Related Commercial Resources

CHAPTER 46

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 pneumatic, mechanical, electrical, electronic, and/or direct digital control (DDC) devices.

This chapter addresses control of typical HVAC systems, design of controls for system coordination and for energy conservation, and control system commissioning. Chapter 15 of the 2005 *ASHRAE Handbook—Fundamentals* covers details of component hardware and the basics of control.

SYSTEM TYPES

A pneumatic or electronic-based system has several physical control loops, with each loop including a combination of a sensor, controller, and actuator. DDC systems can more easily share a sensor value with several control loops or have multiple control loops selectively activate an actuator.

DDC systems allow information such as system status or alarms to be collected in a central operator workstation (OWS) and shared between HVAC systems, enabling advanced, energy-saving, system-level applications through a common communication protocol. ASHRAE *Guideline* 13 and *Standard* 135 have more detailed discussions of networking and interoperability.

Without a network and interoperability, DDC controllers are isolated: they collect information, but are unable to share that information across systems; they generate alarms, but are unable to display them at a central location where they can be more easily seen and addressed. In general, distributed processing locates local loop (e.g., variable-air-volume reheat coil) control near devices most immediately involved in that loop, and broader strategies (switching from heating to cooling) at a campus level, but applications can be located in any controller.

HEATING SYSTEMS

Heating systems include boilers, fired by either fuel combustion or electric resistance, direct flame-to-air furnaces, and electric resistance air heaters. 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. Flame cycling and modulation can be handled by the boiler control package, or the DDC can send commands to the boiler controls. The control designer decides under what circumstances to turn boilers on and off in sequence and, for hot-water boilers, at what temperature to control the boiler supply water.

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 for multiple zones. Other controls to be considered include (1) minimum water flow through boilers, (2) protecting boilers from temperature shock and condensation in the flue, 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 1 shows a system for load control of a gas or oil-fired boiler. Boiler safety controls usually include flame-failure, high-temperature, and other cutouts. Field-installed operating controls must allow safety controls to function in all modes of operation. Intermittent burner firing usually controls capacity, although burner modulation is common in larger systems. In most cases, the boiler is controlled to maintain a constant water temperature, although an outside air thermostat or other control strategies can reset the temperature if the boiler is not used for domestic water heating. A typical outside air reset schedule is shown in Figure 1. To minimize condensation of flue gases and boiler damage, water temperature should not be reset below that recommended by the manufacturer, typically 140°F, or condensation may occur and lead to corrosion-related failure. Condensing boilers are specifically designed to allow flue gases to condense, and can operate at lower water temperatures. Systems with sufficiently high pump operating costs can use variable-speed pump drives to reduce secondary pumping capacity to match the load and conserve energy. ASME (2004) requires that, for boilers above a certain size, a manually operated remote shutdown switch be located outside the boiler room door, to disconnect power to burner controls.

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 2 can be used with either low-pressure steam or boiler water from 200 to 360°F. The supply water temperature sensor controls a modulating two-way valve in the steam (or hot-water) supply line. An outside temperature sensor usually resets the supply water temperature downward as load decreases, to improve the controllability of heating valves at low load and to reduce piping losses. This reset is typical for constant-flow systems; resetting the water temperature in variable-flow systems should be carefully studied, because it increases the pumping energy. A flow switch interlock should close the

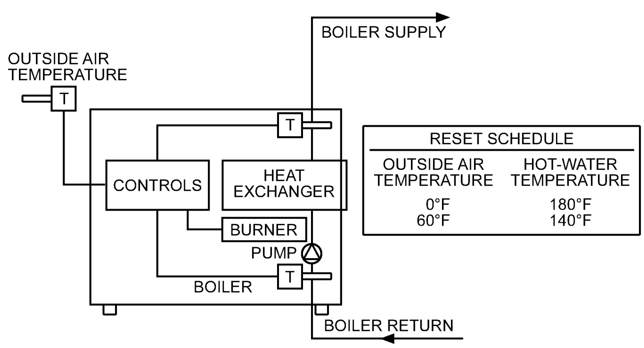

Fig. 1 Boiler Control

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

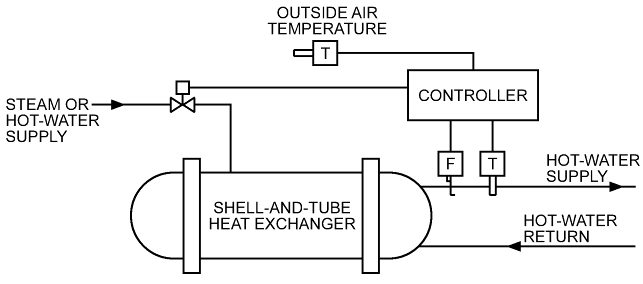

Fig. 2 Steam-to-Water Heat Exchanger Control

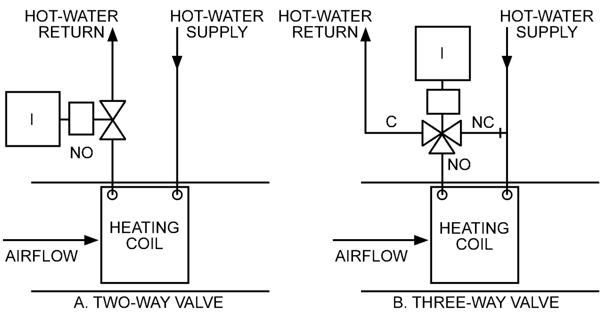

Fig. 3 Control of Hot-Water Coils

two-way valve when the hot-water pump is not operating. Ensure that the flow switch will operate as expected at minimum flow rate on variable-flow systems. With an integrated DDC system, feedback from zone heating valves can be used to control starting and stopping of the hot-water pumps. On constant-flow systems, feedback can be used to reset the hot-water temperature to the lowest temperature that meets zone requirements. When shutting down a steam converter system, close the steam valves and allow the water to circulate for a while, to remove residual heat in the converter and prevent the pressure relief valve from opening.

