Related Commercial Resources

CHAPTER 48. DESIGN AND APPLICATION OF CONTROLS

AUTOMATIC control of HVAC systems and equipment usually includes control of temperature, humidity, pressure, and flow rates of air and water. Automatic controls can sequence equipment operation to meet load requirements and to provide safe equipment operation using direct digital control (DDC), electronic, electrical, mechanical, and/or pneumatic devices. Automatic controls are only fully effective when applied to well-designed mechanical systems; they cannot compensate for misapplied systems, excessive under- or oversizing, or highly nonlinear processes.

This chapter addresses control of typical HVAC systems, design of controls for system coordination and for energy conservation, and control system commissioning. Chapter 7 of the 2021 *ASHRAE Handbook—Fundamentals* covers details of component hardware and the basics of control.

1. SYSTEM TYPES

A building automation system (BAS) with direct digital, electronic, or pneumatic controls has several physical control loops, with each loop including a controlled variable (e.g., temperature), controlled device (e.g., actuator), and the process to be controlled (e.g., heating system). A BAS with DDC controllers can share sensor values with several control loops or have multiple control loops selectively activate an actuator.

A BAS with DDC controllers allows information such as system status or alarms to be shared through a common communication protocol, between HVAC systems or within building systems and services, thereby enabling advanced, energy-saving, system-level applications. ASHRAE *Guideline* 13 and *Standard* 135 have more detailed discussions of networking and interoperability.

2. DESIGN CONSIDERATIONS AND PRINCIPLES

In designing and selecting the HVAC system for the entire building, the type, size, use, and operation of the structure must be considered. Subsystems such as fan and water supply are normally controlled by local automatic control or a local loop control, which includes the sensors, controllers, and controlled devices used with a single HVAC system and excludes any supervisory or remote functions such as reset and start/stop. However, local control is frequently extended to central control to diagnose malfunctions that might result in damage from delay, or that might increase labor and energy costs. Special modes of operation may be required to allow for load shedding, purge, warm-up, cooldown, and lockdown. Initiators may be manual or automatic, based on weather, announcements of extraordinary events, high concentrations of expected and therefore measured hazardous gases, or daily schedules reset by outdoor air temperature.

Distributed processing using microprocessors has enabled computer use at many locations other than the central control point. It is more common to use DDC instead of pneumatic or electric thermostats for local loop controllers. The local loop control is integrated with energy management functions performed by upper-level DDC devices to form a complete BAS.

Because HVAC systems are designed to meet maximum design conditions, they nearly always function at partial capacity. The system must be adjusted and operated for many years, so the simplest control that produces the necessary results is usually the best.

Mechanical and Electrical Coordination

Even a pneumatic control system includes wiring, conduit, switchgear, and electrical distribution for many electrical devices. The mechanical designer must inform the electrical designer of the total electrical requirements if controls are to be wired by the electrical contractor. Requirements include (1) the devices to be furnished and/or connected, (2) electrical load, (3) location of electrical items, (4) a description of each control function, and (5) whether the control system needs to be on emergency power or UPS.

Coordination is essential. Proper coordination should produce a control diagram that shows the interface with other control elements to form a complete and usable system. As an option, the control engineer may develop a complete performance specification and require the control contractor to install all wiring related to the specified sequence. The control designer must run the final checks of drawings and specifications. Both mechanical and electrical specifications must be checked for compatibility and uniformity.

Sequences of Operation

A BAS requires that the engineer define how the system is to be controlled. Sequences define the control logic and functional requirements of the equipment and system. They are developed by the BAS designer and implemented by the controls programmer. Sequences can be depicted as textual descriptions in plain language and are often supported by logic diagrams. Sequences relate directly to equipment points list requirements and support commissioning of the system through identifying points to be commissioned and validated.

These sequences are then programmed into the system by the system installer. Writing clear, unambiguous, concise, yet comprehensive sequences of controls is key to the energy and comfort performance of the HVAC system, yet it is very difficult to do well. It requires a clear understanding of how controls work, the limitations of the specific DDC hardware specified and the HVAC system design, and a knack for clear thinking and writing. Techniques for writing successful control sequences are discussed in ASHRAE *Guideline* 13. Sequences for typical HVAC systems can be found in the ASHRAE *Guideline* 36.

3. CONTROL PRINCIPLES FOR ENERGY CONSERVATION

After a building's general needs have been established and the building and system subdivision has been made, the mechanical system and its control approach can be considered. Designing systems that conserve energy requires knowledge of (1) the building, (2) its operating schedule, (3) the systems to be installed, and (4) ASHRAE *Standard* 90.1. Some principles or approaches that conserve energy are as follows:

- *Run equipment only when needed*. Schedule HVAC unit operation for occupied periods. Run heat at night only to maintain internal temperature at around 13 and 16°C to prevent freezing. Start morning warm-up as late as possible to achieve design internal temperature by occupancy time, considering residual space temperature, building construction and mass, outdoor temperature, and equipment capacity (optimum start control). Under most conditions, heating and cooling equipment can be shut down some time before the end of occupancy, depending on internal and external load and space temperature (optimum stop control). Calculate shutdown time so that space temperature does not drift out of the selected comfort zone before occupancy ends and ensure that ventilation is provided throughout occupancy.
- *Sequence heating and cooling*. Do not supply heating and cooling simultaneously unless it is required for humidity control. Central fan systems should use cool outdoor air, if available, in sequence between heating and cooling. Zoning and system selection should eliminate, or at least minimize, simultaneous heating and cooling. Also, humidification and dehumidification should not take place concurrently.
- *Provide only the heating or cooling actually needed*. Reset the supply temperature of hot and cold air (or water). In air systems that support variable-speed fans, reset duct static pressure to provide thermal comfort at the lowest possible fan speed and energy consumption.
- *Supply heating and cooling from the most efficient source*. Use free or low-cost energy sources first, then higher-cost sources as necessary.
- *Apply outdoor air control*. When on minimum outdoor air, supply no less than that recommended by ASHRAE *Standard* 62.1 and, on VAV systems, include controls that ensure that outdoor air rates are maintained under all expected supply air operating conditions.

System Selection

The mechanical system significantly affects the control of zones and subsystems. System type and number and location of zones influence the amount of simultaneous heating and cooling that occurs. For perimeter areas, heating and cooling should be controlled in sequence to minimize simultaneous heating and cooling. In general, this sequencing must be accomplished by the control system because only a few mechanical systems (e.g., two-pipe and single-coil) can prevent simultaneous heating and cooling. Systems that require engineered control systems to minimize simultaneous heating and cooling include the following:

- *VAV cooling with zone reheat*. Reduce cooling energy and/or air volume to a minimum before applying reheat.
- *Four-pipe heating and cooling for unitary equipment*. Sequence heating and cooling.
- *Dual-duct systems*. Condition only one duct (either hot or cold) at a time. The other duct should supply a mixture of outdoor and return air.
- *Single-zone heating/cooling*. Sequence heating and cooling.

Some exceptions exist, such as dehumidification with reheat.

Control zones are determined by the location of the thermostat or temperature sensor that sets the requirements for heating and cooling supplied to the space. Typically, control zones are for a room or an open area of a floor.

Energy standards such as ASHRAE *Standard* 90.1 no longer allow constant-volume systems that reheat cold air or that mix heated and cooled air, except in special applications such as hospitals. If used, they should be designed for minimal use of reheat through zoning to match actual dynamic loads and resetting cold and warm air temperatures based on the zone(s) with the greatest demand. Heating and cooling supply zones should be structured to cover areas of similar load. Areas with different exterior exposures should have different supply zones.

Systems that provide changeover switching between heating and cooling prevent simultaneous heating and cooling. Some examples are hot or cold secondary water for fan-coils or single-zone fan systems. They usually require small operational zones, which have low load diversity, to allow changeover from warm to cold water without occupant dissatisfaction.

Systems for building interiors usually require year-round cooling and are somewhat simpler to control than exterior systems. These interior areas normally use all-air systems with a constant supply air temperature, with or without VAV control. Proper control techniques and operational understanding can reduce the energy used to treat these areas. General load characteristics of different parts of a building may lead to selecting different systems for each.

Load Matching

With individual room control, the environment in a space can be controlled more accurately and energy can be conserved if the entire system can be controlled in response to the major factor influencing the load. Thus, water temperature in a water-heating system, steam temperature or pressure in a steam-heating system, or delivered air temperature in a central fan system can be varied as building load varies. Control of the entire system relieves individual space controls of part of their burden and provides more accurate space control. Also, modifying the basic rate of heating or cooling input in accordance with the entire system load reduces losses in the distribution system.

The system must always satisfy the area with the greatest demand. Individual controls handle demand variations in the area the system serves. The more accurate the system zoning, the greater the control, the smaller the distribution losses, and the more effectively space conditions are maintained by individual controls.

Buildings or zones with a modular arrangement can be designed for subdivision to meet occupant needs. Before subdivision, operating inefficiencies can occur if a zone has more than one thermostat. In an area where one thermostat activates heating while another activates cooling, the terminals should be controlled from a single thermostat until the area is properly subdivided.

Size of Controlled Area

No individually controlled area should exceed about 465 m^2 because of the difficulty of obtaining good distribution and of finding a representative location for space control increases with zone area. Each individually controlled area must have similar load characteristics throughout. Equitable distribution, provided through competent engineering design, careful equipment sizing, and proper system balancing, is necessary to maintain uniform conditions throughout a controlled area. The control can measure conditions only at its location; it cannot compensate for nonuniform conditions caused by improper distribution or inadequate design. Areas or rooms having dissimilar load characteristics or different conditions to be maintained should be controlled individually. The smaller the controlled area, the better the control and the performance and flexibility.

Location of Space Sensors

Space sensors and controllers must be located where they accurately sense the variables they control and where the condition is representative of the area (zone) they serve. In large open areas having more than one zone, thermostats should be located in the middle of their zones to prevent them from sensing conditions in surrounding zones. Typically, space temperature controllers or sensors are placed in the following locations.

- **Wall-mounted thermostats** or **sensors** are usually placed on inside walls or columns in the space they serve. Avoid outdoor wall locations. Mount thermostats at generally accessible heights according to the Americans with Disabilities Act (ADA) (USDOJ 1994) (usually 1220 mm) and in locations where they will not be affected by heat from sources such as direct sun rays, wall pipes or ducts, convectors, or direct air currents from diffusers or equipment (e.g., copy machines, coffeemakers, refrigerators). The wall itself should be sealed tightly if it penetrates a pressurized supply air plenum either under the floor or overhead. Air circulation should be ample and unimpeded by furniture or other obstructions, and the thermostat should be protected against mechanical damage. Thermostats in spaces such as corridors, lobbies, or foyers should be used to control those areas only.
- **Return air thermostats** can control floor-mounted unitary conditioners such as induction or fan-coil units and unit ventilators. On induction and fan-coil units, the sensing element is behind the return air grille. On classroom unit ventilators that use up to 100% outdoor air for natural cooling, however, provide a forced-flow sampling

chamber for the sensing element, which should be located carefully to avoid radiant effect and to ensure adequate air velocity across the element.

If return air sensing is used with a central fan system, locate the sensing element as near as possible to the space being controlled to eliminate any influence from other spaces and the effect of any heat gain or loss in the duct. Where supply/return light fixtures are used to return air to a ceiling plenum, the return air sensing element can be located in the return air opening. Be sure to offset the set point to compensate for heat from the light fixtures.

- **Diffuser-mounted thermostats** usually have sensing elements mounted on circular or square ceiling supply diffusers and depend on aspiration of room air into the supply airstream. They should be used only on high-aspiration diffusers adjusted for a horizontal air pattern. The diffuser on which the element is mounted should be in the center of the occupied area of the controlled zone.
- **CO_2 sensors** for DCV are usually located in spaces with high occupant densities (e.g., conference rooms, auditoriums, courtrooms). Locating the sensor in return air ducts/plenums that serve multiple spaces measures average concentrations and does not provide information on CO_2 levels in rooms with the highest concentrations. CO_2 sensors should be located in the breathing zone of the occupied space (see ASHRAE [2011] and USGBC [2009]).

Commissioning

Commissioning is the process of ensuring that systems are designed, installed, functionally tested, and capable of being operated and maintained in conformity with the design intent. Commissioning HVAC systems begins with planning and includes design, construction, start-up, acceptance, and training, and can be applied throughout the life of the building.

For HVAC systems, **functional performance testing (FPT)** is an important part of the commissioning process. FPT is the process of determining the ability of HVAC system to deliver heating, ventilating, and air conditioning in accordance with the final design intent. Commissioning is team-oriented and generally involves cooperation of various parties, including the owner, design engineers, and contractors and subcontractors. A commissioning authority (the designated person, company, or agent who implements the overall commissioning process) generally leads the process. Each commissioning project must have a plan that defines the commissioning process and is developed in increasing detail as the project progresses. The most useful tool used to challenge (simulate changes to) systems operation is the control system itself. A BAS provides the added convenience of central execution of test steps, and the ability to record responses. The BAS must be commissioned before it can be used to validate the HVAC systems. Commissioning a BAS is discussed in ASHRAE *Guideline* 11. Commissioning HVAC systems is recommended for construction of new buildings, and should be repeated periodically in existing buildings. See Chapter 43 and ASHRAE *Guidelines* 0 and 11 for more information.

Third-Party Performance Certification

For owners who want to verify that their buildings conserve energy, third-party building performance rating systems such as ASHRAE Building EQ, LEED®, Green Globes, and the Living Building Challenge provide guidance on designing and building commercial, institutional, and government buildings to produce quantifiable benefits for occupants, owners, and the environment. During design and construction, these rating systems provide opportunities to use a BAS, not only for comfort and energy efficiency, but also for sustainability verification throughout the facility's lifetime.

Applications. It is recommended that the design team choose the controls consultant/contractor and involve them with the team early in design, allowing for controls input on the control schemes.

For any facility, the BAS provides thermal comfort and routine programming (e.g., occupied/unoccupied). When working with third-party rating programs, the controls consultant or contractor uses the BAS for benchmarking and alarming when reference points are exceeded (e.g., monitoring facility electrical use). Once a baseline is established, programs with established alarm limits can alert the owner if the facility exceeds the baseline by 10% or more, so corrections can be made and continued higher utility expenses avoided. Depending on the facility, this may be monitored for hourly, daily, weekly, or monthly comparison.

The BAS can also

- Turn solar panels for optimum sun exposure
- Adjust blinds and awnings
- Keep indoor air quality (IAQ) acceptable by adjusting outdoor air (OA) without overventilating
- Open and close windows for natural ventilation
- Monitor and adjust indoor and outdoor lighting

- Control irrigation based on weather
- Collect weather data
- Track scheduling hours for occupancy
- Monitor equipment run times and set points
- Track electricity use profiles
- Monitor efficiency of large equipment performance (e.g., chillers, variable-frequency-drive [VFD] pumping, boilers)

All of these factors affect building sustainability.

Because the BAS operates in real time and can compare real-time data to previously defined baselines, it is a valuable tool. Open protocols allow the BAS to monitor, control, and provide critical alarming for non-HVAC equipment (e.g., for power monitoring, chiller performance, VFDs for VAV systems or variable-speed pumping, water efficiency, emergency generators, indoor and outdoor lighting, boilers) for a minimal investment.

The BAS provides an excellent tool for commissioning both HVAC and other equipment, as required by third-party performance rating systems. Its data acquisition abilities allow comparisons of daily performance as building use changes, thereby allowing the commissioning agent to determine whether equipment is operating properly, and allowing the owner to compare real-time data to previously defined benchmarks as part of a **measurement and verification (M&V)** program. Typical operation sequences, such as optimizing outdoor air quantities or chiller efficiency, and providing multiple sensing points for thermal comfort, are still used and are critical for maintaining sustainability.

Perhaps the most important aspect of a successful control system is training the owner. Many owners do not fully realize the capabilities of BAS, many of which are intangible and somewhat obscure, so it is critical that the owner be given proper training in using the controls system. The order in which training takes place is equally important: mechanical equipment training should come first, then operation and maintenance (O&M) and controls layout configurations, use of the controls system to provide thermal comfort and maintain equipment, and, finally, written M&V procedures to maintain sustainability.

If the facility cannot justify a full-time energy manager, then the owner should consider third-party contracting to ensure the facility performs at its designed energy efficiency.

4. AIR SYSTEMS

Air systems use fans to move air within a space or building. The simplest ones use a small fan enclosed in a casing to circulate air within a single room, whereas more complex systems can use multiple fans and duct systems to circulate air throughout an entire building. Air systems can provide heating and/or cooling using multiple energy sources. Regardless of which components are included in an air system, there are common control schemes that can be used to categorize air system operation.

Constant-Volume (CV) Systems

In a constant-volume system, supply and return fan airflow rates are manually set to meet the airflow requirements for peak thermal load and ventilation for full occupancy. Once set, the airflow does not change; it is constant except when no airflow is needed and the unit is off. The air-handling unit’s economizer outdoor air dampers, heating coil (where applicable), and cooling coil are controlled in sequence to maintain the **supply air temperature (SAT)** at a set point. For a single-zone constant-volume system, the SAT may also be reset based on outdoor air or return air temperature. For a constant-volume system serving multiple zones, the SAT may be reset to satisfy the zone with the maximum load or the cumulative change represented by the return air temperature. For a SAT reset strategy, the minimum cooling SAT set point should be set no lower than the design cooling coil’s leaving air temperature, to prevent excessive chilled-water temperature reset requests that can reduce chiller plant efficiency. The maximum cooling SAT set point is typically 15.6°C in humid climates and 18°C in milder climates.

If unoccupied warm-up mode is used, the outdoor damper is closed and the supply air set point is adjusted upward to the desired value.

Control strategies for economizers, demand-controlled ventilation, morning warm-up, and night cooldown are the same as for VAV systems.

Variable Air Volume (VAV)

Variable-air-volume systems provide thermal control by varying the amount of airflow delivered. A reduction in demand for cool air means the supply fan can operate at a reduced speed, saving energy. Most VAV systems have pressure-independent terminals, which means a separate airflow control loop operates each terminal damper.

Terminal Units

A terminal unit (also called a constant-air volume [CAV] or variable-air-volume [VAV] box) is the zone-level control device for constant- or variable-volume systems. At a minimum, a terminal unit consists of a calibrated air damper, though different types also include components such a heating coil (in reheat boxes), automatic actuator controlling the calibrated air damper (in VAV boxes), and an integral fan in fan-powered terminal units.

A system is considered to be variable volume if primary airflow to the space varies. Total airflow to the space (primary air + plenum air) may be constant for some terminal units, even in a variable-volume system. Space set point is maintained by changing the temperature of the air delivered to the space or, with constant low-temperature supply air, by limiting the amount of the cool air that enters the space, or both. A space temperature of 24°C and 50% rh requires air supplied from the AHU that has typically been cooled to 13°C for moisture removal. Minimum ventilation may require so much cool air that the space is below the desired temperature. Zonal reheat coils at the terminal units heat the supply air at the zone to meet the space temperature set point.

VAV systems typically serve fewer than 15 zones per AHU. A terminal unit typically serves 1 to 8 outlets. The system should serve zones with similar thermal loads (e.g., all internal zones or all zones on the same exterior exposure), so that the unit is not continually switching between heating and cooling. To ensure minimum outdoor air ventilation is maintained, outdoor airflow must be controlled (see the section on Minimum Outdoor Air Control). A simple fixed damper position typically is not adequate.

Individual systems and their respective zones should be grouped together for easy scheduling and system-wide override. The system and zone groups all work in the same operating mode based on their operating schedule, occupancy status, and deviations from set point. See ASHRAE *Guideline* 13 for best practices in locating zone group operating mode programming logic based on network architecture. Depending on region, climate, and application, terminal unit zone groups should consider the following operating modes:

- Occupied
- Warmup
- Cooldown
- Setback
- Freeze protection setback
- Setup

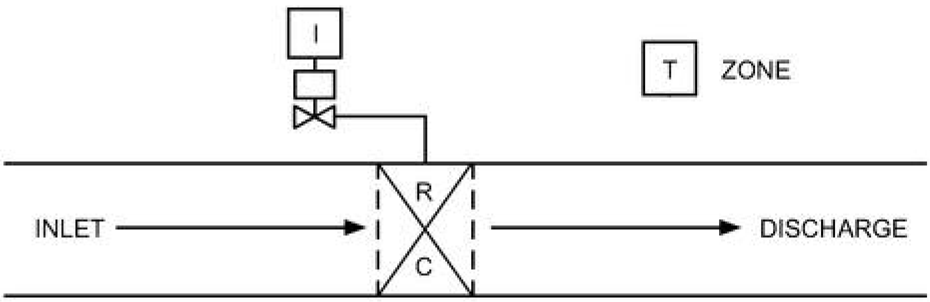

Figure 1. Single-Duct Constant-Volume Zone Reheat

Single-Duct, Constant-Volume. Reheat terminals use a single constant-volume fan system that serves multiple zones (Figure 1). All of the system's supply air is cooled to satisfy the greatest zone cooling load. Air delivered to other zones is then reheated with heating coils (hot-water, steam, or electric) in individual zone ducts. The reheat coil valve (or electric heating element) modulates as required to maintain the space condition. Because these systems consume more energy than VAV systems, they are generally limited by energy standards to applications with fixed ventilation needs, such as hospitals, special processes, or laboratories.

No fan control is required for constant-volume terminal units, because the design, selection, and adjustment of fan components determine the air volume and duct static pressure. The same temperature air is supplied to all zones. However, the controller can vary the supply temperature to respond to demand from the zone with the greatest cooling load, thus conserving energy, and demand-controlled ventilation may be implemented where applicable.

Single-Duct Variable-Volume. A **throttling VAV terminal** has an inlet damper that controls the flow of primary supply air (Figure 2). For spaces with exterior exposures or a high airflow requirement of ventilation air requiring heating, a reheat coil can be installed in the discharge. With pressure-independent controls, the space temperature sensor does not control the inlet damper directly. The space temperature control loop output is used to reset the primary airflow delivered to the space between a maximum and minimum rate. Direct control of airflow makes the VAV box independent of variations in duct static pressure.

The currently recommended control sequence is the dual maximum sequence in Figure 3. As the space goes from design cooling load to design heating load, the airflow set point is first reset from the cooling maximum to the minimum value needed for ventilation. Then the supply air temperature is reset from minimum (e.g., 13°C) to maximum (e.g., 32°C), and the reheat coil is modulated to maintain the supply air temperature at set point. Lastly, the airflow set point is reset from the minimum up to the heating maximum. The minimum flow rated for ventilation may be a constant, but is more likely adjusted by occupancy. The minimum flow rate may be further adjusted according to a measured concentration of some air constituent, typically carbon dioxide.

