Table 6 Duct Leakage Classification[a]

Duct Type	Sealed[b,c] Predicted Leakage Class C_L	Sealed[b,c] Leakage Rate, L/(s·m²) at 250 Pa	Unsealed[c] Predicted Leakage Class C_L	Unsealed[c] Leakage Rate, L/(s·m²) at 250 Pa
Metal (flexible excluded)				
Round and flat oval	4	0.14	42 (8 to 99)	1.5 (0.3 to 3.6)
Rectangular	17	0.62	68 (17 to 155)	2.5 (0.6 to 5.6)
Flexible				
Metal, aluminum	11	0.40	42 (17 to 76)	1.5 (0.6 to 2.8)
Nonmetal	17	0.62	30 (6 to 76)	1.5 (0.2 to 2.8)
Fibrous glass				
Round	4	0.14	NA	NA
Rectangular	8	0.29	NA	NA

[a]Leakage classes here are averages based on tests conducted by AISI/SMACNA (1972), ASHRAE/SMACNA/TIMA (1985), and Swim and Griggs (1995).
[b]"Sealed" leakage classes assume that, for metal ducts, all transverse joints, seams, and openings in duct wall are sealed.
[c]Leakage classes anticipate about 0.82 joints per metre of duct. For systems with a high fitting-to-straight-duct ratio, greater leakage occurs in both sealed and unsealed conditions.

Table 7 Leakage as Percentage of Airflow[a,b]

Leakage Class	System L/s per m² Duct Surface[c]	Static Pressure, Pa 125	250	500	750	1000	1500
68	10	15	24	38	49	59	77
	12.7	12	19	30	39	47	62
	15	10	16	25	33	39	51
	20	7.7	12	19	25	30	38
	25	6.1	9.6	15	20	24	31
34	10	7.7	12	19	25	30	38
	12.7	6.1	9.6	15	20	24	31
	15	5.1	8.0	13	16	20	26
	20	3.8	6.0	9.4	12	15	19
	25	3.1	4.8	7.5	9.8	12	15
17	10	3.8	6	9.4	12	15	19
	12.7	3.1	4.8	7.5	9.8	12	15
	15	2.6	4.0	6.3	8.2	9.8	13
	20	1.9	3.0	4.7	6.1	7.4	9.6
	25	1.5	2.4	3.8	4.9	5.9	7.7
8	10	1.9	3	4.7	6.1	7.4	9.6
	12.7	1.5	2.4	3.8	4.9	5.9	7.7
	15	1.3	2.0	3.1	4.1	4.9	6.4
	20	1.0	1.5	2.4	3.1	3.7	4.8
	25	0.8	1.2	1.9	2.4	3.0	3.8
4	10	1.0	1.5	2.4	3.1	3.7	4.8
	12.7	0.8	1.2	1.9	2.4	3.0	3.8
	15	0.6	1.0	1.6	2.0	2.5	3.2
	20	0.5	0.8	1.3	1.6	2.0	2.6
	25	0.4	0.6	0.9	1.2	1.5	1.9

[a]Adapted with permission from *HVAC Air Duct Leakage Test Manual* (SMACNA 1985, Appendix A).
[b]Percentage applies to airflow entering a section of duct operating at an assumed pressure equal to average of upstream and downstream pressures.
[c]Ratios in this column are typical of fan volumetric flow rate divided by total system surface. Portions of systems may vary from these averages.

HVAC Duct Construction Standards (2005). Fibrous glass ducts and their closure systems are covered by UL *Standards* 181 and 181A. For fibrous glass duct construction standards, consult NAIMA (2002) and SMACNA (2003). Flexible duct performance and installation standards are covered by UL *Standards* 181 and 181B and ADC (2003). Soldered or welded duct construction is necessary where sealants are not suitable. Sealants used on exterior ducts must be resistant to weather, temperature cycles, sunlight, and ozone.

Shaft and compartment pressure changes affect duct leakage and are important to health and safety in the design and operation of contaminant and smoke control systems. Shafts should not be used for supply, return, and/or exhaust air without accounting for their leakage rates. Airflow around buildings, building component leakage, and the distribution of inside and outside pressures over the height of a building, including shafts, are discussed in Chapters 16 and 24.

