the space:

$$\Phi_g = P_a \qquad (6.3)$$

For precise details of efficiencies, which will vary with motor type, speed, performance and the character of the drive, reference should be made to manufacturers' data. For preliminary system design, in the absence of such data, reference may be made to Tables 6.13 and 6.14.

'High efficiency' motors are designed to minimise the inherent losses of the motor by using more copper in the stator and low-loss steel in the rotor. The improvement in efficiency is greatest at part-load, see Figure 6.3[17], particularly for loads below 50%.

Table 6.13 Average efficiencies for electric motors

Motor output rating	Average motor efficiencies, η_m / %			
	DC motors	AC motors		
		Single phase	Two-phase	Three-phase
0.75	76	65	73	74
3.75	83	78	84	85
7.50	86	81	87	88
15	88	83	88	90
38	90	85	91	91
56	92	86	92	92

Table 6.14 Average drive efficiencies

Drive	Drive efficiency, η_d / %
Plain bearings	95–98
Roller bearings	98
Ball bearings	99
Vee-belts	96–98
Spur gears	93
Bevel gears	92

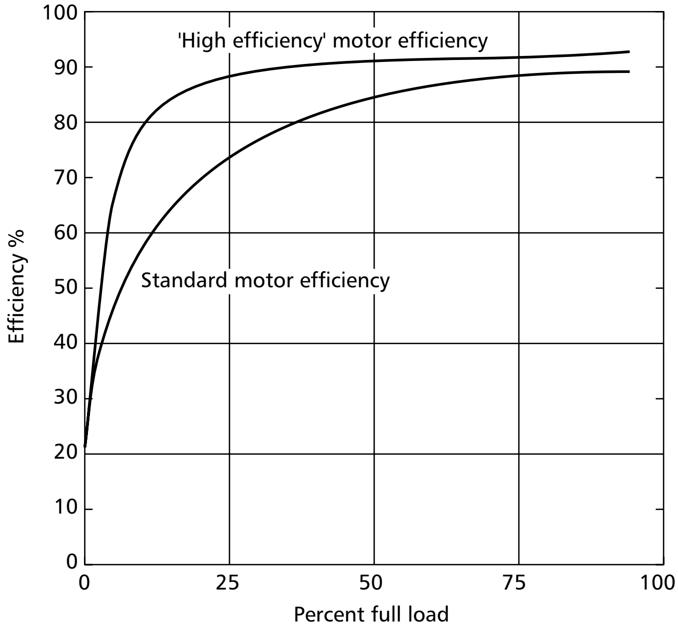


Figure 6.3 Comparison of efficiencies of standard and 'high efficiency' motors[17]

6.6.2 Escalator motors

It may be assumed that all the input power to the escalator motor will be converted to heat (ignoring the potential energy gained by ascending passengers). However, the motor will normally run at less than the motor rating and guidance should be sought from the manufacturer.

6.6.3 Lift motors

It may be assumed that all the input power to the lift motor will be dissipated as heat within the lift motor room. The motor will not work continuously, nor at constant load. Table 6.15 may be used for preliminary systems design, in the absence of manufacturers' data.

6.7 Cooking appliances

6.7.1 General

Heat gain estimates for cooking appliances are subjective due to the variety of appliances, applications, time in use and types of installation. In estimating appliance loads, the probability of simultaneous use and operation for different appliances located in the same area must be considered.

To estimate heat gains from cooking appliances the actual energy input rating supplied by manufacturers should be used, suitably modified by appropriate usage factors, efficiencies or other judgmental factors. When preliminary assessment is required prior to the establishment of detailed design, Appendix 6.A2 provides typical data[6] for a wide range of appliances. Such preliminary assessments should be checked once manufacturers' information is available.

6.7.2 Hooded appliances

Laboratory tests of hooded cooking appliances have indicated that the heat gains from effective hooded cooking

Table 6.15 Measured average power consumption of passenger lift motors

Drive type	Speed / $m \cdot s^{-1}$	Number of passengers									
		8		10		13		16		21	
		Motor rating / kW	Average power / kW	Motor rating / kW	Average power / kW	Motor rating / kW	Average power / kW	Motor rating / kW	Average power / kW	Motor rating / kW	Average power / kW
Geared variable voltage	1.0	10	2.4	10	2.4	2.4	10	12	2.9	15	3.6
	1.6	15	3.6	15	3.6	15	3.6	17.6	4.2	22	5.3
Geared variable frequency	1.0	5.5	1.0	7.5	1.3	9.5	1.6	11	1.8	15	2.4
	1.6	9.5	1.6	11	1.8	13	2.2	18.5	3.0	22	3.6
Gearless static direct drive	2.5	—	—	—	—	—	3.8	—	4.2	—	5.2
	4.0	—	—	—	—	—	6.7	—	7.6	—	8.6

appliances is primarily radiant, and that latent and convective heat are exhausted and do not enter the space[18].

The radiant heat gain from hooded cooking appliances varies from 15 to 45% of the actual energy consumption of the appliance[19,20]. This may be expressed as a radiation factor, F_r, which depends on the appliance type and the fuel used by the appliance. The rate of heat gain to the space, Φ_h, is obtained by multiplying the average rate of energy consumption for the appliance by the radiation factor. The average rate of energy consumption for the appliance is obtained from the manufacturer's rated energy input, Φ_i, by applying a usage factor, F_u. Therefore:

$$\Phi_h = F_r \, (F_u \times \Phi_i) \qquad (6.4)$$

where Φ_h is the rate of sensible heat gain to the space from a hooded appliance (W), Φ_i is the manufacturer's input rating or nameplate rating (W), F_r is the radiant factor and F_u is the usage factor.

Values for F_r and F_u for the main types of cooking equipment are given in Table 6.16.

The usage factor, F_u, is the ratio of the standby or idle energy input to manufacturer's input rating. For appliances with a hood not listed in Table 6.16, typical values are 0.5 for types of equipment which cycle or require a constant temperature to be maintained, 0.4 for refrigerators and freezers and 1.0 for all other types of equipment.

The radiation factor, F_r, is the ratio of maximum room heat gain to the idle energy input of a hooded appliance. An average value is 0.32.

6.7.3 Appliances without hoods

For cooking appliances not installed under an extract hood nor connected directly to an exhaust duct, a usage factor of 0.5 should be assumed, regardless of the type of energy or fuel used by the appliance. On average, 34% of the total heat gain may be assumed to be latent and 66% sensible heat[6].

For the purposes of estimating cooling loads, appliances served by hoods which are not exhausted to outside should be treated as appliances without hoods.

6.8 Hospital and laboratory equipment

Hospital and laboratory equipment can be major sources of heat gain in conditioned spaces. As this equipment is highly specialised, heat outputs for the specific pieces of equipment intended to occupy the space should be obtained from manufacturer. Care must be taken in evaluating the probability and duration of simultaneous usage when components are concentrated in one area.

For laboratories, the heat gains from equipment will vary widely according to the type of laboratory and the equipment likely to be installed and specific data should be obtained from the equipment suppliers. Heat gains of 50 to 200 $W \cdot m^{-2}$ are common for laboratories with high concentrations of equipment[21].

References

1 *A review of small power provision and occupation densities in office buildings* Stanhope Position Paper (London: Stanhope) (2001) (www.stanhopeplc.com)

2 *BCO Guide 2005: Best practice in the specification for offices* (London: British Council for Offices) (2005) (www.bco.org.uk)

3 Komor P Space cooling demands from office plug loads *ASHRAE J.* **39** (12) 41–44 (December 1997)

4 Knight I and Dunn G Evaluation of heat gains in UK office environments *Proc. CIBSE/ASHRAE Conf., Edinburgh, 2003* (2003)

5 *Energy use in offices* Energy Consumption Guide ECG019 (The Carbon Trust) (www.thecarbontrust.co.uk)

6 *Nonresidential cooling and heating load calculations* ASHRAE Handbook: *Fundamentals* (ch. 30) (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (2005)

7 Bedocs L and Hewitt H Lighting and the thermal environment *J. Inst. Heating and Ventilating Eng.* **37** 217–231 (January 1970)

8 *Energy Efficiency in Offices* Good Practice Case Studies Nos. 1/14/15/16/17/18/19/20/21/62 (Garston: Building Research Energy Conservation Support Unit) (various dates)

9 Probe 1: Tanfield House *Building Serv., CIBSE J.* **17** (9) 38–41 September 1995; Probe 2: 1 Aldermary Square *Building Serv., CIBSE J.* **17** (12) 29–33 December 1995; Probe 3: C&G Chief Office *Building Serv., CIBSE J.* **18** (2) 31–34 February 1996;

Table 6.16 Useage and radiation factors for hooded cooking appliances[1]

Appliance	Usage and radiation factors for appliances using stated fuel			
	Electrical appliance		Gas appliances	
	Usage factor (F_u)	Radiation factor (F_r)	Usage factor (F_u)	Radiation factor (F_r)
Griddle	0.16	0.45	0.25	0.25
Fryer	0.06	0.43	0.07	0.35
Convection oven	0.42	0.17	0.42	0.20
Charbroiler	0.83	0.29	0.62	0.18
Open top range, without oven	0.34	0.46	0.34	0.17
Hot-top range:				
— without oven	0.79	0.47	—	—
— with oven	0.59	0.48	—	—
Steam cooker	0.13	0.30	—	—

Probe 4: Queens Building *Building Serv., CIBSE J.* **18** (4) 35–38 April 1996; Probe 5: Cable and Wireless College *Building Serv., CIBSE J.* **18** (6) 35–39 June 1996; Probe 6: Woodhouse Medical Centre *Building Serv., CIBSE J.* **18** (8) 35–38 August 1996; Probe 7: Gardner House *Building Serv., CIBSE J.* **18** (10) 39–43 October 1996

10 *Code for Lighting* (London: Society of Light and Lighting) (2004)

11 Hosni M H, Jones B W and Xu H Experimental results for heat gain and radiant/convective split from equipment in buildings (RP-1055) *ASHRAE Trans.* **105**(2) 527–539 (1999)

12 *Energy Star Office Equipment Product Specification, Attachment A* (Washington DC: US Environmental Protection Agency) (www.energystar.gov)

13 Council Decision of 8 April 2003 concerning the conclusion on behalf of the Community of the Agreement between the Government of the United States of America and the European Community on the coordination of energy-efficient labelling programmes for office equipment (2003/269/EC) *Official J. of the European Union* **L99** (46) 47–48 (17.04.2003)

14 Wilkins C K and Hosni M H Heat gain from office equipment *ASHRAE J.* **42** (6) 33 (June 2000)

15 Wilkins C K and McGaffin N Measuring computer equipment loads in office buildings *ASHRAE J.* **36** (8) 21–24 (1994)

18 Wilkins C K, Kosonen R and Laine T An analysis of office equipment load factors *ASHRAE J.* **33** (9) 38–44 (September 1991)

17 *Energy efficiency in buildings* CIBSE Guide F (London: Chartered Institution of Building Services Engineers) (2004)

18 Marn W L *Commercial gas kitchen ventilation studies* Research Bulletin No. 90 (Cleveland OH: Gas Association Laboratories) (March 1962)

19 Gordon E B, Horton D J and Parvin F A Development and application of a standard test method for the performance of exhaust hoods with commercial cooking appliances *ASHRAE Trans.* **100** (2) 988–999 (1994)

20 Smith V A, Swierczyna R T and Claar C N Application and enhancement of the standard test method for the performance of commercial kitchen ventilation systems *ASHRAE Trans.* **101** (2) (1995)

21 *Laboratories* ASHRAE Handbook: *Applications* (ch. 14) (ISBN 1 883413 23 0) (Atlanta GA: American Society of Heating, Refrigerating and Air-conditioning Engineers) (1995)

Appendix 6.A1: Rate of heat emission from animal bodies

The sensible and latent heat emissions from the bodies of a variety of animals of average mass are listed in Table 6.17.

In the absence of experimental results, an approximation to the basal metabolic rate (BMR) from an animal may be established using the expression:

$$h = 3.2\, m^{0.75} \tag{6.5}$$

where h is the basal metabolic rate (W) and m is the mass of the animal (kg).

The basal metabolic rate is the rate at which heat is emitted from a body at rest in a warm environment. The equation should be modified to take account of any physical activity.

Table 6.17 Estimated sensible and latent heat emissions from animal bodies at normal body temperature

Creature	Average body weight / kg	Rectal temp. / °C	BMR / W	Typical occupancy per 10 m² floor area	Rate of heat emission* / W	
					Sensible	Latent
Mouse	0.02	36.5	0.175	2000	0.5	0.3
Hamster	0.12	36.9	0.483	1350	1.6	0.4
Rat	0.30	37.3	1.32	485	3.7	1.2
Guinea pig	0.41	39.1	1.7	400	4.6	2.2
Rabbit	2.6	39.4	5.6	32	8.5	2.5
Cat	3.0	38.6	7.35	16	11	3.8
Monkey	4.2	38.8	10	16	24	14
Dog	16	38.9	26	5	40	13
Goat	36	39.2	41	5	62	21
Sheep	45	38.8	56	5	81	29
Pig	250	39.3	210	1.5	317	106
Pigeon	0.27	43.3	1.35	400	2.2	0.5
Chicken	2.0	41.4	5.6	195	9.2	1.8

* Based on a 24-hour average; during periods of high activity the heat production may be double that of the estimated 24-hour average

Appendix 6.A2: Rate of heat gain from restaurant/cooking equipment

The following table is reproduced from ASHRAE Handbook: *Fundamentals*(6) by kind permission of the American Society of Heating, Refrigerating and Air Conditioning Engineers.

Table 6.17 Typical rates of heat gain from restaurant and cooking equipment(6) (Copyright 2005, ©American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (www.ashrae.org). Reprinted by permission from 2005 ASHRAE Handbook— Fundamentals. This text may not be copied nor distributed in either paper or digital form without ASHRAE's permission)

Appliance	Size	Energy rate / W		Rate of heat gain / W			
		Rated	Standby	Without hood			With hood
				Sensible	Latent	Total	Sensible
(*a*) Electric, no hood required							
Barbeque (pit), per kg of food capacity	36–136 kg	88	—	57	31	88	27
Barbeque (pressurised), per kg food capacity	20 kg	210	—	71	35	106	33
Blender, per litre of capacity	1.0–3.8 litre	480	—	310	160	470	150
Braising pan, per litre of capacity	102–133 litre	110	—	55	29	84	40
Cabinet, large, hot (holding)	0.46–0.49 m^3	2080	—	180	100	280	85
Cabinet, large, hot (serving)	1.06–1.15 m^3	2000	—	180	90	270	82
Cabinet, large (proofing)	0.45–0.48 m^3	2030	—	180	90	270	82
Cabinet, small, hot (holding)	0.09–0.18 m^3	900	—	80	40	120	37
Cabinet, very hot, (holding)	0.49 m^3	6150	—	550	280	830	250
Can opener		170	—	170	0	170	0
Coffee brewer	12 cup/2 burners	1660	—	1100	560	1660	530
Coffee heater, per boiling burner	1–2 burners	670	—	440	230	670	210
Coffee heater, per warming burner	1–2 burners	100	—	66	34	100	32
Coffee/hot water boiling urn, per litre of capacity	11 litre	120	—	79	41	120	38
Coffee brewing urn (large), per litre of capacity	22–38 litre	660	—	440	220	660	210
Coffee brewing urn (small), per litre of capacity	10 litre	420	—	280	140	420	130
Cutter (large)	460 mm bowl	750	—	750	0	750	0
Cutter (small)	360 mm bowl	370	—	370	0	370	0
Cutter and mixer (large)	28–45 litre	3730	—	3730	0	3730	0
Dishwasher (hood type, chemical sanitising), per 100 dishes/h	950–2000 dish/h	380	—	50	110	160	50
Dishwasher (hood type, chemical sanitising), per 100 dishes/h	950–2000 dish/h	380	—	56	123	179	56
Dishwasher (conveyor type, chemical sanitising) per 100 dishes/h	5000–9000 dish/h	340	—	41	97	138	44
Dishwasher (conveyor type, water sanitising) per 100 dishes/h	5000–9000 dish/h	340	—	44	108	152	50
Display case (refrigerated), per m^3 of interior	0.17–1.9 m^3	1590	—	640	0	640	0
Dough roller (large)	2 rollers	1610	—	1610	0	1610	0
Dough roller (small)	1 roller	460	—	460	0	460	0
Egg cooker	12 eggs	1800	—	850	570	1420	460
Food processor	2.3 litre	520	—	520	0	520	0
Food warmer (infrared bulb), per lamp	1–6 bulbs	250	—	250	0	250	250
Food warmer (shelf type), per m^2 of surface	0.28–0.84 m^3	2930	—	2330	600	2930	820
Food warmer (infrared tube), per metre length	1.0–2.1 m	950	—	950	0	950	950
Food warmer (well type), per m^3 of well	20–70 litre	37 400	—	12 400	6360	18 760	6000
Freezer (large)	2.07 m^3	1340	—	540	0	540	0
Freezer (small)	0.51 m^3	810	—	320	0	320	0
Griddle/grill (large), per m^2 of cooking surface	0.43–1.1 m^2	29 000	—	1940	1080	3020	1080
Griddle/grill (small), per m^2 of cooking surface	0.20–0.42 m^2	26 200	—	1720	970	2690	940
Hot dog boiler	48–56 hot dogs	1160	—	100	50	150	48
Hot plate (double burner, high speed)		4900	—	2290	1590	3880	1830
Hot plate (double burner, stockpot)		4000	—	1870	1300	3170	1490
Hot plate (single burner, high speed)		2800	—	1310	910	2220	1040
Hot water run (large), per litre of capacity	53 litre	130	—	50	16	66	21
Hot water run (small), per litre of capacity	7.6 litre	2.30	—	87	30	117	37
Ice maker (large)	100 kg/day	1090	—	2730	0	2730	0
Ice maker (small)	50 kg/day	750	—	1880	0	1880	0
Microwave oven (heavy duty, commercial)	20 litre	2630	—	2630	0	2630	0
Microwave oven (residential type)	30 litre	600–1400	—	600–1400	0	600–1400	0
Mixer (large), per litre of capacity	77 litre	29	—	29	0	29	0
Mixer (small), per litre of capacity	11–72 litre	15	—	15	0	15	0
Press cooker (hamburger)	300 patties/h	2200	—	1450	750	2200	700
Refrigerator (large), per m^3 of interior space	0.71–2.1 m^3	780	—	310	0	310	0
Refrigerator (small) per m^3 of interior space	0.17–0.71 m^3	1730	—	690	0	690	0
Rotisserie	300 burgers/h	3200	—	2110	1090	3200	1020
Serving cart (hot), per m^3 of well	50–90 litre	21 200	—	7060	3530	10 590	3390
Serving drawer (large)	252–336 rolls	1100	—	140	10	150	45

Table continues

Table 6.17 Typical rates of heat gain from restaurant and cooking equipment[6] — *continued*