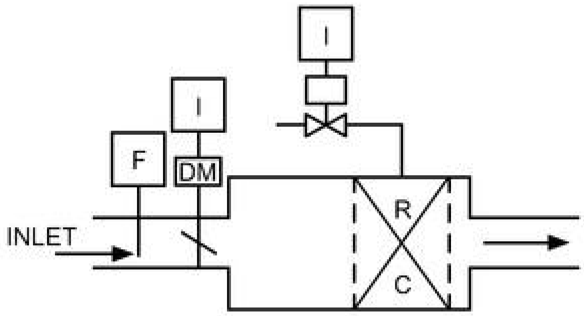

Figure 2. Throttling VAV Terminal Unit

Previously, it was common to keep the primary airflow rate always high enough to handle the maximum heating load. ASHRAE *Standard* 90.1 and some other energy codes do not allow that practice because it increases simultaneous heating and cooling.

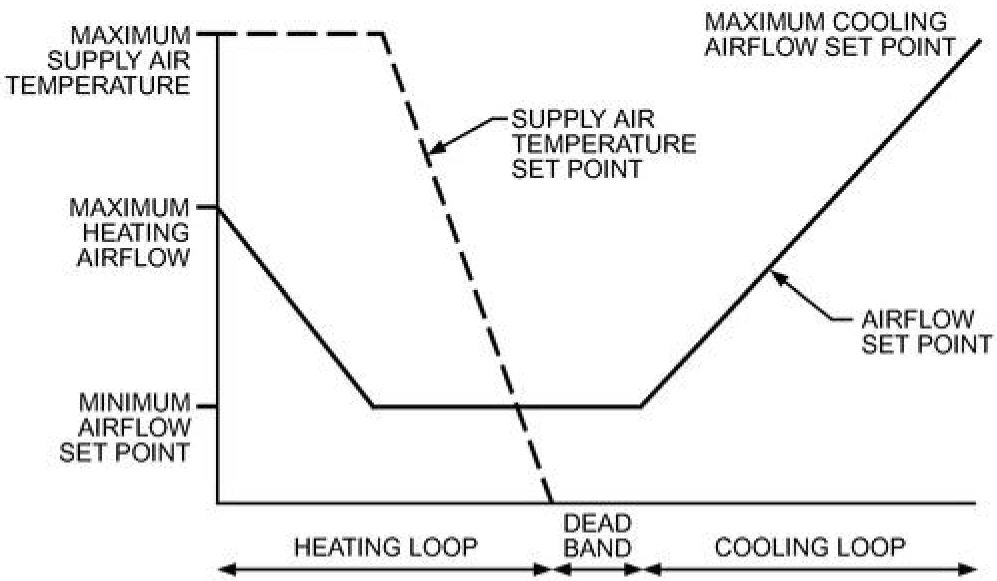

Figure 3. Throttling VAV Terminal Unit: Dual Maximum Control Sequence

An **induction VAV terminal** controls space temperature by reducing supply airflow to the space and by inducing return air from the plenum into the airstream for the space (Figure 4). Both dampers are controlled simultaneously, so as the primary air opening decreases, the return air opening increases. When space temperature drops below the set point, the supply air damper begins to close and the return air damper begins to open.

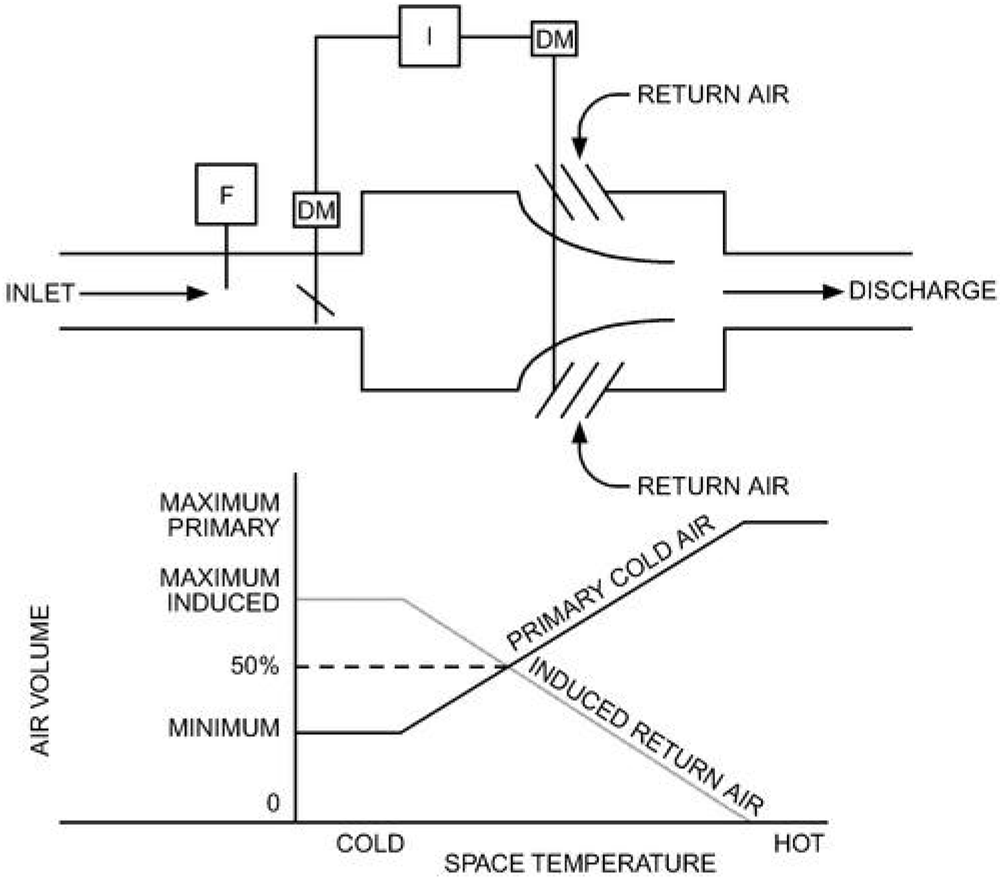

Figure 4. Induction VAV Terminal Unit

A **series fan-powered terminal unit** has an integral fan in series with the primary supply air damper that supplies to the space (Figure 5). These terminals can be either constant or variable volume. In addition to enhancing air distribution in the space, a reheat coil can be added for space heating and to maintain a minimum temperature in the space when the primary system is off, for strategies such as setback, warm-up, and demand-controlled ventilation. When the space is occupied, the fan runs to provide air to the space. The fan can draw air from the return plenum to compensate for reduced supply air volume. As temperature in the space decreases below the cooling set point, the supply air damper begins to close and the fan draws more air from the return plenum. For zones with a reheat coil, when supply air reaches its minimum volume and the space temperature begins to drop below the heating set point, the valve to the reheat coil begins to open. Depending on the fan and motor, it may be important to start the terminal fan before the central air handler. If primary air is flowing when the terminal fan is off, it can spin the fan backwards. This can damage the motor when the terminal fan starts.

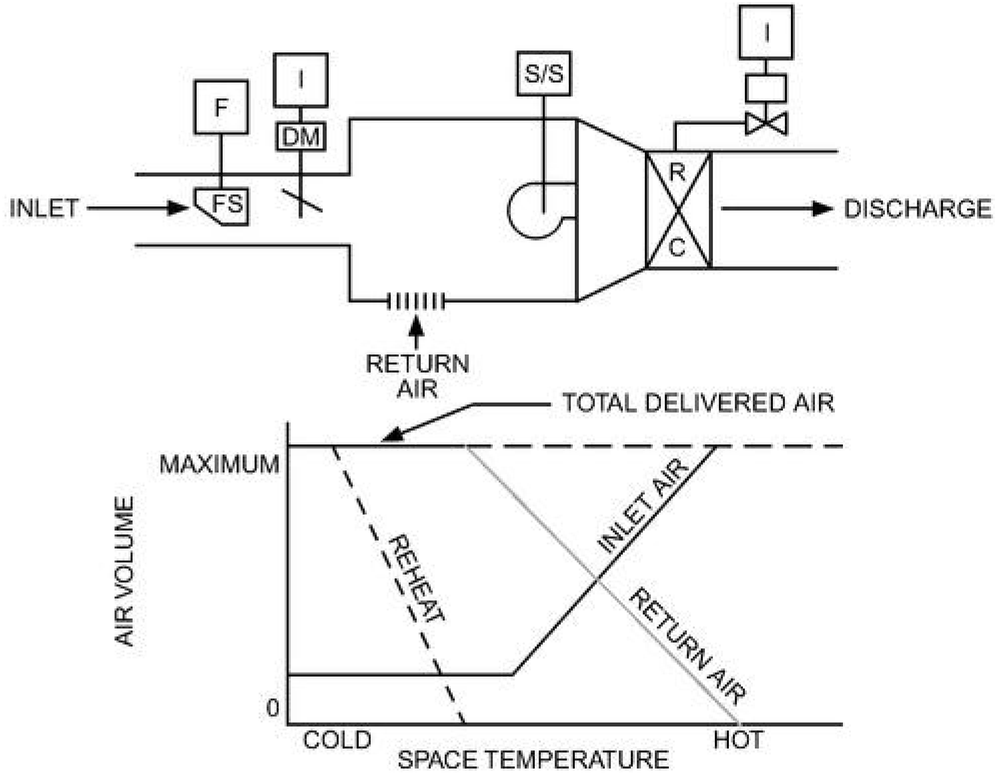

Figure 5. Series Fan-Powered VAV Terminal Unit

A **parallel fan terminal unit** is similar to the series fan terminal, except that the fan is in parallel with the primary supply air VAV damper (Figure 6). These terminals can be either constant or variable volume. A reheat coil may be placed in the discharge to the space or in the return plenum opening. The fan is intended to operate primarily in heating mode, but may also operate to maintain a minimum airflow to the space, allowing reduced primary airflow rates. Total airflow to the space is the sum of the fan output and supply air quantity. When space temperature drops below the cooling set point, the supply air damper begins to reduce the quantity of supply air entering the terminal. Once the supply damper reaches its minimum position and the space temperature begins to drop below the heating set point, the reheat coil valve starts to open. When the space is unoccupied and requires heating for setback or warm-up, the supply air damper is closed, the fan turns on, and the reheat coil valve modulates to maintain the unoccupied set point.

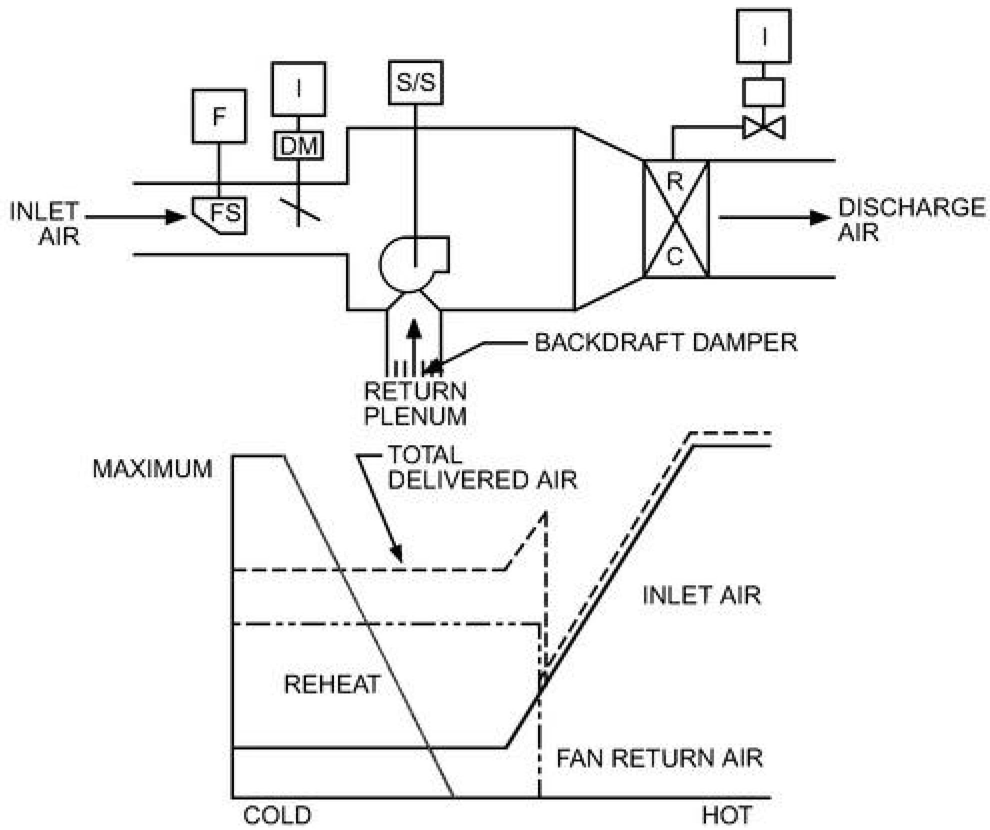

Figure 6. Parallel Fan Terminal Unit

Variable-volume, dual-duct terminal units (Figure 7) have inlet dampers (with individual damper actuators and airflow controllers) on the cooling and heating supply ducts. The space thermostat resets the airflow controller set points in sequence as the space load changes. The airflow controllers maintain adjustable minimum flows for ventilation. If the heating supply has sufficient ventilation air, there need not be any overlap of damper operations (one snaps closed and the other snaps open at the heat/cool changeover point), resulting in no simultaneous heating and cooling in the terminal unit. On systems where the heating supply does not have sufficient ventilation air (e.g., on some dual-fan dual-duct applications), the cooling damper can be controlled to a minimum for ventilation.

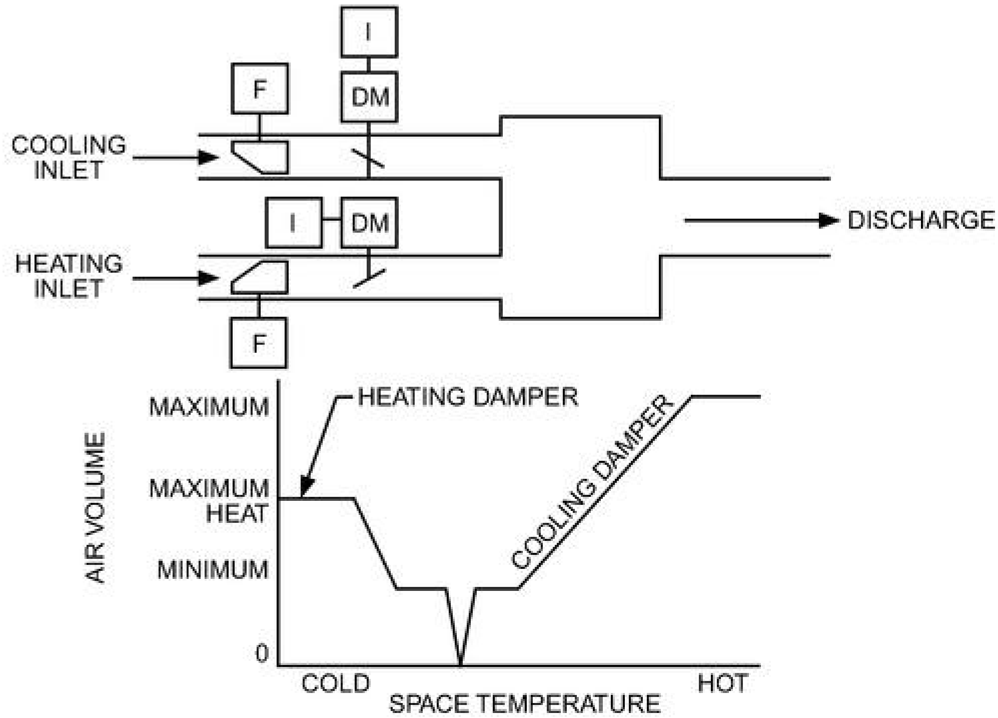

Figure 7. Variable-Volume Dual-Duct Terminal Unit

There are two main control strategies for dual-duct terminal units. **Snap acting control** is the most efficient control logic and does not require dual-duct boxes with mixing sections that have a high pressure drop. It eliminates the need for mixing plenum because airflows do not mix. However, snap action control logic is not recommended for demand control, because it can cause the zone to oscillate between cooling and heating. It may also cause low supply air temperature on systems with high outdoor airflows and no preheat coils because it cannot mix hot and cold air. **Mixing control logic** is the preferred option for applications with demand-control ventilation or application with high minimum airflow rates.

Chilled-beam terminal units are available as active or passive. Active chilled beams are supplied with constant-volume primary air from a dedicated outdoor air system (DOAS) unit or other air system. This air flows through the chilled beam and induces room air. Passive chilled beams do not receive any system air; see Chapter 20 of the 2020 *ASHRAE Handbook—HVAC Systems and Equipment*. Chilled beams can be two or four pipe, and are controlled similarly to other radiant elements. Compared to some other HVAC systems, chilled beams with DOAS can simplify room control sequences because temperature control and ventilation control are separated. This separation helps HVAC designers when planning to meet the individual loads and makes it easy to design control sequences.

Modulating and two-position valves have been applied for temperature control with chilled beams. For temperature control, the system needs at least one flow control valve for each temperature control zone. One control valve may serve multiple chilled beams. Mechanical sizing issues or piping arrangements may favor driving several flow control valves in unison from one temperature controller.

Preventing condensation on cooling surfaces is important in design and operation of chilled beams and radiant cooling devices. This is mainly accomplished by coordinating the primary systems delivering air and chilled water to the room. The general approach is to dry the supply air sufficiently to keep the space dew point a couple of degrees below the chilled-water supply temperature. Depending on loads and sizing, this may be possible without active control. Often, the solution is to sense relative humidity in some or all of the spaces, calculate the dew point, and deliver that information to the primary air and water control systems, where reset strategies prevent condensation.

As a back-up, condensation detectors at the chilled beams are recommended. When the switch closes, indicating condensing conditions, the system closes the control valve to the cooling coil. This prevents damage by condensation, but also disables cooling in the space. It is a backup safety measure; coordinating air and water distribution temperatures is the primary strategy. Sometimes, the energy advantage that leads to a chilled-beam design depends on optimizing features in the primary systems. In the cooling plant, the water-side economizer and other special strategies that deliver relatively warm water to the chilled beams may be critical to efficiency. Overall system efficiency may also depend on reducing use of reheat in the rooms. The supply air temperature reset strategy that minimizes reheat and controls moisture in the space may be sophisticated. Typically, the simple room control strategies must deliver coordinating data into relatively advanced primary plant controls.

Air-Handling Unit Controls

As VAV terminals vary the amount of air provided to each space, the associated air-handling unit should vary the amount of air provided to the system to ensure that only the minimum amount of air necessary is provided to the system. Airflow regulation is achieved through supply fan control. See ASHRAE *Guideline* 36 for high-performing sequence of operation for typical air handling units.

Supply Fan Control. The VAV supply fan controller

- Ensures pressure in the duct is enough to serve the terminals
- Prevents excessive pressure from disrupting terminal flow loops
- Reduces the risk of excessive pressure from damaging duct systems
- Allows for reduced energy consumption at the fan
- Keeps the fan in a stable region of the pressure-flow curve

Historically, various mechanisms (e.g., bypass damper, variable inlet vanes) have been used to regulate fan output. These methods vary widely in efficiency and energy consumption. Currently, variable-speed drives (VSDs) and electronically commutated (EC) motors are the most common because their low energy consumption and low first cost makes them cost effective. They are prescriptively required for most VAV systems by energy standards such as ASHRAE *Standard* 90.1.

The most common variable-airflow method is a closed-loop **proportional-with-integral (PI)** control, using the pressure measured at a selected point in the duct system. Historically, the set point was a constant, selected by the designer and confirmed by the balancer during system commissioning. However, this control strategy is based on the readings of a single sensor that is assumed to represent the pressure available to all VAV boxes. Choosing duct pressure sensor location can be difficult: if it malfunctions or is placed in a nonrepresentative location, operating problems will result; if it is located too close to the fan, the sensor will not sufficiently indicate service of the terminals. This usually leads to excessive energy consumption. Some have reported that placing the sensor at the far end of the duct system couples fan control too closely with the action of a single terminal, making it difficult to stabilize the system. Experience indicates that performance is satisfactory when the sensor is located at 75 to 100% of the distance from the first to the most remote terminal (Figure 8). ASHRAE *Standard* 90.1 prescriptively requires that the location result in a set point no higher than one-third of the total system static pressure drop.

Even with a good sensor location, fixed-pressure set point uses more energy than necessary. There are many operating hours when the fan pushes air through a system full of partly closed dampers. Many energy standards such as ASHRAE *Standard* 90.1 prescriptively requires automatically adjusting duct pressure based on zone demand as system load varies for systems with DDC at the zone level integrated with the air handler control. Airflow to zones is still regulated by flow loops in the terminal controllers and is unaffected, but all else being equal, the system meets the load more efficiently with the terminal dampers closer to open. This reduces energy consumption at the fan. Ideally, pressure is reduced to the point that at least one of the dampers opens all the way. Any further supply fan speed reduction reduces airflow at the terminals.

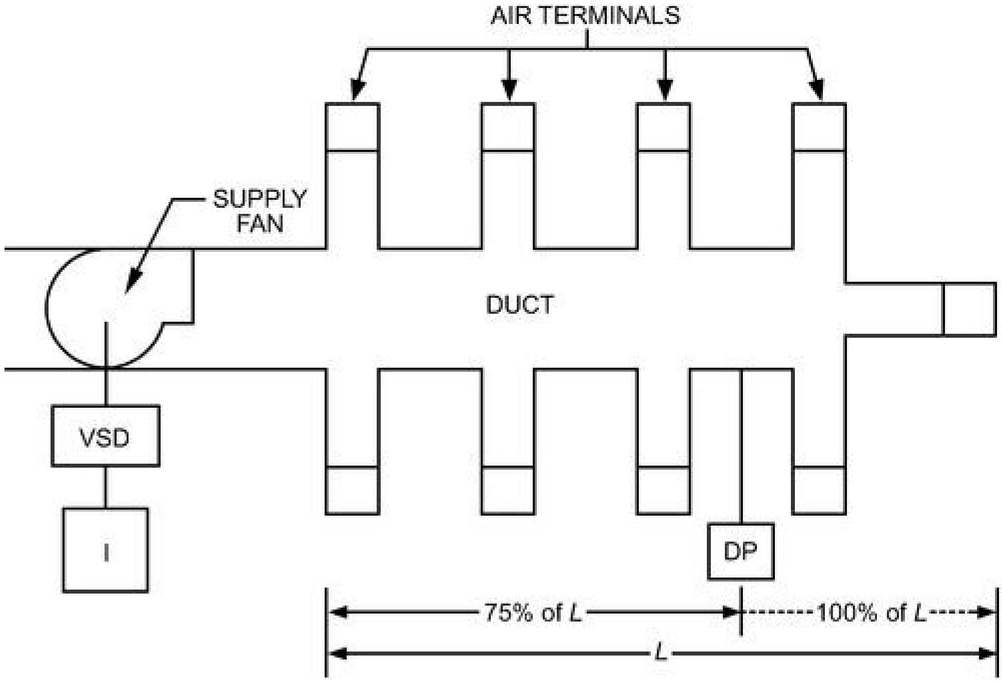

Figure 8. Duct Static-Pressure Control

Many methods have been published to automatically reset duct pressure (Ahmed 2001; EDR 2007; Englander and Norford 1992). Reported energy savings, monitored over weeks or months, have ranged from 30 to 50% of fan energy used by the same system running with a constant-pressure set point. All of these reset designs use data from terminal controllers to alter fan operation.