Heating Coils

Heating coils that are not subject to freezing can be controlled by simple two- or three-way modulating valves (Figure 3). Steam-distributing coils are required to ensure proper steam coil control. (For information on air-side 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 open to allow heating if control power fails. In many systems, the outside air temperature resets the heating discharge air controller.

To provide unoccupied heating or preoccupancy warm-up, a heating coil can be added to the central fan system. During warm-up or unoccupied periods, a constant supply-duct heating temperature is maintained and the cooling-coil valve is kept closed. Once the facility has attained the minimum required space temperature, the central air handler reverts back to occupied mode.

Heating coils in central air-handling units preheat, reheat, or heat, depending on the climate and the amount of minimum outside air needed.

Preheating coils using steam or hot water must have protection against freezing, unless (1) the minimum outside air quantity is small enough to keep the mixed air temperature above freezing 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.

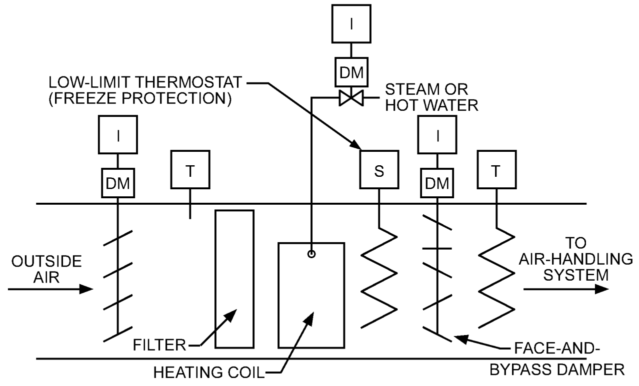

Fig. 4 Preheat with Face-and-Bypass Dampers

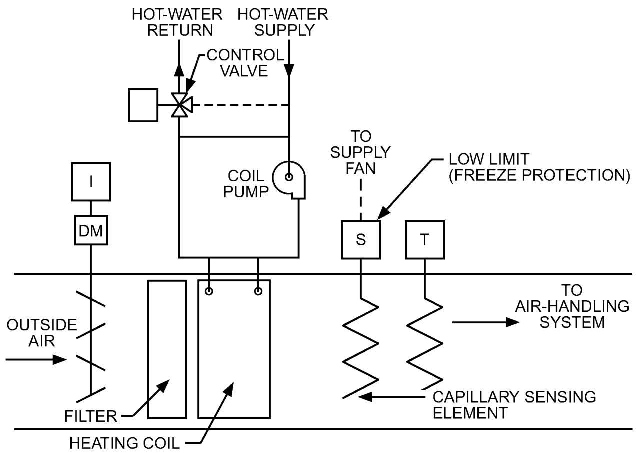

Fig. 5 Coil Pump Piped Primary/Secondary

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 outside (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 4). 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 outside air temperature is safely above freezing (roughly 35°F), the bypass damper is full open to the coil face and the coil valve can be modulated to improve controllability.

Hot-water coils must maintain a minimum water velocity in the tubes (on the order of 3 fps) to prevent freezing. A two-position valve combined with face and bypass dampers can be used (Figure 4), but a coil pump is more common. There are many coil pump piping schemes; the most common are shown in Figures 5 and 6. In each scheme, the control valve modulates to maintain the desired coil air discharge temperature and the pump maintains the minimum tube water velocity when the outside air is below freezing. Pumped coils can freeze, so additional low-temperature detection measures such as glycol-based fluids should be used in cold regions. Another protective device 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 38°F), it provides an alarm to shut the outside damper and open the return damper, or shut down the fan.

Figure 5 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

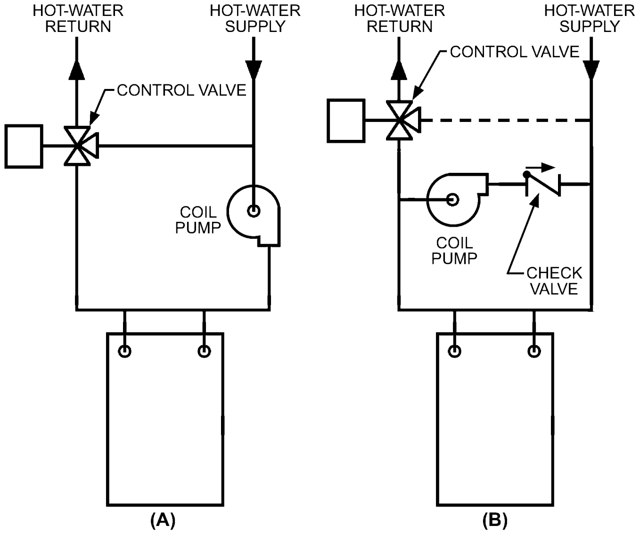

Fig. 6 Pumped Hot-Water Coil Variations: (A) In-Line and (B) Parallel

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 6A, the coil pump is decoupled from the primary pumps by the bridge. This prevents the control and hydraulic problems that can be associated with series pumping. Flow to the coil is constant volume and flow to the system is variable (for a two-way valve) or constant (for a three-way valve). Pressure drop through the control valve is handled by the system pumps.

Figure 6B 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 either the primary pump or 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. This design also requires a little more care in selecting the heating coil and sizing the pipe entering and leaving the coil: at design conditions (when the pump is on and the control valve is wide open), flow through the coil significantly exceeds the design primary water flow. The entering water temperature is less than that of the entering primary water, and thus may require a larger coil heat transfer area (fin and/or rows). 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 42 of the 2004 *ASHRAE Handbook—HVAC Systems and Equipment*).