Appliance	Size	Energy rate / W		Rate of heat gain / W			
		Rated	Standby	Without hood			With hood
				Sensible	Latent	Total	Sensible
Serving drawer (small)	84–168 rolls	800	—	100	10	110	33
Skillet (tilting), per litre of capacity	45–125 litre	180	—	90	50	140	66
Slicer, per square metre of slicing carriage	0.06–0.09 m^2	2150	—	2150	0	2150	680
Soup cooker, per litre of well	7–11 litre	130	—	45	24	69	21
Steam cooker, per m^3 of compartment	30–60 litre	214 000	—	17 000	10 900	27 900	8120
Steam kettle (large), per litre of capacity	76–300 litre	95	—	7	5	12	4
Steam kettle (small), per litre of capacity	23–45 litre	260	—	21	14	35	10
Syrup warmer, per litre of capacity	11 litre	87	—	29	16	45	14
Toaster (bun toasts on one side only)	1400 buns/h	1500	—	800	710	1510	480
Toaster (large conveyor)	720 slices/h	3200	—	850	750	1600	510
Toaster (small conveyor)	360 slices/h	2100	—	560	490	1050	340
Toaster (large pop-up)	10 slice	5300	—	2810	2490	5300	1700
Toaster (small pop-up)	4 slice	2470	—	1310	1160	2470	790
Waffle iron	0.05 m^2	1640	—	700	940	1640	520
(*b*) Electric, exhaust hood required							
Broiler (conveyor infrared), per m^2 of cooking area/minute	0.19–9.5 m^2	60 800	—	—	—	—	12 100
Broiler (single deck infrared), per m^2 of broiling area	0.24–0.91 m^2	34 200	—	—	—	—	6780
Charbroiler, per linear metre of cooking surface	0.6–2.4 m	10 600	8900	—	—	—	2700
Fryer (deep fat)	15–23 kg oil	14 000	850	—	—	—	350
Fryer (pressurised), per kg of fat capacity	6–15 kg	1010	—	—	—	—	38
Griddle, per metre length of cooking surface	0.6–2.4 m	18 800	3000	—	—	—	1350
Oven (full-size convection)		12 000	5000	—	—	—	850
Oven (large deck baking with 15.2 m^3 decks) per m^3 of oven space	0.43–1.3 m^3	17 300	—	—	—	—	710
Oven (roasting), per m^3 of oven space	0.22–0.66 m^3	28 300	—	—	—	—	1170
Oven (small convention), per m^3 of oven space	0.04–0.15 m^3	107 000	—	—	—	—	1520
Oven (small deck baking with 7.7 m^3 decks), per m^3 of oven space	0.22–0.66 m^3	28 700	—	—	—	—	1170
Open range (top), per 2 element section	2–10 elements	4100	1350	—	—	—	620
Range (hot top/fry top), per m^2 of cooking surface	0.36–0.74 m^2	22 900	—	—	—	—	8500
Range (oven section), per m^3 of space	0.12–0.32 m^3	40 600	—	—	—	—	1660
(*c*) Gas, no hood required							
Broiler, per m^2 of broiling area	0.25	46 600	190*	16 800	9030	25 830	3840
Cheese melter, per m^2 of cooking surface	0.23–0.47	32 500	190*	11 600	3400	15 000	2680
Dishwasher (hood type), chemical sanitising), per 100 dish/h	950–2000 dish/h	510	190*	150	59	209	67
Dishwasher (hood type, water sanitising), per 100 dish/h	950–2000 dish/h	510	190*	170	64	234	73
Dishwasher (conveyor type, chemical sanitising), per 100 dish/h	5000–9000 dish/h	400	190 *	97	21	118	38
Dishwasher (conveyor type, water sanitising), per 100 dish/h	5000–9000 dish/h	400	190*	110	23	133	41
Griddle/grill (large), per m^2 of cooking surface	0.43–1.1 m^2	53 600	1040	3600	1930	5530	1450
Griddle/grill (small), per m^2 of cooking surface	0.23–0.42 m^2	45 400	1040	3050	1610	4660	1260
Hot plate	2 burners	5630	390 *	3430	1020	4450	1000
Oven (pizza), per m^2 of hearth	0.59–1.2 m^2	14 900	190 *	1970	690	2660	270
(*d*) Gas, exhaust hood required							
Braising pan, per litre of capacity	102–133 litre	3050	190 *	—	—	—	750
Broiler, per m^2 of broiling area	0.34–0.36 m^3	68 900	1660	—	—	—	5690
Broiler (large conveyor, infrared), per m^2 of cooking area/minute	0.19–9.5 m^2	162 000	6270	—	—	—	16 900
Broiler (standard infrared), per m^2 of broiling area	0.22–0.87 m^2	61 300	1660	—	—	—	5040
Charbroiler (large), per metre length of cooking area	0.6–2.4 m	34 600	21 000	—	—	—	3650
Fryer (deep fat)	15–23 kg	23 500	1640	—	—	—	560
Oven (bake deck), per m^3 of oven space	0.15–0.46 m^3	79 400	190*	—	—	—	1450
Griddle, per metre length of cooling surface	0.6–2.4 m	24 000	6060	—	—	—	1540
Oven (full-size convection)		20 500	8600	—	—	—	1670
Oven (pizza), per m^2 of oven hearth	0.86–2.4 m^2	22 800	190*	—	—	—	410
Oven (roasting), per m^3 of oven space	0.26–0.79 m^3	44 500	190 *	—	—	—	800
Oven (twin bake deck), per m^3 of oven space	0.31–0.61 m^3	45 400	190 *	—	—	—	810

Table continues

Table 6.17 Typical rates of heat gain from restaurant and cooking equipment[1] — *continued*

Appliance	Size	Energy rate / W		Rate of heat gain / W			
		Rated	Standby	Without hood			With hood
				Sensible	Latent	Total	Latent
Range (burners), per 2 burner section	2–10 burners	9840	390	—	—	—	1930
Range (hot top or fry top), per m^2 of cooking surface	0.26–0.74 m^3	37 200	1040	—	—	—	10 700
Range (large stock pot)	3 burners	29 300	580	—	—	—	5740
Range (small stock pot)	2 burners	11 700	390	—	—	—	2290
Range top, open burner (per 2 element section)	2–6 elements	11 700	4000	—	—	—	640
(*e*) Steam							
Compartment steamer, per kg of food capacity/h	21–204 kg	180	—	14	9	23	7
Dishwasher (hood type, chemical sanitising), per 100 dish/h	950–2000 dish/h	920	—	260	110	370	120
Dishwasher (conveyor, water sanitising), per 100 dish/h	950–2000 dish/h	920	—	290	120	410	130
Dishwasher (conveyor, chemical sanitising), per 100 dish/h	5000–9000 dish/h	350	—	41	97	138	44
Dishwasher (conveyor, water sanitising), per 100 dish/h	5000–9000 dish/h	350	—	44	108	152	50
Steam kettle, per litre of capacity	12–30 litre	160	—	12	8	20	6

7 Moisture transfer and condensation

7.1 Introduction

This chapter discusses the factors that determine the risk of surface condensation and mould growth within buildings and the effect of interstitial condensation on the performance of building elements. It presents methods for the prediction of surface and interstitial condensation and guidelines on how to minimise these problems. Some of the basic physics of moisture movement and psychrometrics are also discussed.

Moisture gives rise to two types of problems:

- condensation, or more importantly, mould growth, on internal surfaces; as discussed in chapter 8, moulds and their spores are one of the most important causes of respiratory problems in buildings, especially in housing
- accumulation of moisture within a structure in areas where it may cause corrosion of metal components, decay of timber based components or reduction of the performance of insulants.

Minimisation of these problems depends on:

- good thermal design of the building fabric
- consideration of moisture production and ventilation within the building
- use of combinations of materials that allow for the possibility of storage and movement of moisture within the structure
- use of materials and detailing appropriate to the location and use of the building.

The effects of natural ventilation and HVAC systems on the humidity of the air within buildings are discussed in chapter 5.

7.2 Notation

The notation used in this section follows that in BS EN ISO 7345(1) and BS EN ISO 9346(2).

c_p	Specific heat capacity at constant pressure ($J{\cdot}kg^{-1}{\cdot}K^{-1}$)
c_{pa}	Specific heat capacity at constant pressure for air ($J{\cdot}kg^{-1}{\cdot}K^{-1}$)
d	Thickness (m)
f_{Rsi}	Surface temperature factor appropriate to value of inside surface resistance (—)
g	Mass flow rate per unit area ($kg{\cdot}m^{-2}{\cdot}s^{-1}$)
g_k	Mass flow rate per unit area through interface k ($kg{\cdot}m^{-2}{\cdot}s^{-1}$)
g_s	Rate of condensation at a surface ($kg{\cdot}m^{-2}{\cdot}s^{-1}$)
h_c	Convective heat transfer coefficient ($W{\cdot}m^{-2}{\cdot}K^{-1}$)
h_e	Specific latent heat of evaporation ($J{\cdot}kg^{-1}$)
n	Number of layers in an element
p	Vapour pressure (Pa)
p_k	Vapour pressure of interface k (Pa)
p_s	Saturation vapour pressure (Pa)
p_{ss}	Saturation vapour pressure at surface temperature (Pa)
p_t	Total pressure (Pa)
R	Thermal resistance ($m^2{\cdot}K{\cdot}W^{-1}$)
R_j	Thermal resistance of layer j ($m^2{\cdot}K{\cdot}W^{-1}$)
R'_k	Cumulative thermal resistance from inside to interface k ($m^2{\cdot}K{\cdot}W^{-1}$)
R_{si}	Internal surface resistance to heat transfer ($m^2{\cdot}K{\cdot}W^{-1}$)
R'_T	Total thermal resistance of component ($m^2{\cdot}K{\cdot}W^{-1}$)
r_v	Vapour resistivity ($MN{\cdot}s{\cdot}g^{-1}{\cdot}m^{-1}$)
R_w	Molar gas constant for water ($J{\cdot}K^{-1}$)
s_d	Water vapour diffusion-equivalent air layer thickness (m)
T	Temperature (K)
t	Time (h)
u	Moisture content of a material ($kg{\cdot}kg^{-1}$)
V	Volume (m^3)
v	Moisture content of air by volume ($g{\cdot}m^{-3}$)
x	Moisture content of air by mass ($g{\cdot}kg^{-1}$)
Z	Vapour resistance ($MN{\cdot}s{\cdot}g^{-1}$)
Z_j	Vapour resistance of layer j ($MN{\cdot}s{\cdot}g^{-1}$)
Z'_k	Cumulative vapour resistance from inside to interface k ($MN{\cdot}s{\cdot}g^{-1}$)
Z'_T	Total vapour resistance of component ($MN{\cdot}s{\cdot}g^{-1}$)
β_v	Surface mass transfer coefficient ($m{\cdot}s^{-1}$)
Δp_v	Vapour pressure difference (Pa)
δ_a	Vapour permeability of still air ($kg{\cdot}m^{-1}{\cdot}s^{-1}{\cdot}Pa^{-1}$)
δ_p	Vapour permeability of a material ($kg{\cdot}m^{-1}{\cdot}s^{-1}{\cdot}Pa^{-1}$)
θ_d	Dew-point temperature (°C)
θ_e	External air temperature (°C)
θ_i	Internal air temperature (°C)
θ_k	Temperature at interface k (°C)
θ_{si}	Internal surface temperature (°C)
μ	Water vapour resistance factor (—)
ρ_a	Density of air ($kg{\cdot}m^{-3}$)
ϕ_e	External relative humidity (%)
ϕ_i	Internal relative humidity (%)

7.3 Internal water vapour loads

In order to select the internal design conditions for condensation calculations, it is necessary to have some idea of the moisture content or vapour pressure of the air in a building. This will be determined mainly by the sources of moisture and the ventilation rate. Table 7.1 gives estimates for the amounts of moisture produced by various sources in housing.

Table 7.1 Sources of moisture within buildings

Source	Moisture produced
Combustion in room heaters/cookers without flues:	
— paraffin	0.1 kg·h^{-1}·kW^{-1}
— natural gas	0.16 kg·h^{-1}·kW^{-1}
— butane	0.12 kg·h^{-1}·kW^{-1}
— propane	0.13 kg·h^{-1}·kW^{-1}
Household activities:	
— cooking (3 meals)	0.9–3.0 kg·day^{-1}
— dish washing (3 meals)	0.15–0.45 kg·day^{-1}
— clothes washing	0.5–1.8 kg·day^{-1}
— clothes drying (indoors)	2–5 kg·day^{-1}
— baths/showers	0.2–0.5 kg·person^{-1}·day^{-1}
— floor washing	0.5–1.0 kg per 10 m^{-2}
— indoor plants	0.02–0.05 kg·plant^{-1}·day^{-1}
Perspiration and respiration of building occupants	0.04–0.06 kg·h^{-1}·person^{-1}

BS 5250[3] suggests a typical daily moisture production rate of 6 kg for a five person family but clothes washing and the use of moisture-producing (i.e. non-electric) room heaters can increase this to 15 kg. The instantaneous moisture production will vary with the activities, e.g. a maximum will usually occur during cooking and clothes washing.

Industrial buildings present special problems due to the rate of production of moisture by some processes. The building services engineer should discuss the proposed use of the building with the client to enable any likely problems to be anticipated. For example, in the textiles industry it is estimated that about half a kilogram of water vapour is produced for each kilogram of wool that is scoured, dyed and washed.

Swimming pools have particularly high internal moisture loads because of the high air temperatures for the comfort of users, and the large exposed surface of heated water.

Animal houses need special consideration since chickens produce about 0.003 kg·h^{-1} (per bird) of moisture, sheep produce about 0.04 kg·h^{-1} (per animal) and pigs about 0.15 kg·h^{-1} (per animal).

7.4 Moisture content of materials

Most materials will take up water when exposed to moist air, the equilibrium quantity depending on the nature of the material, its pore structure and the relative humidity of the air. This phenomenon is important when assessing the thermal conductivity of building and insulating materials, see chapter 3, Appendix 3.A1. Equilibrium moisture contents for various materials are given in Table 7.2. More complete information is given elsewhere[4,5].

The moisture absorption is largely, though not solely, due to capillary forces. The vapour pressure over a concave surface is less than that over a plane surface. Water will condense on any surface having a radius of curvature such that the corresponding vapour pressure is less than that in the ambient air. If the radius is sufficiently small, condensation will occur from unsaturated atmospheres.

Table 7.2 Equilibrium moisture content of materials

Material	Density / kg·m^{-3}	Moisture content at ambient air relative humidity of 50% / % by mass
Brick	1600	0.4
	2000	0.3
Cellular concrete	230	3
	640	4
Concrete	1200	4
	2300	1.5
Cement mortar	2000	2.5
Plaster:		
— lime sand	1750	1
— cement sand	2000	1
Limestone	2700	0.2
Sandstone	1800	2
Cork	200	5
Expanded Polystyrene	30	4
Glasswool slab	30	0.5
Mineral wool	30	0.5
Urethane foam	25	7
Hardwood	750	10
Softwood	420	10
Plywood	600	10
Strawboard	—	10
Woodwool/cement slab	360	10

Note: for newly constructed buildings, moisture contents will be higher than stated until 'drying-out' is completed

Table 7.3 Pore radius for hygroscopic equilibrium

Radius of curvature of pore / nm	Ambient air relative humidity for equilibrium / %
2.1	60
5	80
10	90
100	99

Table 7.3 shows the relative humidity of the ambient air that is in equilibrium with a concave surface.

If the relative humidity is greater than that given in Table 7.3, water will condense on the surface. For example, a dry material containing pores of radius 2.1 nm will take up water from an atmosphere with a relative humidity of 60% and these, and any smaller pores, will become filled with water. This process can contribute to the movement of water vapour through building materials. If the opposite face of the material is exposed to an atmosphere with a relative humidity of 40%, moisture will evaporate from the pore menisci and a state of dynamic equilibrium is set up whereby water condenses at one side, moves under capillary forces through the material to the other side and there evaporates.

The traditional picture of interstitial condensation, in which water vapour diffuses through, but does not interact with, a material until it reaches an area where the local temperature and vapour pressure combine to give a relative humidity of 100%, can be seen to be a simplification. However, as discussed below, it still provides a practical method of risk assessment in many cases.

7.5 Mechanisms of moisture movement

7.5.1 Surface moisture transfer

The moisture mass transfer rate at a surface is derived from Fick's diffusion law[6] which can be written as:

$$g_s = \frac{(p_a - p_{ss})\,\beta_v}{R_w\,T} \qquad (7.1)$$

where g_s is the mass flow rate per unit area of moisture ($kg{\cdot}m^{-2}{\cdot}s^{-1}$) (condensation if positive, evaporation if negative), p_a is the vapour pressure of the water vapour in the air (Pa), p_{ss} is the saturation vapour pressure at the surface temperature (Pa), β_v is the surface mass transfer coefficient ($m{\cdot}s^{-1}$), R_w is the molar gas constant for water (= 461) ($J{\cdot}kg^{-1}{\cdot}K^{-1}$) and T is the temperature (K).

Mass transfer is analogous to heat transfer and the surface mass transfer coefficient is numerically related to the convective heat transfer coefficient by the Lewis relation[6]:

$$\beta_v = h_c / \rho_a\, c_{pa} \qquad (7.2)$$

where, h_c is the convective heat transfer coefficient ($W{\cdot}m^{-2}{\cdot}K^{-1}$), ρ_a is the density of air ($kg{\cdot}m^{-3}$) and c_{pa} is the specific heat capacity of air ($J{\cdot}kg^{-1}{\cdot}K^{-1}$).

Values of the convective heat transfer coefficient can be derived from the appropriate expressions given in chapter 3 of CIBSE Guide C[7]. Some common values are given in Table 7.4.

Table 7.4 Values of convective heat transfer and surface mass transfer coefficients

Direction of heat flow	Convective heat transfer coefficient, h_c / $W{\cdot}m^{-2}{\cdot}K^{-1}$	Surface mass transfer coefficient, β_v / $m{\cdot}s^{-1}$
Downward	1.5	1.25×10^{-3}
Horizontal	3.0	2.5×10^{-3}
Upward	4.3	3.6×10^{-3}

7.5.2 Diffusion

Diffusion is the movement of molecules from high to low concentration. Most solid materials permit the diffusion of water vapour to some extent and, whenever there is a difference in the vapour pressure across the material, a movement of water takes place. This is analogous to the flow of heat through a material when subjected to a temperature difference and this similarity is exploited in the calculation methods described.

Under steady state conditions, the rate of mass transfer per unit area through an element of a given material is given by:

$$g = \Delta p_v / Z \qquad (7.3)$$

where g is the mass flow rate per unit area ($kg{\cdot}m^{-2}{\cdot}s^{-1}$), Δp_v is the vapour pressure difference (Pa) and Z is the vapour resistance in terms of vapour pressure ($m^2{\cdot}s{\cdot}Pa{\cdot}kg^{-1}$) .

The vapour resistance of a sample of a given material is defined by:

$$Z = r_v\, d \qquad (7.4)$$

where r_v is the vapour resistivity ($m{\cdot}s{\cdot}Pa{\cdot}kg^{-1}$) and d is the thickness of the element (m).

Vapour resistivity is the reciprocal of vapour permeability, δ_p ($kg{\cdot}m^{-1}{\cdot}s^{-1}{\cdot}Pa^{-1}$).

To provide more manageable numbers, vapour resistivity is normally quoted in $MN{\cdot}s{\cdot}g^{-1}{\cdot}m^{-1}$ and resistance in $MN{\cdot}s{\cdot}g^{-1}$. Values for common materials are quoted in Tables 3.43 to 3.46 of chapter 3 of this Guide and in Appendix C of BS 5250[3].

European standards (including BS EN ISO 13788[8], the standard for condensation calculations) and manufacturers' data sheets commonly quote vapour resistivities or resistances in the form of the dimensionless water vapour resistance factor, μ, or the equivalent air layer thickness, s_d, in metres. These are defined by :

$$\mu = \delta_a / \delta_p \qquad (7.5)$$

and:

$$s_d = \mu\, d \qquad (7.6)$$

where δ_a is the vapour permeability of still air ($kg{\cdot}m^{-1}{\cdot}s^{-1}{\cdot}Pa^{-1}$), δ_p is the vapour permeability of the material ($kg{\cdot}m^{-1}{\cdot}s^{-1}{\cdot}Pa^{-1}$) and d is the thickness of a sample of material (m).

Values of μ and s_d are tabulated in BS EN 12524[9].

The permeability of air, δ_a, varies with temperature and atmospheric pressure (further details are given in BS EN ISO 12572[10]); however a value of 0.2 $g{\cdot}m{\cdot}MN^{-1}{\cdot}s^{-1}$ should be taken as typical of UK conditions. Therefore to convert a μ-value to a vapour resistivity in the units given in chapter 3 of this Guide and Appendix C of BS 5250 ($MN{\cdot}s{\cdot}g^{-1}{\cdot}m^{-1}$), divide by 0.2.

Similarly, to convert an s_d value in metres into a vapour resistance in the units given in chapter 3 of this Guide and Appendix C of BS 5250 ($MN{\cdot}s{\cdot}g^{-1}$), divide by 0.2.

Table 7.5 gives an indication of the likely resistivities of fibrous and open-celled materials and may be used in the absence of data for specific materials. Table 7.5 also gives a value for the vapour resistivity of still air in cavities within composite structures. However, as there will always be some degree of air movement within cavities, driven by pressure or temperature differences, it is conventional to assume that cavity vapour resistivity is zero in calculations of interstitial condensation risk.

Vapour control layers are usually thin materials and it is more convenient to classify them by their vapour resistance than by their thickness and vapour resistivity. Table 7.6 gives approximate values of the vapour resistances of various membranes. It should be noted that

Table 7.5 Approximate values of vapour resistivity for fibrous or open-celled materials and for air spaces within structures

Density / $kg \cdot m^{-3}$	Vapour resistivity / $MN \cdot s \cdot g^{-1} \cdot m^{-1}$
Air space	5
600	20
800	30
1000	40
1500	100
2000	220
2500	520

Table 7.6 Vapour resistance of membranes

Material	Thickness / mm	Vapour resistance / $MN \cdot s \cdot g^{-1}$
Polythene film	0.05	125
	0.1	200
	0.15	350
Mylar film	0.025	25
Gloss paint (average)	—	15
Interior paint	—	1
Varnish (phenolic, epoxy, polyurethane)	0.05	5
Roofing felt	—	400–1000
Aluminium foil	—	4000

these values apply to undamaged membranes and the presence of any perforations may reduce the vapour resistance considerably.

As discussed in section 7.4, many materials are porous and absorb liquid water from the air at a rate that depends on the ambient relative humidity; this water then moves through the structure under relative humidity gradients. This process can be represented as an equivalent vapour diffusion, with a vapour permeability that increases with relative humidity, with a rapid increase above 80% relative humidity. To reflect this, it is common to quote both 'dry cup' and 'wet cup' values of vapour resistivity or permeability, which are appropriate to mean relative humidities across the material below or above 80% respectively. In practice, it is safe to use the 'wet cup' value for materials between the insulation layer and the outside air and the 'dry cup' value for materials between the insulation layer and the inside air.

7.5.3 Air movement

Moisture is transferred by air movement through gaps at the junctions between elements of the construction and through cracks within the elements. In a typical masonry wall with windows or other openings, the mass flow of moisture due to air movement through gaps can be as much as an order of magnitude greater than that produced by diffusion. This is especially true in the case of pitched roofs, where the moisture transfers are dominated by wind and stack driven air flows from the house into the loft, through gaps in the ceiling and from the loft to outside via installed ventilators and laps in the undertiling membrane. Risk assessments that do not take account of these flows are not reliable. Air flows also dominate moisture transports though metal cladding and roofing systems.