Most reset strategies use zone control data to adjust the set point of the duct pressure control loop. This makes the location of the pressure sensor much less important.

Other reset strategies (Hartman 1993) eliminate the pressure control loop, using data from zones to drive the fan directly.

Reset strategies may be categorized according to the type of data collected from the terminal controllers. At least three approaches are in commercial use. The terminal controllers may deliver

- Damper position (or damper position and flow error)
- Flow set point
- Saturation signal (terminal indicates that the pressure is insufficient)

Data available for coordinating a fan control system vary with the model of terminal controller. Most have both a flow set point and a damper position value, though the suitability of the data for coordination varies. Control system designers should ensure that the data available from terminal controllers, the fan control strategy, and network data capacity are compatible.

The signal selected for coordination can determine the data communication load that the fan control strategy places on the network. Flow set points and saturation signals tend to change less often than damper position or flow measurements, so using them may be more practical with lower available bandwidth, especially in systems with many terminals. Saturation signals are binary, so they do not indicate their distance from the critical point. This can affect reset algorithm design.

One approach (using **damper position data**) is based on the idea that the desired mechanical operating point occurs when at least one damper is fully (or almost fully) open, and all terminals deliver the required flow. The fan controller processes damper position data from each terminal and adjusts duct pressure (or fan speed) to drive one damper open. To ensure that the open box is not starved, the reference may be set a little lower (95% open, for example), or the controller may check flow (or flow error) data from the terminal controllers. Floating actuator application methods may result in unreliable damper position values for some terminal controllers. It is important to take this into account when selecting a reset method. Rogue zones can provide false feedback and keep the fan at a higher speed than is necessary, and should be identified and corrected or removed from the control logic. Rogue zones can be caused by improper box sizing, false thermostat readings, or constantly loaded spaces.

Another approach (using **flow set points**) is based on the fact that the required pressure depends on the distribution system (ducts, terminals, diffusers) and required flow. One way is to add the flow set points from each terminal and then use an empirically determined function to set the pressure. A more exact approach puts the individual flow set points into a calibrated model of the duct network, and calculates the pressure needed at fan discharge to drive the required flow to each terminal (Kalore et al. 2003). This online optimization applies the same calculations used to size a fan in real time. The pressure control loop then adjusts the fan speed to maintain the calculated pressure, which results in all terminals being satisfied, with one critical damper fully open (Ahmed 2001, 2002). This method is now in commercial use. In contrast to a reset based on damper position or saturation signals, a reset based on flow set point is open loop; this means that performance depends on careful calibration, but is inherently stable.

A third approach (using a **saturation signal**) distributes more of the logic. Each terminal controller uses flow data, damper position data, timing, or other information to decide whether its local loop is sufficiently supplied by the fan. If not, the saturation signal is activated. If a saturation signal is available, then the fan control algorithm depends less on the details of the terminal control than other methods. These signals are typically mated with a fan algorithm that ramps pressure up or down according to the number of unsatisfied terminals or resets static pressure set point using trim-and-respond logic (Taylor 2007).

To specify a pressure reset system, a designer can select the fan control algorithm, data that integrate terminal controllers, and characteristics of the communication network. Alternatively, the designer can specify the logic in performance terms (i.e., that the intended mechanical operating point is the lowest pressure that satisfies the terminals with at least one damper wide open). A performance-based specification allows proposals from vendors with a wider variety of equipment and algorithms.

Duct Static Pressure Limit Control. In larger fan systems, or where fire or fire/smoke dampers could close off a significant percentage of airflow, static pressure limit controls are recommended. When the high limit set point is reached (or low limit on the suction side of the fans for systems with economizer dampers), the fan is deenergized. Limit controls should be manually reset. On large fans, inertia of the fan wheel could damage the ductwork even after the fan is deenergized. Additional protection for the ductwork (e.g., duct pressure relief doors or mechanical relief dampers) is needed in these situations.

Space Pressure Control. Differential static-pressure control, differential airflow (CFM offset), and directional bleed airflow are methods used to control pressurization of a space relative to adjacent spaces or the outdoors. Typical applications include pressure barriers for any occupied space to prevent infiltration of moist unfiltered and untreated air, or to maintain interior comfort conditions. Applications requiring higher-performance controls include cleanrooms (positive pressure to prevent infiltration; see Chapter 18), laboratories and health care infection control (positive or negative, depending on use; see Chapter 8), and various manufacturing processes, such as spray-painting rooms (see Chapters 31 and 32 for industrial applications). The pressure controller usually modulates fan speed, dampers or airflow valves to maintain the desired pressure relationship or bleed airflow direction as exhaust volumes change. An alternative is to supply sufficient makeup air and to modulate a separate exhaust system to maintain space pressurization flow as auxiliary exhausts in the space are turned on or off.

Health Care Pressurization Codes, Regulations, and Application Design Guides. The Facility Guidelines Institute (FGI 2014) incorporated ASHRAE/ASHE *Standard* 170 into their guidelines. These guidelines include requirements for differential pressure or differential flow control for rooms such as positive and negative isolation rooms. Refer to the guidelines for details.

Building Pressurization. A slight positive building pressure (1 to 20 Pa) is generally desired to reduce infiltration of unconditioned outdoor air. Pressure results from the development of a pressurization flow between adjacent pressure zones. A zone is positive to an adjacent zone if the pressurization flow across the zone barrier is positive. Generally, outdoor air is required to pressurize the building as a whole.

Building static pressure control is one method for control of the relief or exhaust fan; this requires direct measurement of the space and outdoor static pressures. The inside static pressure measuring location must be selected carefully, away from openings to the outdoors, elevator lobbies, and other locations where it can be affected by wind pressure and drafts. Stack effect also affects the reading for tall buildings in hot or cold weather; multiple pressure zones with independent sensors controls may be required to maintain positive pressure on all floors without overpressurizing some. The outdoor static pressure measuring location must also be selected carefully and oriented to minimize wind effects from all directions. Even with good sensor port locations, pressure readings can fluctuate and should be buffered before using for control. If multiple fan systems serve areas that are open to one another, a single pressure control loop should be used to prevent instability.

The amount of minimum outdoor air for pressurization varies with building permeability and relief or exhaust fan operation. Control of building pressurization can affect the amount of outdoor air entering the building.

Proper return fan control for VAV systems is required for building pressurization. In one approach, outlined in ASHRAE *Guideline* 16, the return fan is controlled to maintain the return air plenum pressure while exhaust (relief) air dampers are controlled to maintain building static pressure (see Figure 9). For relief fan systems, the relief fan speed is generally directly controlled by building pressure.

Direct-measurement pressurization flow compares an interior static pressure location to an outdoor reference to modulate relief fan speed or relief dampers. This control allows for greater operational repeatability, and improved energy savings potential where there are natural relief paths such as operable windows.

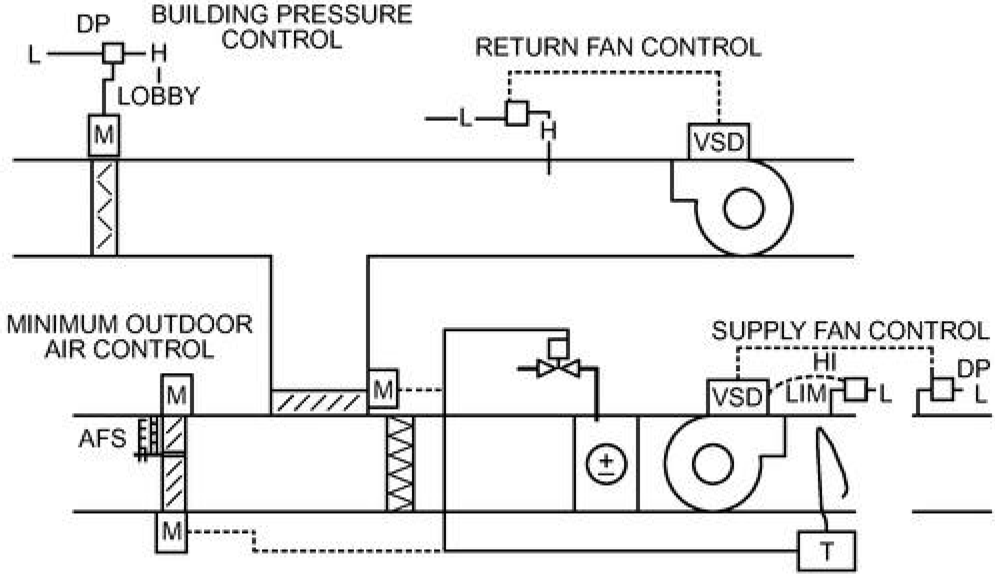

Figure 9. Supply/Return Fan Control

Indirect building pressure control uses duct or fan airflow measurements to control a fixed differential air volume by modulating dampers, fan speed, or discharge rates (Figure 10). Because return air is typically the controlled variable and its rate is set to track the normal changes in VAV supply at a fixed rate, this method is referred to as return fan or airflow tracking. The airflow differential set point is often determined empirically during commissioning as that needed to maintain a slight positive pressure with doors and windows closed.

Using fixed-differential air volume to maintain pressurization flow, rather than measured space static pressure, results in very stable control. Direct pressure control systems can be impacted by fluctuating pressures from gusts of wind, opening doors and windows, and multiple air-handling systems serving interconnected areas that interact. However, the fixed-differential control is indirect, so actual space pressure varies (e.g., with stack effect as outdoor air temperature changes). Also, fan tracking is less reliable than direct-measurement pressurization control because the cumulative error of the two airflow measurements can be large, particularly at low supply/return airflow rates (*Advanced VAV Design Guidelines*).

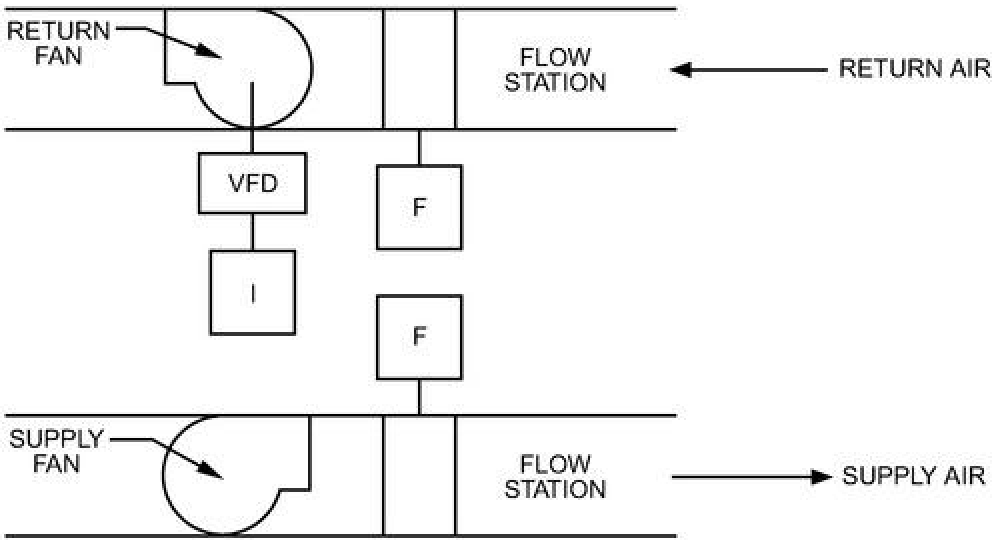

Figure 10. Airflow Tracking Control

Airflow quantity is indicated in Figure 11 by Q. Q_P is leakage in or out of the room, driven by the net pressure differential. Note that each surface may have a different ΔP because this value is relative to the pressure in the space on the other side of the wall.

When the control strategy changes from occupied (ventilation air required) to unoccupied warm-up, which does not require ventilation but needs thermal control to change the air-balancing requirements, warm-up is accomplished by setting return airflow equal to or just slightly less than the supply fan airflow, with toilet and other exhaust fans turned off and limiting supply fan volume to return fan capability. If exhaust fans remain running, then the supply fan must deliver sufficient outdoor air to make up the exhaust and still have a slightly pressurized space. During night cooldown, when using large quantities of outdoor air, the return fan operates in the normal mode (Kettler 1995).

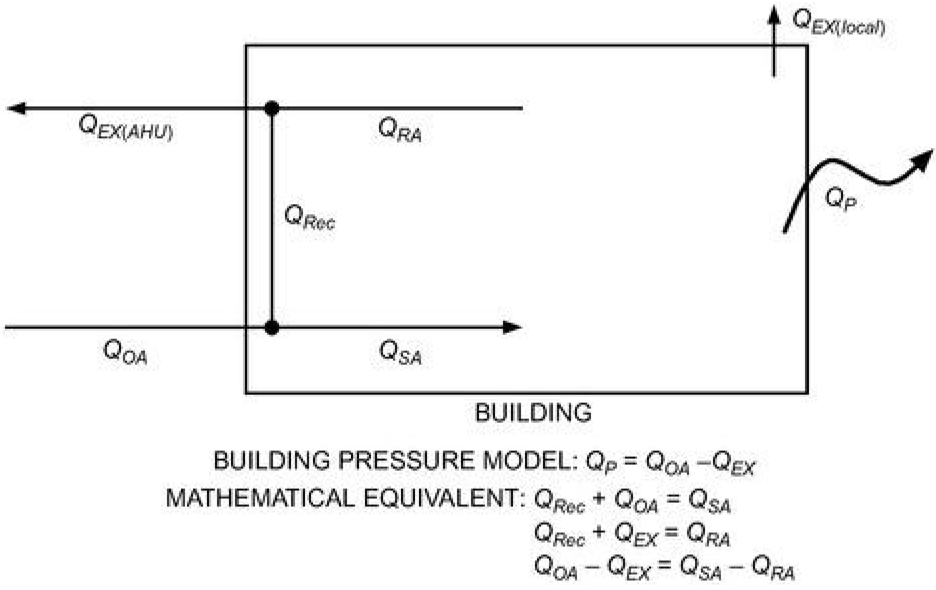

Figure 11. Building Pressure Model

Unstable fan operation in VAV systems can usually be avoided by proper fan sizing. However, if airflow reduction is large (typically over 60%), fan sequencing is often required to maintain airflow in the fan's stable range. Zone-based static pressure set point reset also allows the fan system with variable-speed drives to almost completely avoid the unstable region of its operating curves until fan speed is so low that instabilities are minor. This logic can allow very large VAV fans to serve very small airflows, during off-hours for instance.

Supply air temperature reset can be used to improve energy performance in most multiple-zone systems, including VAV systems, and is prescriptively required by energy standards such as ASHRAE *Standard* 90.1 for systems that have simultaneous heating and cooling at the zone level. In cool weather, supply air temperature can be reset upward based on zone demand, similar to static pressure reset. This reduces reheat energy losses and extends economizer operation, reducing mechanical cooling energy. In warmer weather, when space heat is not needed, supply air temperature should be reduced to reduce fan energy (EDR 2007).

Minimum Outdoor Air Control. Fixed minimum outdoor airflow control provides dilution air for ventilation, pressurization flow (usually exfiltration), and makeup air for exhaust fans. In some circumstances, minimum outdoor air may also provide combustion air for processes converting fuel to heat.

Several variations of minimum outdoor airflow control for VAV systems are possible (ASHRAE 2021; Felker and Felker 2010; Kettler 2000):

- Differential pressure is measured across the outdoor air intake louver or two-position minimum outdoor air damper. The differential pressure set point correlating to the minimum outdoor airflow is determined by measuring intake airflow directly upstream of the outdoor air damper in the field. This set point is maintained by modulating the return damper when not in economizer operation (Figure 12).
- A dedicated outdoor air injection fan with airflow station (Figure 13).
- An airflow station installed in the minimum outdoor air section with a minimum flow rate maintained by modulating the intake and return dampers in sequence (Figure 14). In this case, the intake opening should be sized for velocities high enough to facilitate measurement; some airflow sensors have relatively high minimum velocity requirements.

According to the *Standard 62.1 User's Manual* (ASHRAE 2021), VAV systems require one of the preceding methods for minimum outdoor air control or similar dynamic airflow controls for compliance; a fixed minimum damper position or a fixed-speed outdoor-air fan without control devices will not maintain rates within the required accuracy without overventilating.

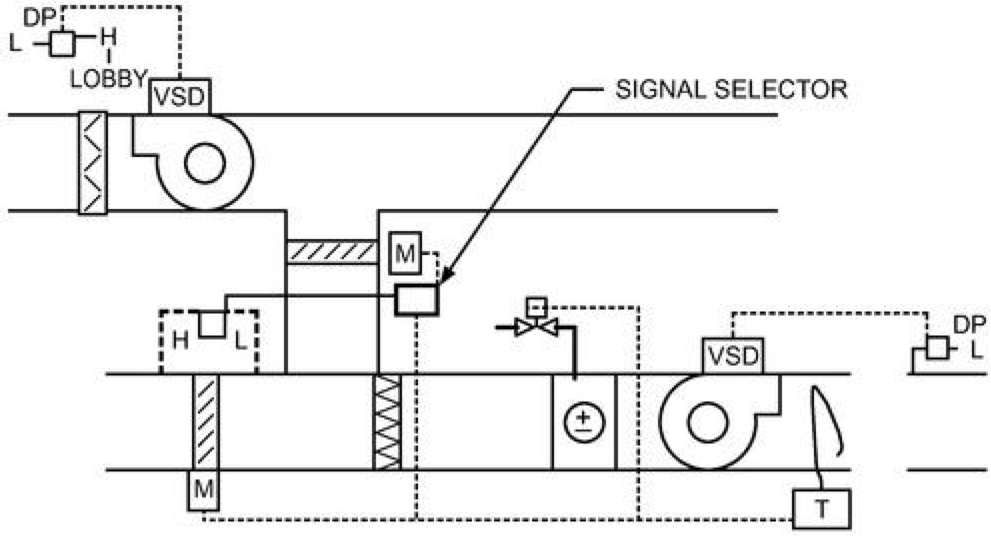

Figure 12. Minimum Outdoor Air Control Using Differential Pressure Controls

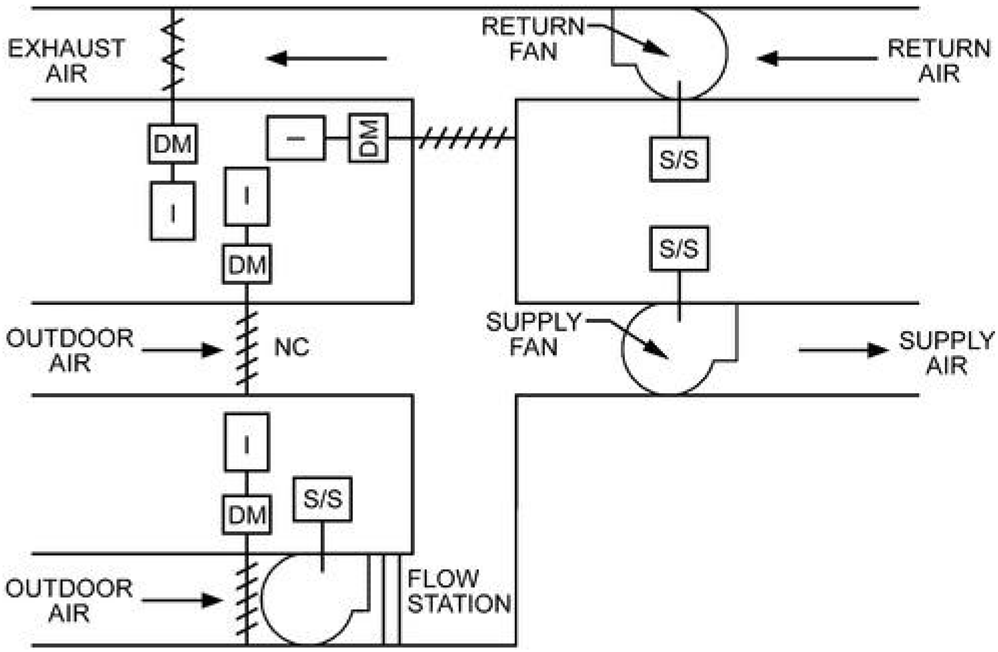

Figure 13. Minimum Outdoor Air Control with Outdoor Air Injection Fan

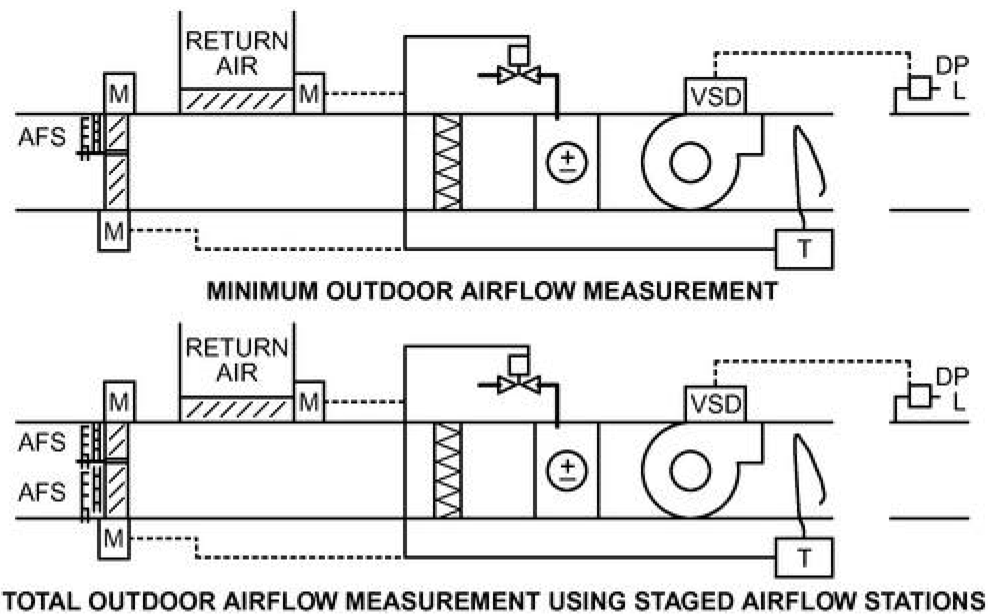

Figure 14. Outdoor Air Control with Airflow Measuring Stations

ASHRAE research project RP-980 (Krarti et al. 2000) and the *Standard 62.1 User's Manual* suggest that return fan or airflow tracking are unsatisfactory for minimum outdoor ventilation control because even small errors in measurements of total supply airflow and total return flow can cause significant errors in the determination and control of minimum outdoor airflow rates. Although airflow tracking may be an option for building pressurization control, minimum outdoor air must be controlled independently.