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. 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. Step controllers provide cam-operated sequencing control of up to 10 stages of electric heat. Each stage may require a contactor, depending on the step controller contact rating. 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 make cycling so rapid that control appears proportional; therefore, face-and-bypass dampers are not used. Use

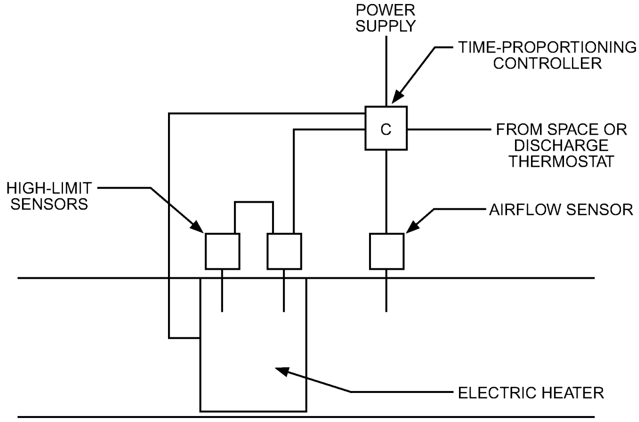

Fig. 7 Electric Heat: Solid-State Controller

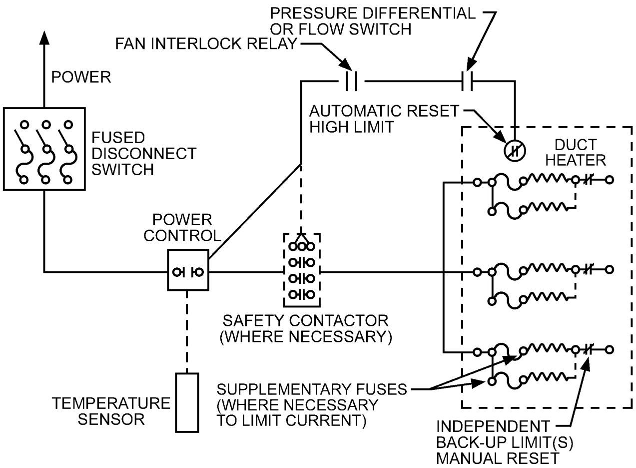

Fig. 8 Duct Heater Control

of electric heating coils is restricted in some areas because of energy consumption. Code compliance should be checked before using this application. A control system with a solid-state controller and safety controls is shown in Figure 7.

Current in individual elements of electric duct heaters is normally limited to a maximum safe value established by the *National Electrical Code*® (NFPA 2005) or local codes. Two safety devices in addition to the airflow interlock device are usually applied to duct heaters (Figure 8). The automatic reset high-limit thermostat normally turns off the control circuit. 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 back-up high-limit safety device is generally set independently to interrupt all current to the heater if other control devices fail. An electric heater must have a minimum airflow switch and two high-temperature limit sensors: one with manual reset and one with automatic reset. If still energized, electric coils and heaters can be damaged through overheating when air stops flowing around them. Therefore, control and power circuits must interlock with heat transfer devices (pumps and fans) to shut off electrical energy when the device 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. Limit thermostats should also be installed to turn off heaters when temperatures exceed safe operating levels.

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 42 of the 2004 *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 than a comparable water coil. In low-temperature applications, two-position control should be used, as discussed previously.

Radiant Heating and Cooling

Radiation 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. The thermostat can be placed in the unit's return air or on a wall at occupant level; return air control is generally less accurate and results in wider space-temperature fluctuations. Unit-mounted thermostats and packaged controls allow lower component cost and better assembly surety, and avoid the cost and coordination of a second trade for remote sensor installation. When the space is controlled for the comfort of seated occupants, wall-mounted thermostats give the best results.

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), outside reset of the water temperature to the radiator should be considered. Radiation can be zoned by exposure, and the compensating outside sensor can be located to sense compensated inside (outside) temperature, solar load, or both.

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 VAV systems, with the radiant panel operated as a one-zone, constant-temperature system. The one-zone radiant heating panel system is often operated from an outside temperature reset system to vary panel temperature as outside temperature varies. Where hydronic tubing or electric heating elements are imbedded in concrete, the rate of slab temperature change must be limited to prevent the concrete from cracking from thermal expansion.

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; as internal latent loads increase, the chilled-water temperature for the radiant cooling panels should be reset upward if the dew point becomes too high.

Direct-Fired Gas Heat Makeup Air Units

Direct-fired gas heat makeup air units are controlled from either a two-position or modulating sensor/controller or thermostat. Control can be from discharge air temperature, space temperature, outside air temperature, or combinations of these (e.g., discharge control with space or outdoor temperature override or reset). Outdoor temperature may also be used to enable heating. Temperature set point can be fixed, or controlled by a manual adjustment or a time clock for night setback. Sometimes switches mounted on outside (overhead) doors are used either to turn the unit on or off or to change the set point to compensate for changes in heating load.

Units have manufacturer-supplied controls and safeties for flame proving, airflow proving, outside air temperature reset of gas flow, and discharge air low limit, and 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. Any added controls must not interfere with the manufacturer's controls and safeties.

When a line voltage space thermostat is provided, it must have an adequate volt-ampere rating for the total load it must carry. The amperage ratings of all relay coils, valve-solenoid coils, and other components should be totaled and the thermostat's rating matched to the total.

COOLING SYSTEMS

Chillers

The manufacturer almost always supplies chillers with an automatic control package installed. 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 2004 *ASHRAE Handbook—HVAC Systems and Equipment*, Chapter 38 describes various chillers (e.g., centrifugal, reciprocating, screw, and scroll), and Chapter 12 covers variations in various 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 either.

Chiller plants are generally one of two types: variable flow (Figures 9 and 10) or constant flow (Figure 11). The figures show a parallel-flow piping configuration. Series-flow chiller configurations are often used in variable-primary-flow applications (Figure 9). 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.