Certain buildings, such as operating theatres and clean rooms are deliberately operated at an overpressure to minimise ingress of contaminants. These are especially vulnerable to severe interstitial condensation caused by air infiltrating the structure.

7.6 Surface condensation and mould growth

7.6.1 Psychrometry of condensation of water vapour

The vapour pressures and temperatures at which water and air are in equilibrium are uniquely related by the saturation line, which applies whether the water vapour is present on its own or mixed with air. The equilibrium condition can be thought of in two ways:

(*a*) when the temperature of the air equals the dew-point temperature corresponding to the partial pressure of the water vapour in the mixture

(*b*) when the partial pressure of the water vapour equals the saturation pressure corresponding to the temperature of the mixture.

The first is useful when considering surface or superficial condensation on surfaces cooler than the room or ambient air. The second is more appropriate when considering internal or interstitial condensation within a building construction through which water vapour is moving under the influence of a difference between internal and external partial pressures.

The conventional psychrometric relationships are given in chapter 1 of Guide C[7]. The following relationships between temperature and saturated vapour pressure can be used for the calculation of surface and interstitial condensation risk at the temperatures commonly found in buildings.

For $\theta \geq 0$:

$$p_s = 610.5 \exp\left(\frac{17.269 \times \theta}{237.3 + \theta}\right) \tag{7.7}$$

For $\theta < 0$:

$$p_s = 610.5 \exp\left(\frac{21.875 \times \theta}{265.5 + \theta}\right) \tag{7.8}$$

where θ is the temperature (°C) and p_s is the saturated vapour pressure (Pa).

The equation for $\theta < 0$ gives the saturation vapour pressure values over ice.

These equations are consistent with BS 5250[3] and BS EN ISO 13788[8] and may be usefully inverted to calculate dew-point temperature, θ_d, from the vapour pressure in Pa.

For $p \geq 610.5$ Pa:

$$\theta_d = \frac{237.3 \ln\left(\frac{p}{610.5}\right)}{17.269 - \ln\left(\frac{p}{610.5}\right)} \tag{7.9}$$

For $p < 610.5$ Pa:

$$\theta_d = \frac{265.5 \ln\left(\frac{p}{610.5}\right)}{21.875 - \ln\left(\frac{p}{610.5}\right)} \tag{7.10}$$

where θ_d is the dew-point temperature (°C) and p is the vapour pressure (Pa).

The moisture content of the air by volume, v (g·m^{-3}), may be calculated from the vapour pressure using:

$$v = 2.17\, p / (\theta + 273.3) \tag{7.11}$$

where v is the moisture content of the air (g·m^{-3}), p is the vapour pressure (Pa) and θ is the temperature (°C).

The moisture content of the air by mass, x (g·kg^{-1} of dry air), may be calculated from the vapour pressure using:

$$x = 0.622\, p / (P - p) \tag{7.12}$$

where x is the moisture content of the air (g·kg^{-1} of dry air) and P is the total atmospheric pressure (Pa).

7.6.2 Surface condensation

Condensate frequently occurs on:

— single glazing in bedrooms overnight or in kitchens and bathrooms at any time

— double glazing, especially near to the frames, in rooms with high humidities

— on WC cisterns or cold pipes in bathrooms or kitchens

— on the walls of hallways and stairs in buildings of heavy masonry construction after a change from cold, dry weather to mild, wet weather

— on massive floors in offices or industrial buildings, which remain cold after a change to warmer, more humid weather, or when heating is turned on in the morning.

Condensate is often only a nuisance. However, more serious consequences can result from, for example:

— condensate from glazing promoting decay in the wooden window frames or condensate running from sills onto the wall below, damaging the décor

— condensate dripping from roofs onto food preparation processes or sensitive electronic equipment

— condensate on certain floor types, leading to a slip hazard.

It is sometimes possible to deal with the condensate by drainage or by mopping up before it collects and runs to vulnerable areas. However, persistent, severe condensation on glazing, especially double glazing, in many rooms, suggests that there may be excessive moisture production or inadequate ventilation within the dwelling; this may lead to the more serious problems.

7.6.3 Mould growth

Mould growth is a source of health problems within buildings, increasing the incidence of asthma and other respiratory allergies. Mould spores exist in large numbers in the atmosphere and the critical factor for their germination and growth is the moisture conditions at surfaces and the length of time these conditions exist. Studies have shown that moulds can germinate and grow under steady state conditions if the relative humidity at a surface is above 80%[11]. In reality, the relative humidity within houses and at wall surfaces fluctuates greatly, especially in rooms, such as kitchens and bathrooms, where moisture production is intermittent. Little information is available concerning the growth of moulds under these circumstances, however it can be assumed that, if the surface relative humidity is over 80% for less than three hours a day, mould growth is unlikely and if it is over 80% for more than six hours a day, mould growth is very likely.

The surface relative humidity criterion of 80% for mould growth imposes a considerably more severe constraint on the thermal design of the building fabric than the 100% RH required for surface condensation. Table 7.7 shows the surface temperatures that must be achieved to avoid condensation and mould growth with an internal temperature of 20 °C and a range of internal relative humidities.

Table 7.7 Surface temperatures necessary to avoid condensation and mould assuming an internal air temperature of 20 °C and various internal relative humidities

Internal relative humidity / %	Surface temp. to avoid condensation / °C	Surface temp to avoid mould / °C
40	6.0	9.3
50	9.3	12.6
60	12.0	15.4
70	14.4	17.9

In the winter, the internal surfaces of external walls will be colder than the air temperature within buildings, the relative humidity at the wall will, therefore, be about 10% higher than in the centre of a room. This temperature and relative humidity difference will be reduced if the walls are well insulated or greater at thermal bridges, see section 7.6.4.

A further consideration is the thermal mass of the wall. The surface temperature of a lightweight construction, or a wall with internal insulation will respond rapidly to changes in internal air temperature, limiting the rise of internal surface relative humidity on the occasions when there are simultaneous inputs of heat and water vapour. Conversely a massive masonry wall, with external insulation will respond very slowly to changes in temperature, increasing the risks of mould growth in houses that are intermittently heated.

As a guide, however, it can often be assumed that, if the average relative humidity within a room stays at above 70% for a long period of time, the relative humidity at the external wall surfaces will be high enough to support the growth of moulds.

7.6.4 Thermal bridges

Thermal bridges are areas of the building fabric where, because of the presence of high conductivity materials or the geometry of the detail, there is significantly higher heat loss than through surrounding areas. Besides leading to increased energy use, they lower the internal surface temperature and are therefore sites for condensation and mould growth. Thermal bridges fall into two categories.

(a) *Repeating thermal bridges*: e.g. timber joists, mortar joints, mullions in curtain walling. These have a significant effect on heat loss, and are required to be taken into account in the calculation of *U*-values, using the methods specified in chapter 3. They are, however, rarely severe enough to cause surface temperatures to fall low enough to cause surface condensation or mould growth.

(b) *Non-repeating bridges*: which commonly occur around openings such as lintels, jambs and sills and at wall–roof junctions, wall–floor junctions and where internal walls or floors penetrate the outer building fabric. If details to minimise thermal bridges are not used, they can add 10–15% to the total heat loss from the building besides causing condensation and mould.

The severity of a thermal bridge, in terms of its effect on internal surface temperatures may be expressed by the surface temperature factor, f_{Rsi}, defined under steady state conditions by:

$$f_{Rsi} = \frac{(\theta_{si} - \theta_e)}{(\theta_i - \theta_e)} \tag{7.13}$$

where f_{Rsi} is the surface temperature factor, θ_{si} is the internal surface temperature (°C), θ_i is the internal air temperature (°C) and θ_e is the external air temperature (°C).

For plane areas, where one dimensional heat flow may be assumed:

$$f_{Rsi} = 1 - U\,R_{si} \tag{7.14}$$

where U is the U-value of the element ($W \cdot m^{-2} \cdot K^{-1}$) and R_{si} is the internal surface heat transfer coefficient ($m^2 \cdot K \cdot W^{-1}$).

The temperature factor will be close to 1.0 for a well insulated structure, but will fall towards 0.5 or less at severe thermal bridges. As it depends on the properties of the construction detail alone and is independent of the environmental conditions, once the temperature factor has been found for a thermal bridge, it may be used to calculate the internal surface temperature from any set of environmental temperatures, see section 7.7.1 (equation 7.15).

This may then be used to calculate the internal surface relative humidity and risk of mould growth, if the internal humidity is known.

The calculated surface temperature, and therefore the temperature factor, f_{si}, is very sensitive to the assumed value of the internal surface heat transfer coefficient, R_{si}. (The subscript 'Rsi' is added to the *f*-value, to emphasise this.) Therefore, it is important that the appropriate value for R_{si} be used at all stages. Because problems commonly occur in corners or behind furniture, where heat transfer from the room air to the wall surface will be restricted, BS EN ISO 13788[8] recommends using a relatively high value for R_{si} of 0.25 $m^2 \cdot K \cdot W^{-1}$ for calculating the risk of surface condensation and mould growth.

Table 7.8 shows the critical values of surface temperature factor that should be achieved to avoid surface condensation or mould growth in building with different occupancies.

The surface temperature factor of a thermal bridge may be found from thermal bridge catalogues[12,13] or by calculation, using software that complies with BS EN ISO 10211-1[14].

Table 7.8 Critical values of surface temperature factor to avoid problems of condensation and mould in buildings of different occupancy

Problem	Critical value of surface temperature factor, f_{Rsi}
Surface condensation in:	
— storage buildings	0.30
— offices, retail premises	0.50
— sports halls, kitchens, canteens and buildings with unflued gas heaters	0.80
Buildings with high internal humidity, such as swimming pools, laundries and breweries	0.90
Mould growth in:	
— dwellings, residential buildings and schools	0.75
— swimming pools, including pools in housing	0.90

7.6.5 Interstitial condensation

As has been noted above (section 7.4) the interactions between building materials and moisture are complex, with absorption of liquid water into pores and subsequent movement under relative humidity gradients. This means that calculations of the effect of moisture on structures can be extremely complex. In many cases however it is possible to carry out useful assessments by assuming that moisture transfer is purely by vapour diffusion, until 'interstitial condensation' occurs in areas where the local relative humidity is equal to 100%.

A building element will have a temperature gradient and a vapour pressure gradient across it due to the differing conditions on either side. In some cases these gradients may be zero but more usually, particularly for external elements in the winter, they are non-zero. Intermediate values of vapour pressure and temperature can be calculated inside the structure. Interstitial condensation, i.e. free liquid water, will be predicted if the calculated vapour pressure at any point is greater than the saturated vapour pressure corresponding to the calculated temperature at that point.

The effect of interstitial condensation can be inconsequential; for example, the amount of condensate that occurs on the outer leaf of a masonry cavity wall is usually small compared to the effect of wetting by rain; condensation regularly occurs on the underside of the outer sheet of metal roofs overnight and evaporates again during the day. However condensation can cause:

— decay of timber leading to structural failure
— corrosion of metal coverings and components
— reduction in the thermal performance of insulation
— dimensional changes
— migration of salts and liberation of chemicals
— electrical failure can also result
— staining of internal decoration and damage to equipment, if it drips into the building.

In the UK climate, problems of interstitial condensation have usually occurred in the winter, when the air temperature and vapour pressure are higher inside buildings than outside. However in warm humid climates and increasingly in the UK in the summer, the use of air conditioning can reverse this situation and lead to severe problems in buildings designed for cold climates with, for example, impermeable vapour control layers on the inside of the insulation. Calculations of interstitial condensation risk and design of structures should always take account of the direction of the vapour pressure gradient.

7.7 Inside and outside design conditions

Inside and outside conditions are chosen to suit the purpose of the analysis, bearing in mind that the simpler calculation methods assume steady state conditions. Usually the purpose is to determine either the long term build-up of condensation within the thickness of a construction or the short term rate of condensation on one of its exposed surfaces. Relatively less extreme conditions will be more appropriate for the former purpose and relatively more extreme for the latter.

7.7.1 Design conditions to avoid mould growth

The risk of mould growth on the internal surfaces of buildings depends on the combination of the internal surface temperature of external walls and the internal humidity. Because of the thermal inertia of structures and the time moulds take to germinate and grow, monthly mean conditions are often felt to be a sufficiently accurate predictor.

The surface temperature, θ_{si}, depends on the internal air temperature, θ_i, the thermal properties of the structure, summarised as f_{si}, and the external temperature, θ_e.

$$\theta_{si} = \theta_e + f_{Rsi}\,(\theta_i - \theta_e) \qquad (7.15)$$

If the internal temperature is kept more or less constant in a heated building, the lowest monthly mean internal surface temperatures will occur in the month with the coldest external temperature.

The internal vapour pressure depends on the external vapour pressure and the vapour pressure excess, Δp, which depends on a combination of moisture production within the building, the building volume and the ventilation rate:

$$p_i = p_e + \Delta p \qquad (7.16)$$

where p_i is the internal vapour pressure (Pa), p_e is the external vapour pressure (Pa) and Δp is the vapour pressure excess (Pa).

BS EN ISO 13788[8] contains a method for categorising buildings into ‘climate classes’ depending on their likely vapour pressure excess. The fact that ventilation rates vary as more windows are opened in warmer weather is allowed for by assuming that (*a*) the vapour pressure excess is constant below 0 °C as all windows are closed, and (*b*) the vapour pressure excess falls linearly to zero at an external temperature of 20 °C, when the building is assumed to be well ventilated. This gives the boundaries between the classes shown in Figure 7.1, which also shows the internal relative humidity at an outdoor temperature of 0 °C and an indoor temperature of 20 °C.

Typical buildings in each class are summarised in Table 7.9.

In the case of air conditioned buildings in which the internal humidity is controlled independently of the external environment the set values of the temperature and relative humidity should be used to calculate the internal moisture load.

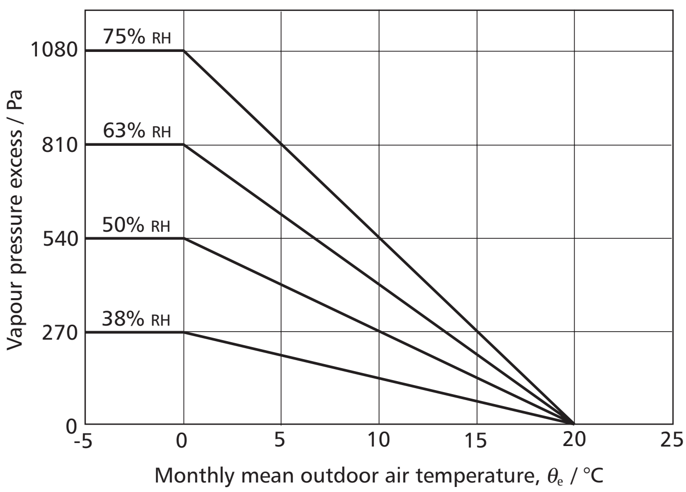


Figure 7.1 Variation of internal humidity classes with external temperature

Table 7.9 Typical buildings in each humidity class

Humidity class	Building
1	Storage areas
2	Offices, shops
3	Dwellings with low occupancy
4	Dwellings with high occupancy, sports halls, kitchens, canteens; buildings heated with un-flued gas heaters
5	Special buildings, e.g. laundry, brewery, swimming pool

The complex relationship between inside vapour pressure and outside temperature that results from the use of climate classes, means that the highest surface humidities, and therefore risk of mould growth, may not occur in the coldest months of the year. This is in accord with the common experience that mould is often worse in November or December when external humidities are high, than in the colder but drier conditions in January or February.

7.7.2 Design conditions to avoid condensation on windows and their frames

Windows and their frames or other similar components with little thermal inertia, respond very rapidly to changes in outside temperature. The daily outside minimum temperature should therefore be used to calculate the risk of condensation. Table 7.10 shows the temperatures that the daily minimum falls below for different numbers of days per year at London, Manchester and Edinburgh. These can be used for design depending on the acceptable frequency of the occurrence of condensation.

Similar values can be derived from the available records for other locations.

Table 7.10 Temperature that the daily minimum temperature falls below on various numbers of occasions per year (1983–2002)

No. of occasions	Temperature below which minimum daily temperature falls for stated number of occasions		
	London (Heathrow)	Manchester (Ringway)	Edinburgh (Turnhouse)
Minimum recorded temperature	–9.6	–11.1	–15.0
1 day/year	–6.1	–6.4	–9.3
2 days/year	–4.7	–5.0	–7.9
5 days/year	–3.4	–3.8	–5.7
10 days/year	–2.1	–2.6	–4.0
20 days/year	–0.6	–1.2	–2.5

7.7.3 Design conditions to avoid interstitial condensation

The method for interstitial condensation assessment described in BS EN ISO 13788[8], uses as external conditions the monthly mean temperature and vapour pressure over a full year. Internal conditions are derived from the appropriate climate class of the building as described in section 7.7.1.

Monthly mean temperature data are readily available for many locations; however vapour pressure data are much more difficult to obtain. The only useful source is the CD-ROM: *International Station Meteorological Climate Summary, Version 3.0*, available from the US National Climate Data Centre*, which contains information for 43 UK stations and many more around the world. Table 7.11 summarises the mean temperatures and relative humidities, calculated from the mean temperature and vapour pressure, for London, Manchester and Edinburgh.

* www.ncdc.noaa.gov/oa/ncdc.html

Table 7.11 Monthly mean temperature and relative humidity for interstitial condensation calculations

Month	London (Heathrow)		Manchester (Ringway)		Edinburgh (Turnhouse)	
	Temp. / °C	RH / %	Temp. / °C	RH / %	Temp. / °C	RH / %
Jan	4.9	84	4.2	83	3.5	83
Feb	4.7	82	4.1	80	3.7	81
Mar	6.9	77	5.8	76	5.3	78
Apr	8.8	71	7.8	71	7.0	75
May	12.6	69	11.3	68	9.9	75
Jun	15.7	69	14.1	71	12.8	75
Jul	17.9	68	16.1	72	14.7	76
Aug	17.6	70	15.8	74	14.4	78
Sep	14.9	75	13.3	77	12.1	80
Oct	11.2	81	10.3	81	9.2	82
Nov	7.6	84	6.7	82	5.8	83
Dec	5.9	86	5.2	84	4.3	84

The data in Table 7.11 are derived from long-term means and can be used for calculating parameters such as the long-term energy performance of a building, but are less appropriate for calculating the potential damage due to condensation. Any construction that just passes the BS EN ISO 13788[8] criteria, using these external climates, will fail in half the years. It is more satisfactory to use the climate, which is more severe in condensation risk terms, that may recur once in *N* years, where *N* is a number appropriate to the likely consequences of condensation occurring. In most buildings a once in ten-years risk might be appropriate; however, in particularly sensitive buildings such as computer centres etc., a once in 50-years risk might be more appropriate.

Condensation risk years with various return periods can be constructed by changing the monthly temperatures and relative humidities from a mean year with the corrections shown in Table 7.12.

Table 7.12 Corrections to monthly mean temperatures and relative humidities from a mean year to achieve condensation risk years with various return periods

Risk	Temp. correction / °C	RH correction / %
1 in 5	–1	+2
1 in 10	–1	+4
1 in 20	–2	+4
1 in 50	–4	+6

At present there are no standard methods for transforming data measured at airfield locations to buildings in city centres or in distant locations or at different altitudes. The best possible method at present is to define a number of regions similar, to the degree-day regions, and specify a condensation risk year for each, with perhaps a correction for altitudes above 100 m.

Condensation within ground floors is relatively unusual, however there is a risk in some constructions types and cases of floors collapsing due to condensation have occurred. The appropriate external conditions to use depend on the type of floor. If an insulated suspended floor with a ventilated subfloor is to be assessed, the temperature and humidity within the subfloor should be used as the external conditions. The monthly temperature in the subfloor can be estimated by adding the values

Table 7.13 Values to be added to the annual mean external air temperature to estimate the monthly temperature in a ventilated subfloor below an insulated floor.

Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
2.3	2.3	2.5	2.7	3.0	3.3	3.4	3.4	3.2	2.9	2.5	2.3

shown in Table 7.13 to the annual mean external air temperature. The relative humidity should be taken as 95% throughout the year.

The condensation risk in the centre of a solid ground bearing floor may be assessed by including 2 metres of soil with thermal conductivity 2 $W \cdot m^{-1} \cdot K^{-1}$ below the floor, and taking the external conditions as constant at the annual mean external air temperature and 99% RH. However, the most severe risk of condensation will be adjacent to an external corner where the temperature below the insulation in winter will be considerably lower than that in the centre. In this case the most appropriate calculation is to take a path such as that shown in Figure 7.2, down though the floor to below the insulation layer and then horizontally out through the wall, with the normal external temperatures and humidities as the design conditions.

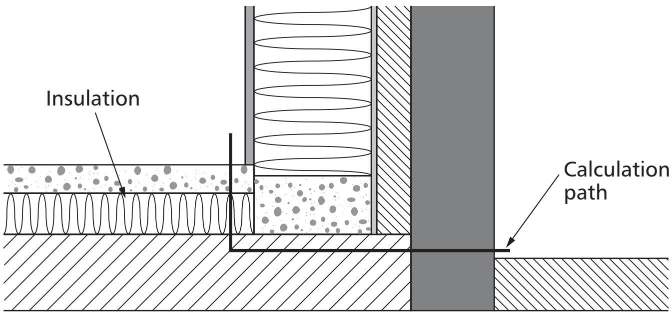


Figure 7.2 Recommended calculation path for assessment of condensation risk at the edge of a solid ground floor

7.8 Condensation calculations

7.8.1 Calculations of the risk of surface condensation and mould growth

BS EN ISO 13788[(8)] specifies a procedure for design of structures to avoid mould growth, surface condensation or corrosion, where relevant. The principal steps in the design procedure are to determine the internal air humidity and then, based on the required relative humidity at the surface, to calculate the acceptable saturation vapour pressure, p_s, at the surface. From this value, a minimum surface temperature and hence a required 'thermal quality' of the building envelope (expressed by f_{Rsi}), is established.