Table 8 Typical Design Velocities for HVAC Components

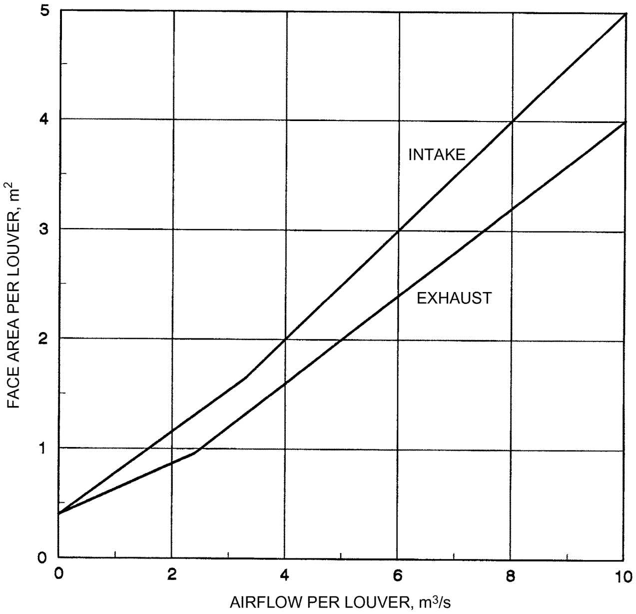
Duct Element	Face Velocity, m/s
Louvers[a]	
Intake	
3300 L/s and greater	2
Less than 3300 L/s	See Figure 14
Exhaust	
2400 L/s and greater	2.5
Less than2400 L/s	See Figure 14
Filters[b]	
Panel filters	
Viscous impingement	1 to 4
Dry-type, extended-surface	
Flat (low efficiency)	Duct velocity
Pleated media (intermediate efficiency)	Up to 3.8
HEPA	1.3
Renewable media filters	
Moving-curtain viscous impingement	2.5
Moving-curtain dry media	1
Electronic air cleaners	
Ionizing type	0.8 to 1.8
Heating Coils[c]	
Steam and hot water	2.5 to 5 1 min., 8 max.
Electric	
Open wire	Refer to mfg. data
Finned tubular	Refer to mfg. data
Dehumidifying Coils[d]	2 to 3
Air Washers[e]	
Spray type	Refer to mfg. data
Cell type	Refer to mfg. data
High-velocity spray type	6 to 9

[a]Based on assumptions presented in text.
[b]Abstracted from Ch. 28, 2008 *ASHRAE Handbook—HVAC Systems and Equipment*.
[c]Abstracted from Ch. 26, 2008 *ASHRAE Handbook—HVAC Systems and Equipment*.
[d]Abstracted from Ch. 22, 2008 *ASHRAE Handbook—HVAC Systems and Equipment*.
[e]Abstracted from Ch. 40, 2008 *ASHRAE Handbook—HVAC Systems and Equipment*.

System Component Design Velocities

Table 8 summarizes face velocities for HVAC components in built-up systems. In most cases, the values are abstracted from pertinent chapters in the 2008 *ASHRAE Handbook—HVAC Systems and Equipment*; final selection of components should be based on data in these chapters or, preferably, from manufacturers.

Use Figure 14 for preliminary sizing of air intake and exhaust louvers. For air quantities greater than 3300 L/s per louver, the air intake gross louver openings are based on 2 m/s; for exhaust louvers, 2.5 m/s is used for air quantities of 2400 L/s per louver and greater. For smaller air quantities, refer to Figure 14. These criteria are presented on a per-louver basis (i.e., each louver in a bank of louvers) to include each louver frame. Representative production-run louvers were used in establishing Figure 14, and all data used were based on AMCA *Standard* 500-L tests. For louvers larger than



Parameters Used to Establish Figure	Intake Louver	Exhaust Louver
Minimum free area (1220 mm square test section), %	45	45
Water penetration, mL/($m^2 \cdot 0.25$ h)	Negligible (less than 0.3)	N/A
Maximum static pressure drop, Pa	35	60

Fig. 14 Criteria for Louver Sizing

1.5 m^2, the free areas are greater than 45%; for louvers less than 1.5 m^2, free areas are less than 45%. Unless specific louver data are analyzed, no louver should have a face area less than 0.4 m^2. If debris can collect on the screen of an intake louver, or if louvers are located at grade with adjacent pedestrian traffic, louver face velocity should not exceed 0.5 m/s.