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

The decision to reset chilled-water temperature in a variable-primary-flow chiller system should be based on facility requirements. In some cases, the chilled-water set point should be reset to a lower rather than a higher temperature, because the fan and pump power savings may outweigh the additional chiller power consumption at a lower temperature. Chilled-water reset may not be appropriate in variable-flow systems with tight space temperature and humidity requirements, such as hospitals, computer data centers, museums, or some manufacturing facilities. These facilities should operate with a constant chilled-water supply temperature. In a variable-primary-flow system, when flow is below the chiller minimum and the bypass valve is open, the chilled-water temperature can be reset until system flow matches chiller minimum flow with

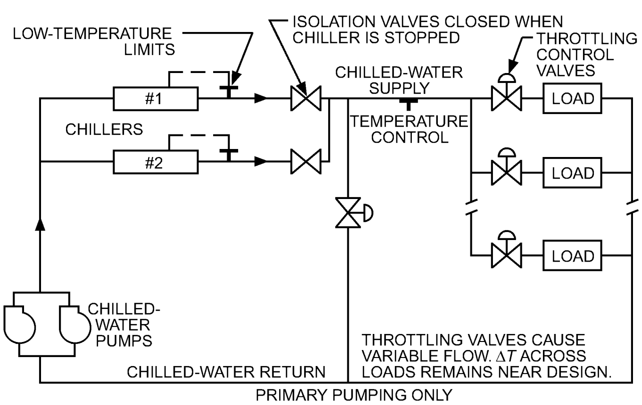

Fig. 9 Variable-Flow Chilled-Water System (Primary Only)

no additional pump power penalty. The chilled-water supply temperature typically establishes the base flow rate. To improve energy efficiency, the set point is reset for the zone with the greatest load (load reset) or other variances. Maintaining a high ΔT for the chillers improves plant efficiency. Properly sized coils and control valves help maintain a high ΔT.

The constant-flow system (Figure 11) is only constant flow under each combination of chillers on line; a major upset occurs whenever a chiller is turned on or off. The load reset function ensures that the zone with the largest load is satisfied, whereas supply or return water control treats average zone load. Use of constant-flow systems may be limited to smaller systems by energy codes. Three-way valves result in a lower ΔT at part load.

Chiller Plant Operation Optimization

Chapter 41 contains an extensive discussion on optimized control of chiller plants and a detailed description of 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.

Additional chiller(s) should be sequenced on when the existing online chillers can no longer meet the required cooling load, which in practice is determined when the chilled-water temperature supplied to the cooling coils rises above its defined set point. When additional chiller capacity is required, the projected power of all valid chiller combinations with their associated chilled- and condenser water pumps should be evaluated and the chiller/pump combination with the least projected power should be selected to operate. Valid chiller combinations involve chillers that are not in alarm or locked out, or have load ratios below a low limit (e.g., 30%) or above a high limit (e.g., 100%).

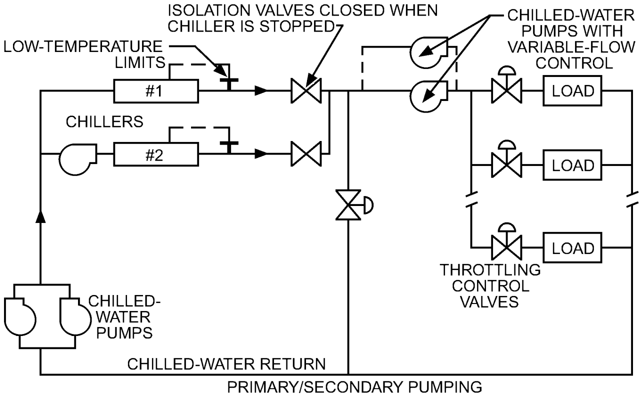

Fig. 10 Variable-Flow Chilled-Water System (Primary/Secondary)

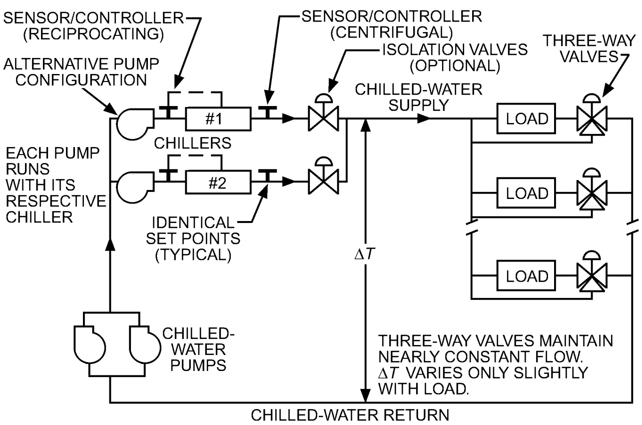

Fig. 11 Constant-Flow Chilled-Water System (Primary Only)

Multiple chillers 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 of the chillers. For the general case of chillers with different part-load characteristics, multiple chillers should be operated at equal marginal coefficients of performance. 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 constant-speed-drive centrifugal 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- or condenser water pumps are piped in a parallel (headered) arrangement or 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. In general, the optimal point of operation for each plant component (chillers, pumps, cooling tower fans, and air handler fans) that results in minimum plant power consumption occurs when each component is operating at equal marginal performance.

Chiller efficiency is a function of the percent of full load on the chiller and the difference in refrigerant pressure between the condenser and evaporator. In practice, the pressure is represented by condenser water exit temperature minus chilled-water supply temperature. To reduce the refrigerant pressure differential, the chilled-water supply temperature must be increased and/or the condenser water temperature decreased. An energy saving of 1 to 2% is obtained for each 1°F reduction in condenser water temperature supplied to a chiller. The savings are even greater when using variable-speed drives on centrifugal chillers.

The following methods are used to reduce the refrigerant pressure differential:

- Use chilled-water load reset to raise chilled-water supply set point as load decreases (use manufacturer's recommendations). Note that additional pump and fan power of VAV air handlers must be considered in calculating net energy savings.
- Lower condenser water temperature to lowest safe temperature (use manufacturer's recommendations), operate at full condenser water pump capacity, and maintain water temperature within design cooling tower approach (see Chapter 36 of the 2004 *ASHRAE Handbook—HVAC Systems and Equipment*).

Because cooling tower performance is tightly coupled to chiller performance, the cooling tower fans should be controlled to minimize the sum of the chiller and tower power consumption. The section on Supervisory Control Strategies and Tools in Chapter 41 describes a near-optimal algorithm for optimizing control of the tower fans according to this criterion. In this algorithm, condenser water temperature is allowed to float between high and low limits, and tower fans are controlled according a mathematical formula involving the part-load ratio (normalized load) measured on the chilled-water side of the plant and design parameters from the cooling tower and chiller manufacturer.

Cooling Tower

The most common packaged mechanical-draft cooling towers for comfort air-conditioning applications are counterflow induced-draft and forced-draft. They are controlled similarly. The 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 12 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. 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).