Measuring the total outdoor airflow range of a VAV design from a minimum of less than 50% to maximum design capacity requires a measurement tool that can provide the needed reliability across the entire anticipated temperature and velocity range. One way to do this with pitot arrays is to subdivide the intake and use dual airflow stations sized for 1/4 and 3/4, or 1/3 and 2/3, of the maximum opening size. This increases the velocity pressure for the pitot array to ensure accurate measurement at minimum pressure drop (Kettler 2000). Thermal velocity sensors, which have a much lower minimum velocity than pitot devices, may be used without creating damper sections.

Regardless of the type of system, pressurization flow rate and outdoor airflow rate are controlled separately: the two functions are related but must be independently controlled.

The outdoor airflow set point for dilution ventilation should be established using ASHRAE *Standard* 62.1. In addition, the outdoor air set point for pressurization should be established by adding the pressurization flow requirement to the sum of the local exhausts in the zones served by the air-handling system. The greater of the two dictates the outdoor air set point.

Traditional economizer controls call for the outdoor air and recirculation dampers to be modulated inversely to maintain set point: one opens as the other closes. A more energy-efficient approach for VAV systems is to decouple the outdoor air and recirculation dampers by individually actuating each. The outdoor airflow rate is then controlled by sequencing the dampers (ASHRAE *Guideline* 16). This reduces pressure drop and thus reduces fan energy.

Dynamic Reset of Minimum Outdoor Air Intake Rates. Demand-controlled ventilation (DCV) is a control scheme designed to reduce minimum outdoor air levels when the spaces served have less than design occupancy. The most common scheme is to use CO_2 concentration to reset the occupant component of the minimum outdoor air rate required by ASHRAE *Standard* 62.1.

Ventilation optimization or **reset control** is a related control scheme for resetting outdoor air and minimum supply air rates as system ventilation efficiency changes because of operational changes in the system. Both control schemes are required by ASHRAE *Standard* 90.1 for many applications and are described in detail in the user's manual (ASHRAE 2021) for ASHRAE *Standard* 62.1.

Table 1 Economizer Damper Type and Sizing

Relief System	Damper	Blade Type	Face Velocity, m/s
Return fan	Relief/exhaust	Opposed	5.1 to 7.6
	Outdoor air	Parallel	2.0 to 5.1
	Return air	Parallel	Per ΔP across damper ~7.6
Relief fan or barometric	Outdoor air	Parallel	2.0 to 5.1
	Return air	Parallel	4.1 to 5.1

When implementing dynamic ventilation reset schemes that reduce outdoor air intake, ensure that pressurization flow is maintained (i.e., the relationship between outdoor and exhaust air is maintained). When outdoor air dew point approaches or exceeds 16°C, a net positive pressurization flow is required to prevent transport of water and outdoor air contaminants into the building or its envelope (ASHRAE *Standard* 62.1).

Air-Side Economizer Cycle. Economizer-cycle control reduces cooling costs when outdoor air is cool and dry enough to be used as a cooling medium. The economizer is enabled when outdoor air conditions are below the high-limit device setting. When enabled, the economizer return and outdoor air dampers modulate to maintain a supply air temperature in sequence with the mechanical cooling. Typically, the economizer is controlled in sequence with the mechanical cooling, using the same supply air temperature control loop. Figure 15 shows integrated control, in which the economizer and mechanical cooling can be active at the same time. This is prescriptively required in most applications by ASHRAE *Standard* 90.1. When the outdoor air temperature exceeds the economizer high-limit set point, the economizer is disabled and only minimum outdoor air is supplied.

ASHRAE *Guideline* 16 addresses the sizing and selection of dampers for outdoor air economizer systems. Table 1 summarizes the guideline's recommendations as a function of the relief air system. Refer to the guideline for additional details and rationale.

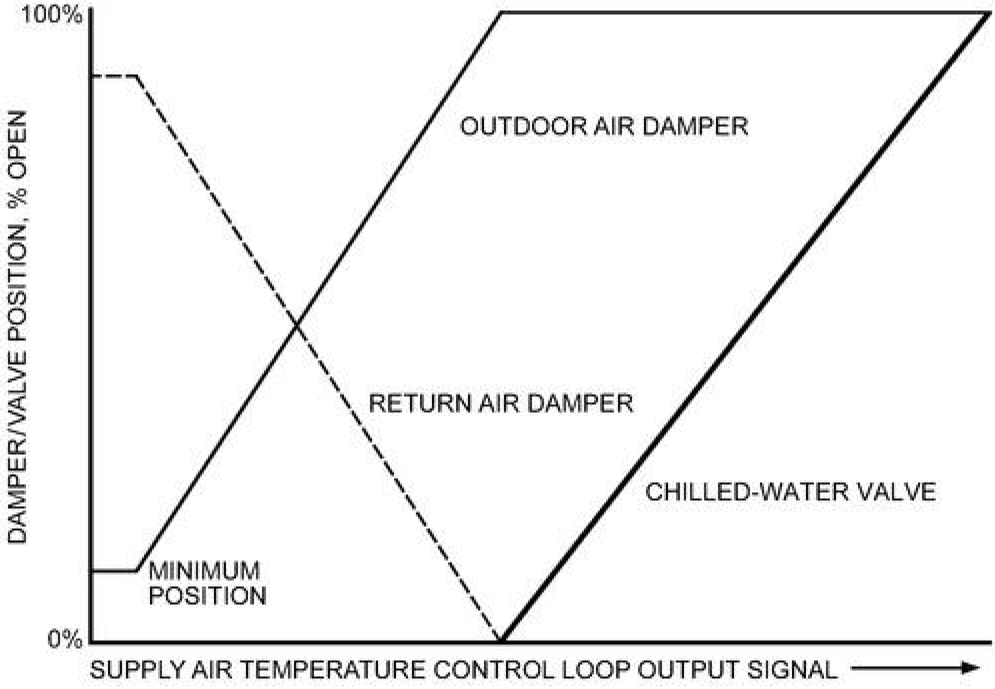

Figure 15. "Integrated" Economizer Cycle Control

High-limit controls are intended to disable the economizer when supplying outdoor air would use more energy than recirculating air. Common high-limit controls are

- Fixed dry-bulb temperature (compares outdoor air dry bulb to a fixed set point)
- Differential dry-bulb temperature (compares outdoor air dry bulb to return air dry bulb)
- Fixed enthalpy (compares outdoor air enthalpy to a fixed set point)
- Differential enthalpy (compares outdoor air enthalpy to return air enthalpy)
- Electronic enthalpy (compares outdoor air temperature and humidity to a set point that is a curve on the psychrometric chart)
- Combinations of these controls

ASHRAE *Standard* 90.1 includes some limitations on which controllers can be used and controller set points based on climate zone. The most energy-efficient high limit theoretically is a combination of differential enthalpy and differential dry-bulb temperature. However, it effectively requires four sensors (one temperature and one humidity in each of the outdoor air and return airstreams), all of which have inaccuracy and can get out of calibration, in particular the humidity sensors. Sensor error may result in increased energy usage relative to other, less expensive high-limit controls. In practice, the simplest, least expensive, and most reliable high-limit control is a fixed outdoor air dry-bulb temperature sensor set to the set point prescriptively required by ASHRAE *Standard* 90.1.

The relief air system should be enabled during economizer operation because the large quantities of outdoor air should leave the building along a planned path of flow and not an unplanned path, such as entry doors that may be pushed open.

VAV warm-up control during unoccupied periods requires no outdoor air if exhaust fans are off; typically, outdoor and exhaust dampers remain closed. Where a return fan is installed, the supply fan and return airflows are offset to maintain zero differential airflow.

Where outdoor conditions allow, night cooldown control provides 100% outdoor air for cooling during unoccupied periods. The space is cooled to the space set point, typically 5 K above outdoor air temperature. Limit controls prevent operation if outdoor air is above space dry-bulb temperature, if outdoor dew-point temperature is excessive, or if outdoor dry-bulb temperature is too cold (typically 10°C or below). When outdoor air conditions are acceptable and the space requires cooling, the cooldown cycle is the first phase of the optimum start sequence.

During unoccupied mode, with air-handling units off or not providing outdoor air, offgassing from building contents and construction materials can accumulate in the space. A **preoccupancy purge** sequence may be used to dilute the resultant volatile organic compounds (VOCs) before initial or daily scheduled occupancy. Purge damper settings are a fixed set point and should be adjusted equivalent to the building floor area component of the minimum outdoor air damper settings.

Humidity Control

Humidity control relies on the output of a humidity sensor located either in the space or in the return air duct. Most comfort cooling involves some natural but uncontrolled dehumidification. The amount of dehumidification is a function of the effective coil surface temperature and is limited by the coolant's freezing point. If water condensing out of the airstream freezes on the coil surface, airflow is restricted and, in severe cases, may be shut off. The practical limit is about 5°C dew point on the coil surface. As indicated in Figure 16, this results in a relative humidity of no less than 30% at a space temperature of 24°C. When lower humidity is needed for a process application (e.g., dryroom), a desiccant dehumidifier is required.

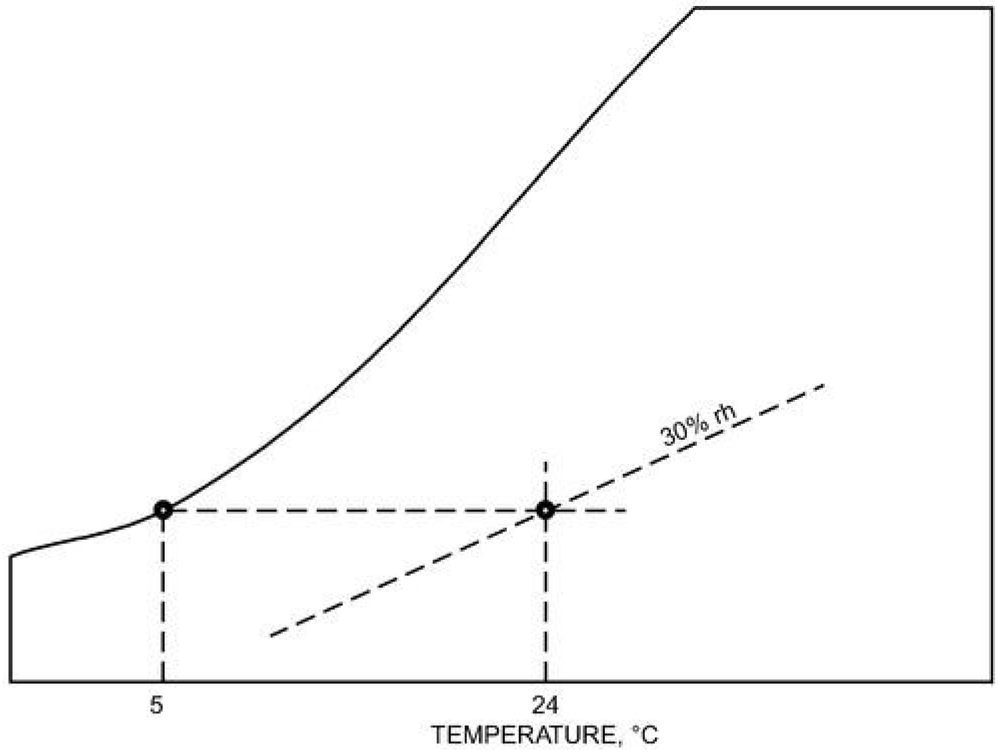

Figure 16. Psychrometric Chart: Cooling and Dehumidifying, Practical Low Limit

Although simple cooling by refrigeration typically provides dehumidification as a by-product of the cooling process, without additional equipment, it does not directly control space humidity. **Dehumidification** can be directly controlled in several ways. If relative humidity is the critical measure, adding heat to the space decreases the relative humidity, but usually the object is to remove moisture and lower the dew point. One method is to control the cooling coil based on relative humidity, not space temperature. The supply air temperature leaving the coil is lowered until enough moisture is removed from the supply air to maintain the humidity set point. When a relative humidity limit is required, a space or return air humidistat is provided in addition to the space thermostat. A control function selects the higher of the output signals from the two devices and controls the cooling coil valve to provide either temperature or humidity control. A low-temperature coil used to remove moisture from an airstream may lower a space temperature below the desired set point. The process line in Figure 16 shows a significant loss of sensible temperature before moisture is removed. As space temperature decreases, the relative humidity increases even though the absolute humidity (dew point, grams of moisture, kilograms of water to kilograms of air) decreases.

A reheat coil is required to maintain the space temperature if moisture removal results in too low a supply air temperature. This coil may be located in the AHU, as shown (Figure 17), or a space-temperature-controlled reheat coil may be provided at the room terminal unit.

Sprayed-coil dehumidifiers have been used for dehumidification. Space relative humidity ranging from 35 to 55% at 24°C can be obtained with this equipment; however, the costs of maintenance, reheat, and removal of solid deposits on the coil make the sprayed-coil dehumidifier less desirable than other methods.

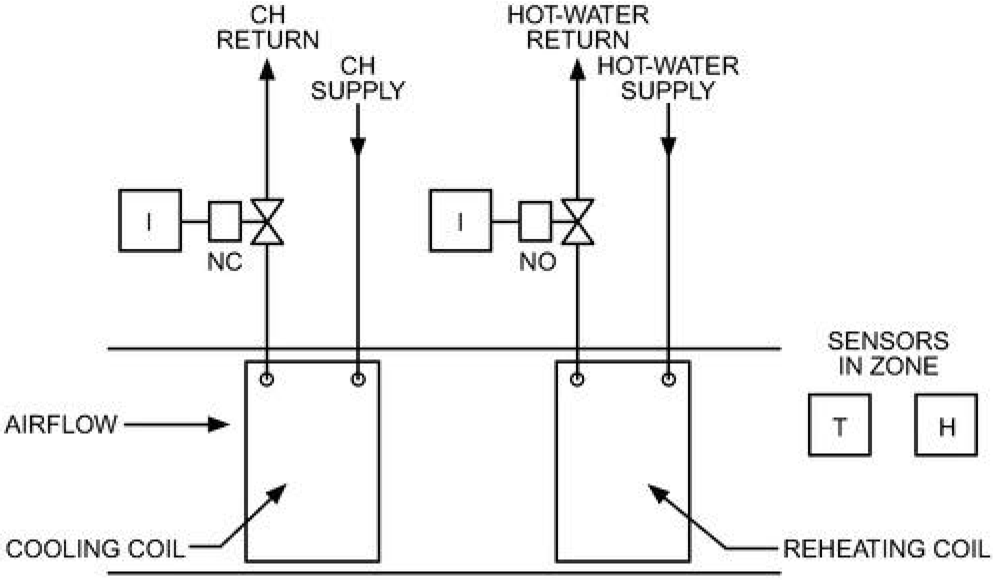

Figure 17. Cooling and Dehumidifying with Reheat

Air washers with a cold-water spray (down to 5°C) do not require the coil for cooling, though mist eliminators add to the equipment in the air handler. The spray cools the air, removes moisture down to the temperature of the spray water, and collects and concentrates particulates in the spray pan. As with the spray coil, this is messy and slower to respond than other methods.

A **desiccant-based dehumidifier** can lower space humidity below that possible with cooling/dehumidifying coils. This device adsorbs moisture using silica gel or a similar material. For continuous operation, heat is added to the adsorbent material out of the dehumidification airstream to evaporate moisture and regenerate the material. This is often in a wheel configuration that rotates the gel from the wet side, where it absorbs moisture, into the heated dryer side, where moisture is driven off and exhausted from the system. The adsorption process adds heat to the dehumidified air. Cooling is required, but at a warmer temperature to limit the need for lower-temperature cooling coils. The psychrometric process is shown in Figure 18, and Figure 19 shows a typical control. There are two control loops to consider: (1) the heating source must be hot enough to regenerate the media for effective moisture removal, and (2) the cooling coil needs only lower the temperature to a point that provides a comfortable space condition. Because the moisture has been removed by adsorption, the conditioned air to the space may be 5 to 7 K warmer than required for an AHU that uses only cooling to control moisture. When the outdoor air is drier, regeneration can use a lower-temperature source, saving heating and subsequent cooling energy.

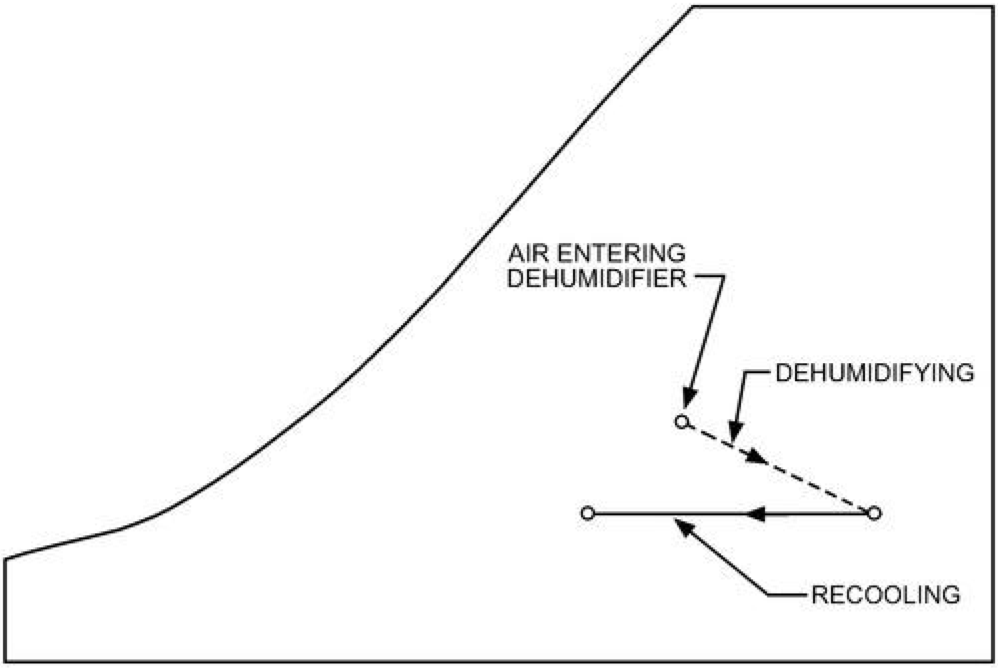

Figure 18. Psychrometric Chart: Desiccant-Based Dehumidification

Humidification can be achieved by adding moisture to supply air, using evaporative pans (usually heated), steam injection, or atomizing spray tubes. A space or return air humidity sensor provides the necessary signal for the controller. A humidity sensor in the duct should be used to minimize moisture carryover or condensation in the duct

(Figure 20). With proper use and control, humidifiers can achieve high space humidity, although they more often are used to maintain design minimum humidity during the heating season. Atomized fine droplets make it easier to mix water in to an airstream, but converting a droplet into water vapor requires heat for the change of state. Because evaporative cooling occurs with an atomized method, additional heat must be provided. Steam is already a vapor, so additional heat is not required for the humidification process. It is important to have not only a space or return sensor to call for adding humidity, but also a duct sensor just downstream of the humidifier to limit the moisture concentration being injected to less than 85%, to avoid condensation on the duct walls.

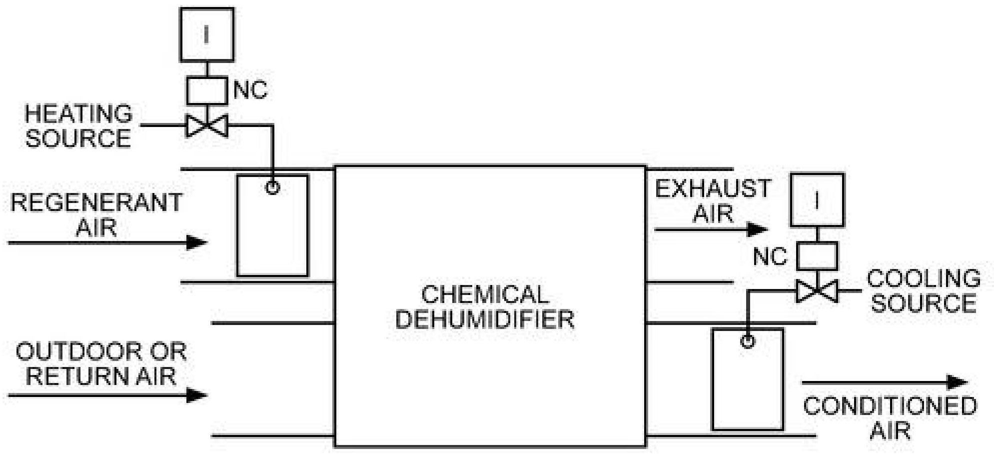

Figure 19. Desiccant Dehumidifier

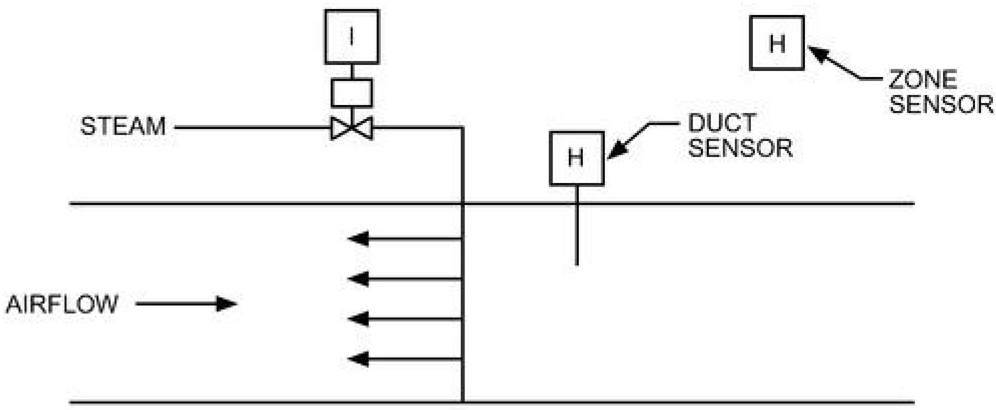

Figure 20. Steam Injection Humidifier

Single-Zone Systems

A single-zone system (Figure 21) is an air handler serving one area of a building that has similar loads and occupancy throughout the zone. Single-zone systems do not require terminal boxes, because zone temperature can be maintained by modulating fan speed and using the heating and cooling control valves (and optional economizer dampers) to control discharge air temperature, as indicated in Figure 22. The fan is typically variable speed, which is now required for many applications by energy standards such as ASHRAE *Standard* 90.1.