For each month of the year, perform the following steps:

(*a*) Define the external air temperature and vapour pressure as discussed in section 7.7.1.

(*b*) Define the internal temperature according to the use of the building.

(*c*) Calculate the internal vapour pressure, p_i, from p_e and the vapour pressure excess, Δp, appropriate to the use of the building (see section 7.7.1) or from a constant relative humidity for a conditioned environment.

(*d*) Calculate the minimum acceptable saturation vapour pressure at the internal surface, $p_s(\theta_{si})$:

$$p_s(\theta_{si}) = p_i / 0.8 \tag{7.17}$$

where $p_s(\theta_{si})$ is the minimum acceptable saturation vapour pressure at the internal surface and p_i is the internal vapour pressure (Pa).

The factor 0.8 applies to the risk of mould growth at 80% relative humidity. Other factors, such as 1.0 for surface condensation or 0.6 for corrosion, can be used where appropriate.

(*e*) Determine the minimum acceptable surface temperature, $(\theta_{si})_{min}$, from the minimum acceptable saturation vapour pressure.

(*f*) From the minimum acceptable surface temperature, $(\theta_{si})_{min}$, assumed internal air temperature, θ_i, and external temperature, θ_e, calculate the minimum temperature factor, $(f_{Rsi})_{min}$, using equation 7.13.

The month with the highest required value of $(f_{Rsi})_{min}$ is the critical month. The temperature factor for this month is $(f_{Rsi})_{max}$, and the building element must be designed so that $(f_{Rsi})_{max}$ is always exceeded; i.e. $f_{Rsi} > (f_{Rsi})_{max}$.

Table 7.14 shows an example of this process with the vapour pressure excess Δp taken from the line between climate classes 3 and 4 in Figure 7.1. The largest value of $(f_{Rsi})_{min}$ is 0.714 and occurs in November. This defines the lowest acceptable thermal quality of the building envelope to avoid mould growth.

Table 7.14 Example calculation of the minimum value of f_{si} necessary to avoid mould growth

Month	θ_i / °C	θ_e / °C	p_e / Pa	Δp / Pa	p_i / Pa	$p_s(\theta_{si})$ / Pa	θ_{si} / °C	$(f_{Rsi})_{min}$
January	20	2.8	660	697	1357	1696	14.9	0.705
February	20	2.8	657	697	1354	1692	14.9	0.703
March	20	4.5	709	628	1337	1671	14.7	0.658
April	20	6.7	788	539	1327	1658	14.6	0.592
May	20	9.8	941	413	1354	1693	14.9	0.499
June	20	12.6	1162	300	1462	1827	16.1	0.471
July	20	14.0	1302	243	1545	1931	17.0	0.493
August	20	13.7	1317	255	1572	1965	17.2	0.560
September	20	11.5	1183	344	1527	1909	16.8	0.620
October	20	9.0	1017	446	1463	1828	16.1	0.645
November	20	5.0	820	608	1428	1784	15.7	0.714
December	20	3.5	719	668	1387	1734	15.3	0.713

7.8.2 Calculations of the risk of interstitial condensation

7.8.2.1 Principle of the method

BS EN ISO 13788[8] contains a method for establishing the annual moisture balance and calculating the maximum amount of accumulated moisture due to interstitial condensation within a structural element. The method should be regarded as an assessment tool, suitable for comparing different constructions and assessing the effects of modifications rather than as an accurate prediction method. It does not provide a realistic assessment of moisture conditions within the structure under service conditions, and is not suitable for calculation of drying out of built-in moisture.

Starting with the first month in which any condensation is predicted, the monthly mean external conditions are used to calculate the amount of condensation or evaporation in each of the twelve months of a year. The accumulated mass of condensed water at the end of those months when condensation has occurred is compared with the total evaporation during the rest of the year.

One-dimensional, steady state conditions are assumed. Air movements through or within the building elements are not considered. In building elements where there is air flow through or within the element, the calculated results can be very unreliable and great caution should be used when interpreting the results.

7.8.2.2 Starting month

Starting with any month of the year (the trial month), calculate the temperature, saturated vapour pressure and vapour distributions through the component as specified below. Determine whether any condensation is predicted.

If no condensation is predicted in the trial month, repeat the calculation with successive following months until either:

(*a*) no condensation has been found in any of the twelve months, then report the component as free from condensation, or

(*b*) a month is found with condensation; this is the starting month.

If condensation is predicted in the trial month, repeat the calculation with successively earlier months until either:

(*a*) condensation is predicted in all twelve months; then, starting in any month, calculate the total annual accumulation of condensation as specified below, or

(*b*) a month is found with no condensation; then take the following month as the starting month.

In climates outside the tropics, with well defined seasons, choosing a trial month two or three months before the coldest period of the year will normally enable the starting month to be found rapidly.

7.8.2.3 Monthly calculations

The procedure below assumes that condensation is present at only one interface between layers; in some structures it is possible for condensation to occur at two or more interfaces. The equations below can be simply extended to cover this case (see BS EN ISO 13788[8]).

The procedure is described in terms of the more usual UK situation of a heated building in winter, where the vapour pressure is higher inside than out. It can equally be applied to a cooled building in summer when the vapour pressure gradient is reversed.

For each month of the year, starting with the month identified as specified above:

(1) Divide the structure to be analysed into a series of parallel layers, j, each with uniform thermal resistance, R_j, and vapour resistance, Z_j. The layers will usually be chosen to each consist of a separate material. However, a monolithic construction or an individual, very thick, material layer can be subdivided if there is felt to be a chance of condensation occurring within the layer.

(2) Calculate the accumulated thermal and vapour resistance from the inside to each interface between layers, k:

$$R'_k = R_{si} + \sum_{j=1}^{k} R_j \tag{7.18}$$

$$Z'_k = \sum_{j=1}^{k} Z_j \tag{7.19}$$

The total thermal and vapour resistances are:

$$R'_T = R_{si} + \sum_{j=1}^{N} R_k + R_{se} \tag{7.20}$$

$$Z'_T = \sum_{j=1}^{N} Z_j \tag{7.21}$$

(3) Calculate the temperature at each interface between layers using:

$$\theta_k = \theta_i - \frac{R'_k}{R'_T}\left(\theta_i - \theta_e\right) \tag{7.22}$$

Calculate the saturated vapour pressure corresponding to the temperature at each interface, $p_s(\theta_k)$, using equations 7.7 or 7.8 as appropriate.

(4) If there is accumulated condensate from the previous month, go to step 8. If there is no accumulated condensate calculate the vapour pressure at each interface between layers using:

$$p_k = p_i - \frac{Z'_k}{Z'_T}\left(p_i - p_e\right) \tag{7.23}$$

If the vapour pressure, p_k, is less than the saturated vapour pressure $p_s(\theta_k)$, at all interfaces, the structure is free from condensation during this month; proceed to the next month.

(5) If the vapour pressure, p_k, is greater than the saturated vapour pressure $p_s(\theta_k)$, at one or more interfaces, determine the interface with the greatest value of $(p_k - p_s(\theta_k))$. This is the condensation interface (c). Set the vapour pressure equal to the saturated vapour pressure at this interface $p_s(\theta_c)$, and recalculate the vapour pressure at the other interfaces in two stages:

(*a*) from the inside to the condensation interface:

$$p_k = p_i - \frac{Z'_k}{Z'_c}\left(p_i - p_s(\theta_c)\right) \quad (7.24)$$

(*b*) from the condensation interface to the outside:

$$p_k = p_s(\theta_c) - \frac{Z'_k - Z'_c}{Z'_T - Z'_c}\left(p_s(\theta_c) - p_o\right) \quad (7.25)$$

In some constructions, it is possible that further condensation planes, i.e. interfaces with $p_k > p_s(\theta_k)$, will be found after this procedure. The stage specified above should then be repeated until no further condensation planes are identified (see BS EN ISO 13788[8]).

(6) Calculate the amount of condensate at the condensation interface from:

$$g_{c,m} = 0.0864\, n_d \left(\frac{p_i - p_s(\theta_c)}{Z'_c} - \frac{p_c(\theta_c) - p_e}{Z'_T - Z'_c}\right) \quad (7.26)$$

where $g_{c,m}$ is the amount of condensate at the condensation interface for month m ($g \cdot m^{-2}$) and n_d is the number of days in the month.

With similar equations for any further condensation interfaces.

(7) The amount of condensate, $g_{c,m}$, is equal to the accumulated total for that interface $g'_{c,m}$.

Proceed to the next month.

(8) If, in any month, there is accumulated condensate from the previous month at any interface, (c), set the vapour pressure at that interface equal to $p_s(\theta_c)$, and calculate the vapour pressure at the remaining interfaces using equations 7.24 and 7.25.

(9) Calculate the amount of condensate at the interface for the month, $g_{c,m}$, using equation 7.26. This will be positive if further condensation is occurring and negative if evaporation is occurring.

(10) Calculate the accumulated amount of condensate at the interface using:

$$g'_{c,m} = g'_{c,m-1} + g_{c,m} \quad (7.27)$$

If $g'_{c,m}$ is negative (i.e. the amount of evaporation predicted in a month is more than the amount of condensate present at the start of the month), set $g'_{c,m}$ to zero.

Proceed to the next month.

7.8.2.4 Criteria for assessing constructions

There are three criteria for assessing the results

(*a*) No condensation predicted at any interface in any month.

In this case the structure is reported as being free of interstitial condensation.

(*b*) Condensation occurs at one or more interfaces but, for each interface concerned, all the condensate is predicted to evaporate during the summer months.

In this case the maximum amount of condensation that occurred at each interface, and the month during which the maximum occurred are reported. Also, the risk of degradation of building materials and deterioration of thermal performance as a consequence of the calculated maximum amount of moisture are considered according to regulatory requirements and other guidance in product standards.

(*c*) Condensation at one or more interfaces does not completely evaporate during the summer months.

In this case the structure has failed the assessment, and the maximum amount of moisture that occurred at each interface together with the amount of moisture remaining after 12 months at each interface is reported

7.8.2.5 Sources of error

The procedure specified in BS EN ISO 13788[8] and summarised above is one-dimensional, and ignores the effects of moisture storage within materials and air flow through the structure. Moisture transfer is assumed to be pure water vapour diffusion, and the thermal conductivity and the thermal resistance are assumed constant and the specific heat capacity of the materials not relevant. Heat sinks/sources due to phase changes are neglected.

Calculation methods according to this principle are often called 'Glaser' methods. More advanced methods, which are not currently standardised, are available and are described more fully elsewhere[15,16].

There are several sources of error caused by these simplifications:

— The thermal conductivity depends on the moisture content, and heat is released/absorbed by condensation/evaporation. This will change the temperature distribution and saturation values and affect the amount of condensation/drying.

— The use of constant material properties is an approximation.

— Capillary suction and liquid moisture transfer occur in many materials and this can change the moisture distribution.

— Air movements through cracks or within air spaces can change the moisture distribution by moisture convection. Rain or melting snow can also affect the moisture conditions.

— The real boundary conditions are not constant over a month.

- Most materials are at least to some extent hygroscopic and can absorb water vapour.
- One-dimensional moisture transfer is assumed.
- The effects of solar and long-wave radiation are neglected.

Due to the many sources of error, this calculation method is less suitable for building components in which there is significant storage of water and which can experience large diurnal changes in temperature.

In some constructions, especially tiled pitched roofs and metal sheeted walls and roofs, moisture transport is dominated by wind and stack driven air flows. In these cases calculations using both 'Glaser' vapour diffusion models and more complex models that account for liquid water movement through pores are very inaccurate. Some specialist models that take account of air flow are being developed, however these are not generally available.

7.8.2.6 Examples

Tables 7.15 to 7.19 show examples of the calculation of the risk of interstitial condensation in three wall structures.

Example 7.1: Masonry wall with partially filled cavity

Table 7.15 shows the material properties of the individual layers of the wall and the temperature and vapour pressures at the interfaces during January when the internal and external conditions are 20 °C/58% RH and 3.9 °C/85% RH. The initial calculation of the vapour pressures shows that $p_k(1) > p_s(\theta_k)$ at the interface between the cavity and the external leaf of brickwork. The vapour pressure at the other interfaces $p_k(2)$ is recalculated with $p_c(2) = p_s(\theta_c)$ at this point.

Table 7.16 shows the monthly internal and external conditions, the condensation and evaporation and the accumulated condensate, assuming a house at the top of

Table 7.15 Example 7.1: material properties and interface conditions for a masonry wall with partially filled cavity

Layer	d / mm	λ / W·m^{-1}·K^{-1}	$R_{t,j}$ / m^2·K·W^{-2}	$R'_{t,k}$ / m^2·K·W^{-2}	θ_k / °C	$p_s(\theta_k)$ / °C	$r_{v,1}$ / MN·s·g^{-1}·m^{-1}	Z_j / MN·s·g^{-1}	Z'_k / MN·s·g^{-1}	$p_k(1)$ / Pa	$p_k(2)$ / Pa
Inside air					20.00	2337				1355	1355
Boundary layer			0.13				0	0			
Interface				0.13	19.18	2221			0	1355	1355
Plaster	12	0.16	0.075				10	0.12			
Interface				0.205	18.71	2157			0.12	1332	1331
Blockwork	105	0.2	0.525				12	1.26			
Interface				0.73	15.41	1750			1.38	1084	1075
Insulation	50	0.025	2				20	1			
Interface				2.205	6.14	943			2.38	887	872
Cavity*	50		0.18								
Interface				2.385	5.01	872			2.38	887	872
Brick	102	0.752	0.136				10	1.02			
Interface				2.521	4.15	822			3.4	686	686
Boundary layer			0.04				0	0			
Outside air				2.561	3.90	807			3.4	686	686

* Air cavities have a standard thermal resistance (see chapter 3, Table 3.3) and no vapour resistance

Table 7.16 Example 7.1: monthly environmental conditions and accumulated condensate for masonry wall with partial cavity fill

Month	θ_i / °C	ϕ_i / %	θ_1 / °C	ϕ_e / %	$g_{c,m}$ / g·m^{-2}	$g'_{c,m}$ / g
October	20	62	10.6	83	0	0
November	20	59	6.4	85	0	0
December	20	58	4.6	86	59.0	59.0
January	20	58	3.9	85	98.0	157.0
February	20	57	3.9	82	7.3	164.3
March	20	56	5.7	79	–261.0	0.0
April	20	56	8.0	75	0	0
May	20	58	11.3	73	0	0
June	20	64	14.2	75	0	0
July	20	67	15.8	75	0	0
August	20	68	15.7	77	0	0
September	20	65	13.5	80	0	0

climate class 3 in Manchester. Condensation starts in December, reaches a peak accumulation of 164.3 g·m^{-2} in February and all evaporates in March. As the condensate does not persist for the whole year and the peak accumulation is small compared to the amount of driving rain that will impact the wall from the outside in any case, the construction would be considered to have passed the assessment criteria.

Example 7.2: Timber framed wall with no vapour control layer

Table 7.17 shows the material properties of the individual layers of the wall and the temperature and vapour pressures at the interfaces during January when the internal and external conditions are 20 °C/58% RH and 3.9 °C/85% RH. The initial calculation of the vapour pressures shows that $p_k(1) > p_s(\theta_k)$ at the interface between the mineral wool and plywood sheathing. The vapour pressure at the other interfaces $p_k(2)$ is recalculated with $p_c(2) = p_s(\theta_c)$ at this point.

Table 7.18 shows the monthly internal and external conditions, the condensation and evaporation and the accumulated condensate, assuming a house at the top of climate class 3 in Manchester. Condensation is present in October, reaches a peak accumulation of 3048 g·m^{-2} in April and does not all evaporate in September, with 776 g·m^{-2} remaining. As the condensate persists for the whole year and the peak accumulation very high on the plywood, a vulnerable material, the construction fails the assessment criteria

Table 7.17 Example 7.2: material properties and interface conditions for a masonry wall with partially filled cavity

Layer	d / mm	λ / W·m^{-1}·K^{-1}	$R_{t,j}$ / m^2·K·W^{-2}	$R'_{t,k}$ / m^2·K·W^{-2}	θ_k / °C	$p_s(\theta_k)$ / °C	$r_{v,1}$ / MN·s·g^{-1}·m^{-1}	Z_j / MN·s·g^{-1}	Z'_k / MN·s·g^{-1}	$p_k(1)$ / Pa	$p_k(2)$ / Pa
Inside air					20.00	2337				1355	1355
Boundary layer			0.13				0	0			
Interface				0.13	19.33	2242			0	1355	1355
Plasterboard	12	0.167	0.07				45	0.54			
Interface				0.20	18.97	2191			0.54	1326	1211
Glassfibre	100	0.04	2.50				10	1			
Interface				2.70	6.15	945			1.54	1273	945
Plywood	12	0.143	0.08				450	5.4			
Interface				2.79	5.72	917			6.94	982	817
Breather								0.4			
Interface				2.79	5.72	917			7.34	960	807
Cavity	50		0.18				0	0			
Interface				2.97	4.80	860			7.34	960	807
Brick	102	0.752	0.14				50	5.1			
Interface				3.10	4.11	819			12.44	686	686
Boundary layer			0.04				0	0			
Outside air				3.14	3.9	807			12.44	686	686

Table 7.18 Example 7.1: monthly environmental conditions and accumulated condensate for a timber framed wall without a vapour control layer

Month	θ_i / °C	ϕ_i / %	θ_1 / °C	ϕ_e / %	$g_{c,m}$ / g·m^{-2}	$g'_{c,m}$ / g
October	20	62	10.6	83	13.5	401.5
November	20	59	6.4	85	412.4	813.9
December	20	58	4.6	86	582.2	1396.0
January	20	58	3.9	85	651.1	2047.1
February	20	57	3.9	82	546.0	2593.1
March	20	56	5.7	79	368.5	2961.6
April	20	56	8.0	75	86.6	3048.2
May	20	58	11.3	73	–281.4	2766.8
June	20	64	14.2	75	–470.4	2296.4
July	20	67	15.8	75	–643.6	1652.7
August	20	68	15.7	77	–576.1	1076.6
September	20	65	13.5	80	–300.7	775.9

Table 7.19 Example 7.1: material properties and interface conditions for a timber framed wall with a vapour control layer

Layer	d / mm	λ / $W \cdot m^{-1} \cdot K^{-1}$	$R_{t,j}$ / $m^2 \cdot K \cdot W^{-2}$	$R'_{t,k}$ / $m^2 \cdot K \cdot W^{-2}$	θ_k / °C	$p_s(\theta_k)$ / °C	$r_{v,1}$ / $MN \cdot s \cdot g^{-1} \cdot m^{-1}$	Z_j / $MN \cdot s \cdot g^{-1}$	Z'_k / $MN \cdot s \cdot g^{-1}$	$p_k(1)$ / Pa
Inside air					20.00	2337				1355
Boundary layer			0.13				0	0		
Interface				0.13	19.33	2242			0	1355
Plasterboard	12	0.167	0.07				45	0.54		
Interface				0.20	18.97	2191			0.54	1354
Vapour control								250		
Interface				0.20	18.97	2191			250.54	716
Glassfibre	100	0.04	2.50				10	1		
Interface				2.70	6.15	945			251.54	714
Plywood	12	0.143	0.08				450	5.4		
Interface				2.79	5.72	917			256.94	700
Breather membrane								0.4		
Interface				2.79	5.72	917			257.34	699
Cavity	50		0.18				0	0		
Interface				2.97	4.80	860			257.34	699
Brick	102	0.752	0.14				50	5.1		
Interface				3.10	4.11	819			262.44	686
Boundary layer			0.04				0	0		
Outside air				3.14	3.90	807			262.44	686

Example 7.3: Timber framed wall with vapour control layer

Table 7.19 shows the material properties of the individual layers of the wall, including a vapour control layer with vapour resistance 250 $MN \cdot s \cdot g^{-1}$ between the plasterboard and insulation, and the temperature and vapour pressures at the interfaces during January when the internal and external conditions are 20 °C/58% RH and 3.9 °C/85% RH. The initial calculation of the vapour pressures shows that $p_k(1) > p_s(\theta_k)$ at all the interfaces. No condensation is therefore predicted. Repeating this calculation for each month gives the same result.

7.9 Control of condensation

7.9.1 Condensation assessment

Design for the control of both surface and interstitial condensation depends upon obtaining a satisfactory relationship between air conditions (internal and external air temperatures and humidities) and the thermal moisture related properties of the external elements of construction.

Condensation control should be considered as part of the design process. Successful control will depend on the interaction of the factors that determine moisture production, ventilation, thermal insulation and the heating system. Some of these are under the control of the occupants, others depend on good design of the building and its systems. All these aspects, therefore, should be considered carefully and, as they are interdependent to a greater or lesser degree, they should be considered together.