On larger towers, 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 in operation, each fan's speed should sequenced 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.

Systems with multiple cooling towers may have isolation valves to allow unneeded towers to be shut down. When towers are piped on a common header, valves should be sequenced with condenser water pumps to ensure that pumps are not deadheaded or that flow to operating towers is not reduced below the manufacturer's minimum flow.

In climates where the wet-bulb temperature could be higher than the condenser water supply temperature (tower approach), a wet-bulb high limit should be provided so that the controls are not operating fans at increasingly higher speeds with no increase in cooling. This high limit is useful when using the towers on hydronic free cooling systems.

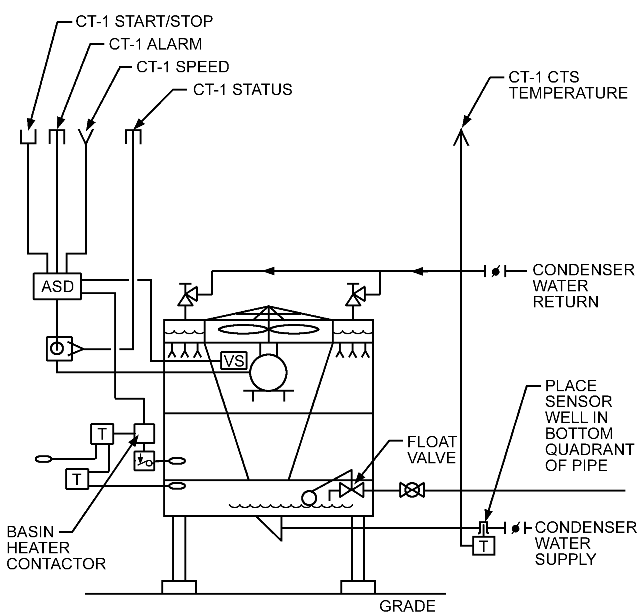

Fig. 12 Cooling Tower

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 outside 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.

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 outside air approaches freezing, and refill the tower on the next call for cooling.

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 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. Flow in air-cooled chillers does not turn down as much as with other types of chillers. Minimum flow varies by manufacturer, but may be as high as 60%. 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.

Water-Side Economizers

Water-side economizers are typically flat-plate heat exchangers where one side is cold tower water and the other side is chilled water. 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. 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 often makes starting the chiller difficult. 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 DDC system. 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. The pressure signal used for modulating the valve should be directly from a pressure transmitter. Relying on pressures obtained from the chiller controller through the network makes for difficult control, because data refresh rates may not be sufficient. Another challenge in modulating chiller valves is valve sizing. If line-sized valves are used, they are oversized for good control. Valves correctly sized for control add a considerable pressure drop to the condenser water loop.

Cooling Coil

Chilled-water or brine (glycol) cooling coils are controlled by two- or three-way valves (Figure 13). 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 or hot-gas bypass valves. Most DX systems for commercial application have two to six stages.

Most often, these feedback loops are set up to cycle the cooling stages on and off as air temperature swings up and down around the set point. Depending on the capacity of the stages and the allowable error in the feedback loop, it is sometimes possible to set up the loop so that it stabilizes (rather than cycling), with some number of stages active and a steady-state temperature error.

When set up properly, a cycling DX system under a steady load operates like a two-position control loop (see Chapter 15 of the 2005 *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 DDC 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 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. This 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 driving a staging module.

If the DX system serves a single zone, the feedback signal is usually the space temperature, which usually varies by 1 to 4°F. If the DX equipment serves multiple zones, as in a VAV air handler, the feedback signal is often 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.

AIR SYSTEMS

Variable Air Volume (VAV)

Supply Fan Control. Most VAV systems have pressure-independent terminals, which means a separate airflow control loop operates each terminal damper. In normal operation, these loops combine to set the airflow though the supply fan. So, within limits, the fan controller has no effect on airflow to the zones. The VAV fan controller

- Ensures the pressure in the duct is enough to serve the terminals
- Prevents excessive pressure disrupting the terminal flow loops
- Avoids unnecessary energy consumption at the fan
- Keeps the fan in a stable region of the pressure-flow curve

Historically, a variety of mechanisms (e.g., bypass damper, movable inlet vanes) have been used to regulate fan output. These methods vary widely in efficiency and energy consumption. A bypass damper (Figure 14), for example, does not change the fan's output, but can allow the fan to accommodate distribution system flow variations without fan instability. Some methods, such as discharge dampers, are restricted by energy codes. Variable-speed drives (VSDs) are frequently used because their low energy consumption and falling first cost makes them cost effective.

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. Most often, the set point is a constant, selected by the designer and confirmed by the balancer during system commissioning. If the pressure loop and flow loops are sufficiently decoupled, this is a solid, stable design. 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 15).

Though widely used, this fixed-pressure design uses more energy than necessary. Even when applied carefully, there are many

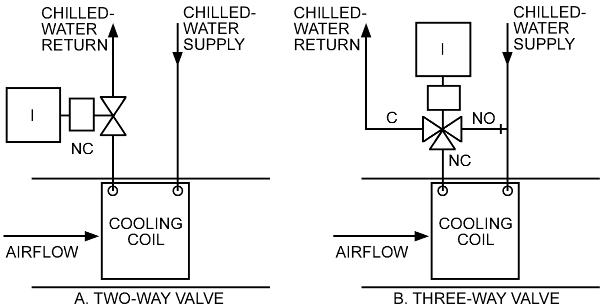

Fig. 13 Control of Chilled-Water Coils

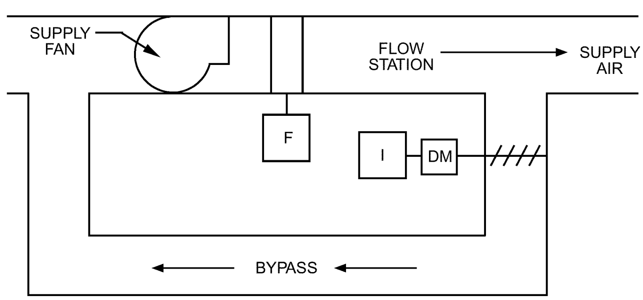

Fig. 14 Fan Bypass Control to Prevent Supply Fan Instability

operating hours when the fan pushes air through a system full of partly closed dampers. Many building codes and ASHRAE *Standard* 90.1 require automatically adjusting duct pressure based on zone requirements as system load varies. Airflow to zones is still regulated by flow loops in the terminal controllers and is unaffected, but the system meets the load more efficiently with terminal dampers closer to open and a lower fan speed. This reduces energy consumption at the fan. Ideally, pressure is reduced to the point that some of the dampers open all the way. Any further supply fan speed reduction reduces airflow at the terminals.