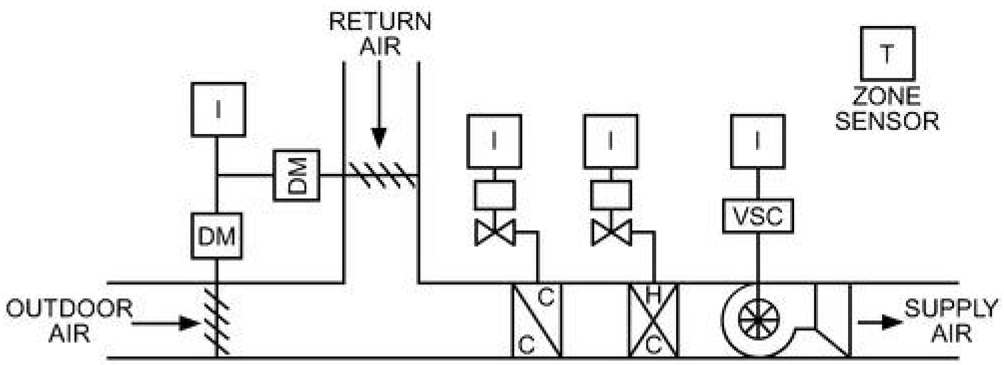

Figure 21. Single-Zone Fan System

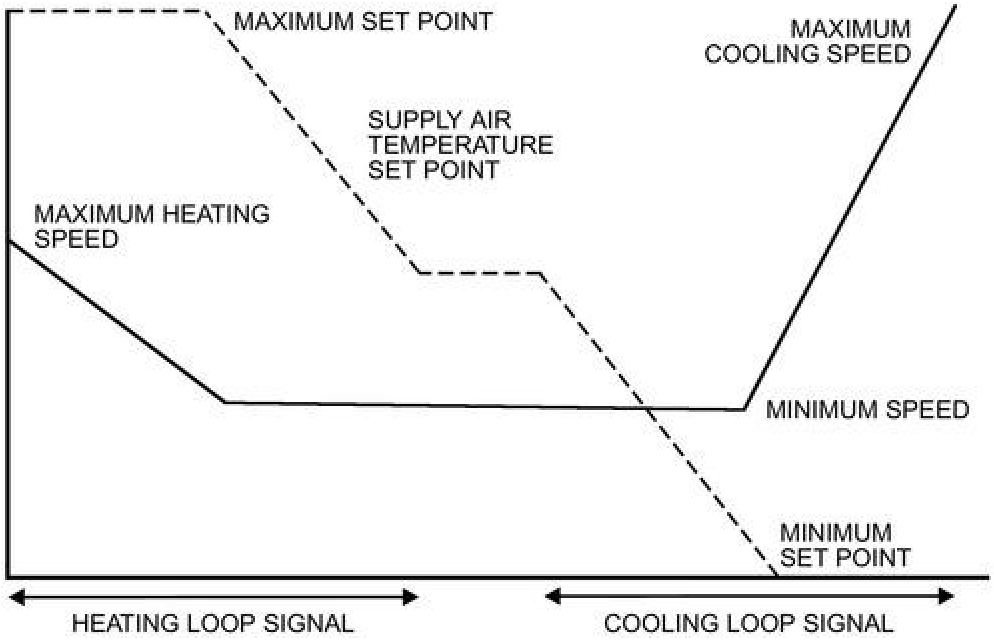

Figure 22. Single-Zone VAV Control

A **unit ventilator (UV)** is designed to heat and ventilate a space by introducing up to 100% outdoor air. Optionally, it can cool and dehumidify with a cooling coil (either chilled water or direct expansion). Heating can be by a gas furnace, hot water, steam, or electric resistance. Control of these coils can be by valves or by face-and-bypass dampers.

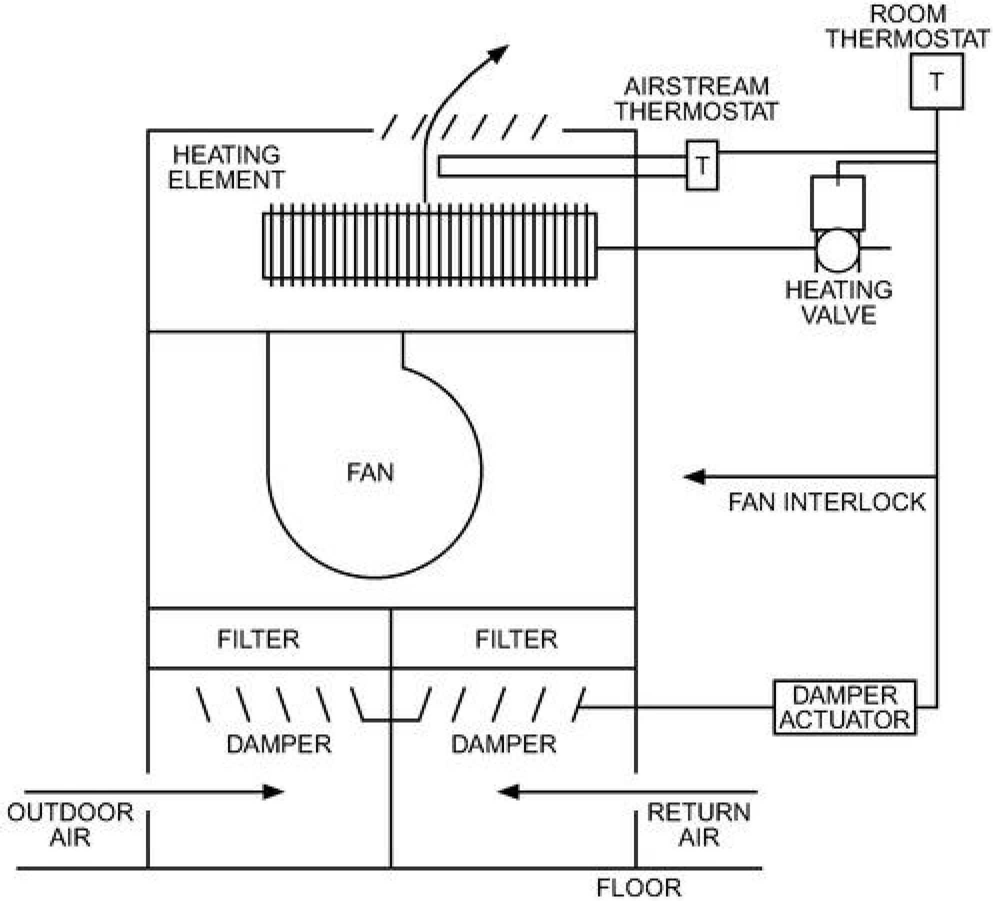

Figure 23. Unit Ventilator Control Arrangements

A typical control sequence for a UV is shown schematically in Figure 23 and logically in Figure 24. During the heating stage, this approach (Figure 23) supplies a set minimum quantity of outdoor air. Outdoor air is gradually increased as required for cooling. During warm-up, the heating valve is open, the outdoor air (OA) damper is closed, and the return air (RA) damper is open. As space temperature rises into the operating range of the space thermostat, ventilation dampers move to their set minimum ventilation positions. The heating valve and ventilation dampers are operated in sequence as required to maintain space temperature. The airstream thermostat can override space thermostat action on the heating valve and ventilation dampers to prevent discharge air from dropping below a minimum temperature. Figure 24 shows the relative positions of the heating valve and ventilation dampers with respect to space temperature.

Makeup Air and Dedicated Outdoor Air Systems (DOAS)

Makeup air units (Figure 25) are 100% outdoor air units that replace air exfiltrated or exhausted from the building through toilet, laboratory, industrial, and combustion processes. Makeup air is often supplied at or near space conditions, but may also provide space heating and cooling. The makeup air fan is usually turned on, either manually or automatically, whenever exhaust fans are turned on. However, the fan should not start until the outdoor air damper is fully open as proven by an end switch. The two-position outdoor air damper remains closed when the makeup fan is not in operation. The outdoor air limit control opens the preheat coil valve when outdoor air temperature drops to the point where the air requires heating to raise it to the desired supply air temperature. To prevent low-temperature shutdowns on start-up, the heating coil should begin circulation and prove heating is available before the outdoor damper opens and the fan starts. A capillary element freezestat located adjacent to the coil shuts the fan down for low-temperature detection if air temperature approaches freezing at any spot along the sensing element.

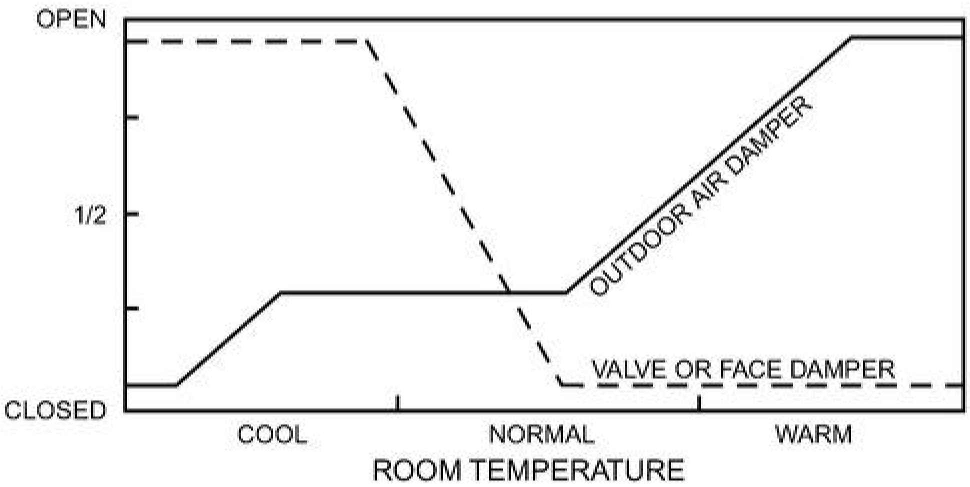

Figure 24. Valve and Damper Positions with Respect to Room Temperature

Fuel-fired makeup air units have staged or modulating fuel-fired direct or indirect furnaces. Units have manufacturer-supplied controls and safeties for flame proving, airflow proving, and discharge air low limit to meet the ANSI *Standard* Z83.4/CSA *Standard* 3.7 combined safety standard. The manufacturer's controls either include temperature controls or provide an interface for the control contractor's temperature controls.

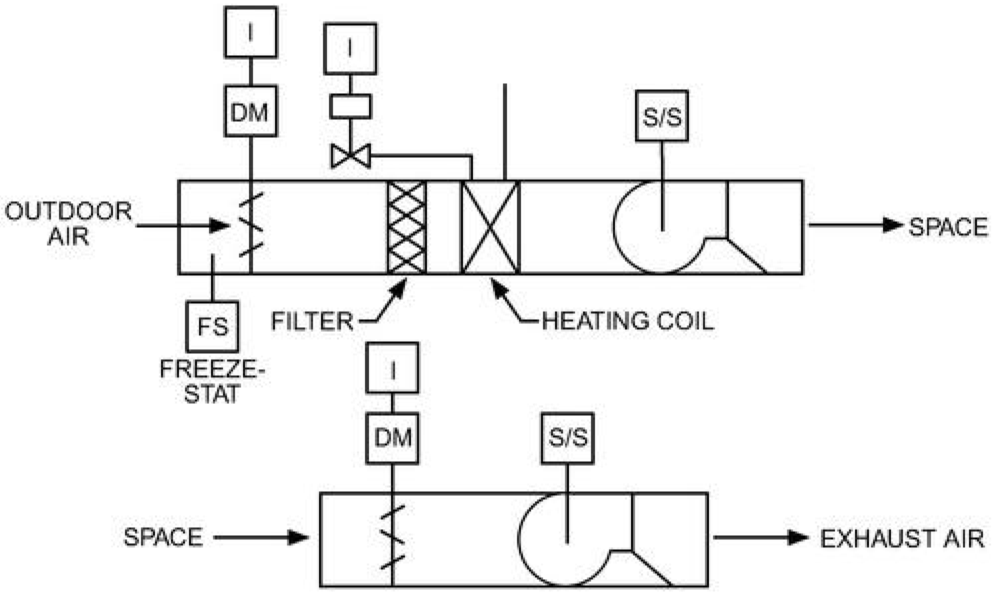

Figure 25. Makeup Air Unit

Dedicated outdoor air systems (DOAS) are 100% outdoor air units that provide conditioned ventilation air to building spaces or to other air systems. DOAS are used to decouple outdoor air cooling and heating loads from space cooling and heating loads, and are specially designed for conditioning 100% outdoor air. Airflow from a DOAS can be constant volume or VAV. The decision to use constant or variable volume is typically driven by whether the system will use demand controlled ventilation (DCV) or ventilation optimization, as described in ASHRAE *Standard* 90.1. Outdoor air reduction must be carefully examined to ensure enough air is available to maintain pressure in the building; however, reducing outdoor air volume allows for energy savings due to reduced cooling and heating of the ventilation air and reduced fan energy. In VAV DOAS, fan speed is controlled based on duct static pressure, similar to a central VAV AHU system, and pressure set point can be reset based on zone outdoor air damper position.

DOAS are typically controlled to maintain a supply air dew-point temperature, based on the dehumidification needs of the spaces served. Supply air temperature can be held constant for simplicity or reset as loads change (e.g., based on sensed space humidity, outdoor air temperature, or number of zones needing reheat). Older DOAS supplied air at space-neutral temperature to avoid overcooling, but more recent trends are to supply air below space-neutral temperature to avoid losing the benefit of cooling energy used for dehumidification. Supplying colder air allows space terminal unit cooling coils to be downsized, but could result in reheat loads in some spaces. DOAS supply air temperature needs to be carefully selected to minimize reheat at both the DOAS equipment and space level while maximizing the benefit of cooling provided at the DOAS.

DOAS typically include energy recovery in systems where exhaust air can be routed to the DOAS. Energy recovered from exhaust air reduces the energy required for cooling and heating ventilation air. DOAS can also include hot gas reheat where the required supply air dew point results in a need for reheat at the DOAS unit. Using heat generated by the dehumidification process provides “free” reheat, though sometimes the modulation capacity of the reheat coil is limited.

For more information on control sequences for DOAS, please see the ASHRAE (2017) *Design Guide for Dedicated Outdoor Air Systems.*

Multiple-Zone, Dual-Duct Systems

A **single-fan, dual-duct system** uses a single fan to supply separate heating and cooling ducts (Figure 26). Dual-duct terminal mixing boxes are used to control the zone temperature. For VAV terminals, static control is similar to that in VAV single-duct systems except that static pressure sensors are needed in each supply duct. A controller allows the sensor detecting the lowest pressure to control the fan output, thus ensuring that there is adequate static pressure to supply the necessary air for all zones.

The hot deck has its own heating coil, and the cold deck has its own cooling coil. Each coil is controlled by its own **discharge air temperature controller**. The hot deck set point may be reset from the zone with the greatest heating demand, and the cold deck set point may be reset from the zone with the greatest cooling demand.

Cooling supply air temperature control is similar to that in single-duct systems, with economizer dampers sequenced with the cooling coil. The economizer causes the supply air temperature entering the hot deck to be as cold as the cold deck, increasing heating energy usage. Because of this inefficiency, single-fan, dual-duct systems with air economizers are not allowed by many energy standards such as ASHRAE *Standard* 90.1.

Dual-fan, dual-duct systems (Figure 27) use separate supply fans for the heating and cooling ducts. This eliminates the economizer inefficiency of single-fan, dual-duct systems. Static-pressure control is similar to that for VAV single-fan, dual-duct, systems, except that each supply fan has its own static pressure sensor and control.

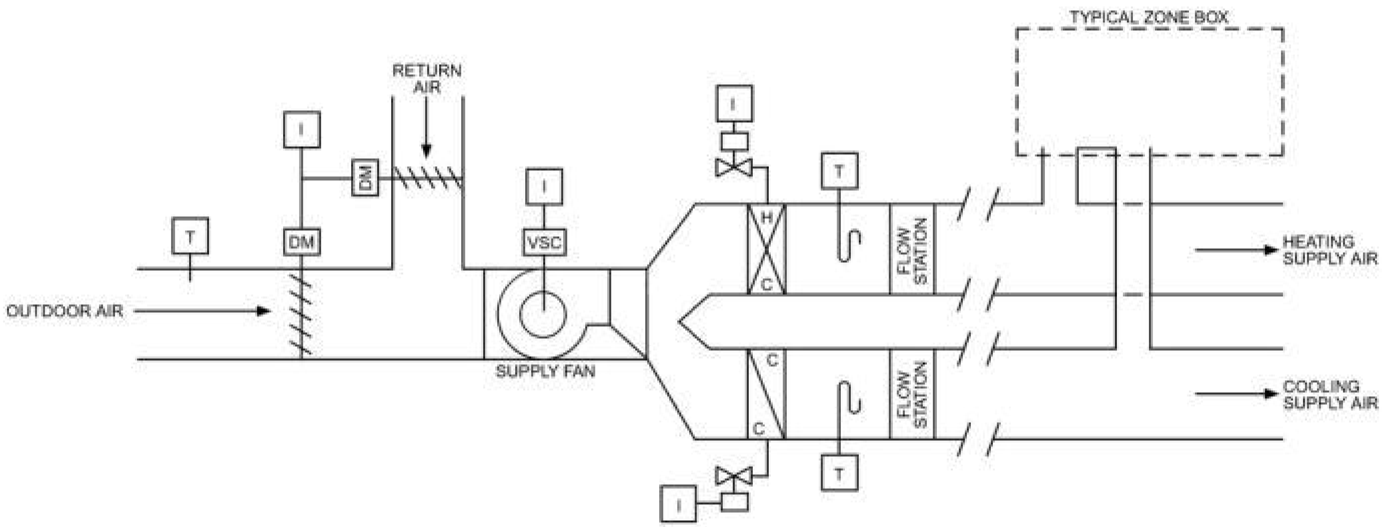

Figure 26. Single-Fan, Dual-Duct System

5. HEATING SYSTEMS

Heating systems include boilers, fired by either fuel combustion or electric resistance, furnaces, electric resistance air heaters, and heat pumps. Systems involving combustion require multiple safeties, which should always follow local codes and manufacturers’ guidelines. Load affects the required rate of heat input to a heating system. The rate is controlled by cycling a fixed-intensity energy source on and off, or by modulating the intensity of the heating process. Heating systems may come with factory-mounted controls capable of handling safeties, cycling, and modulation. In addition, a BAS can be used to interface with the heating system or to control it directly. The designer decides under what circumstances to turn equipment on and off in sequence and at what temperature set point to maintain the supply air or water.

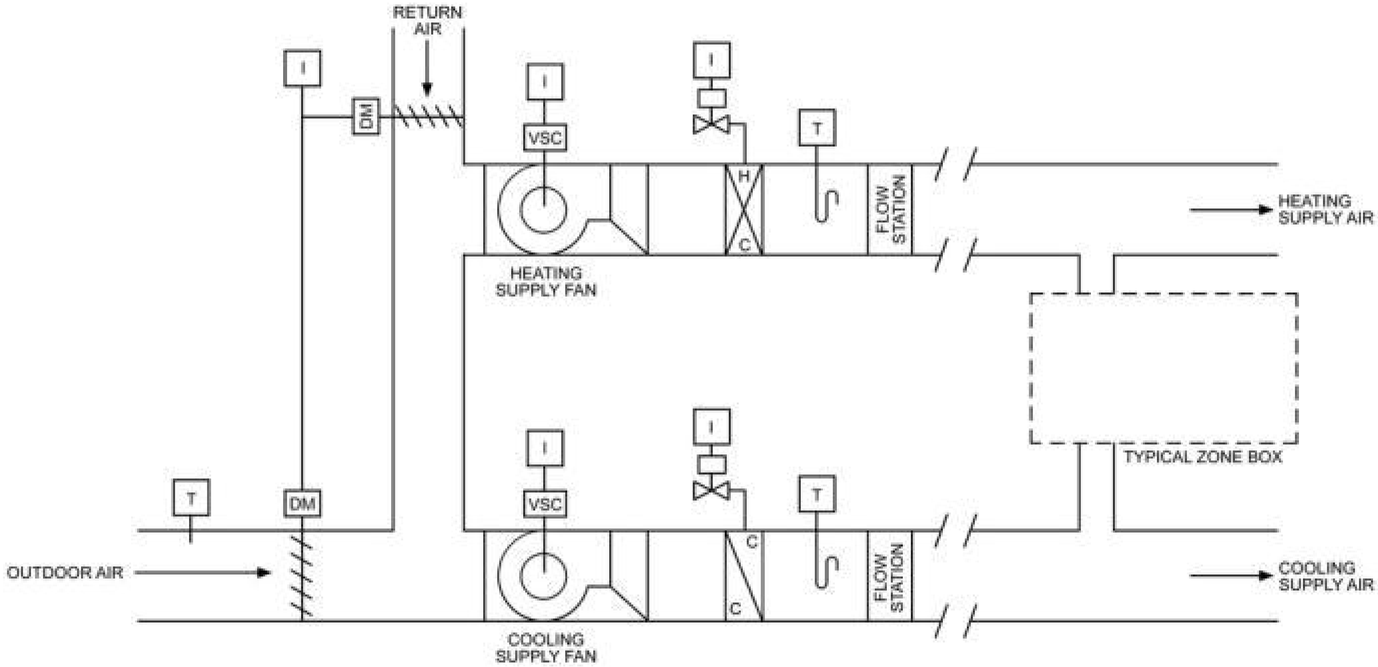

Figure 27. Dual-Fan, Dual-Duct Systems

Heating Coils

Heating coils that are not subject to freezing can be controlled by simple two- or three-way modulating valves (Figure 28). (For information on the air side of coils, see the section on Air Systems.) The modulating valve is controlled by coil discharge air temperature or by space temperature, depending on the HVAC system. In cold regions, valves are set to normally open to allow heating if control power fails. In many systems, the supply air temperature set point is reset based on the outdoor air temperature, zone damper positions, return air temperature, or some other load proxy.

Heating coils in central air-handling units preheat, reheat, or heat, depending on the climate and the amount of minimum outdoor air needed. They can also provide morning warm-up on systems with limited zone heating capacity.

The equipment heating coil that first receives the outdoor air intake, even if mixed with indoor air, must have protection against freezing in cold climates, unless (1) the minimum outdoor air quantity is small enough to keep the mixed air temperature above freezing in all expected operating conditions and (2) enough mixing occurs to prevent stratification. Even when the average mixed-air temperature is above freezing, inadequate air mixing may allow freezing air to impinge on small areas of the coil, causing localized freezing. This blocks flow and, without a heat source, the rest of the coil and equipment downstream is at risk. Preheating coils that heat 100% outdoor air always need (1) protection against freezing and (2) constant water or steam flow in cold climates.

Steam preheat coils should have two-position valves and vacuum breakers to prevent condensate build-up in the coil. The valve should be fully open when outdoor (or mixed) air temperature is below freezing. This causes unacceptably high coil discharge temperatures at times, necessitating face-and-bypass dampers for final temperature control (Figure 29). The bypass damper should be sized to provide the same pressure drop at full bypass airflow as the combination of face damper and coil does at full airflow. When the outdoor air temperature is safely above freezing (roughly 1.7°C), the bypass damper is full open to the coil face and the coil valve can be modulated to improve controllability.