As condensation depends on the interactions between the occupant, the building and the outside weather, it is essentially a stochastic process. Measures that can be taken will reduce the risk but may not eliminate it in all circumstances. This is the approach adopted in BS 5250(3), which talks throughout of 'minimising the risk', not preventing condensation.

7.9.2 Controlling surface condensation

To minimise surface condensation or mould growth, it is necessary to achieve low internal vapour pressures by limiting moisture input to the building and providing effective ventilation and to achieve high surface temperatures by providing more insulation, limiting thermal bridges and, if necessary, increasing the heat input.

The following factors should be taken into account in design:

— *Occupant activity and heating/ventilation regime*: occupants and the activities and processes within buildings, including some domestic appliances, generate moisture. Some industrial processes, canteens, kitchens, laundries, shower rooms or swimming pools generate very large amounts of moisture. In poorly insulated buildings, fuel costs can limit the heat input, leading to cold surface temperatures. The function of a building could change completely, e.g. a building built as a warehouse could be changed to a wet process factory.

— *Built-in water*: some constructions, such as massive concrete floor slabs, contain a large of amount of built-in water, often known as 'construction water'. This will take many months to dissipate and should be considered as a significant source of water vapour within the building during this period.

— *Ventilation*: ventilation removes the water vapour produced, but also has energy costs. Intermittent use of mechanical extract in areas such as kitchens and bathrooms, where moisture production is concentrated, can remove a large proportion of the water vapour produced. Mechanical extract systems with heat recovery are more complex and expensive to install, but bring real benefits. Passive stack ventilators and supply systems installed in lofts provide unobtrusive background ventilation.

— *Thermal insulation*: the more a part of the structure is insulated, the warmer the internal surface will be for the same room heat input and, consequently, the risk of surface condensation or mould growth will be lower. However, layers to the outside of any extra insulation will be colder, and therefore more prone to interstitial condensation, and frost damage (see below).

— *Thermal bridging*: thermal bridging, which lowers surface temperatures, should be minimised by careful design and detailing of insulation in vulnerable areas such as wall–floor junctions, roof eaves and areas around window and door openings.

— *Thermal response*: the thermal response of the internal layers should be matched with the proposed heating and activity regime. High mass elements will warm and cool slowly (slow thermal response) and they are therefore more suitable for buildings which are heated for long periods. Low mass elements will warm and cool quickly (fast thermal response) and are particularly suitable for infrequent or intermittent heating.

7.9.3 Controlling interstitial condensation

7.9.3.1 General

To minimise interstitial condensation, it is necessary to do one or more of the following:

(*a*) obtain low vapour pressures by ventilation and/or reduced moisture input to the building

(*b*) seal any gaps to limit air movement from the warm side into the structure

(*c*) use materials of high vapour resistance near to the warmer side of the construction

(*d*) use material of low vapour resistance, or provide ventilated cavities, near the colder side of the construction

(*e*) use materials of low thermal resistance near to the warmer side of the construction

(*f*) use materials of high thermal resistance near to the colder side of the construction.

If condensation is judged to be harmful, then steps should be taken to limit the amount of moisture reaching the colder elements by using vapour control layers or inner layers of relatively high vapour resistance or by the inclusion of a ventilated air space between the insulation and the outer elements.

7.9.3.2 Vapour control layers

Where a vapour control layer is specified, it should be of appropriate vapour resistance and should be situated on the warm side of the insulation. A vapour control layer placed within the insulation will be colder and a possible site for condensation in a high humidity environment.

It is extremely difficult to construct an impervious layer in practice. For example, a vapour control layer laid above a roof deck can be constructed to have a high vapour resistance, but if the same material is fixed to the soffit, it will be much more difficult to achieve the same resistance. The performance of a vapour control layer depends upon the design life of the building, the material selected, workmanship and 'buildability'. Any holes, fixings, pipes, electrical fittings, etc. will downgrade performance and should be considered in the design.

Joints in a flexible sheet vapour control layer should be lapped to a minimum of 50 mm and sealed with an appropriate sealant and should be made over a solid backing member or substrate. Similarly, tears and splits should be repaired using an overlay of the same material, jointed as above. If polyethylene sheeting is used, it should be protected from heat and sunlight to reduce the risk of degradation occurring.

Where a vapour control layer is incorporated in or on a rigid board or profiled metal liner sheet, joints between adjacent boards should be sealed with an appropriate sealant or tape or covered or otherwise closed to avoid mass transfer of water vapour due to air leakage.

A vapour control layer should extend over the whole internal roof and wall areas. Side and end joints should be kept to a minimum. This requirement to achieve a functional vapour control layer must be carefully considered at the design stage. Great care and importance must be attached to the design of different construction elements and the connections required between different materials. Vapour control layers should be integrated with and sealed to other building elements such as masonry, upstands and glazing systems.

References

1 BS EN ISO 7345: 1996: *Thermal insulation. Physical quantities and definitions* (London: British Standards Institution) (1996)

2 BS EN ISO 9346: 1996: *Thermal insulation. Mass transfer. Physical quantities and definitions* (London: British Standards Institution) (1996)

3 BS 5250: 2002: *Code of practice for control of condensation in buildings* (London: British Standards Institution) (2002)

4 Hansen K K *Sorption isotherms — a catalogue* Technical University of Denmark Building Materials Laboratory Report 162/86 (Lyngby: Technical University of Denmark) (1986)

5 Kumaran M K *Final Report: IEA-Annex 24 Task 3: Material Properties* (Paris: International Energy Agency) (1996)

6 Billington N S *Building Physics: Heat* (London: Pergamon Press) (1967)

7 *Reference data* CIBSE Guide C (London: Chartered Institution of Building Services Engineers) (2001)

8 BS EN ISO 13788: 2001: *Hygrothermal performance of building components and building elements. Internal surface temperature to avoid critical surface humidity and interstitial condensation. Calculation methods* (London: British Standards Institution) (2001)

9 BS EN ISO 12524: 2000: *Building materials and products. Hygrothermal properties. Tabulated design values* (London: British Standards Institution) (2000)

10 BS EN ISO 12572: 2001: *Hygrothermal performance of building materials and products. Determination of water vapour transmission properties* (London: British Standards Institution) (2001)

11 Pasanen A L et al. Laboratory studies on the relationship between fungal growth and atmospheric temperature and humidity *Environment International* **17** 225–228 (1991)

12 *Limiting thermal bridging and air leakage: Robust construction details for dwellings and similar buildings* DEFRA/DTLR Report (London: The Stationary Office) (2001)

13 *Minimising thermal bridging in new dwellings* Good Practice Guide GPG174 (Action Energy) (1996) (www.thecarbontrust.co.uk)

14 BS EN ISO 10211: *Thermal bridges in building construction. Heat flows and surface temperatures*: Part 1: 1996: *General calculation methods* (London: British Standards Institution) (1996)

15 Hens H *Final Report: IEA-Annex 24 Task 1: Modelling* (Paris: International Energy Agency) (1996)

16 *Modelling and controlling interstitial condensation in buildings* BRE Information Paper IP2/05 (Garston: Building Research Establishment) (2005)

8 Health issues

This chapter consists of extracts from CIBSE TM40: *Health issues in building services*, a copy of which is included in PDF format on the CD-ROM than accompanies this Guide

8.1 Introduction

The constitution of the World Health Organisation defines good health as 'a state of complete physical, mental and social well being, not merely the absence of disease and infirmity'. While the indoor environment should be managed in such a way as to promote health, not merely to avoid illness, in the first instance it is necessary to ensure that the environment does not contribute to ill health and undesirable stress (see chapter 1)

The boundaries between health, welfare and safety are vague. Neglect of health or poor welfare can reduce the safety. Discomfort introduces stress, which if maintained can affect health. Surveys by the Health and Safety Executive show that stress-related absenteeism has risen 107% between 1996 and 2001 and we must ensure that our building services are not contributing significantly to the problem. The health hazard is the ill effect upon the person. The risk of ill health is a function of the likelihood of the effect. Occupational health specialists express their type of concern as the relationship between the two, see Figure 8.1.

The Workplace (Health, Safety and Welfare) Regulations 1992[1] set out in general terms how workplaces should be managed to ensure that they meet the health, safety and welfare needs of those in the workplace.

8.2 Thermal conditions for stress

8.2.1 Legal background

The Fuel and Electricity (Heating) (Control) Order 1974[2] and The Fuel and Electricity (Heating) (Control) (Amendment) Order 1980[3] prohibit the use of fuels to heat premises above 19 °C. This Order does not apply where contributions from heat sources (other than from space heating) take the room to higher temperatures.

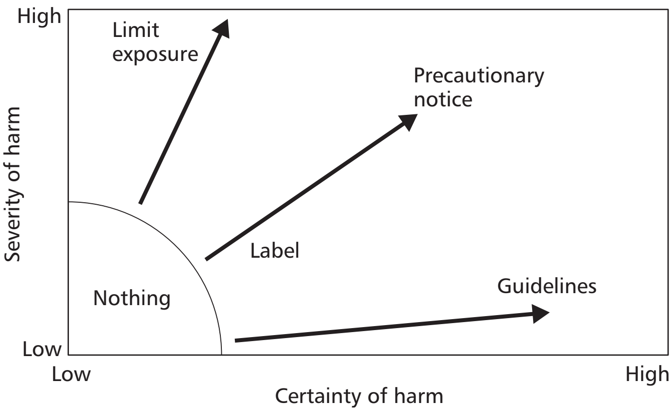


Figure 8.1 Risk analysis for occupational hazards; the relationship between certainty and severity determine the action

The Workplace (Health Safety and Welfare) Regulations 1992[1] require the workplace temperature to be reasonable without the need for special clothes, that the method of heating shall not be injurious nor offensive and that sufficient thermometers shall be provided to enable the workforce to determine the indoor temperature. Approved Code of Practice L24[4] requires workplace temperatures to be normally at least 16 °C unless much of the task is severe physical work, in which case 13 °C is the minimum acceptable temperature. For temperatures below 13 °C the employer is required to provide personal protective equipment.

The Provision and Use of Work Equipment Regulations 1992[5] requires protection from hot surfaces, particularly when siting temporary heaters. Care home and hospitals are required to maintain all exposed heater surfaces at or below 43 °C[6]. Temperatures above and around 43 °C will begin to cause skin damage if the skin contact is of sufficient duration. Below this temperature discomfort, pain sensation and burns will be avoided.

Employers are required to provide hot and cold water for washing. Sustained outbreaks of Legionnaire's disease, however, have been attributed to lukewarm water aerosols from hot and cold water supplies, evaporative cooling towers or whirlpool spas. Guidance is now in place to ensure that water supplies for those in employment are, at the cold tap, below 20 °C within two minutes and, at the hot tap, above 50 °C within one minute. Any hot water calorifier should store the hot water at 60 °C or above. Risk analysis to control the risk of legionellosis is required for any other water outlet which is capable of generating an aerosol accessible to persons and which operates between 20 °C and 50 °C[7].

Specific guidelines are provided for institutions such as prisons, hospitals, schools, and care homes to limit the hot water temperature to around 44 °C so that the occupants of these institutions do not harm themselves or others. Such institutions require thermostatically controlled valves to ensure that the storage and distribution temperature is hot enough to control *legionella* but that the water is thermally safe at the tap outlets[6,8–10]. The water temperature of bidets in hospital and health premises is limited to 38 °C to minimise the shock that may occur when the hot water touches the skin. Unexpected thermal shock while using a bidet may unbalance the user and cause a fall. Free flowing hot water, as in showers or washbasins, is regulated to a maximum 41 °C[6].

8.2.2 Heat stress and heat exhaustion

Working in a hot climate can lead to a number of disorders. Heat stroke is potentially the most serious and is life threatening. Heat stroke is an acute condition caused by the body temperature rising above 40.5 °C (normally 37.6 °C). Symptoms may include mental confusion, mottled or cyanotic skin, loss of consciousness and/or convulsions. Without treatment, it is fatal. Heat stroke will have a lasting effect on a person's ability to tolerate heat.

Heat syncope is a less severe heat illness than heat stroke and results from the pooling of blood in dilated vessels of the skin and/or in the lower extremities. Fainting may occur, particularly if the person is standing up and immobile. It usually affects un-acclimatised individuals. Recovery of consciousness is rapid.

Heat exhaustion can occur either through the loss of salt or excess loss of water after heavy perspiration for several hours. Oral temperature may be normal or low, but rectal temperature is usually elevated (37.5–38.5 °C). Heat cramps, skin eruptions and behavioural disorders are also commonly associated with heat exposure. Those who are regularly exposed to heat stress develop an increased tolerance to heat, known as acclimatisation.

8.2.3 Cold stress

In cold environments, cold air will generally produce severe discomfort before any effects on health occur. However, if no remedial action is taken and cold exposure persists, then body temperature will decline. The rate and total amount of heat loss from the body will depend upon a variety of factors including air temperature, air velocity, activity level and the clothing of the individual. Therefore, there is no simple relationship between hypothermia and room temperature (see chapter 1).

If the core temperature of an individual falls to approximately 35 °C (normally 37.6 °C) then this is generally recognised as hypothermia. A person with hypothermia will often not realise they are in trouble, or act counter to common sense and further precipitate cooling and may also become aggressive making treatment difficult.

The World Health Organisation (WHO) has identified possible health risks associated with cold temperatures[(11)]. No risk is expected between 18 °C and 24 °C for the normal population, although 20 °C is the minimum recommended temperature for the very old and the very young. A review of relevant literature concluded that conditions below 16 °C and above 65% relative humidity impose additional hazards particularly from respiratory diseases and allergic responses to moulds, fungi, house dust and allergens from domestic animals. Temperatures below 12 °C may pose a health risk for pre-school children and the elderly, the sick and those handicapped. The elderly and the very young may be at special risk when bedroom temperatures fall at night[(11)]. Infection is believed to increase below 16 °C[(12)].

8.2.4 Effects of hot and cold stress

When individuals work under hot or cold conditions their performance can be affected. Performance in the cold deteriorates largely for physiological reasons, although pain and decreased motivation induced by thermal extremes play a part. At low temperatures manual dexterity may be affected (e.g. tactile finger sensitivity may be affected resulting in sensations of numbness. Strength, speed of movement (particularly flexion and extension of fingers) is also affected). Slower performance and an increase in mistakes are likely as a result. Cold sensation may also act as a distraction and result in a worker rushing to get out of the cold. This may also lead to an increase in the number of mistakes made.

In mild heat stress, where the body can maintain thermal equilibrium by sweating, numerous field studies show that there is a decrease in performance, although the level at which this occurs is widely disputed. One review of the literature[(13)] proposes the model summarised in Figure 8.2. Tasks are broken down to illustrate the kinds of work studied. The effect of temperature on performance will vary with a number of factors including acclimatisation of the individual, the specific task and conditions and the skill of the subject etc.

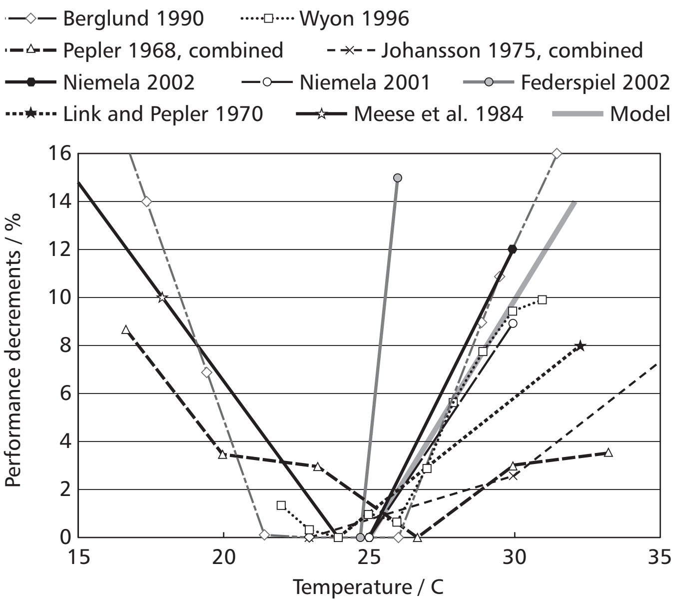


Figure 8.2 Illustrative relationship between room temperature and performance[(13)]

Factors including increased sweating, which can make gripping of objects difficult, increased irritability and increased drowsiness/decreased arousal can all lead to an increase in the number of mistakes. Again, as with work in the cold, the fact that the environment may be perceived as unpleasant can result in people rushing to finish the job in order to get out of that environment[(14)]. The field evidence indicates that the performance of the more skilled individuals suffer less than that of the less skilled in carrying out well rehearsed procedures.

In time of a heat-wave some individuals become dehydrated, overheated, exhausted and experience heat stroke. Those who are susceptible include the old, babies and children, those with mental problems, those with chronic conditions such as breathing and heart problems, those

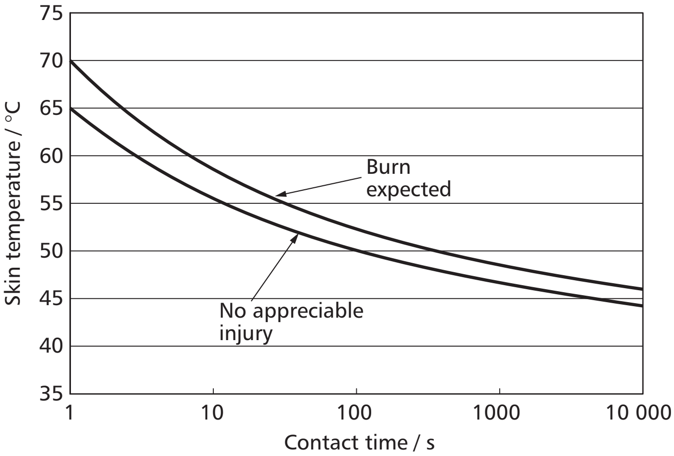


Figure 8.3 The relationship between temperature and time to cause a scald[17]

with a high temperature from infection, those with mobility problems and those physically active such as manual workers and sports people.

8.2.5 Scalding

The rate of skin damage from a scald is a function of temperature and exposure time, regardless of the area of skin affected. Very young children are more sensitive. Reported scalds occur mainly to the very young (1–3 years old) and the very old, and usually occur in the bath. Note that scalding can occur after prolonged exposures to temperatures that would not normally be considered scalding temperatures in everyday terms.

There is a wide difference of opinion on the perception of water temperature but Figure 8.3 shows the relationship between temperature, time and degree of scalding for an adult[15].

8.2.6 Burns

There is a possible danger in care homes or hospitals of a person falling and lying against a hot surface for several hours and experiencing a slow but still damaging burn. Low temperature protective shields are fitted to hot water central heating radiators in such areas to ensure that no exposed surface exceeds 43 °C[6].

8.3 Humidity

8.3.1 Introduction

As mentioned in chapter 1, section 1.3.1.3, humidity has a significant effect on thermal comfort only at high temperatures when high humidity impedes the ease of evaporative cooling through sweating. However, the combination of high temperature and high humidity does introduce a feeling of sultriness or oppression which occurs at 70% RH at 21 °C and 60% RH at 23 °C[16].

Humidity has many other important roles although less well defined than is that for temperature. In normal circumstances, humidity in the range 40–70% RH is acceptable but with the optimum being around 65% RH at comfort temperature. There are wide differences between individuals. Moving suddenly into high humidity zones can provide transient warmth from the adsorption of moisture into the clothing. Equally, moving from a humid zone to a dry one can provide a chilling effect for the reverse process of desorption of water from the clothing fabric. The thermal insulation of clothing and bedding is strongly influenced by the moisture content of the materials used. Damp materials associated with high humidities lose much of their thermal insulation.

Moisture control is particularly important in dwellings and is considered in detail in chapter 7.

8.3.2 Problems with high humidity

Allergies and respiratory illnesses have long been associated with mould growth and moisture, particularly for asthma and rhinitis. Asthma is a difficulty in breathing, often associated with coughing. Rhinitis is an inflammation of the nose which causes similar effects. Both are particularly distressing if they occur at night and interrupt sleep. Both asthma and rhinitis are now closely associated with an allergic response to the house dust mites and their waste products. The incidence reaches it peak in teenagers and thereafter declines with age.

House dust mites prosper in warm damp conditions providing there is a supply of food in the form of human skin. Mites can be controlled if their immediate microclimate can be maintained below 73% RH. However the insulating properties of carpets can mean that there can be a temperature gradient within the carpet with a corresponding increase in RH at the lower temperature. Measurements suggest that this is of the order of 10% RH[17]. This requires maintaining the room condition below 63% RH at all times to prevent mite multiplication.

Buildings in which there are prolonged high air humidities are likely to experience problems such as airborne fungi and house dust mites, particularly if room humidities exceed 70% RH for long periods. Fungi generally grow on damp organic material but they do not necessarily require either high air humidity or high air temperature for growth if the substrate conditions are suitable. The growth rate is dependant upon the nutrient, the temperature and the humidity. Each mould has its special growth characteristics. Once mould has started to grow it is difficult to stop by lowering the humidity because one of the metabolic products of growth is water, which enables the mould to continue to grow in drier conditions.

For the purposes of designing air conditioning systems, a maximum room relative humidity of 60% within the recommended range of summer design dry resultant temperatures would provide acceptable comfort conditions for human occupancy and minimise the risk of mould growth and house dust mites. Condensation should be avoided within buildings on surfaces that could support microbial growth or be stained or otherwise damaged by moisture. This may be achieved by ensuring that, where possible, all surfaces are above the dew-point of the adjacent air.