Louvers require special treatment because the blade shapes, angles, and spacing cause significant variations in louver-free area and performance (pressure drop and water penetration). Selection and analysis should be based on test data obtained from the manufacturer in accordance with AMCA *Standard* 500-L, which presents both pressure drop and water penetration test procedures and a uniform method for calculating the free area of a louver. Tests are conducted on a 1220 mm square louver with the frame mounted flush in the wall. For water penetration tests, rainfall is 100 mm/h, no wind, and the water flow down the wall is 0.05 L/s per linear metre of louver width.

AMCA *Standard* 500-L also includes a method for measuring water rejection performance of louvers subjected to simulated rain and wind pressures. These louvers are tested at a rainfall of 76 mm/h falling on the louver's face with a predetermined wind velocity directed at the face of the louver (typically 13 or 20 m/s). Effectiveness ratings are assigned at various airflow rates through the louver.

System and Duct Noise

The major sources of noise from air-conditioning systems are diffusers, grilles, fans, ducts, fittings, and vibrations. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* discusses sound control for each of these sources, as well as methods for calculating required sound attenuation. Sound control for terminal devices consists of selecting devices that meet the design goal under all operating conditions and installing them properly so that no additional sound is generated. The sound power output of a fan is determined by the type of fan, airflow, and pressure. Sound control in the duct system requires proper duct layout, sizing, and provision for installing duct attenuators, if required. Noise generated by a system increases with both duct velocity and system pressure.

Testing and Balancing

Each air duct system should be tested, adjusted, and balanced. Detailed procedures are given in Chapter 37 of the 2007 *ASHRAE Handbook—HVAC Applications*. To properly determine fan total (or static) pressure from field measurements taking into account fan system effect, see the section on Fan/System Interface. Equation (38) allows direct comparison of system resistance to design calculations and/or fan performance data. It is important that system effect magnitudes be known prior to testing. If necessary, use Equation (17) to calculate fan static pressure knowing fan total pressure [Equation (38)]. For TAB calculation procedures of numerous fan/ system configurations encountered in the field, refer to AMCA (2007b).

DUCT DESIGN METHODS

Duct design methods for HVAC systems and for exhaust systems conveying vapors, gases, and smoke are the equal-friction method, the static regain method, and the T-method. The section on Industrial Exhaust System Duct Design presents the design criteria and procedures for exhaust systems conveying particulates. Equal friction and static regain are nonoptimizing methods, and the T-method is a practical optimization method introduced by Tsal et al. (1988).

To ensure that system designs are acoustically acceptable, noise generation should be analyzed and sound attenuators and/or acoustically lined duct provided where necessary.

Equal-Friction Method

In the equal-friction method, ducts are sized for a constant pressure loss per unit length. The shaded area of the friction chart (see Figure 9) is the suggested range of friction rate and air velocity. When energy cost is high and installed ductwork cost is low, a low-friction-rate design is more economical. For low energy cost and high duct cost, a higher friction rate is more economical. After initial sizing, calculate total pressure loss for all duct sections, and then resize sections to balance pressure losses at each junction.

Static Regain Method

This design method is only applicable to supply air systems. The objective is to obtain the same static pressure at diverging flow junctions by changing downstream duct sizes. This design objective can be developed by rearranging Equation (7a) and setting $p_{s,2}$ equal to $p_{s,1}$ (neglecting thermal gravity effect term). This means that the change in static pressure from one section to another is zero, which is satisfied when the change in total pressure is equal to the change in velocity pressure. Thus,

$$p_{s,1} - p_{s,2} = \Delta p_{t,1\text{-}2} - \left[\frac{\rho V_1^2}{2} - \frac{\rho V_2^2}{2}\right] \qquad (41)$$

and

$$\Delta p_{t,1\text{-}2} = \frac{\rho V_1^2}{2} - \frac{\rho V_2^2}{2} \qquad (42)$$

where $\Delta p_{t,1\text{-}2}$ is total pressure loss from upstream of junction 1 to upstream of junction 2. Junction 2 can be a terminal section, where the total pressure is zero. For each main section, the straight-through and branch sections immediately downstream of the main duct section are determined by iteration of that section's size until Equation (42) is satisfied. However, there could be cases when the straight or branch sections need to be larger than the upstream section to satisfy Equation (42). Fittings in the 2009 *ASHRAE Duct Fitting Database* have not been tested under these conditions, and making downstream sections larger than upstream sections is not practical. The largest straight-through or branch size should be limited to that of the upstream section. The imbalance that occurs is resolved during total-pressure balancing of the system.