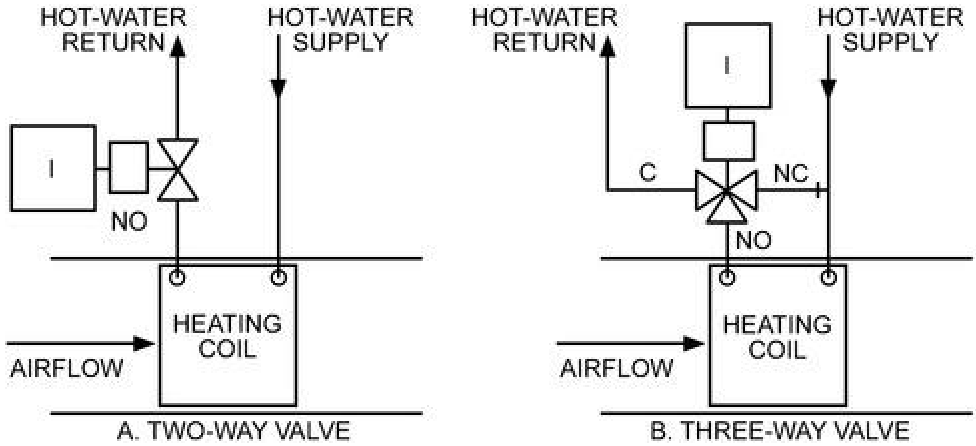

Figure 28. Control of Hot-Water Coils

Hot-water coils must maintain a minimum water velocity in the tubes (on the order of 0.9 m/s) to avoid stratification and ensure proper heat transfer by maintaining turbulent flow. A two-position valve combined with face-and-bypass dampers (Figure 29) or a coil pump can be used. There are many coil pump piping schemes; the most common are shown in Figures 30 and 31. In each scheme, the control valve modulates to maintain the desired coil air discharge temperature and the pump maintains the minimum tube water velocity needed when the outdoor air is below freezing. Pumped coils can still freeze in very cold regions, so additional low-temperature protection measures such as freezestats, glycol-based fluids, and default-to-open valves should be used.

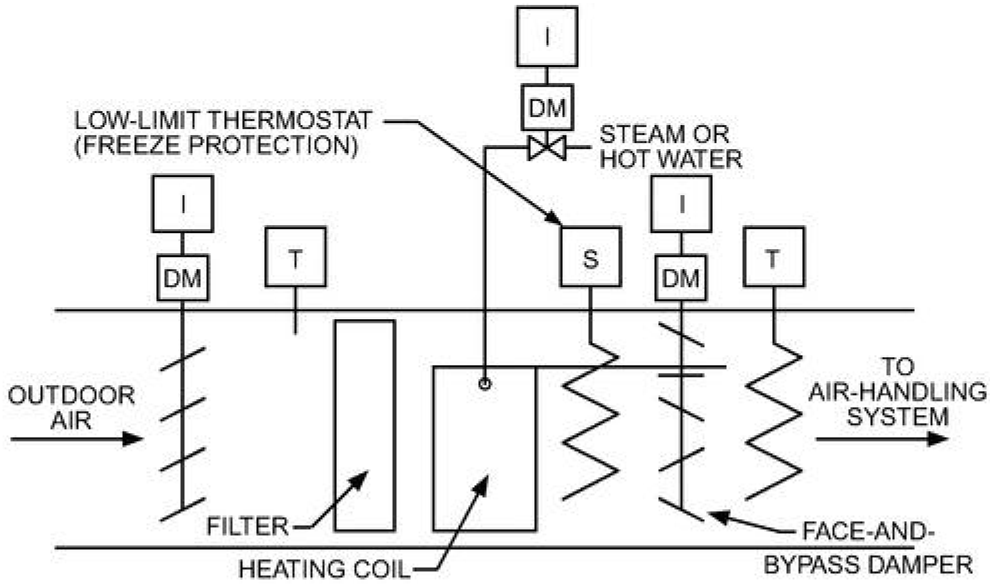

Figure 29. Preheat with Face-and-Bypass Dampers

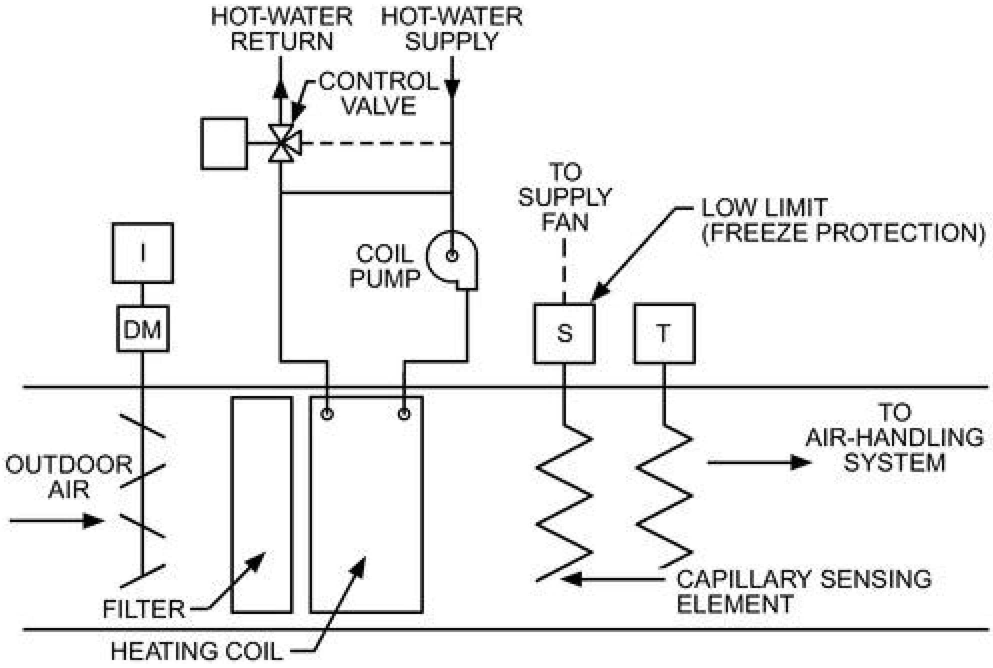

Figure 30. Coil Pump Piped Primary/Secondary

A low-temperature detector (commonly called a **freezestat**), is a long, refrigerant-filled capillary tube used as a low-temperature sensing switch. If any short section of the tube is exposed to a low temperature (typically 3.3°C), it can provide an alarm or a hardwired interlock to shut the outdoor damper and open the return damper, or shut down the fan.

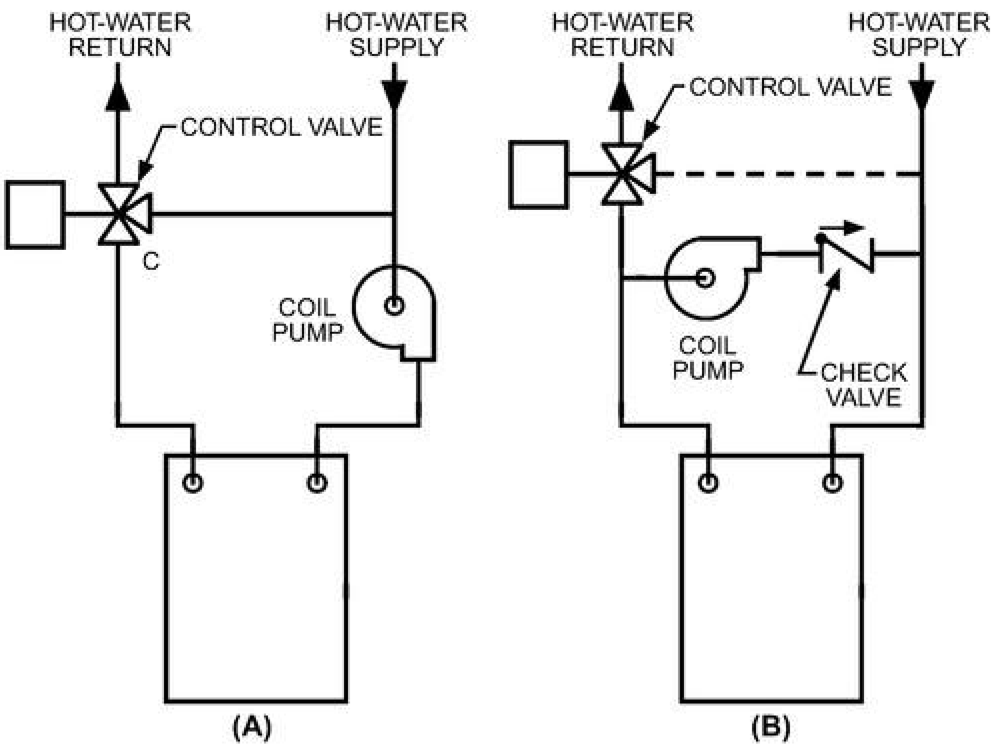

Figure 31. Pumped Hot-Water Coil Variations: (A) Series and (B) Parallel

Figure 30 shows the conventional primary/secondary (or secondary/tertiary) arrangement where the coil pump and the pumps feeding the coil are hydraulically independent. It results in constant flow through the coil and in either variable flow through the primary loop, if a two-way valve is used, or in constant flow through the primary loop, if a three-way valve is used (shown dashed in the figure).

In Figure 31A, the coil pump is in series with the primary pumps; this results in variable primary flow, which can affect flow through parallel coils that do not have pumps when the three-way valve moves, unless the primary system has variable-speed pumps and proper pressure control. The pump is decoupled when the three-way valve is closed to the system. The three-way valve must be oriented with the common port connected to the coil so its flow is not affected by the valve position.

Figure 31B shows the coil pump piped in parallel with the primary pumps. This design has the advantage that hot-water flow can be achieved through the coil if the coil pump fails. This design results in coil flow that varies from the pump design flow rate (when the control valve is closed) up through the sum of the pump flow rate plus the primary system flow rate (when the valve is wide open). Unlike the options in the previous two figures, the primary pump must be sized for the pressure drop through the coil at this high flow rate. Similarly, the coil pump must be sized for that pressure drop and not only for the flow it supplies. Flow through the primary circuit may be variable, if a two-way valve is used, or constant, if a three-way valve is used.

Some systems may use a glycol solution in combination with any of these methods; however, glycol affects control valve sizing (see Chapter 46 of the 2020 *ASHRAE Handbook—HVAC Systems and Equipment*) and requires additional maintenance and careful handling.

Steam Coils. Modulating steam coils are controlled in much the same way as water coils. Control valve size and characteristics are important to achieve proper control (see Chapter 46 of the 2020 *ASHRAE Handbook—HVAC Systems and Equipment*). Because the entering steam is hotter than the entering temperature of most water coils, a steam coil typically responds more rapidly and is smaller than a comparable water coil. In low-temperature applications, two-position control should be used, as discussed previously. For applications that require precise control at low loads, valves should be in a 1/3 and 2/3 arrangement.

Electric heating coils (duct heaters) are controlled in either two-position or modulating mode. Two-position operation uses power relays with contacts sized to handle the amperage of the heating coil. Step controllers can provide sequencing control of multiple stages of electric heat. Each stage may require a contactor, depending on the step controller contact rating. Timed two-position control requires a timer and contactors. The timer can be electromechanical, but it is usually electronic and provides a time base of 1 to 5 min. Thermostat demand determines the percentage of on-time. Because rapid cycling of mechanical or mercury contactors can cause maintenance problems, solid-state controllers are preferred. These devices may make cycling so rapid that control appears proportional; therefore, face-and-bypass dampers are not used. Use of electric heating coils is restricted in some areas by energy standards; check code compliance before using this application. A system with a solid-state controller and safeties is shown in Figure 32.

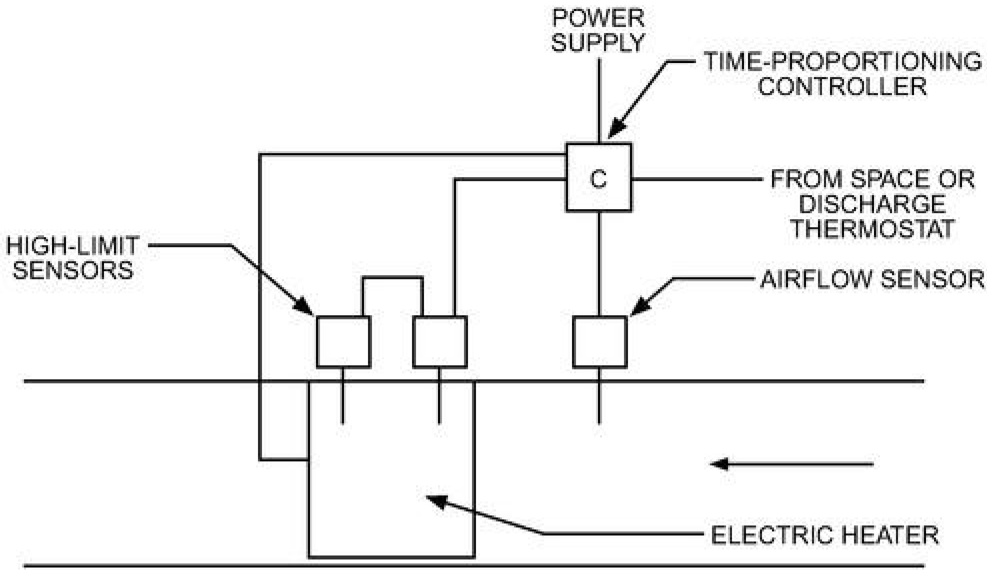

Figure 32. Electric Heat: Solid-State Controller

Current in individual elements of electric duct heaters is normally limited to a maximum safe value established by the *National Electrical Code®* (NFPA *Standard* 70) or local codes. An electric heater must have a minimum airflow switch and two high-temperature limit sensors: one with manual reset and one with automatic reset (Figure 33). The automatic reset high-limit thermostat normally turns off the control circuit when temperatures exceed safe operating levels. If the control circuit has an inherent time delay or uses solid-state switching devices, a separate safety contactor may be desirable. The manual reset backup high-limit thermostat is generally set independently to interrupt all current to the heater if other control devices fail. If still energized, electric coils and heaters can be damaged through overheating and potentially start a fire when air stops flowing around them. Control and power circuits must interlock with the associated fan to shut off electrical energy when it shuts down. Flow or differential pressure switches may be used for this purpose; however, they should be calibrated to energize only when there is airflow. This precaution shuts off power if a fire damper closes or duct lining blocks the air passage. The switch contact should also be calibrated to avoid chattering, which can damage mechanical or mercury contacts.

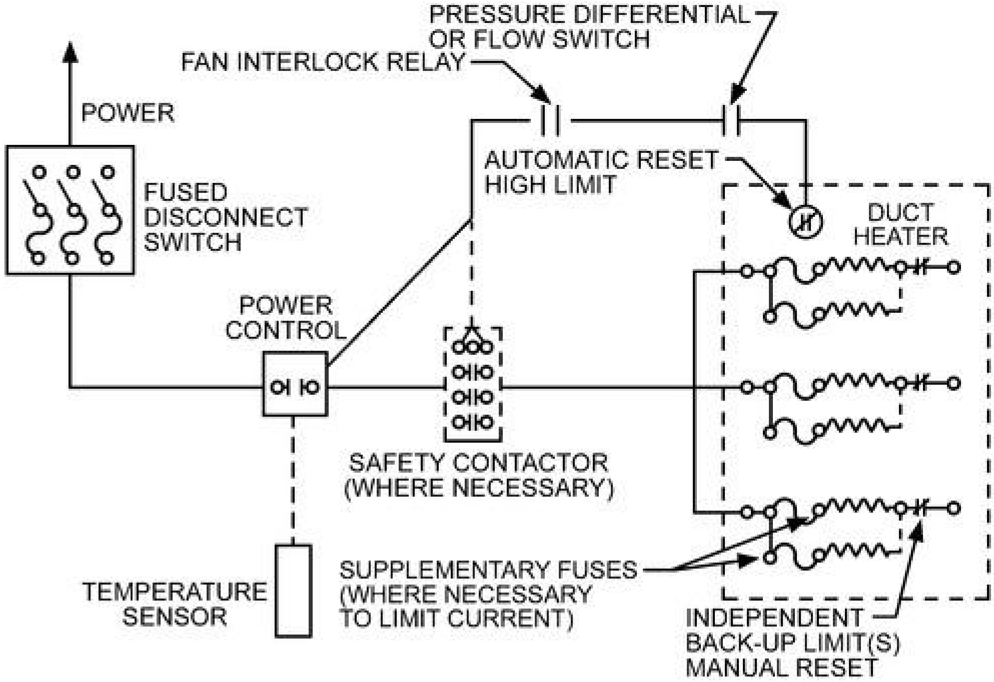

Figure 33. Duct Heater Control

Radiant Heating and Cooling

Radiators (more accurately called **convectors**) can be used either alone or to supplement another heat source. The control strategy depends on the function performed. For a radiation-only heating application, rooms are usually controlled individually; each radiator and convector is equipped with an automatic control valve. Depending on room size, one thermostat may control one valve or several valves in unison. Unit-mounted thermostats and packaged controls allow lower component cost, better assembly quality, and avoid the cost and coordination of a second trade for remote sensor installation. Wall-mounted thermostats give the best results when controlling the space for the comfort of seated occupants.

For supplemental heating applications, where perimeter radiation is used to offset perimeter heat losses (the zone or space load is handled separately by a zone air system), sequence the radiation control with the main zone system to ensure there is no simultaneous heating and cooling. In the past, it was common to control the radiant system based on outdoor air temperature reset of the water temperature perhaps zoned by exposure with a solar compensating outdoor sensor, but this can result in “fighting” between the radiator and the main zone system and is disallowed by energy standards such as ASHRAE *Standard* 90.1.

Radiant panels combine controlled-temperature room surfaces with central air conditioning and ventilation. The radiant panel can be in the floor, walls, or ceiling. Panel temperature is maintained by circulating water or air, or by electric resistance. The central air system can be a basic one-zone, constant-temperature, constant-volume system, with the radiant panel operated by individual room control thermostats, or it can include some or all the features of dual-duct, reheat, multizone, or variable-air-volume (VAV) systems, with the radiant panel operated as a one-zone, constant-temperature system. Where hydronic tubing or electric heating elements are embedded in concrete, the rate of slab temperature change must be limited to prevent thermal expansion from cracking the concrete.

Radiant panels for both heating and cooling require controls similar to those for a four-pipe heating/cooling fan-coil. To prevent condensation, ventilation air supplied to the space during the cooling cycle should have a dew point below that of the radiant panel surface. The dew point should be actively controlled to prevent condensation. Outdoor air intake dehumidification and a tight building envelope or positive pressure are required. When internal latent loads increase, the chilled-water temperature for the radiant cooling panels should be reset upward if the dew point becomes too high.

Hot-Water Distribution Systems

Hot water is distributed using variable flow (primarily two-way valves at coils) or constant flow (three-way valves at coils). An example constant-flow system is shown in Figure 34. Variable-flow systems are similar to the chilled-water distribution systems shown in Figures 35 and 36. Some boilers require constant flow or very high minimum flow rates. They typically are piped using a primary/secondary system (see Figure 36). These boilers are usually required by their listing to have flow switches to enable the boiler only when flow is proven. Boilers that require small (or zero) minimum flow rates are usually piped in a primary-only configuration with a bypass to maintain minimum flow (see Figure 36). A flow meter in the boiler circuit is usually installed to control the bypass valve. The bypass can also be controlled to maintain minimum boiler entering water temperatures for noncondensing boilers.

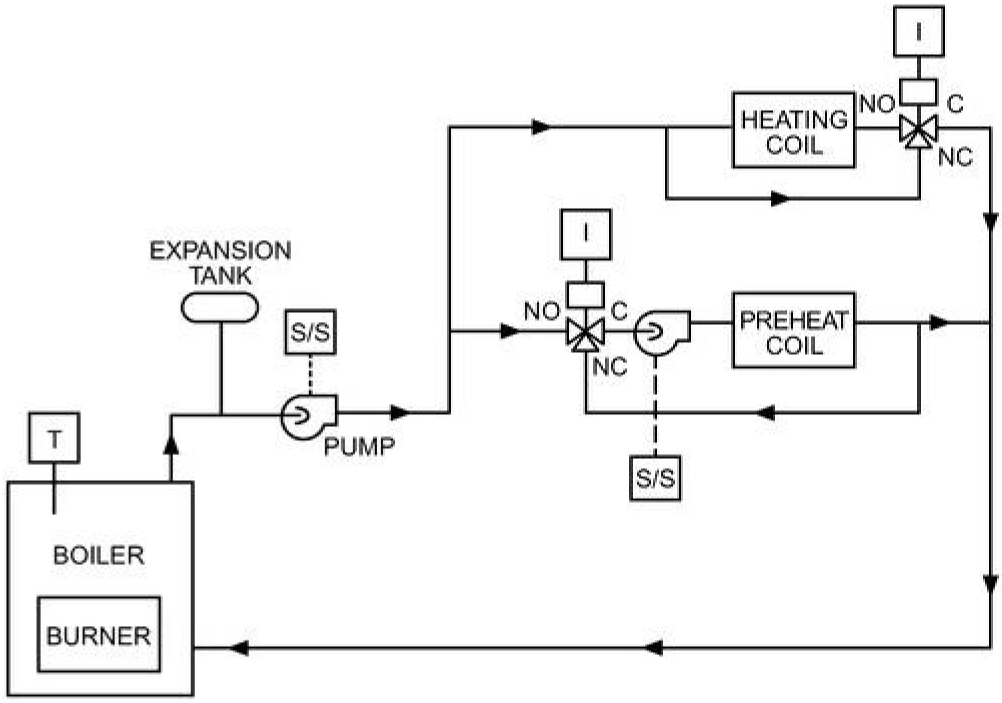

Figure 34. Load and Zone Control in Constant Flow System

Hot-Water and Steam Boilers

Hot-water distribution control includes temperature control at hot-water boilers or the converter, reset of heating water temperature, and control of pumps and distribution systems. Other controls to be considered include (1) minimum water flow through boilers, (2) protecting boilers from temperature shock and condensation on the heat exchanger, and (3) coil low-temperature detection. If multiple or alternative heating sources (e.g., condenser heat recovery, solar storage) are used, the control strategy must also include a way to sequence hot-water sources or select the most economical source.