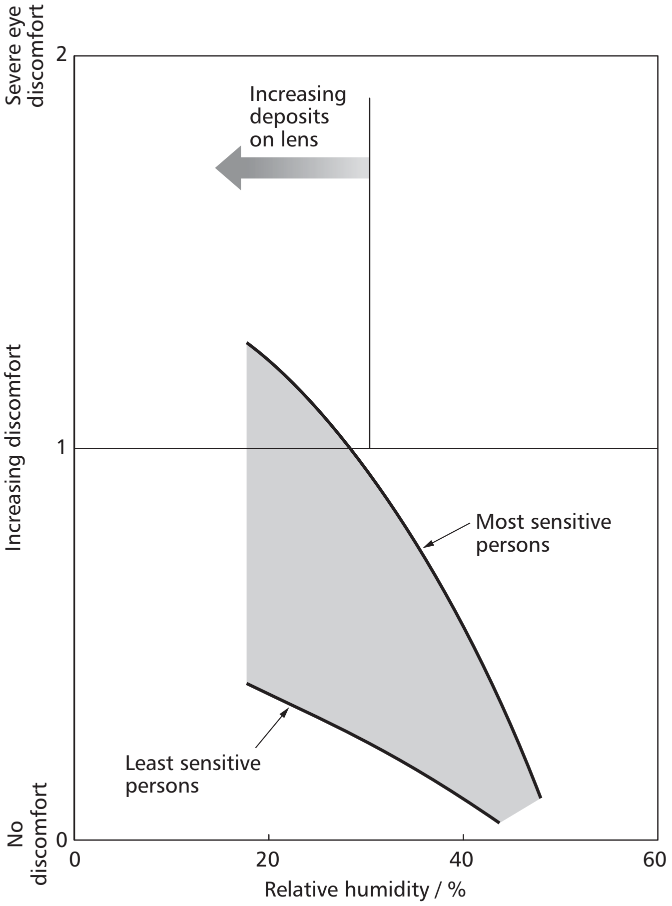


Figure 8.4 Influence of relative humidity on eye discomfort with contact lens[19]

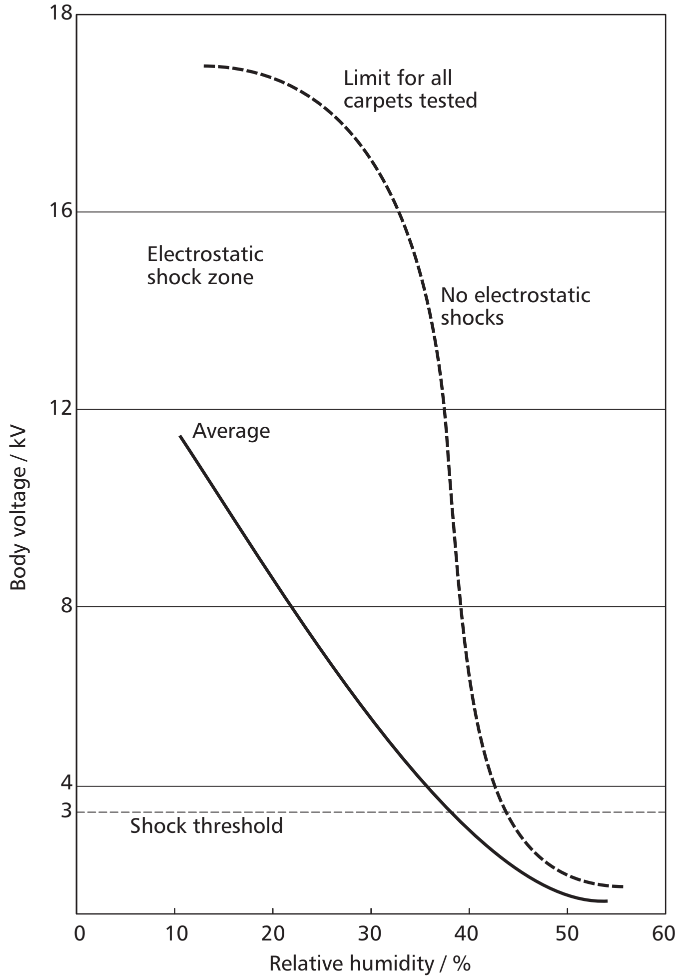


Figure 8.5 Relationship between relative humidity and electrostatic shocks[20]

8.3.3 Problems with low humidity

Complaints of dryness are frequently found in office workers but not clearly linked to the actual humidity. It is likely that there is a second effect where low humidities interact with some other pollutant and increase its effect. Lowering the humidity increases people's perception to smells and makes the smell more objectionable. It also increases the irritation of cigarette smoke. It also increases the smoke production from cigarettes and permits fine particles to stay suspended in the atmosphere and prolong the nuisance[14]. More recent research shows that the concept of freshness is linked to reduced humidities[18].

There is some evidence from recent investigations into problem buildings that there is a correlation between low room humidity and symptoms associated with dryness and irritation of the mucosa. It has been suggested that low room moisture content increases evaporation from the mucosa and can produce micro-fissures in the upper respiratory tract which may act as sites for infection. The reduction in mucous flow inhibits the dilution and rejection of dust, micro-organisms and irritant chemicals such as formaldehyde. This is a particular problem for wearers of contact lenses, see Figure 8.4[19].

If possible, at the design temperatures normally appropriate to sedentary occupancy, the room humidity should be above 40% RH. Lower humidity is often acceptable for short periods. Humidity of 30% RH or below may be acceptable but precautions should be taken to limit the generation of dust and airborne irritants, such as tobacco smoke, and to prevent static discharge from occupants.

Note that for heated-only buildings in the UK, the humidity can remain below 40% RH during periods of sustained cold weather.

Shocks due to static electricity are unlikely with humidities above 40% RH or at lower humidities if special precautions are taken in the specification of materials to prevent the build-up of static electricity. Static electricity can lead to shocks when occupants are not adequately earthed via their shoes and the floor covering. The incidence of electrostatic shocks depends on the electrical resistance of the floor covering and the speed and distance walked. The resistance is a function of the material itself and its moisture content. The carpet becomes more insulating in dryer conditions. Dryer conditions, faster walking, and longer walked distances increase the voltage rise and risk of electrostatic shocks.

At low room humidity, some types of carpet can become highly charged and electrostatic shocks may be experienced. Typical body voltages are shown in Figure 8.5[20] as a function of room percentage saturation. Extreme values, as reported, and the shock voltage threshold are also shown. Women are more sensitive to shocks than men. In general, shocks are unlikely above 40% saturation. Carpeted buildings with underfloor heating have particularly dry carpets and require humidity to be above 55% saturation to avoid electrostatic shocks.

Electrostatic discharges can be dangerous in the presence of hazardous gases or explosive substances. In such conditions an atmosphere above 65% RH is considered safe. Electrostatic problems from machinery in manufacturing are reported at RH below 40%. Humidification can be used to eliminate this effect to protect the process and the person.

8.4 Air quality and ventilation

8.4.1 Regulatory guidelines

8.4.1.1 Residential

There are no air quality standards in Britain for residential buildings. The prescriptive provision for residential ventilation requires controllable ventilation slots of a given size under windows for background ventilation, planned ventilation in the bathroom and kitchen (usually by mechanical extract fans) to extract moisture and by openable panels, usually windows, in each room for occasional, high flow needs such as a warm summer day to prevent overheating[21].

8.4.1.2 Workplaces

Workplace ventilation addresses the particular question of contaminants released at work, either within the building or around it. The Workplace (Health, Safety and Welfare) Regulations 1992[1] set out in general terms the requirements for ventilation of workplaces.

Workplaces need to be adequately ventilated and the introduced air should be drawn from an area outside the workplace that is not contaminated, e.g. by flues or chimneys. Ventilation should remove and dilute warm and/or humid air and there should be sufficient air movement to provide a sense of freshness without it being draughty. If there are processes carried out in the workplace which create heat, dust, fumes or vapours, additional ventilation may be required.

Adequate ventilation may be provided by windows or other openings but additional mechanical ventilation may also be required.

As a general rule, the fresh air supply rate should not fall below between 5 and 8 litres per second per occupant but this will depend on various other factors including floor area per occupant, processes carried out, equipment used and whether the work is strenuous[4]. (8 L/s fresh air is equivalent to an elevation of 600 ppm of carbon dioxide (CO_2) which, when added to the normal outdoor CO_2 of 400 ppm, gives an internal CO_2 concentration of 1000 ppm; 5 L/s would be equivalent to 1350 ppm internally.) The higher ventilation rate of 8 L/s per person is recommended[22]. More general guidance on workplace ventilation can be found in various HSE publications[22,23].

There is more specific HSE guidance for specialist industries such as catering, woodworking, chemical and microbiological work, welding and for working in dusty conditions. Details can be found on the HSE website (www.hse.gov.uk).

Part L of the Building Regulations includes standards of airtightness to minimise air infiltration, and minimum energy efficiency standards for air conditioning and mechanical ventilation equipment[24,25]. This means that, in future, the bulk of the ventilating air will come through planned routes and by means of an efficient supply system.

8.4.1.2 Schools

Schools have prescribed ventilation rates of 3 L/s per person for background ventilation and 8 L/s per person when required[26].

8.4.2 Human sensitivity to inhalation of pollutants

A substance that enters the nasal cavity may be sensed by two largely separate detection systems:

— *the olfactory sense*: responsible for odour detection.

— *the general chemical sense*: sensitive to irritants.

The general chemical sense is located all over the mucous membranes, in the eyes as well as the nose.

The two senses may interact. For example, it is possible for an odour to be disguised by irritation and vice versa[27] or a single substance may evoke both odour and irritant sensations. Humans are known to adapt to odours with time, whereas irritation may increase with time[28,29].

There are two kinds of adaptation to odour. Over periods of about 30 minutes people become less sensitive to any odours present. Over much longer periods (i.e. weeks or months) people come to accept an odour as normal and harmless and therefore become less aware of it. Conversely, over a period of minutes or hours, the discomfort from exposure to irritants will normally increase. Over a longer period, adaptation is possible but this may be largely behavioural (e.g. by ceasing to wear contact lenses). The more likely outcome is to become sensitised so that the same concentration of an irritant has a greater effect. Sensitisation is also possible when a substance exerts its effect through the immune system (e.g. allergic reactions).

In the specific case of exposure to environmental tobacco smoke, one study[30] has found that irritation intensity increases by a factor of two during the first hour of exposure, after which steady state occurs. The same study found that perceived odour intensity declined by a factor of 50% and levelled out after only a few minutes.

Many everyday occurrences result in the release of odours, some of which may be perceived as pleasant and some unpleasant. Some evolve from the release of potentially harmful substances but the airborne contaminants likely to be encountered in non-industrial buildings do not usually result in irreversible health effects. However, the exceptions include *legionella* bacteria, radon gas and lead and benzene from motor vehicle exhaust emissions.

Building occupants may be exposed to a mixture of hundreds, or thousands, of airborne contaminants. The air within a modern office may contain chemicals and micro-organisms, which have originated from numerous sources, both inside and outside the building. Concentrations of individual contaminants are frequently in the order of one thousandth of published occupational exposure limits, or less, but may still be above odour detection thresholds[31].

For comfort, indoor air quality may be said to be acceptable if[32,33]:

— not more than 50% of the occupants can detect any odour, and
— not more than 20% experience discomfort, and
— not more than 10% suffer from mucosal irritation, and
— not more than 5% experience annoyance, for less than 2% of the time.

These comfort-based criteria do not account for potential effects on health of the contaminants found in buildings. Some of these, e.g. radon and its progeny, are odourless and do not affect comfort but may have serious effects on the health of any individuals exposed to them.

The following measures, in sequential order, should be adopted to eliminate or reduce exposure of occupants to airborne contaminants in buildings:

(1) eliminate contaminant(s) at source
(2) substitute with sources that produce non-toxic or less malodorous contaminants
(3) reduce emission rate of substance(s)
(4) segregate occupants from potential sources of toxic or malodorous substances
(5) improve ventilation, e.g. by local exhaust (if source of contamination is local), displacement or dilution
(6) provide personal protection.

These measures are not mutually exclusive, and some combination will usually be necessary. Adequate ventilation will always be required.

Published limits for indoor air pollutant requirements fall into two categories:

— those that have been derived from studies of health effects
— those based on the sensory effects.

8.4.2.1 Exposure limits based on effects on health

Occupational exposure limits (OELs) for the UK are published annually by the Health and Safety Executive in EH40: *Occupational exposure limits*[34]. These limits are levels used to demonstrate compliance with the Health and Safety at Work etc. Act 1974[35] and the COSHH Regulations 1994[36]. This legislation applies not just to industrial workplaces but to all workplaces, including offices. In practice, in most circumstances the levels of exposure and the modes of exposure do not present a significant risk to the occupants of non-industrial workspaces such as offices.

The occupational exposure limits listed in EH40 are not exclusive; absence from the list does not imply that a substance has no ill effects on health nor that it is safe to use without control. Where a particular substance does not have an OEL the employer, in carrying out a risk assessment, should also determine an adequate level of control for the substance and, in effect, set an 'in-house' OEL.

It is not appropriate to use OELs to calculate the required outside air supply. The provision of sufficient outside air is important but it is only one of a combination of measures required to provide adequate control of exposure. Such measures are outside the scope of this Guide and will often require specialist advice.

8.4.2.2 Exposure limits based on effects on senses

In practice, exposure of workers in non-industrial environments to the same concentrations of malodorous substances that occur in industry would not be acceptable. This is primarily because expectations are generally much higher amongst occupants of non-industrial buildings. Odour detection and hence comfort are not primary considerations in setting occupational exposure limits.

Sensory comfort guidelines[33] are available for only a small number of single substances, see Table 8.2 (page 8-8). These are based on the odour detection threshold for given averaging times. These values can be used to calculate dilution rates when it is known that a specific substance may be responsible for odour annoyance.

However, the ideal is for the substance to be eliminated at source. For substances which do not appear in Table 8.2, an exposure limit for non-industrial applications can be estimated by multiplying the relevant EH40[34] occupational exposure limit by a factor of 0.1.

8.4.3 Outdoor air

In all the Regulations relating to ventilation the assumption is that the outdoor air is clean and wholesome. However, as a result of urban pollution, outdoor air can no longer be automatically considered as a clean air source suitable for diluting indoor pollutants. Therefore the quality of outdoor air must be considered in the design of ventilation and air conditioning systems. An analysis of the most important pollutants should be carried out if there is any cause for concern about the quality of the air that can enter the building via windows or ventilation air intakes. The local environmental health department should be consulted to determine whether monitoring has already been carried out at a location with a similar environment close to the site under consideration. Data on atmospheric pollution in the UK are published annually[1,39]. Guidance on the design and positioning of ventilation air intakes is given in CIBSE TM21: *Minimising pollution at air intakes*[40].

The guideline values given in Table 8.2 (page 8-8) apply to both indoor and outdoor pollutants. If a local survey indicates that these concentrations are likely to be exceeded in the incoming ventilation air on a regular basis then consideration should be given to specific filtration of the offending pollutants.

If external pollutant concentrations rise above the standards during a typical day, then it may be possible to reduce ventilation rates during peak times provided that such periods are sufficiently short that higher ventilation rates at other times will provide adequate compensation. This will require continuous sensing of a key indicator of outdoor air quality, such as carbon monoxide.

The external air may need treatment before being used in the building. It is prudent to ensure that the air quality of the incoming air to a building meets at least the approved DEFRA outdoor Air Quality Standards (see www.defra.gov.uk/environment/airquality/aqs/index.htm). Those areas above the recommended maximum pollution are identified as an Air Quality Management Area. One of the pollutants often in excess is fine dust particulates from diesel traffic. Careful selection and proper installation of air filters can reduce the incoming pollution. Local authorities managing such areas require an 'environmental impact assessment' for new building work in these areas. See www.airquality.co.uk/archive/laqm/laqm.php for a map of Air Quality Management Areas.

8.4.3.1 Pollutants

A great deal of work has been done in recent years to identify and quantify the human health impacts of outdoor air pollutants. The most important pollutants in ambient air are generally considered to be airborne particles (e.g. PM10, PM2.5*), ozone, nitrogen dioxide, carbon monoxide and sulphur dioxide.

The recommendations of the UK Expert Panel on Air Quality Standards (EPAQS) have largely driven the development of 'air quality objective levels' within the UK's domestic Air Quality Strategy for ambient air[41], enforced through the Air Quality (England) Regulations 2000[37]. Because of the duties on local authorities to manage and control ambient air pollution, a considerable amount of measurement and modelling is conducted, especially in urban areas, which can be used to help determine whether particular buildings are in high pollution areas and therefore might warrant extra consideration regarding the quality of incoming air.

Conventional filters can remove particulates providing the quality of filtration matches the particulates to be removed. More sophisticated carbon adsorption filters are available for gaseous contaminants are and often used in smelly areas, for example in airport terminals to minimise the smell of paraffin from unburnt fuel for the planes.

8.4.3.2 Filtration strategy

If the main form of outdoor pollution is particulates, the pollution concentration of the incoming air can be reduced by passing the air through fabric or electrostatic filters. Reducing the concentration of gases and vapours requires additional equipment, usually in the form of adsorption filters, see CIBSE Guide B, chapter 3: *Ductwork*[42].

The grade of filtration required depends on the following factors:

— external pollution levels

— exposure limits for the protection of occupants or processes within the building

— degree of protection required for the internal surfaces of the building, air handling plant and air distribution system.

Table 8.1[43] gives the recommended classification for different applications. If high dust loadings are expected, it is wise to install coarse (i.e. G1 to G3) pre-filters upstream of the main filters. This will increase the replacement interval for the downstream higher efficiency (and therefore more costly) filters.

Air filters are designed to collect and retain particulate matter. This includes micro-organisms and mould spores. The presence of moisture in the vicinity of such filters can enable the micro-organisms to grow through the filter medium and contaminate the downstream air supply. Such material can also introduce unpleasant odours into the air intake. The design of filters must ensure that they are changed in accordance with the manufacturers' instructions and are located in a dry part of the ductwork.

Eurovent recommends the use of two filters in series. The 'pre-filter' (F5 or better) will stop the coarse particulates. The second filter (F7 or better) stops the bulk of the finer dust.

Table 8.1 Classification of filters as defined in BS EN 779[43]

Classification	Average arrestance, A_m (%)	Average efficiency, E_m (%)
G 1	$A_m < 65$	—
G 2	$65 \leq A_m < 80$	—
G 3	$80 \leq A_m < 90$	—
G 4	$A_m \geq 90$	—
F 5	—	$40 \leq E_m < 60$
F 6	—	$60 \leq E_m < 80$
F 7	—	$80 \leq E_m < 90$
F 8	—	$90 \leq E_m < 95$
F 9	—	$E_m \geq 95$

8.4.4 Indoor air quality

The World Health Organisation (WHO) has published guidelines for air quality[33], targeted at ambient air pollutants but intended to cover indoor air considerations where relevant. National governments and other bodies often take these values as a starting point when establishing their own health based air quality standards.

The values given in Table 8.2[33] are based on exposure to single airborne chemicals through inhalation alone. They do not take account of additive, synergistic or antagonistic effects or exposure through routes other than inhalation. The basis for derivation is different for each chemical, hence they cannot be compared with each other within an overall hierarchy of exposure effects. For each chemical the WHO guidelines provide information on typical sources, occurrence in air, typical concentrations reported, routes of exposure, metabolic processes, proven and suspected health effects and an evaluation of human health risks.

In 1998, the Department of Health's Committee on the Medical Effects of Air Pollutants (COMEAP) published its report *The Quantification of the Effects of Air Pollution on Health in the United Kingdom*[44], which has led to various

* PM10 and PM2.5 are particulate matter with aerodynamic diameters of 10 μm or less and 2.5 μm or less, respectively.