Many methods have been published to automatically reset duct pressure (Ahmed 2001; CEC 2003; 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. In many cases, it makes the most sense to measure duct pressure near the fan outlet.

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 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. Many controllers do not have saturation signals, but they have become more common since they were added to the LONMARK profile (LONMARK 2002). Control system designers should ensure that 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 is more practical, 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.

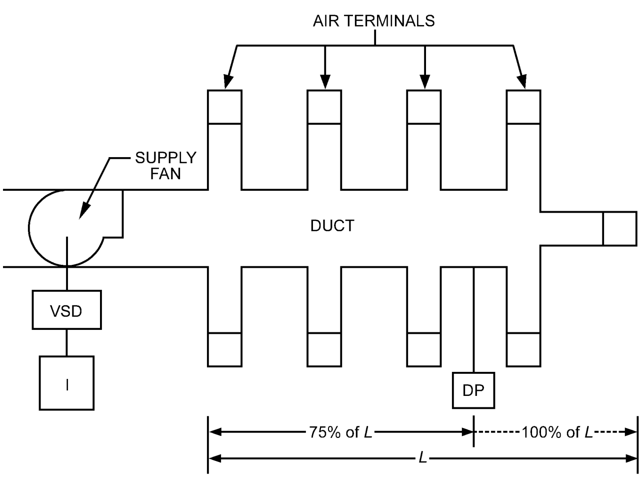

Fig. 15 Duct Static-Pressure Control

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. Some terminal controllers do not always have reliable values for the damper position because of the way they apply floating actuators. It is important to take this into account when selecting a reset method.

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.

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 that the intended mechanical operating point is the lowest pressure that satisfies the terminals. Verifying this condition is fairly straightforward for commissioning agents or building operators. A mechanically specified system allows proposals from vendors with a wider variety of equipment and algorithms.

Supply fan warm-up control for systems having a return fan (Figure 16) must prevent the supply fan from delivering more airflow than the return fan maximum capacity during warm-up mode. If pressure on fan discharge goes negative, supply fan speed should be reduced.

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 (low limit on the suction side of the fans), 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., mechanical relief dampers) is needed in these situations.

Space Pressure Control. Differential static-pressure control, differential airflow, and directional bleed airflow are methods used to pressurize a space relative to adjacent spaces or the outside. Typical applications include pressure barriers for any occupied space to maintain interior comfort conditions, to prevent infiltration of moisture unfiltered and untreated air. Applications requiring higher-performance controls include cleanrooms (positive pressure to prevent infiltration; see Chapter 16), laboratories and health care

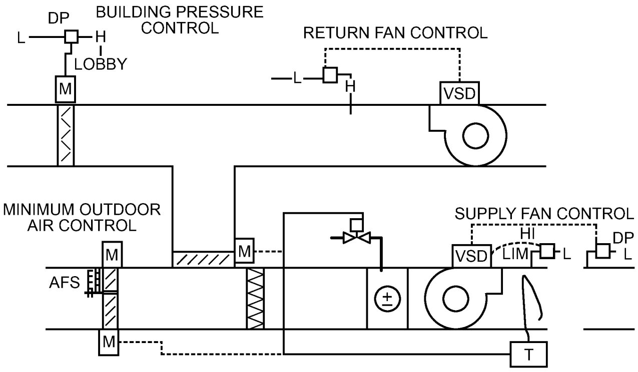

Fig. 16 Supply/Return Fan Control

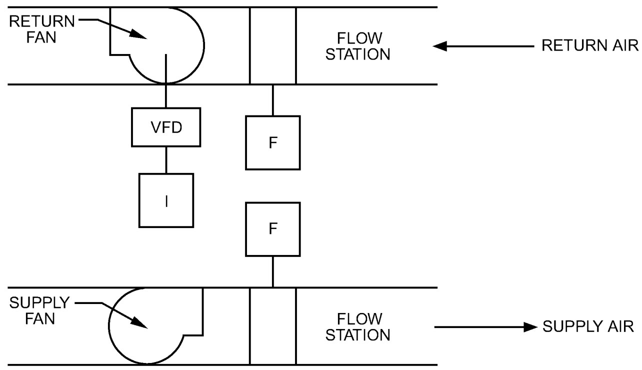

Fig. 17 Airflow Tracking Control

infection control (positive or negative, depending on use; see Chapter 7), and various manufacturing processes, such as spray-painting rooms (see Chapters 29 and 30 for industrial applications). The pressure controller usually modulates fan speed or dampers 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. Health care spaces have specific pressure control guidelines for minimum air change rates, differential pressure, flow, or leakage rates. States typically adopt standards for HVAC system design, ventilation, filtration, and pressurization to ensure an adequate environment for health care occupancies. These standards and their application are discussed by the American Institute of Architects (AIA 2001), ASHRAE (2003a), and Chapter 7. The Joint Commission on Accreditation of Healthcare Organizations (JCAHO) also enforces HVAC system design and construction standards listed in the AIA guidelines.

In addition to these regulations, standards and guidelines published by ASHRAE, the AIA, American Society of Hospital Engineers (ASHE), Centers for Disease Control and Prevention (CDC), National Institute of Health (NIH), and National Institute of Occupational Safety and Health (NIOSH) define the expectation of care and services in health care.

Most guidelines do not allow spaces to be used for purposes that require pressurization flow to change directions (e.g., positive to negative flow), even if the space and control components are able to perform the change.