Figure 37 shows a system for load control of a fossil-fuel-fired boiler. Boiler safety controls, usually factory installed with the boiler, include flame-failure, high-temperature, and other cutouts. ASME *Standard* CSD-1-2012 requires a manually operated remote shutdown switch located just outside the boiler room door for boilers with fuel input ratings less than 3633 kW. Field-installed operating controls must allow safety controls to function in all modes of operation. In most cases, energy savings can be significant if the boiler is controlled to reset the water temperature based on heating load or a proxy such as outdoor air temperature. If the boiler is also used for domestic water heating, the reset range will be limited because minimum temperatures required to kill bacteria must be maintained. A typical outdoor air reset schedule is shown in Figure 37. With DDC devices, reset can be controlled from zone demand, which can improve energy performance and ensure all zones are satisfied. To minimize condensation of flue gases and boiler damage in noncondensing boilers, water temperature should not be reset below that recommended by the manufacturer, typically 60°C entering water temperature, or condensation may occur and lead to corrosion-related failure. Condensing boilers are specifically designed to allow flue gases to condense and should operate at lower water temperatures to harness latent energy in the flue gas. Aggressive reset of hot-water temperatures improves the efficiency of condensing boilers, because efficiency is a strong function of boiler entering water temperature. Additional energy savings are achieved using variable-speed pumps controlled to reduce distribution pump capacity to match the load, as allowed by the boiler's maximum flow.

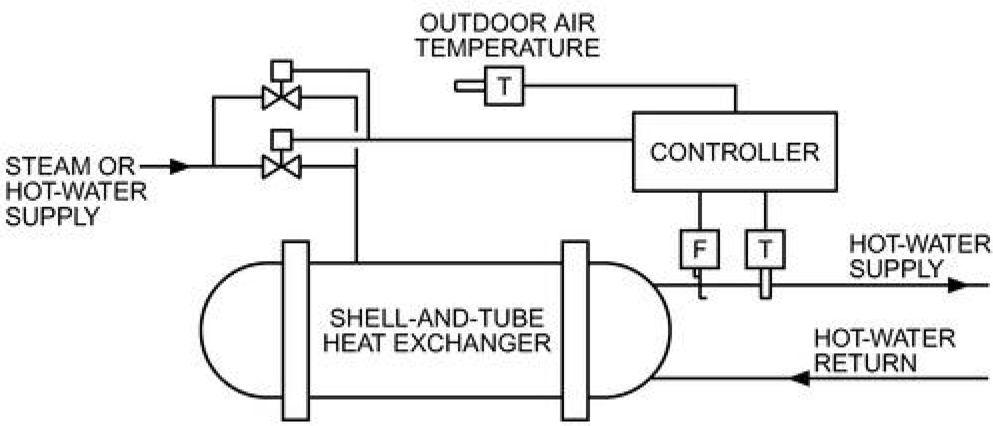

Figure 35. Steam-to-Water Heat Exchanger Control

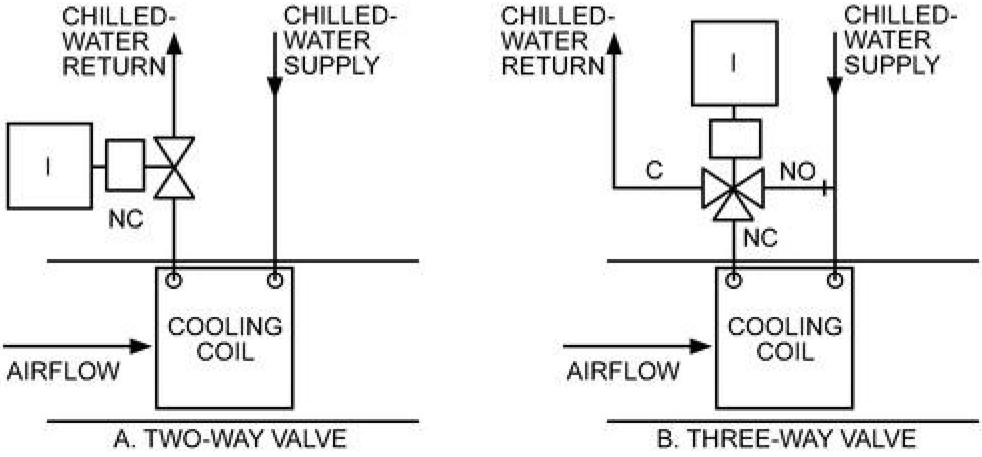

Figure 36. Control of Chilled-Water Coils

Hot-water heat exchangers or steam-to-water converters are sometimes used instead of boilers as hot-water generators. Converters typically do not include a control package; therefore, the engineer must design the control scheme. The schematic in Figure 37 can be used with either low-pressure steam or boiler water from 93 to 180°C. The supply water temperature sensor typically controls two modulating two-way valves in a 1/3 and 2/3 arrangement (because of poor turndown of steam valves) in a steam or high-temperature hot-water supply line. An outdoor temperature sensor (or zone demand, for higher efficiency) can be used to reset the supply water temperature downward as load decreases to improve the controllability of heating valves at low load and to reduce heat losses. A flow or differential pressure switch interlock should close the two-way valve when the hot-water pump is not operating. Ensure that the flow switch operates as expected at minimum flow rate on variable-flow systems. With a BAS, feedback from zone heating valves can be used to control starting and stopping of the hot-water pumps. When shutting down a steam converter or high-temperature hot-water system, close the steam valves and allow the water to circulate long enough to remove residual heat in the converter and prevent the pressure relief valve from opening.

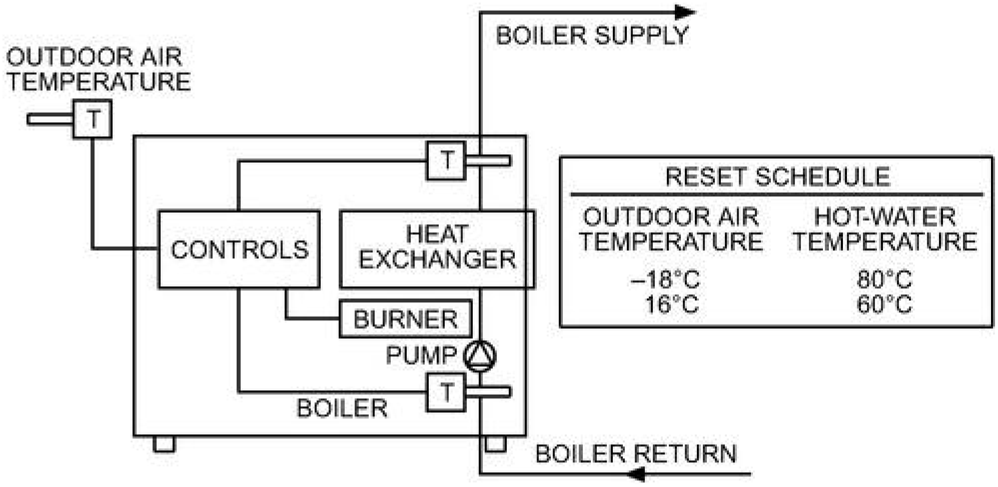

Figure 37. Boiler Control

6. COOLING SYSTEMS

Cooling Coil

Chilled-water or brine (glycol) cooling coils are controlled by two- or three-way valves (Figure 36). These valves are similar to those used for heating control, but are usually closed to prevent cooling when the fan is off. The valve typically modulates to maintain discharge temperature or space temperature set point.

Direct-expansion (DX) cooling equipment is usually controlled by an air, space, or coil discharge temperature feedback loop in discrete stages, by starting and stopping compressors and by applying mechanical unloaders, liquid-line solenoid valves, or hot-gas bypass valves. Most DX systems for commercial application have one to six stages.

When set up properly, a cycling DX system under a steady load operates like a two-position control loop (see Chapter 7 of the 2021 *ASHRAE Handbook—Fundamentals*). Some stages run steadily, and one stage cycles on and off. The behavior of the closed loop can be described by the cycling rate and corresponding swing in controlled air temperature. Most algorithms for DX control address both temperature control (e.g., set points, feedback gains, staging dead bands) and equipment cycling restriction (e.g., minimum-on timer, minimum-off timer, interstage delay). These two characteristics are inextricably linked by thermal sizing and loading conditions. Either characteristic can be affected by adjusting the control algorithm, but it is not possible to affect both characteristics independently: any reduction in temperature swing is accompanied by an increase in cycling rate. Overtightening temperature control will conflict with cycle controls and be rendered irrelevant.

When set up improperly, a DX system may cycle through multiple stages as the temperature oscillates around the set point, rather than having only one stage cycle on and off. Compared to proper operation, both the cycle rate and temperature swing are excessive. This operation can usually be corrected by adjusting parameters in the feedback algorithm.

Some staging systems are arranged so that it takes a greater temperature error to activate the higher stages. The result is that, at higher loads, the system operates at higher temperatures, which is usually not desired; it is usually intended that the system operate in the same temperature range, regardless of loading on the stages. This can be accomplished in many ways, including a proportional-plus-integral (PI) controller output driving a staging module.

If the DX system serves a single zone, the feedback signal is usually the space temperature, which usually varies by 0.5 to 2.2 K. If the DX equipment serves multiple zones, as in a VAV air handler, the feedback signal is usually the coil discharge temperature. Measured at this point, temperature swings appear much larger, though the effect on zone comfort is the same. When adjusting or specifying a DX system for discharge temperature control, it is important to allow the wider range of temperatures. Also, DX staging capacity on VAV applications should be limited based on air-handling unit (AHU) airflow rate, especially at low-part-load conditions, to prevent the coil from freezing.

Heat pumps are a variation of DX systems. (See Chapter 9 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for a detailed description.) Typically, heat pumps are controlled using manufacturer-supplied package controls. A hard-wired space temperature sensor or thermostat is connected to the unit. The controls regulate the unit's staging and cycling of any control valve(s). A BACnet® interface is available on many heat pump units, and can be wired to a BACnet network for monitoring and supervision. Typical BACnet objects include those shown in the table on page 48.19.

Depending on the BACnet device, many more objects may be available. Some objects may be **read-only;** these are generally field input sensors (e.g., temperature) or objects that could cause equipment damage if commanded inappropriately (e.g., commanding compressor directly). **Writable** objects are available to the BAS for commanding (e.g., set points).

Analog Inputs	**Analog Outputs**	**Analog Values**
Space temperature	Space temperature set point	Occupied cooling set point
Supply air temperature	ECM fan override	Occupied heating set point
Effective cooling set point	Fan variable-frequency-drive (VFD) speed control	Unoccupied cooling set point
Effective heating set point		Unoccupied heating set point
Multistate Inputs	**Multistate Outputs**	**Values**
Effective occupancy state	Occupancy command	Cycle fan
Effective mode of operation	Fan command	
Compressor status	Compressor command	
Fan status	Emergency heat command	

Chillers

Manufacturers almost always supply chillers with an automatic control package installed. Their control functions fall into two categories: capacity and safety.

Because of the wide variety of chiller types, sizes, drives, manufacturers, piping configurations, pumps, cooling towers, distribution systems, and loads, most central chiller plants, including their controls, are custom designed. In the 2020 *ASHRAE Handbook—HVAC Systems and Equipment*, Chapter 43 describes various chillers (e.g., centrifugal, reciprocating, screw, scroll), and Chapter 13 covers variations in piping configurations and some associated control concepts. Chiller control strategies should always include an understanding of the chiller limits for minimum flow, minimum temperature, and acceptable rate of change for both.

Chiller plants are generally one of two types: variable flow (Figures 38 and 39) or constant flow (Figure 40). The figures show a parallel-flow piping configuration. Series-flow chiller configurations are often used in variable-primary-flow applications (Figure 40). The higher design water pressure drop of a series configuration is less of a concern in a variable-primary-flow application, because little time is spent at the maximum design flow operating condition. In Figures 38 and 39, the bypass line ensures minimum flow through the chiller(s). In Figure 38, flow is measured by the flow meter, and the BAS modulates the bypass valve as needed to maintain the minimum flow required by the operating chillers.

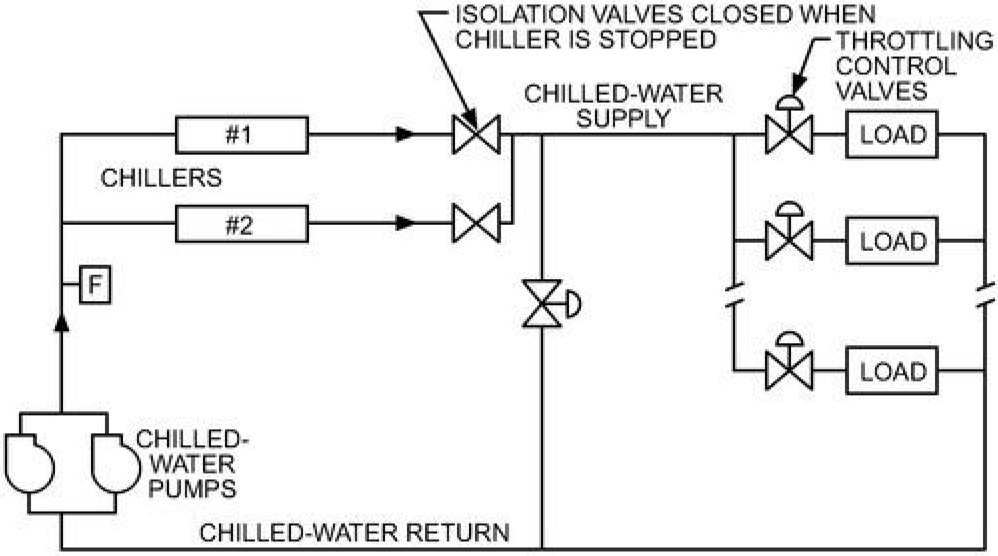

Figure 38. Variable-Flow Chilled-Water System (Primary Only)

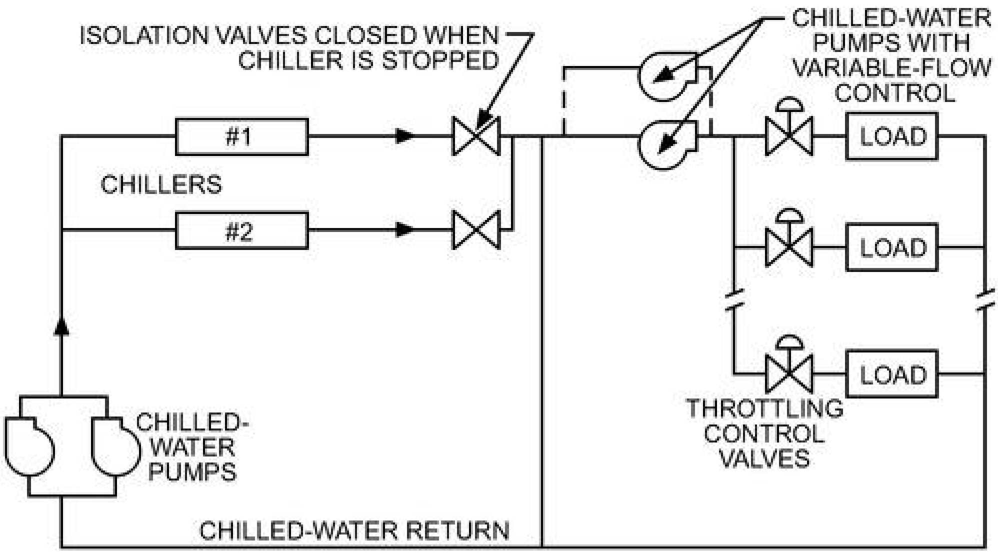

Figure 39. Variable-Flow Chilled-Water System (Primary/Secondary)

Control of the remote load determines which system type should be used. Throttling two-way coil valves vary flow in response to load, and are used with variable-flow systems.

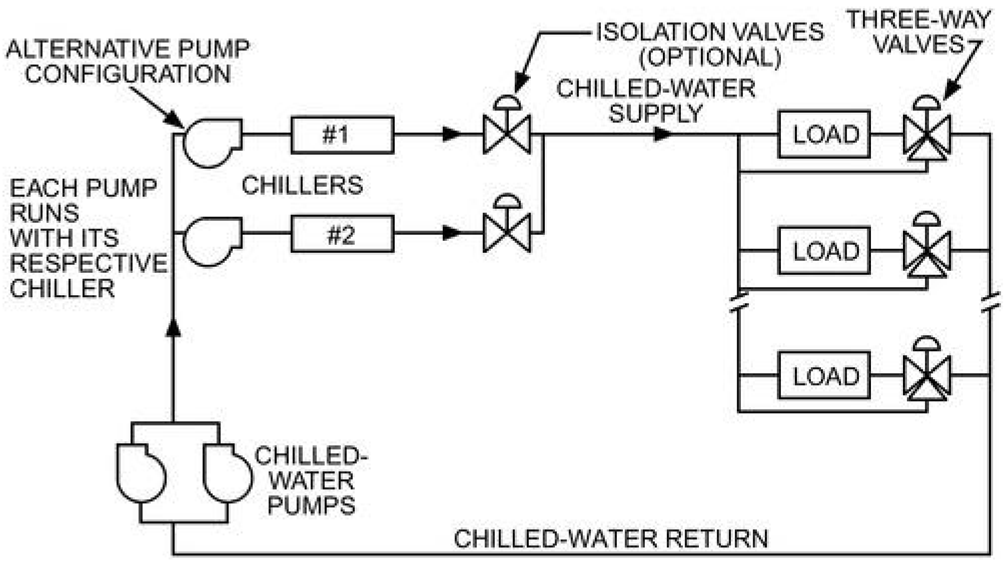

Figure 40. Constant-Flow Chilled-Water System (Primary Only)

Variable primary-only systems require greater care in design of the control system and control sequences than primary-secondary systems (Taylor 2002, 2011) but are usually more efficient. Selecting the bypass control valve and tuning the control loop is sometimes difficult because of the widely ranging differential pressure across the valve caused by its location near the pumps. The valve must be large enough to bypass the minimum chiller flow through it with a pressure drop as low as the differential pressure set point used to control chilled-water (CHW) pump variable-frequency drives (VFDs). This is necessary because, if only a few valves are open in the system, the pressure at the differential pressure sensor location will be nearly equal to the differential pressure at the plant, because there is little pressure drop between these two points due to the low flow rate. This constraint makes the valve oversized for other flow scenarios that can occur, so tuning can be difficult. If the control loop is unstable, cold CHW supply can be fed back into the return intermittently and cause chillers to cycle off because of low load or cold supply water temperatures. However, if the loop is too slow, it may not respond quickly enough to sudden changes in flow (e.g., when a large number of air-handling units [AHUs] shut off at the same time), resulting in insufficient flow through the chillers and causing them to trip on low flow or low temperature.

The decision to reset the chilled-water supply temperature set point should be based on facility requirements. Chilled-water reset may not be appropriate in systems with tight space temperature and humidity requirements (e.g., hospitals, computer data centers, museums, some manufacturing facilities). Resetting the CHW temperature set point upward when loads are low is always an effective energy-saving strategy for constant-flow systems. In variable-flow systems, reset may or may not save energy, depending on plant design. High CHW temperature reduces coil performance, so coils in two-way valve systems demand more chilled water for the same load, degrading ΔT and increasing flow and pump energy requirements. Whether the net energy savings (chiller energy decrease minus pump energy increase) is positive and sufficient to offset the cost of implementing the reset strategy depends on chiller performance characteristics and the nature of coil loads. Resetting the CHW temperature set point also prevents unstable operation of control valves under very low loads. To ensure that no loads are starved, the chilled-water set point should be reset from the zone or system valve with the greatest load (load reset) (Taylor 2011).

The constant-flow system (Figure 40) is only constant flow under each combination of chillers on line; chillers and pumps can be staged if all loads experience similar loads at all times. If that is the case, the plant can supply about 80% of the design flow with just one of the headered pumps, and the chillers can be sequenced based on return temperature. Both pumps would run simultaneously only under peak demand. Staging may be prevented even at low plant loads if some coils are at high load while others are at low load, causing the chillers to operate at low loads, reducing plant efficiency. Use of constant-flow systems may be limited to smaller systems by energy codes.

Air-Cooled Chillers

Air-cooled chillers are controlled similarly to other chillers. If the chiller is to operate during cold weather, it must be equipped with a low-ambient kit. Typically this is a modulating damper or variable-speed fan that limits airflow across the condenser, usually provided as part of the chiller package. In very cold conditions, additional equipment is required. With variable-flow evaporators, careful attention to the manufacturer’s recommended minimum flow must be observed. Chilled-water supply temperature can be reset as described above. The chiller may be equipped with a barrel heater, usually controlled by the chiller’s packaged controls, to prevent the evaporator from freezing in cold weather.

Cooling Tower

Cooling tower fans are typically controlled to maintain condenser water supply (CWS) temperature set point, as described previously. The CWS set point may be reset based on chiller load or, in the case of hydronic free cooling, space conditions. Figure 41 shows a typical cooling tower control schematic.

When the system includes large condenser water sumps, the temperature sensor’s location must be considered. Large sumps introduce significant time delays into the system that must be accounted for in system design and operation. Often, condenser water supply piping does not run full at all times, particularly when draining to a sump. In this case, placing the temperature sensor so that is in contact with the water is important. This may necessitate locating the sensor on the bottom of an elbow or angled into the lower half of the pipe (to avoid mounting at the pipe bottom, where it could be susceptible to moisture collection).

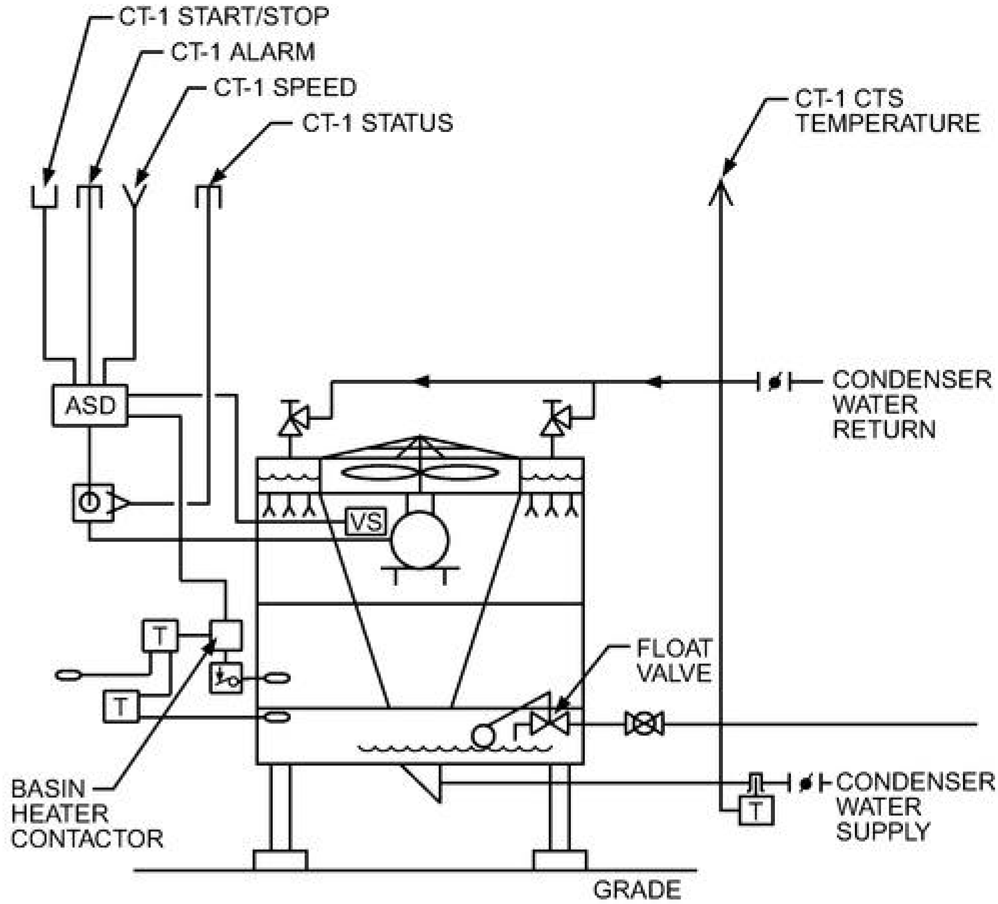

Figure 41. Cooling Tower

Two-speed motors or variable-speed drives can reduce fan power consumption at part-load conditions and stabilize condenser water temperature. Variable-speed drives (VSDs) improve control because fan speed can be better matched to the cooling load. When there are multiple towers, efficiency is maximized when as many cells as possible are active, which increases mass transfer area and reduces required fan speed. It may be necessary, however, to shut off flow to some cells to maintain minimum tower flow rate, as recommended by the tower manufacturer to minimize scaling. Fan speed should be controlled as follows.