Table 8.2 Guideline values for individual substances

Substance	Averaging time	Guideline value concentration in air: By mass	By volume	Source/notes
Arsenic	Lifetime	—	—	Estimated 1500 deaths from cancer in population of 1 million through lifetime exposure of 1 $\mu g \cdot m^{-3}$ (WHO AQG: 1998; see note 1)
Benzene	1 year (running)	5 ppb	16.0 $\mu g \cdot m^{-3}$	AQOL; see note 2
	Lifetime	—	—	Estimated 6 deaths from cancer in population of 1 million through lifetime exposure of 1 $\mu g \cdot m^{-3}$ (WHO AQG: 1998; see note 1)
1,3-butadiene	1 year (running)	1 ppb	2.26 $\mu g \cdot m^{-3}$	AQOL; see note 2
Cadmium	Annual	—	5 $ng \cdot m^{-3}$	WHO AQG: 1998; see note 1
Carbon monoxide	15 min 30 min 1 hour 8 hour (running)	86 ppm 52 ppm 26 ppm 10 ppm	100 $mg \cdot m^{-3}$ 60 $mg \cdot m^{-3}$ 30 $mg \cdot m^{-3}$ 11.6 $mg \cdot m^{-3}$	WHO AQG: 1998; see note 1 WHOAQG: 1998; see note 1 WHO AQG: 1998; see note 1 WHO AQG: 1998; see note 1
Chromium	Lifetime	—	—	Estimated 40,000 deaths from cancer in population of 1 million through lifetime exposure of 1 $\mu g \cdot m^{-3}$ (WHO AQG: 1998; see note 1)
1,2-dichloroethane	24 hours	168 ppb	700 $\mu g \cdot m^{-3}$	WHO AQG: 1998; see note 1
Dichloromethane (methyl chloride)	24 hours	0.84 ppm	3 $mg \cdot m^{-3}$	WHO AQG: 1998; see note 1
Formaldehyde	30 min	80 ppb	100 $\mu g \cdot m^{-3}$	WHO AQG: 1998; see note 1
Hydrogen sulphide	30 min	5 ppb	7 $\mu g \cdot m^{-3}$	WHO AQG: 1998; see note 1
Lead	1 year	—	0.5 $\mu g \cdot m^{-3}$	Based on daily averages. AQOL; see note 2
MMVF–RC (man-made vitreous fibres; refractory ceramic fibres)	Lifetime	—	—	Estimated 40,000 deaths from cancer in population of 1 million through lifetime exposure of 1 $\mu g \cdot m^{-3}$ (WHOAQG: 1998; see note 1
Manganese	1 year	—	0.15 $\mu g \cdot m^{-3}$	WHO AQG: 1998; see note 1
Mercury	1 year	—	1 $\mu g \cdot m^{-3}$	WHO AQG: 1998; see note 1
Nickel	Lifetime	—	—	Estimated 380 deaths from cancer in population of 1 million through lifetime exposure of 1 $\mu g \cdot m^{-3}$ (WHO AQG: 1998; see note 1)
Nitrogen dioxide	1 hour (mean) 1 year	150 ppb 21 ppb	300 $\mu g \cdot m^{-3}$ 42 $\mu g \cdot m^{-3}$	AQOL; see note 2 Based on hourly averages. AQOL; see note 2
Ozone	8 hour	60 ppb	120 $\mu g \cdot m^{-3}$	WHO AQG: 1998; see note 1
PM_{10} (particulate matter < 10 μm diameter)	24 hour	—	50 $\mu g \cdot m^{-3}$	99th. percentile (running). AQOL; see note 2
Radon	Lifetime	—	—	Estimated 36 deaths from cancer in population of 1 million through lifetime exposure of 1 $Bq \cdot m^{-3}$ (WHO AQG: 1998; see note 1)
Styrene	1 week	60 ppb	260 $\mu g \cdot m^{-3}$	WHO AQG: 1998; see note 1
Sulphur dioxide	15 min 24 hour 1 year	100 ppb 46 ppb 19 ppb	270 $\mu g \cdot m^{-3}$ 125 $\mu g \cdot m^{-3}$ 50 $\mu g \cdot m^{-3}$	99.9th. percentile. AQOL; see note 2 WHO AQG: 1998; see note 1 WHO AQG: 1998; see note 1
Tetrachloroethylene	24 hour	—	250 $\mu g \cdot m^{-3}$	WHO AQG: 1998; see note 1
Toluene	1 week	68 ppb	260 $\mu g \cdot m^{-3}$	WHO AQG: 1998; see note 1
Trichloroethylene	Lifetime	—	—	Estimated 4 deaths from cancer in population of 10 million through lifetime exposure of 1 $\mu g \cdot m^{-3}$ (WHO AQG: 1998; see note 1)

Notes:

(1) From WHO *Air Quality Guidelines for Europe* (1998), based on summary document of final consultation meeting held at WHO European Centre for Environment and Health, Bilthoven, Netherlands, 28–31 October 1996. These should be used along with the rationale given in the relevant chapters of the WHO publication[33].

(2) Air quality objective levels (AQOLS) are taken from the Air Quality Regulations 1997[37]. These levels were to be achieved outdoors throughout the UK by 2005. Local authorities have been tasked under Part IV of the Environment Act 1995[38] to review air quality within their areas to achieve these levels. The interpretation to the schedule in the Regulations defines how averaging should be achieved.

Table 8.3 Increase in ill health attributed to each 10 μg/m^3 increase in daily mean outdoor PM10

Effect	Increase in PM10 / %
Exacerbation of asthmatic attacks	3
Increased broncho-dilator use	3
Hospital admissions	1.9
Increase in lower respiratory problems	3
Increase in coughing	1

calculations and other assessments of the heath impacts of pollutants in ambient air, notably of PM10 (see Table 8.3). There is no known safe threshold value for PM10 and so the maximum permitted standard progressively reduce sin future years. The Air Quality Standard for 2004 is an annual mean of 50 μg/m^3 with 35 permitted exceedances per year.

In spite of all the regulatory and other activity surrounding pollution of outdoor air, there is very limited actual or considered regulation and control of indoor air quality and its determinants. Building Regulations ensure adequate ventilation on the basis that the outdoor air is clean, but there are currently no standards covering the quality of indoor air with respect to specific toxic pollutants derived from sources within buildings. This situation may change, however, and serious consideration is now being given to the development of guidelines and/or detailed guidance covering indoor pollutants[(45)]. The only relevant standards that do exist for specific application to indoor air are those that relate to occupational exposure to known hazardous substances[(34)], but these are meant to apply to a 'healthy' worker population (i.e. excluding the old, the sick and other potentially more vulnerable individuals in the general population) and to a typical working day rather than 24-hour exposure.

8.4.5 Indoor/outdoor pollution ratio

In the absence of a relevant indoor source, the concentration of a pollutant inside a building is directly related to the concentration outside. The ratio between indoor and outdoor levels will depend on the amount and nature of ventilation (and/or the 'leakiness' of the building) and on the reactivity and other physico-chemical characteristics of the substance in question. Thus, for example, without an indoor source, ozone values tend to be low inside buildings because it is a very reactive gas; indoor/outside (I/O) ratio will therefore be very low at around 0.3. Carbon dioxide, on the other hand, is an unreactive gas and remains unchanged on entering the building and then adds to the carbon dioxide generated by the breathing of the occupants within the building. Nitrous oxide concentration remains unchanged. Nitrogen dioxide I/O is 0.7[(46)].

Generally speaking, I/O values are around 0.5 for most pollutants. However, very potent indoor sources of pollutants can be present within a building (see section 8.4.6) and it is not unusual in dwellings for concentrations of volatile organic substances (found, for example, in solvents, glues and paints) to be around 10 times higher inside than outside[(47)].

As well as the rate of supply of air to a building, air treatment (e.g. filtering) can obviously have an impact on I/O concentration ratios, and also there can be important and efficient 'sinks' for air pollutants within a building. The ratio of supply rate to the building volume determines the time taken for the indoor pollution to build up. High ventilation rates supplied to small volume buildings, for example with low ceilings, can reach the maximum indoor pollution concentration much faster than low ventilation rates to spacious buildings.

8.4.6 Indoor sources

A building can contain many sources of exposure to air pollutants, both chemical and biological, some of which are very potent. Important sources of pollutants inside buildings include:

— building materials including sealants, adhesives, paint

— cleaning materials, solvents and other consumer products

— furnishings and fabrics

— furniture

— equipment such as photocopiers, printers, and document binders

— gas cookers, heaters and other un-flued fuel-burning appliances

— glues

— house dust mites

— moulds and bacteria

— pesticide products

— pets

— tobacco smoking

— radon seeping in from the ground.

Important pollutants released from these sources are:

— asbestos and man-made mineral fibres particularly in old buildings; although safe while undisturbed, specialist knowledge is needed for safe removal

— bacteria and mould spores particularly in neglected or damp buildings.

— carbon monoxide from neglected unflued appliances such as paraffin heaters

— chlorinated organic compounds and organophosphates from solvents, aerosol products, foaming urethane and degreasing.

— dust mite and pet allergen from moist damp houses and pets

— formaldehyde from insulation, packaging, and compressed wood products

— nitrogen dioxide (and other oxides of nitrogen) from cooking and other unflued combustion devices.

— particles (PM10 and smaller) from photocopiers, combustion products such as cooking, cigarette smoking and burning candles and from disturbed dust from the floor and other surfaces.

- polycyclic aromatic hydrocarbons (PAHs) from fires, vehicle exhaust and coal tar
- volatile organic compounds (VOCs) from people, building materials and furnishings
- ozone from photocopiers, laser printers, and electric motors
- radon (applies to a small number of zones in Britain).

In addition there are the effluents produced by occupants themselves, notably carbon dioxide (CO_2) but also VOCs and various other compounds produced by, or present on, human beings.

The actual pollutants and pollutant sources present in a building will largely be determined by the type of building and its usage. For example, there is likely to be a greater diversity (and certainly a different range) of pollutants and pollutant sources in a home than in an office.

Many of these substances become irritants if concentrations are sufficiently high. Little is known about the possible additive or synergistic effects that may occur when a number of substances combine at low concentrations. This is particularly so in the case of ozone and VOCs.

8.4.7 Health effects of common indoor pollutants

In non-industrial buildings, as with pollutant sources, the range of compounds present in the indoor air is likely to be greatest in dwellings. The most significant compounds in indoor air with respect to overall health impacts are probably carbon monoxide, house dust mites, pet allergens, moulds, formaldehyde, nitrogen dioxide (and other oxides of nitrogen), and possibly particles. While the significance of particulate matter in ambient air is undisputed, there is an unresolved question about the relative toxicities of vehicle derived and other airborne particles, such as may be generated by cooking and heating appliances inside buildings. Carbon dioxide at low concentrations is typically used as a marker for indoor air quality and for ventilation requirements, reflecting the pollutant loading from exhalation by the occupants. The maximum concentration is 5000 ppm in working conditions[34] and it can be an asphyxiant at extremely high concentrations. The role of VOCs in causing ill health is somewhat disputed, although the measurement of total VOCs has in the past been used as a general marker for indoor air quality, and VOCs have been implicated both in 'sick building syndrome' and so-called multiple chemical sensitivity.

The prevention of discomfort from tobacco smoke may be achieved by the prohibition of smoking in rest rooms and rest areas or by providing separate rooms for smokers and non-smokers[4]. There is no known safe limit for continued exposure to tobacco smoke and former smokers can be particularly sensitive to throat and lung irritation from such smoke.

8.4.8 Microbiological contaminants

Increasingly, attention is moving away from 'chemical' pollutants to the role of microbiological contaminants of indoor air. This applies both to the home, where dampness can cause severe infestation, and to commercial buildings with mechanical ventilation, which can harbour moulds and bacteria. Moulds in particular are a current cause for concern, with respect both to their allergenicity and the production of toxic metabolites.

8.4.9 Ventilation systems and health

An American study[48], see Table 8.4, shows that the risk of reported respiratory symptoms tripled if debris lay in the air intake, and if there was poor drainage from the air handling unit condensate pans. The risk of respiratory symptoms more than doubled if the fresh air inlet was within 8 m of an exhaust air outlet, a toilet exhaust ventilator outlet or a rubbish store. Dirty or badly fitting air filters doubled the risk. The air inlet must be placed well away from recognised polluted sources such as toilet vents, chimneys, vehicle exhaust pipes or carry-over from evaporative cooling towers[40]. The ductwork must be clean[4,7,22,49].

A Swedish study of published research[50] showed that higher ventilation rates than previously proposed were welcomed by the occupants, but only if the ventilation system was designed correctly and properly maintained. Ventilation rates below 25 L/s per person increased the risk of sick building syndrome, increased short term sick leave and decreased perceptions of productivity. The literature studied also indicated that occupants of buildings with air conditioning systems may have an increased risk of 'sick building' symptoms compared to naturally or mechanically ventilated buildings and that negligent design, inadequate maintenance and malfunctioning air conditioning systems contribute to increased prevalence of sick building symptoms.

It is likely that the cause of sick building syndrome is multi-factorial. Researchers have identified a statistically significant correlation between symptom prevalence and many different and unrelated factors. It would appear that

Table 8.4 Increase in reported respiratory symptoms by office occupants[48]

Situation	Increase in reported respiratory symptoms / %
Debris lies in the air intake	310
Poor drainage from condensate pans	300
Ductwork not cleaned	280
Air inlet within 8 m of an exhaust ventilator	240
Air inlet within 8 m of an exhaust ventilator	240
Air inlet within 8 m of a standing water	230
Air inlet within 8 m of a toilet exhaust ventilator	220
Moisture in the ductwork	220
Filters not secure	220
Air inlet within 8 m of a rubbish container	200
Air inlet within 8 m of vehicular traffic	190
Filters not clean	190

if environmental conditions are within the comfort limits set out in this Guide, then the risk of occupant dissatisfaction and sick building syndrome is reduced, though not eliminated.

8.4.10 Determination of required outdoor air supply rate

8.4.10.1 General

Ventilation requirements for a wide range of building types are summarised in chapter 1, Table 1.5. Detailed information on specific applications is given in CIBSE Guide B[42], chapter 2: *Ventilation and air conditioning*. For some industrial applications outdoor air may be required both to dilute specific pollutants and to make up the air exhausted through local extract ventilation systems (see CIBSE Guide B, chapter 3: *Ductwork*. Specialist advice should be sought in dealing with toxic and/or high emission pollutants.

8.4.10.2 Prescribed outdoor air supply rates

For applications in which the main odorous pollutants arise due to human activities, e.g. body odour, it is possible to supply a quantity of outdoor air based on the number of occupants in a given space. If smoking is prohibited, as is increasingly the case, then the recommended outdoor air supply rates given in chapter 1, Table 1.5, apply.

Spaces in which smoking is permitted should be regarded as 'smoking rooms' and an outdoor air supply rate of 45 $L \cdot s^{-1}$ per person is suggested for such rooms. However, it should be noted that this recommendation aims only to reduce discomfort and does not ensure health protection.

8.5 Visual environment

See also chapter 1, section 1.8.

8.5.1 Legislation

The three general lighting requirements are as follows:

— Every workplace must have suitable and sufficient lighting (i.e. at an appropriate and even illuminance, free from shadow, direct and reflected glare, clean, well maintained and flicker free and without stroboscopic effects).

— The lighting should, as far as is practicable, be natural light.

— Emergency lighting must be provided when workers would be exposed to danger in the event of failure in the electric lighting. This should be designed to illuminate the escape route, all hazards or hazardous tasks, and the fire fighting equipment. It also can mean providing sufficient lighting to enable processes to be closed down safely at times of power failure.

Illuminance for safety while the normal lighting system is operating is usually satisfactory if the illuminance required for performance and/or pleasantness is maintained. Suitable values, for a wide range of applications, are given in the Society for Light and Lighting (SLL) *Code for lighting*[51]. However, in many instances there are legally enforced minimum standards and the Workplace (Health, Safety and Welfare) Regulations 1992[1] and the Provision and Use of Work Equipment Regulations 1992[5] require that provision be made for 'suitable and sufficient lighting', with a preference for daylight, when available.

The Health and Safety Executive Guide (HSE), *Lighting at work*[52], indicates minimum lighting standards for safety in the workplace. These standards should not be confused with those given in the *Code for lighting*[51], which are the lighting levels required in order to perform tasks quickly and accurately.

Although the main purpose of legislation is to ensure safety, there are situations where health and welfare are the primary considerations. For locations where food is prepared (other than in agriculture), the Food Hygiene (General) Regulations 1995[53] require 'suitable and sufficient means of lighting' in order that proper cleanliness can be maintained and the local authority (which is responsible for enforcement of these regulations) should be consulted regarding the specific standards which apply.

The Health and Safety (Display Screen Equipment) Regulations 1992[54,55] provide protection for users of display screens and keyboards, and require that 'satisfactory lighting conditions' be provided, allowing for background conditions, glare, reflections and the visual requirements of users. Figure 8.6 illustrates the potential for reflections in a display screen[56].

In the cases of schools and hospitals, minimum lighting standards are specified by the Department for Education and Skills and the Department of Health, respectively. Guidance is also provided in CIBSE/SLL Lighting Guides LG5[57] and LG7[56].

The above summary of mandatory requirements is not comprehensive and identifies only a more the most important legislation concerning the provision of lighting for safety. There may be other, more demanding requirements specific to particular applications and it is essential that the relevant authorities (e.g. local authority, HSE Inspectorate etc.) be consulted at an early stage in the design process.

The HSE specifies illuminances for a range of working conditions in terms of minimum and average values, with average values ranging from 20 lux for circulation spaces, 50 lux for the movement of people in hazardous zones, 100 lux for work requiring limited detail such as kitchens, 200 lux for office work and 500 lux for work requiring perception of detail such as drawing offices and electronics. The HSE also specifies the maximum ratio of illuminance between the work areas and those areas adjacent. Guidance also specifies the angular exclusion zone for different types of lamp to avoid discomfort and disability glare[52], see Figure 8.7. Stroboscopic effects produced by discharge lamps operating on an AC supply can be reduced by wiring adjacent luminaries to different phases of the three phase supply of by the use of high frequency control gear. Such hazards are considered in SLL Lighting Guide LG1[58]. Emergency lighting design is outlined in BS 5266-4[59].

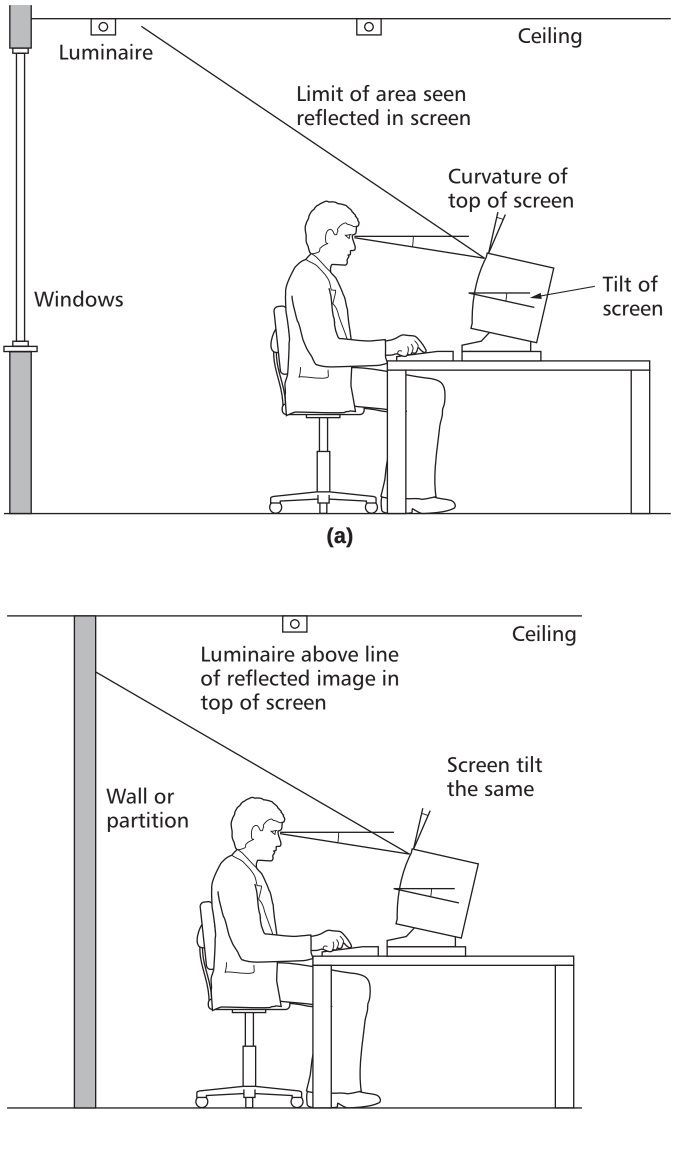


Figure 8.6 Typical arrangement for a display screen; (a) reflections of luminaires and windows far back in office are possible, (b) smaller office in which there is less chance of seeing luminaires reflected in the screen

8.5.2 Lighting and health

8.5.2.1 Light as radiation

Light can affect health simply as electromagnetic radiation, regardless of whether or not it stimulates the visual system. In very large amounts, light as radiation can cause tissue damage to the eye and the skin[60]. The Illuminating Engineering Society of North America has developed a system for assessing light sources in terms of the risk of tissue damage[61]. For most people, incandescent and fluorescent lamps as used in commercial and domestic situations pose no risk but some high wattage discharge and tungsten halogen lamps may do. Further information can be found elsewhere[61,62].

Unfortunately, there are some groups of people who are much more sensitive than the average to light as radiation. One such group consists of very premature babies, particularly those weighing less than 1000 g at birth. These infants have eyes that are still developing and exposure to light is believed to be involved in the retinopathy of

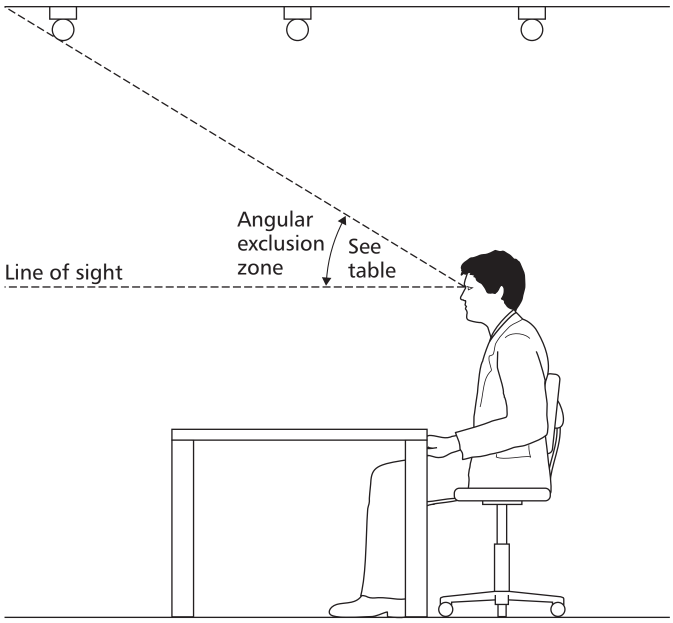


Figure 8.7 Avoidance of direct glare from lamps

prematurity, a visual disorder that can permanently damage the retina of such babies. Proposals to limit the light exposure of babies in neonatal intensive care units have been made[63].