Building Pressurization. Building pressure and outside airflow control must be integrated for proper system performance. 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, outside air is required to pressurize a zone.

Static pressure control is one method for control of the relief or exhaust fan requiring direct measurement of the space and outside static pressures. The location for measuring inside static pressure must be selected carefully, away from openings to the outside or elevator lobbies, and, when using a sensor, in a large representative area that is shielded from wind pressure effects or drafts. The outside location must also be selected carefully, typically 10 to 15 ft above the building and oriented to minimize wind effects from all directions.

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

Return or relief fan control for VAV systems is required for proper building pressurization and minimum outside air control. The return fan is controlled to maintain the return air plenum pressure. The relief fan or exhaust air damper may be controlled to maintain building static pressure (see Figure 16).

Direct-measurement pressurization flow compares an interior static pressure location to an outdoor reference to modulate fan speed. This control allows for greater operational repeatability, greater comfort control, improved energy savings potential and the opportunity to use a small (0.004 in. of water) differential counterflow barrier to the infiltration of moisture that can cause condensation and mold in exterior walls.

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 17). Because return air is typically the controlled variable and because 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 differentials that may be tracked are described in the following pressurization flow model, which proves the equivalency between the differentials of air system flows (intake minus exhaust, or supply minus return, assuming that positive pressurization flow is desired).

Fixed-differential air volume to maintain pressurization flow, rather than measured space static pressure, results in very stable control for several reasons. First, multiple air-handling systems operate independently. Systems based on measured static pressure generally interact with each other in such a way that it is impossible for each fan system to operate independently. As a result, pressurization problems in specific pressure zones are likely. Second, static pressure control can only ensure that pressure is maintained at the measured location and as compared to a reference point. The reference selected may not represent a true neutral position for external static. Finally, transient changes in the building (e.g., doors opening, wind gusts) result in the system going out of control in a static pressure-based system.

Indirect building pressure control, as an independent control loop, avoids system instability and inefficiencies when doors open or close, and when transient environmental changes affect differential pressure and would otherwise cause wide swings in volumetric relationships.

Airflow quantity is indicated in Figure 18 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. The windward wall may have significant infiltration, whereas the door into a very negative infectious isolation room may show exfiltration from the room being measured.

Pressurization flow can be controlled using airflow tracking by maintaining a fixed-volume differential between either total intake and total exhaust, or total supply and total return. Unfortunately, the more direct method of control is usually not practical because of the difficulty in measuring flow rates with most exhaust/relief configurations. Intake airflow rates may be measured reliably

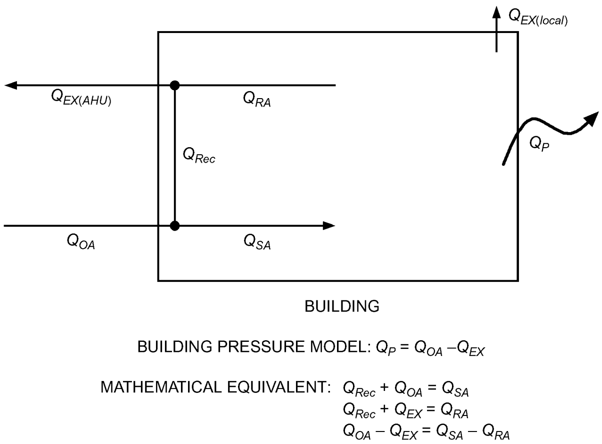

Fig. 18 Building Pressure Model

upstream of modulating intake dampers and with minimal protection from external wind effects, or by reducing the intake duct size sufficiently to overcome the low-velocity limits of differential-pressure-based sensors.

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 then 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 outside air to make up the exhaust and still have a slightly pressurized space. During night cooldown, when using large quantities of outside air, the return fan operates in the normal mode (Kettler 1995).

VAV systems that use return or relief fans require control of airflow through the return or relief air duct systems. Return fans are commonly used in VAV systems when return duct pressure loss is high, and a high percentage of outdoor air is used (Kettler 2004). In a return fan VAV system, there is significant potential for control system instability because of the interaction of control variables (Avery 1992; Kettler 2004), which typically include supply fan speed, supply duct static pressure, return fan speed, mixed air temperature, outside and return air damper flow characteristics, and wind pressure effect on the relief louver. Interaction of these variables and selection of control schemes to minimize or eliminate interaction must be considered carefully. Mixed-air damper sizing and selection are particularly important. Zone pressurization, building construction, and outside wind velocity must be considered. The resultant design helps ensure proper air distribution, especially through the return air duct.

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 usually required to maintain airflow in the fan's stable range.

Supply air temperature reset can be used to avoid fan instability by resetting the cooling coil discharge temperature higher (Figure 19), so building cooling loads require greater airflow. Because of the time lag between temperature reset and demand for more airflow, the value at which reset starts should be selected on the safe side of the fan instability point. When this technique is used, it must be ascertained that zone cooling load and dehumidification requirements can be met with the higher discharge air temperature. Discharge temperature should be reset lower to keep fan speed at the lowest acceptable value for stability. This saves fan power, and only decreases chiller efficiency a small amount.