Two-Speed Motors. The lowest fan speed should be used. For three two-speed towers, staging should be as follows:

Tower	Stage 1	Stage 2	Stage 3	Stage 4	Stage 5	Stage 6	Stage 7
1	Off	Low	Low	Low	High	High	High
2	Off	Off	Low	Low	Low	High	High
3	Off	Off	Off	Low	Low	Low	High

Provisions should be made to decelerate the fan when switching from high to low speeds.

Variable-Speed Drives. All of the operating tower fans should operate at the same speed.

Tower fans should also have a vibration switch hard wired to shut down the fan if excessive vibration is sensed.

Most tower manufacturers do not recommend bypassing water around the tower as a normal temperature control function, because this causes localized drying of the tower media and spot scaling. Bypass valves may be required for some low-temperature operations or hydronic economizer operations. Check with the tower manufacturer for bypass valve applications.

Where year-round tower operation is required in cold regions, cooling towers may require sump heating and/or continuous full flow over the tower to prevent ice formation, and may also require deicing. The cooling tower sump thermostat controls an electric heating element or hot-water or steam valve to keep water in the sump from freezing. Typically, sump heating is locked out during tower operation and when outdoor air temperatures are above freezing. When operating the tower in cold weather, full water flow is needed to prevent localized ice formation. If towers build up ice, reversing the fan rotation and sending air backward through the tower can deice them, either manually or automatically. Many operating personnel prefer to do this operation manually so they can observe the towers while deicing is under way. If deicing cycles are needed, the fan starter or VSD must be able to reverse the fan direction. There should be deceleration time interlocks so that the fans spin to a stop before engaging reverse operation. Automatic deicing control strategies are available from tower manufacturers.

Towers that have internal balancing piping or chambers must be allowed to drain into the basin when the tower is not operating in cold weather.

Auxiliary Control Functions. The control system may be required to cycle a fill valve to maintain sump level, or to monitor water consumption or water treatment systems. If the tower does not have sump heaters, the control system may be required to drain the tower and piping when outdoor air approaches freezing, and refill the tower on the next call for cooling.

Chiller Plant Operation Optimization

Chapter 42 discusses optimized control of chiller plants and details the control strategies that can be applied. This section highlights the general conclusions that can be drawn from the specific optimization strategy in the section on Sequencing and Loading of Multiple Chillers of that chapter. See ASHRAE *Guideline* 36 for high-performance sequence of operation.

Multiple chillers of different characteristics should be operated at a point that minimizes overall plant power consumption, considering the power consumption of the auxiliary chilled- and condenser water pumps and cooling tower fans as well as the consumption of the chillers and the load fans (Hartman 2005). For the general case of chillers with different part-load characteristics, operate multiple chillers at identical supply temperatures but different flows to achieve equal partial derivatives of the individual chillers' power consumption with respect to their loads. That is, the ratio of the incremental change in thermal load to the incremental change in input power should be identical for each online chiller. For identical chillers, this is achieved by loading them equally.

For constant-speed-drive chillers, sequencing is straightforward. It is best to run one chiller until fully loaded before bringing on a second machine. This dynamic changes for chillers with variable-speed drives, or if the chilled-water pumps are piped in a parallel (headered) arrangement or do not have variable-speed drives; for plants of this configuration, often overall plant power consumption is minimized when more chillers are operating at reduced load compared to fewer chillers operating at higher loads.

To meet any given plant load, many possible combinations exist for operating points of system equipment (e.g., chillers, pumps, cooling tower fans, air handler fans). When any given piece of equipment's operation point is changed, the operating point of at least one other piece of equipment must be changed to compensate. This results in one component reducing its power use and the other increasing it. Depending on how much each changes, total power use may increase or decrease. The combination of component operating points that results in minimum plant power consumption occurs when all the equipment in each subset that affect a variable (e.g., temperature, load, flow) operate at points where the power consumption partial derivative to the affected variable for each piece of equipment is equal.

A practical method to obtain the highest possible plant efficiency for current weather and load conditions uses a mathematical plant energy performance model and minimization to find the most efficient set of set points and commands. Alternatively, the most efficient configuration is precalculated for all foreseeable conditions and the control commands are then obtained from a lookup table (Nelson 2012).

When detailed information about the plant components' part-load energy performance for multiple scenarios is unavailable, or when sophisticated controls are beyond budget, use the following strategies. In most cases, they increase energy performance and are relatively easy to implement. However, return-on-investment analysis of controls of different sophistication levels is recommended, because the energy savings tend to quickly pay for the additional cost.

Cooling Tower Relief. Lowering the condenser water temperature by operating the tower fans at a higher speed reduces the compressor lift and the energy consumed by the chillers. Lowering the lift too much, however, may cause the fans to consume more energy than is saved by the chillers. A condenser temperature set point reset (to maintain constant tower approach [when the outdoor wet-bulb temperature is available] or proportional to the heat rejected or based on outdoor air temperature, and limited to the minimum chiller lift as recommended by the chiller's manufacturer), results in chiller energy savings of 1 to 3% for each 1 K reduction in water temperature for constant-speed chillers, and 2 to 5% for each 1 K reduction for variable-speed chillers. Chiller lift is defined as the difference between the chillers' leaving condenser water temperature and leaving evaporator temperature.

Supply Chilled-Water Temperature Reset. Increasing the supply temperature also reduces the chiller lift with similar savings per degree. Consider the effect on the loads and distribution pumps, because variable-speed pumps always (and fans sometimes) need to run harder, offsetting some or all the chiller energy savings. Also, some loads may not get enough cooling and be unable to maintain room temperature. Dehumidification requirements further limit the range of allowed supply temperature increase. As a first approach, the supply temperature set point can be reset between design value and 27 K proportional to the heat load or outdoor air temperature, or to maintain the most open cooling valve in the loads close to full open.

Maximize the Number of Variable-Speed Cooling Towers Used. Depending on the minimum flow needed by the cooling towers, more than one tower per chiller may be used. For example, two variable-speed towers can dissipate the same amount of heat as one tower by running at half the speed, and use about 1/8 of the energy.

Optimize the Number of Chillers Running. If the chillers' performance improves at part load (e.g., variable-speed centrifugal chillers), adding chillers before they are fully loaded results in more efficient configurations. Use the amount of heat removed or the kilowatts consumed by the chillers and, as a first approach, add a chiller when the operating chiller(s) are at 80% of capacity and shed one when they reach 35%.

Maximize (Return Temperature – Supply Temperature). Constant-flow chilled-water systems and constant-speed primary pumps result in more flow than needed for the amount of heat transported. The excess flow causes unnecessary pump energy use and increased compressor lift, which reduces chiller efficiency. Variable primary systems with two-way valves on all loads are the most efficient, but require a system bypass valve and careful control to prevent tripping the chillers (and potentially freezing them) because of low flow, when the load valves close, or when adding chillers. Systems with variable primary and variable secondary pumps and an open decoupler line are the next best in efficiency; they can be controlled to match the primary flow with the secondary, except to maintain the chillers' minimum flow. Next in efficiency are systems with constant primary and variable secondary pumps; these systems are robust and significantly simpler to control, not even requiring flow meters (hence their popularity). Constant-flow systems, which operate at part load most of the time, have the worst energy efficiency in most HVAC systems.

Performance Monitoring. Research (Deng and Burnett 1997; Erpelding 2007; Hartman 2012) shows that chiller plant performance begins to degrade soon after chillers are commissioned unless some form of ongoing commissioning is in place. The cause is usually equipment malfunction, poor maintenance, and/or inappropriate operation, which is difficult to detect in the complexity of plant operation. A performance-monitoring and -diagnosing system, especially an automated one, identifies deviations from expected operation and provides insights into their causes as well as suggesting improvements, thus prompting action and greatly contributing towards plant energy savings throughout its life cycle.

Water-Side Economizers

Water-side economizers use cold water from the cooling tower to produce chilled water without (or with reduced) mechanical refrigeration. This is accomplished by running the cooling towers to produce water typically at or below 7°C while ambient wet-bulb temperatures are low. The cold water is pumped through a low approach water-to-water heat exchanger (HX), usually a plate-and-frame type, to produce chilled water at or below 10°C. The heat exchanger prevents contamination of the chilled water by debris and chemicals found in tower water. The heat exchangers can be piped in series or in parallel. With parallel operation, the heat exchanger functions like another chiller. In series arrangement, the heat exchanger precools the chilled-water return to the chillers. When there is enough heat exchanger capacity, the chillers may be turned off and a bypass opened to direct flow around the chillers. The series arrangement allows for integrated (simultaneous) economizer and chiller operation, which is required by some energy standards such as ASHRAE *Standard* 90.1. Chilled-water temperature is controlled by varying the tower fan speed. When changing from water-side economizer mode to chiller mode, the condenser water is cold, which requires that chiller head pressure control be addressed. Consult the chiller manufacturer for the requirements of each specific machine. To maintain condenser head pressure, many manufacturers recommend self-contained modulating valves or control valves modulated by the BAS or, preferably, by the chiller controller (many have head pressure control outputs as standard). When modulating the condenser water flow to maintain head pressure, the flow switch may need to be bypassed for a short time to keep the machine operating; consult the manufacturer. When the signal used for modulating the valve is controlled by the BAS, it should be directly from a pressure transmitter; relying on pressures obtained from the chiller controller through a network connection can be unstable or unreliable because data refresh rates may be slow or inconsistent.

7. SPECIAL APPLICATIONS

Mobile Unit Control

The operating point of any control that relies on pressure to operate a switch or valve varies with atmospheric pressure. Normal variations in atmospheric pressure do not noticeably change the operating point, but a change in altitude affects the control point to an extent governed by the change in absolute pressure. This pressure change is especially important with controls selected for use in land and aerospace vehicles that are subject to wide variations in altitude. The effect can be substantial; for example, barometric pressure decreases by nearly one-third as altitude increases from sea level to 3000 m.

In mobile applications, three detrimental factors are always present in varying degrees: vibration, shock, and acceleration forces. Controls selected for service in mobile units must qualify for the specific conditions expected in the installation. In general, devices containing mercury switches, slow-moving or low-force contacts, or mechanically balanced components are unsuitable for mobile applications; electronic solid-state devices are generally less susceptible to these three factors.

Explosive Atmospheres

Sealed-in-glass contacts are not considered explosionproof; therefore, other means must be provided to eliminate the possibility of a spark in an explosive atmosphere.

When using electric control, the control case and contacts can be surrounded with an explosionproof case, allowing only the capsule and the capillary tubing to extend into the conditioned space. It is often possible to use a long capillary tube and mount the instrument case in a nonexplosive atmosphere. This method can be duplicated with an electronic control by placing an electronic sensor in the conditioned space and feeding its signal to an electronic transducer located in the nonexplosive atmosphere.

Because pneumatic control uses compressed air, it is safe in otherwise hazardous locations. However, many pneumatic controls interface with electrical components. All electrical components require appropriate explosionproof protection.

Sections 500 to 503 of the *National Electrical Code®* (NFPA *Standard* 70) include detailed information on electrical installation protection requirements for various types of hazardous atmospheres.

Extraordinary Incidents

Building owners and design engineers are sometimes interested in applying the BAS to implement strategies that protect occupants from airborne attack. It is crucial that the engineer does not approach this complex topic as a control system issue. The BAS may include protective features, but only in the context of a comprehensively designed ventilation system. A protective ventilation strategy only makes sense in the context of a thorough risk assessment and an overall security plan. If a protective ventilation strategy is attempted, it is crucial to consider every air movement device and pathway, not just the main fan(s) and damper(s). It is also necessary to consider possible interaction of a protective operation with other emergency control operations, such as the response to a fire/life safety device (e.g., a smoke detector).

ASHRAE (2003), FEMA (2007), and NIOSH (2002), among others, have published references to guide an engineer or building owner in organizing a comprehensive plan. Also see Chapter 59 for more information on this topic.

Integration with Packaged Control Systems

Packaged control systems are controllers that are provided with a piece of equipment. Sometimes these controllers are prewired at the factory, and sometimes parts will need to be field wired. A packaged controller is provided with preprogrammed control sequences, which can be as basic as equipment safety interlocks or as complex as a full suite of air-handling unit control sequences. When packaged equipment is installed in a building with a BAS, the owner will typically want to monitor and/or control the packaged equipment through the BAS. Most manufacturers provide several methods of interfacing with their packaged controllers.

Terminal Strip. A terminal strip is the most basic form of interface between a packaged control system and a BAS. The BAS integrates using multiple wires that are directly connected to terminals through a screw or clamp connection. Each terminal will allow simple electric signals such as on/off, 0 to 10 V, or 4 to 20 mA to pass into or out of the packaged controller. The BAS contains signal generators and receivers that can control or monitor the packaged controller. Each terminal corresponds to a single “point” of information, and each wire carries only one signal to control or monitor that point. Connections to terminal strips must be checked carefully to ensure the correct terminals are used for each point, otherwise damage to the controller or equipment could result.

Serial or IP Interface. Many equipment manufacturers offer serial or IP interfaces between the BAS and their packaged equipment. The benefit of this type of connection is that only one cable is needed, compared to the multiple wires that must be used to connect a terminal strip. There is less room for physical error with a single connection point. However, the BAS and the packaged controller must use the same control protocol to communicate, otherwise control and monitoring is not possible. BACnet™ MS/TP and BACnet IP are standard control protocols that many manufacturers offer in their packaged controllers, and integration guides are typically provided to assist the control programmer in configuring the BAS to read and write value through the serial or IP connection. Some manufacturers require a gateway or network card to enable communication through standard protocols such as BACnet, so the control designer must be careful to specify inclusion of these communication cards.

REFERENCES

ASHRAE members can access *ASHRAE Journal* articles and ASHRAE research project final reports at technologyportal.ashrae.org. Articles and reports are also available for purchase by nonmembers in the online ASHRAE Bookstore at www.ashrae.org/bookstore.

Ahmed, O. 2001. A model-based control for VAV distribution systems. Presented at CLIMA Conference, Napoli.

Ahmed, O. 2002. *Variable-volume, variable-pressure system*. International Facilities Management Association, Research and Development Council, Orlando, FL.

ANSI. 2013. Non-recirculating direct gas-fired industrial air heaters. ANSI *Standard* Z83.4-2013/CSA *Standard* 3.7-2013. American National Standards Institute, Washington, D.C., and Canadian Standards Association, Toronto.

Arens, E., H. Zhang, T. Hoyt, S. Kaam, J. Goins, F. Bauman, Y. Zhai, T. Webster, and B. West. 2014. Thermal and air quality acceptability in buildings that reduce energy by reducing minimum airflow from overhead diffusers. ASHRAE Research Project RP-1515, *Final Report*.

ASHRAE. 2003. *Risk management guidance for health, safety, and environmental security under extraordinary incidents*. Presidential Ad Hoc Committee for Building Health and Safety Under Extraordinary Incidents.

ASHRAE. 2005. *Sequences of operation for common HVAC systems*.

ASHRAE. 2021. *Standard 62.1-2010 user’s manual*.

ASHRAE. 2017. *ASHRAE design guide for dedicated outdoor air systems.*

ASHRAE. 2005. The commissioning process. *Guideline* 0-2005.

ASHRAE. 2007. HVAC&R technical requirements for the commissioning process. *Guideline* 1.1.

ASHRAE. 2009. Field testing of HVAC controls components. *Guideline* 11.

ASHRAE. 2007. Specifying direct digital control systems. *Guideline* 13-2007.

ASHRAE. 2010. Selecting outdoor, return, and relief dampers for air-side economizer systems. ASHRAE *Guideline* 16.

ASHRAE. 2021. High-performance sequence of operation for HVAC systems. *Guideline* 36-2021.

ASHRAE. 2013. Ventilation for acceptable indoor air quality. ANSI/ ASHRAE *Standard* 62.1-2013.

ASHRAE. 2013. Energy standard for buildings except low-rise residential buildings. ANSI/ASHRAE/IES *Standard* 90.1-2013.

ASHRAE 2012. BACnet®—A data communication protocol for building automation and control networks. ANSI/ASHRAE *Standard* 135-2012.

ASHRAE. 2013. Ventilation of health care facilities. ANSI/ASHRAE/ASHE *Standard* 170-2013.

ASME. 2012. *Controls and safety devices for automatically fired boilers*. American Society of Mechanical Engineers, New York.

Deng, S.-M., and J. Burnett. 1997. Performance monitoring and measurement for central air conditioning chiller plants in buildings in Hong Kong. *HKIE Transactions* 4(1).

EDR. 2009. *Advanced variable air volume system design guide*. Energy Design Resources, Pacific Gas and Electric Company.

Englander, S.L., and L.K. Norford. 1992. Saving fan energy in VAV systems—Part 2: Supply fan control for static pressure minimization. *ASHRAE Transactions* 98(1):19-32.

Erpelding, B. 2007. Real efficiencies of central plants. *HPAC Engineering* (May).

Felker, L.G., and T.L. Felker. 2010. *Dampers and airflow control*. ASHRAE.

FEMA. 2007. Reference manual to mitigate potential terrorist attacks against buildings. *Publication* 426, Federal Emergency Management Agency, U.S. Department of Homeland Security, Washington, D.C.

FGI. 2018. *Guidelines for design and construction of health care facilities*. Facility Guidelines Institute, Dallas.

Hartman, T. 1993. Terminal regulated air volume (TRAV) systems. *ASHRAE Transactions* 99(1):791-800, *Paper* CH-93-03-3.

Hartman, T. 2005. Designing efficient systems with the equal marginal performance principle. *ASHRAE Journal* 47(7):64-70.

Hartman, T. 2012. New strategies for optimizing building performance. *HPAC Engineering*. hpac.com/building-controls/new-strategies-optimizing -building-performance.

Hydeman, M. In progress. Advanced control sequences for HVAC systems—Phase 1, air distribution and terminal systems. ASHRAE Research Project RP-1455.

Kalore, P., O. Ahmed, and M. Cascia. 2003. Dynamic control of a building fluid distribution system. Presented at IEEE Conference on Control Applications, Istanbul.

Kettler, J. 1995. Minimum ventilation control for VAV systems—Fan tracking vs. workable solutions. *ASHRAE Transactions* 101(2):625-630. *Paper* SD-95-02-3.

Kettler, J. 2000. Measuring and controlling outdoor airflow. *ASHRAE IAQ Applications* (Winter).

Krarti, M., C.C. Schroeder, E. Jeanette, and M.J. Brandemuehl. 2000. Experimental analysis of measurement and control techniques of outside air intake rates in VAV systems (RP-980). *ASHRAE Transactions* 106(1): 39-52, *Paper* 4369.

Lau, J., G.K. Yuill, and X. Lin. 2013. CO2-based demand controlled ventilation for multiple zone HVAC systems. ASHRAE Research Project RP-1547, *Final Report*.

Nelson, K. 2012. Simulation modeling of central chilled water systems. *ASHRAE Transactions* 118(1), *Paper* CH-12-003.

NFPA. 2014. National electrical code®. ANSI/NFPA *Standard* 70-2014. National Fire Protection Association, Quincy, MA.

NIOSH. 2002. Guidance for protecting building environments from airborne chemical, biological, or radiological attacks. DHHS *Publication* 2002-139. National Institute for Occupational Safety and Health, Department of Health and Human Services, Cincinnati.

Taylor, S. 2002. Primary-only vs. primary-secondary variable flow systems. *ASHRAE Journal* 44(2):25-29.

Taylor, S. 2007. Increasing efficiency with VAV system static pressure setpoint reset. *ASHRAE Journal* 49(6):24-32.

Taylor, S. 2011. Optimizing design and control of chilled water plants, part 1: Chilled water distribution system selection. *ASHRAE Journal* (July).

USDOJ. 1994. *ADA standards for accessible design.* Title III Regulations, Appendix A. 28CFR36. U.S. Department of Justice, Washington, D.C.

USGBC. 2009. *LEED® reference guide for green building design and construction*. U.S. Green Building Council, Washington, D.C.

BIBLIOGRAPHY

ASHRAE. 2013. Commissioning process for buildings and systems. ANSI/ASHRAE/IES *Standard* 202-2013.

Avery, G. 1992. The instability of VAV systems. *HPAC Engineering* (February):47-50.

Federspiel, C.C. 2005. Detecting critical supply duct pressure, *ASHRAE Transactions* 111(1):957-963, *Paper* OR-05-13-1.

Kettler, J. 2004. Return fans or relief fans: How to choose? *ASHRAE Journal* 46(4):28-32.

Mills, E., H. Friedman, T. Powell, N. Bourassa, D. Claridge, T. Haasl, and M.A. Piette. 2004. The cost-effectiveness of commercial-buildings commissioning. *Report* LBNL-56637 (rev.). Lawrence Berkeley National Laboratory. PDF available from sites.google.com/site/evanmillsresearch /home/publications.

Salsbury, T. and B. Chen. 2002. A new sequence controller for multistage systems of known relative capacities. *International Journal of HVAC&R Research* (now *Science and Technology for the Built Environment*) 8(4): 403-428.

Seem, J., M. House, G. Kelly, and C. Klaassen. 2000. A damper control system for preventing reverse airflow through exhaust air dampers. *International Journal of HVAC&R Research* (now *Science and Technology for the Built Environment*) 6(2): 135-148.

Shadpour, F. 2001. *Fundamentals of HVAC direct digital control: Practical applications and design*. Hacienda Blue, Escondido, CA.

U.S. DOE. 2008. Continuous commissioning process. Ch. 8 in *Commissioning for federal facilities*. U.S. Department of Energy, Washington, D.C. energy.gov/node/727996.

The preparation of this chapter is assigned to TC 1.4, Control Theory and Application.