Another population with a problem with exposure to light are post-operative cataract patients who have had their lens removed, i.e., patients who are aphakic. Such patients are much more likely to suffer photochemical retinal damage due to short wavelength visible and ultraviolet radiation exposure than are people with their biological lens intact, unless they are fitted with an ultraviolet-absorbing, intraocular lens.

8.5.2.2 Light operating through the visual system

Light is necessary for the visual system to operate but if used in the wrong way it can be injurious to health. The most common effect of lighting operating through the visual system on health is colloquially known as eyestrain. Eyestrain is the result of prolonged experience of lighting conditions that cause discomfort. The symptoms of eyestrain are irritation of the eyes, evident as inflammation of the eyes and lids; breakdown of vision, evident as blurring or double vision; and referred effects, usually in the form of headaches, indigestion, and giddiness.

Lighting conditions which have been shown to lead to eyestrain are inadequate illuminance for the task, excessive luminance ratios between different elements of a task and lamp flicker, even when it is not visible.

Everyone is likely to experience eyestrain in poor lighting conditions but there are some groups who are particularly sensitive to lighting conditions. One such group are those who suffer from photoepilepsy. Given fluctuating light of the right frequency, covering a large area and at a high percentage modulation, these individuals can be driven into a seizure. The frequency to which people with photoepilepsy are most sensitive is about 15 Hz, although about 50 percent still show signs of a photoconvulsive response at 50 Hz.

A larger but related group are those who suffer from migrane ('migraineurs'). The exact cause of a migraine is not known but what is known is that migraineurs are more sensitive to light than people who do not experience migraine, even when they are headache-free. This means that migraineurs are much more likely to experience glare from luminaires and to complain about high light levels. In addition, migraineurs are likely to be hypersensitive to visual instability, no matter whether it is produced by fluctuations in light output from a light source, or by large area, regular patterns of very different reflectances. One way to ensure that light output fluctuations do not cause trouble is either to use light sources that are inherently low in modulation, such as the incandescent lamp, or, if high modulation discharge light sources are to be used, to operate them from high frequency control gear. There is some evidence to suggest that changing from magnetic to electronic control gear for fluorescent lamps does little for most people but does reduce the frequency of headaches and eyestrain for people who frequently have such symptoms.

Another group who can be expected to be sensitive to fluctuations in light output are people who suffer from autism. The symptoms of autism are repetitive activities, stereotyped movements, resistance to changes in the environment and the daily routine and unusual responses to sensory experiences. Observations of autistic children have demonstrated that repetitive behaviour occur more frequently under fluorescent lighting than under incandescent lighting. This suggests that people with autism may also benefit from the use of electronic control gear for fluorescent lamps. Care should also be taken to avoid lighting control systems that change light levels suddenly.

8.5.2.3 Light operating through the circadian system

Circadian rhythms are a basic part of life. The human circadian system involves three components:

— an internal oscillator

— a number of external oscillators that can entrain the internal oscillator

— a messenger hormone, melatonin, which carries the internal 'time' information to all parts of the body through the bloodstream.

The light–dark cycle is one of the most potent of the external stimuli used for entrainment.

The sleep–wake cycle is the most obvious human circadian rhythm so it is hardly surprising that exposure to bright light at the right time can be used to treat some sleep disorders involving the timing and duration of sleep. Exposure to bright light is also a useful means of stabilising the rest–activity cycle of people with Alzheimer's disease and of relieving the symptoms of seasonal affective disorder[60].

8.5.2.4 Light as a purifier

Ultraviolet radiation has the ability to destroy many types of viruses, bacteria, moulds and yeasts, some of which have the potential to damage human health. The mechanism of destruction is the absorption of UV radiation by the DNA molecule of the target organism. This absorption produces mutation or cell death, both of which stop the organism from multiplying. The light source used to provide the ultraviolet radiation is an electric discharge passed though a low pressure mercury vapour, the vapour being enclosed in either a special glass or a quartz tube that transmits ultraviolet radiation. Ninety-five percent of the energy emitted by these germicidal lamps is at a wavelength of 253.7 nm. The effectiveness of this radiation in destroying micro-organisms depends on many parameters, including the susceptibility of the specific organism which is related to the thickness of the cell wall, the spectrum of the radiation received and the radiant exposure.

Ultraviolet radiation has been used to purify air and liquids. Interest in air purification by ultraviolet radiation has grown in recent years with the emergence of drug-resistant strains of airborne disease, such as tuberculosis. Germicidal lamps are usually placed in one of two locations. Where air conditioning is used, the lamps can be placed in the ductwork. For this location, exposure times are short because of the high air velocity in the duct, so a high irradiance is required. Alternatively, the germicidal lamps can be placed inside the occupied space. This poses a problem because exposure to ultraviolet radiation can lead to photokeratitis and skin damage. There are two solutions to this problem. Where the ceiling height allows (>2.9 m) the lamps can be installed in luminaires that confine the ultraviolet radiation to the air volume above the occupants' heads, the surfaces directly irradiated having a reflectance at 253.7 nm of less than 0.05. Of course, this method relies on sufficient air circulation to move all the air in the occupied space through the irradiated volume frequently. If the ceiling height does not allow overhead purification, then the some form of local air circulation that brings the air of the occupied space through an enclosed fitting containing the germicidal lamp is possible.

8.6 Electromagnetic effects

See also chapter 1, section 1.11.

Both electric and magnetic fields arise from the generation, transmission and use of electricity. Electric fields are related to the voltage and measured in terms of volts per metre. At mains frequency the electric fields do not penetrate the body but do charge up its surface. In the highest fields hair may move and small electric shocks may be observed. Most people will not experience shocks below 25 kV/m. Electric fields can be reduced by metal shielding. Magnetic fields are associated with the current and are measured in units of tesla (T). Alternating magnetic fields cause electric currents to circulate within the body. In very high magnetic fields this current can cause flashes of light in the eye. Magnetic fields pass through most materials. Both fields reduce rapidly with distance. Exposure guidelines propose an investigation whenever the electric field at 50 Hz reaches 12 kV/m or the magnetic field reaches 1.6 mT[64]. Such fields are only found in specialist medical or industrial equipment.

However, since the 1960s there has been concern expressed regarding the possible effects on health of extremely low frequency electromagnetic fields (i.e. below 300 Hz). Some reports have suggested that exposure to these fields, such as might be experienced by those living near high voltage overhead power lines, increases the risk of cancer, particularly leukaemia, especially amongst children. Other studies have raised the possibility that 'electrical' occupations, such as those that entail prolonged proximity to visual display terminals, result in an increased risk of illness.

A review of these studies reveals that all suffer from methodological or other shortcomings but it is not clear whether these are sufficient to explain the results. Experiments with animals have produced conflicting and confusing results, and their relevance to the effect on humans is difficult to assess. No plausible mechanism for carcinogenesis due to exposure to electrical or magnetic fields has yet been deduced. It has been established that such fields affect the function of cardiac pacemakers but this is unlikely to be a hazard at the field strengths normally encountered and most modern pacemakers are designed to cope with high field strengths.

Therefore, current evidence does not permit firm conclusions to be drawn on the relationship between electromagnetic fields and physiological or psychological effects on humans. Until the situation is clarified by further research and provided that no significant cost penalties result, it is suggested that potential fields be minimised. Often this can be achieved by ensuring that line and return cables are in close proximity, as is usual practice for mains wiring.

The NPRB[65] has adopted the recommendations of the International Committee of Non-Ionising Radiation Board (ICNIRB) for limiting exposures for electromagnetic (EMFs) effects from radiation in the frequency range 0–300 GHz. Generally occupational exposures concern healthy adults under controlled conditions. These conditions include the opportunity to apply engineering and administrative measures and, where necessary and practicable, to provide personal protection. For members of the public such controls do not generally exist and individuals of varying ages respond differently to exposures to EMFs. 'The public' includes very young infants, the elderly, and those on medication whose health status may be less robust. For these reasons the exposure restrictions for the public are 20% of those criteria for the working population.

8.6.1 Air ionisation

It has been suggested that the ion balance of the air is an important factor in human comfort in that negative ions tend to produce sensations of freshness and well-being and positive ions cause headache, nausea and general malaise. Present evidence on the effects of air ions and, in particular, the effectiveness of air ionisers is inconclusive and hence no design criteria can be established.

8.6.2 Static electricity

Shocks due to static electricity are unlikely with humidities above 40% RH; see section 8.3.3.

8.7 Noise and vibration

See also chapter 1, sections 1.9 and 1.10.

8.7.1 Hearing damage

Exposure to high noise levels, such as may occur in a plant room, can cause temporary or permanent hearing damage. Where workers are exposed to high levels of noise, the noise levels must be assessed by a qualified person. The Noise at Work Regulations 2005[66] identify two levels of 'daily personal noise exposure' (measured in a manner similar to $L_{Aeq,T}$) at which actions become necessary. These levels are 80 dBA for the lower level and 85 dBA for the higher level, corresponding to advisory and compulsory requirements. In addition, for impulse noise, there is a lower peak level limit of 112 Pa and a higher peak level limit of 140 Pa. These peak action levels control exposure to impulse noise. Suppliers of machinery must provide noise data for machines likely to cause exposure to noise above the action levels.

8.7.2 Vibration

8.7.2.1 General

In the context of building services installations, vibrations arise from reciprocating machines or from unbalanced forces in rotating machines. The vibration is often most noticeable during machine start-up (i.e. low-frequency operation), during which some machines pass through a critical (resonant) speed before reaching their normal operating condition. Vibration associated with start-up may not be important if the machine operates for long periods, since that condition occurs only infrequently. However, machines which switch on and off under thermostatic control, for example, may require special precautions.

Vibrations transmitted from machines through their bases to the building structure may be heard, and sometimes felt, at considerable distances from the plant and, in extreme cases, even in neighbouring buildings. Therefore, adequate isolation is important in those cases where vibration is expected. Vibration isolators must be chosen to withstand the static load of the machine as well as isolate it from the structure. Efficient vibration isolation is the preferred way of controlling structure-borne noise, which occurs when vibration transmitted to building surfaces is re-radiated as noise. Structure-borne noise is enhanced when the excitation frequency corresponds with a structural resonance frequency, causing unexpected noise problems.

8.7.2.2 Response of the human body to vibration

Vibrating motion of the human body can produce both physical and biological effects. The physical effect is the excitation of parts of the body and under extreme conditions physical damage may result. Building vibration, which is at a much lower level, may affect the occupants by reducing both quality of life and working efficiency. Complaints about vibration in residential situations are likely to arise from occupants when the vibration levels are only slightly greater than the threshold of perception.

The levels of complaint resulting from vibration, and acceptable limits for building vibration, depend upon the characteristics of the vibration and the building environment, as well as individual response. These factors are incorporated in guidance given in BS 6472[67],which gives magnitudes of vibrations below which the probability of complaints is low

References

1 The Workplace (Health, Safety and Welfare) Regulations 1992 Statutory Instruments 1992 No. 3004 (London: Her Majesty's Stationery Office) (1992)

2 The Fuel and Electricity (Heating) (Control) Order 1974 Statutory Instruments 1974 No. 2160 (London: Her Majesty's Stationery Office) (1974)

3 The Fuel and Electricity (Heating) (Control) (Amendment) Order 1980 Statutory Instruments 1980 No. 1013 (London: Her Majesty's Stationery Office)

4 *Workplace health, safety and welfare — Workplace (Health, Safety and Welfare) Regulations 1992 — Approved Code of Practice and Guidance* Health and Safety Commission Legislation series L24 (Sudbury: HSE Books) (1994)

5 The Provision and Use of Work Equipment Regulations 1992 Statutory Instruments 1992 No. 2932 (London: Her Majesty's Stationery Office) (1992)

6 *Safe hot water and surface temperatures* NHS Estates Health Guidance Note (London: The Stationery Office) (1998)

7 *Legionnaires disease the control of legionella bacteria in water systems — Approved Code of Practice and Guidance* Health and Safety Commission Legislation series L8 (Sudbury: HSE Books) (2001)

8 The Care Homes Regulations 2001 Statutory Instruments 2001 No. 3965 (London: The Stationery Office) (2001)

9 The Children's Homes Regulations 2001 Statutory Instruments 2001 No. 3967 (London: The Stationery Office) (2001)

10 *Health and safety in care homes* Health and Safety Guidance. HSG220 (Sudbury: HSE Books) (2001)

11 *Health impact of low indoor temperatures* (Copenhagen: World Health Organisation) (1987)

12 Raw G J *Building regulation health and safety* BRE Report BR417 (Garston: BRE) (2001)

13 Seppänen O, Fisk W J and Faulkner D Cost benefit analysis of the night-time ventilative cooling *Proc. Healthy Buildings Conference, Singapore 2003* **3** 394–399 (2003)

14 McIntyre D A *Indoor Climate* (London: Applied Science) (1980)

15 Moritz A R and Henriques F C Studies of thermal injury — Part 2: The relative importance of time and surface temperature in the causation of cutaneous burns *Am. J. Pathology* **23** 695–720 (1947)

16 DIN 1946-2: 1994-2: *Ventilation and air conditioning technical health measurements* (Berlin: Deutsches Institut fur Normung) (1994)

17 Arlain L G Water balance and humidity requirements of house dust mites *Experimental and Applied Acarology* **16** 15–35 (1992)

18 Fang L, Clausen G, Fanger P O Impact of temperature and humidity on perception of indoor air quality *Indoor Air* **8** (2) 80–90 (1998)

19 Nilsson S E G, Andersson L Contact lens wear in dry environments *Acta Opthalmol.* **65** 221–225 (1986)

20 Brundrett G W A review of the factors influencing electrostatic shocks in offices *J. Electrostatics* **2** 295–315 (1977)

21 *Ventilation* Building Regulations 1991 Approved Document F (London: The Stationery Office) (1994)

22 *How to deal with sick building syndrome Guidance for employers, building owners and building managers* Health and Safety Guidance HSG132 (Sudbury: HSE Books) (1995)

23 *General ventilation in the workplace — Guidance for employers* Health and Safety Guidance HSG202 (Sudbury: HSE Books) (2000)

24 *Conservation of fuel and power* The Building Regulations 2000 Approved Document L1/L2 (London: The Stationery Office) (2002)

25 *HVAC strategies for well-insulated airtight buildings* CIBSE TM29 (London: Chartered Institution of Building Services Engineers) (2002)

26 *Guidelines for environmental design in schools* Building Bulletin for schools 87 (2nd. edn.) (London: Department for Education and Skills) (2003)

27 Cain W S *Perceptual characteristics of nasal irritation* (New Haven: John B Pierce Foundation Laboratory) (1989)

28 Cain W S, See L C and Tosun T Irritation and odour from formaldehyde: chamber studies *Proc. ASHRAE Conf. IAQ '86: Managing the Indoor Air for Health and Energy Conservation, Atlanta, USA* (1986)

29 Gunnarsen L and Fanger P O Adaptation to indoor air pollution *Healthy Buildings 88* (ed. Berglund B and Lindvall T) **3** 157–167 (Stockholm: Swedish Council for Building Research) (1988)

30 Clausen G H, Nielsen K S, Sahin F and Fanger P O Sensory irritation from exposure to environmental tobacco smoke *Indoor Air 87* **2** 52–56 (1987)

31 Berglund B The role of sensory reactions as guides for non-industrial indoor air quality (preprint) *Proc. Conf. American Industrial Hygiene, St Louis, USA* (May 1989)

32 ANSI/ASHRAE Standard 62-2001: *Ventilation for acceptable indoor air quality* (Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers) (2001)

33 *Air Quality Guidelines for Europe* European Series No. 91 (Copenhagen: World Health Organization) (2000)

34 *Occupational exposure limits* EH40 (London: The Stationery Office) (published annually)

35 Health and Safety at Work, etc. Act 1974 Elizabeth II. Chapter 37 London: Her Majesty's Stationery Office) (1974)

36 Control of Substances Hazardous to Health Regulations 2002 Statutory Instruments 2002 No. 2677 (London: The Stationery Office) (2002)

37 The Air Quality (England) Regulations 2000 Statutory Instruments 2000 No. 928

38 Environment Act 1995 Elizabeth II. Chapter 25 (London: The Stationery Office) (1995)

39 *Annual digest of environmental statistics* (London: The Stationery Office) (published annually)

40 *Minimising pollution at air intakes* CIBSE TM21 (London: Chartered Institution of Building Services Engineers) (1999)

41 *The national air quality strategy for England, Scotland and Northern Ireland working together for clean air* Cm. 4548 Northern Ireland Assembly papers Session 1 (1999–2000) NIA 7 Scottish Executive papers Session 2 (2000) SE 2000/3 (London: The Stationery Office) (2000)

42 *Heating, ventilating, air conditioning and refrigeration* CIBSE Guide B (London: Chartered Institution of Building Services Engineers) (2001–2)

43 BS EN 779: 2002: *Particulate air filters for general ventilation. Determination of the filtration performance* (London: British Standards Institution) (2002)

44 *Quantification of the effects of air pollution on health in the United Kingdom* Department of Health Committee on the Medical Effects of Air Pollutants Great Britain (London: The Stationery Office) (1998)

45 Harrison P T C *Health effects of indoor air pollutants* in Hester R E and Harrison R M (eds.) *Air Pollution and Health* 101–125(Letchworth: The Royal Society of Chemistry) (1998)

46 Kukadia V, Hall D J, Walker S, Sharples H, Doocey D, Gadian A and Griffiths R F *The effect of external pollution in the context of low energy architecture* BRE Report 81460 for the DETR (Garston: BRE) (April 2000)

47 Berry R W, Brown V M, Coward S K D, Crump D R, Gavin M, Grimes C P, Higham D F, Hull A V, Hunter C A, Jeffrey I G, Lea R G, Llewellyn J W and Raw G J *Indoor Air Quality in Homes: Part 1: BRE Indoor Environment Study* BRE Report BR 299 (London: Construction Research Communications) (1996)

48 Sieber W K, Petersen M R, Staynor L T, Malkin R, Mendell M J, Wallingford K M, Wilcox T G, Crandall M S, and Reed L Associations Between Environmental Factors and Health Conditions *Proc. Indoor Air* **2** 901–906 (1996)

49 *Minimising the risk of Legionnaires' disease* CIBSE TM13 (London: Chartered Institution of Building Services Engineers) (2002)

50 Wargocki P, Sundell J, Bischof W, Brundrett G, Fanger P O, Gyntelberg F, Hanssen S O, Harrison P, Pickering A, Seppänen O and Wouters P Ventilation and health in nonindustrial indoor environments *Indoor Air* **12** 113–128 (2002)

51 *Code for lighting* (London: Society of Light and Lighting) (2004)

52 *Lighting at work* Health and Safety Guidance HSG38 (Sudbury: HSE Books) (1998)

53 The Food Safety (General Food Hygiene) Regulations 1995 Statutory Instruments 1995 No. 1763 (London: Her Majesty's Stationery Office) (1995)

54 Health and Safety (Display Screen Equipment) Regulations 1992 Statutory Instruments 1992 No. 2792 (London: Her Majesty's Stationery Office) (1992)

55 *The law on* VDUs *— an easy guide* Health and Safety Guidance HSG90 (Sudbury: HSE Books) (2003)

56 *Office lighting* SLL Lighting Guide LG7 (London: Society of Light and Lighting) (2005)

57 *Lecture, teaching and conference spaces* CIBSE Lighting Guide LG5 (London: Chartered Institution of Building Services Engineers) (1991)

58 *Industrial lighting* SLL Lighting Guide LG1 (London: Society of Light and Lighting) (2002)

59 BS 5266-4: 1999: *Emergency lighting. Code of practice for design, installation, maintenance and use of optical fibre systems* (London: British Standards Institution) (1999)

60 Boyce P R *Human factors in lighting* (2nd edn.) (London: Taylor and Francis) (2003)

61 *Recommended practice for photobiological safety for lamps and lamp systems* ANSI/IESNA RP-27-96 (New York NY: Illuminating Engineering Society of North America) (1996)

62 Kohmoto K Evaluation of actual light sources with proposed photobiological lamp safety standard and its applicability to guide on lighted environment *Proc. CIE 24th Session, Warsaw, 1999* (Vienna: Commission Internationale de l'Éclairage) (1999)

63 Bullock J and Rea M S Lighting for neonatal intensive care units: some critical information for design *Lighting Res. Technol.* **28** 189–198 (1996)

64 *Restrictions on exposure to static and time varying electromagnetic fields 0–300 GHz* (Didcot: National Radiological Protection Board) (1999)

65 Advice on the limiting exposure to electromagnetic fields from 0–300 GHz *NRPB* **15** (2) 5–35 (2004)

66 Noise at Work Regulations 2005 Statutory Instruments 2005 No. 1643 (London: The Stationery Office) (1989)

67 BS 6472: 1992: *Guide to evaluation of human exposure to vibration in buildings (1 Hz to 80 Hz)* (London: British Standards Institution) (1999)

Index