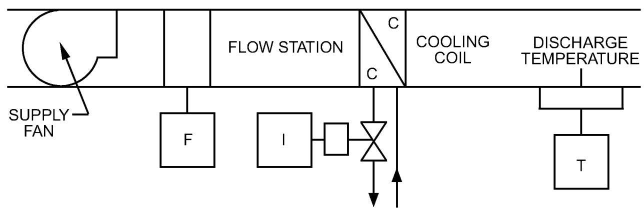

Fig. 19 Coil Reset Control to Prevent Supply Fan Instability

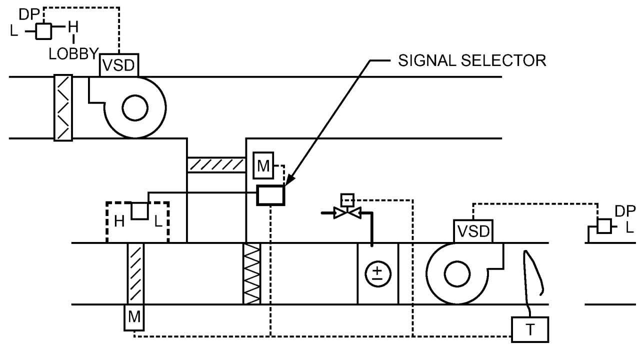

Fig. 20 Minimum Outside Air Control Using Differential Pressure Controls

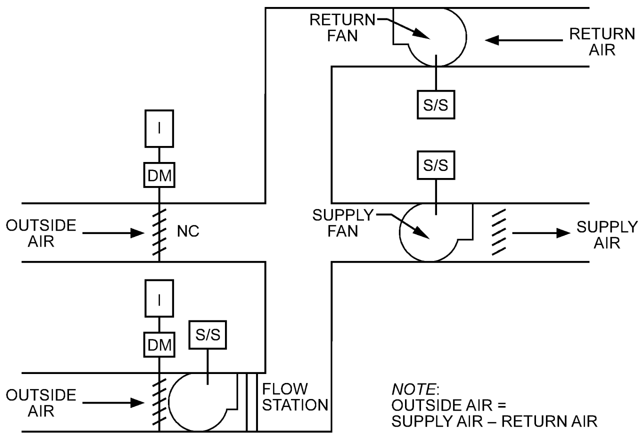

Fig. 21 Minimum Outside Air Control with Outside Air Injection Fan

Minimum Outside Air Control. Fixed minimum outside air control provides dilution air for ventilation, pressurization flow (usually exfiltration), combustion air for processes converting fuel to heat, and makeup air for exhaust fans.

Several variations of minimum outside air control for VAV systems are possible:

- Minimum outside airflow may be determined by measuring intake airflow directly upstream of the outside damper and modulating the return damper to maintain target minimum flow rate as supply airflow is reduced (Figure 20).
- A dedicated outdoor air injection fan may be used (Figure 21).
- An airflow station may be installed in the minimum outside air section and minimum flow rate controlled directly by modulating the intake and return dampers (or return fan) in sequence (Figure 22). In this case, the intake opening should be sized for velocities high enough to facilitate measurement and avoid wind effects, yet low enough to prevent moisture entrainment.

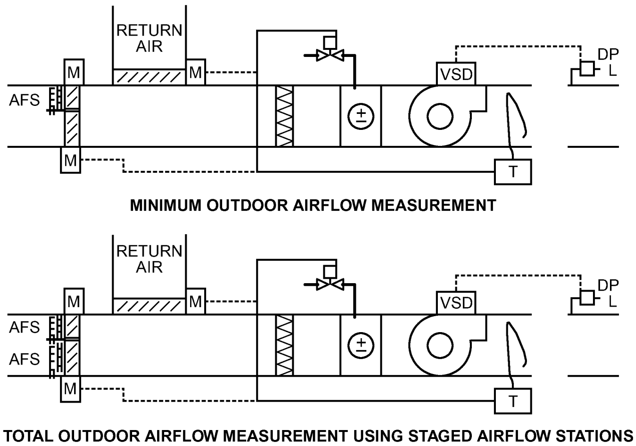

Fig. 22 Outside Air Control with Airflow Measuring Stations

According to the *Standard 62.1 User's Manual* (ASHRAE 2005), most VAV systems require outdoor airflow sensors and modulating dampers or injection fans for compliance; a fixed-speed outdoor-air fan without control devices will not maintain rates within the required accuracy. In addition, ASHRAE *Research Project* RP-980 (Krarti et al. 2000) and the *Standard 62.1 User's Manual* suggest that even small errors in measurements of total supply airflow and total return flow can cause significant errors in the determination and control of the minimum outside airflow rates, making return fan or airflow tracking unsatisfactory for minimum outside ventilation control. Dynamic intake controls require real-time feedback of intake rate changes for accurate system adjustments.

If the total outdoor airflow range of a VAV design from a minimum of less than 50% to maximum design capacity is to be measured, the selected measurement technology should provide the needed reliability across the entire anticipated temperature and velocity range. One method for accomplishing this with pitot arrays is by subdividing the intake and using 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). If more outside air is supplied than the difference between the supply and return fan airflows, a variation of economizer cycle control may be used (Figure 22).

In addition to the capability of lower flow rate measurement, significant consideration should be given to the modulating mechanism's ability to control rates as low as those anticipated. Today's rooftop air-handling units with economizers usually use a single large damper to control both economizer and minimum air intake rates. This single damper must then control at minimum air velocities (often below 50 fpm) under maximum turndown conditions. Traditional damper blade and linkage designs combined with the finite modulating capabilities of actuation devices make stable control flow difficult at these rates. The resistance of the wide-open damper can be expressed as a fraction of the total system resistance, called **damper authority** or **characteristic ratio** (Lizardos and Elantz 2000). Both damper authority and measurement placement in relation to the exterior wall should be evaluated and determined when control of intake flow rates below 200 fpm is required.

In systems using 100% outside air, all air goes to the fan and no air is returned (Figure 23). The outside air damper is interlocked and usually opens before the fan starts. Fan speed may be controlled to dynamically compensate for internal or external environmental conditions that affect mixed air plenum pressure and fan performance, or to compensate for filter loading when constant supply volume is required.

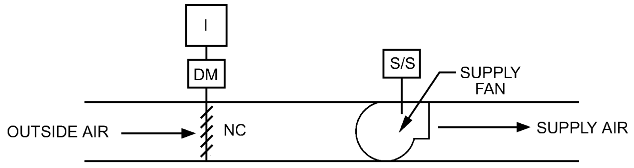

Fig. 23 100% Outside Air Control

Regardless of the type of system, pressurization flow rate and outside airflow rate are controlled similarly. There is generally no relief at the air-handling unit during minimum outside air mode.

The outside airflow set point for dilution ventilation should be established using ASHRAE *Standard* 62.1. In addition, the outside 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 outside air set point. If the outside airflow rate for minimum ventilation significantly exceeds that required for pressurization, the economizer control strategy for pressurization control should be followed.

Because wind and stack pressure variations can significantly affect intake flow rates, outside airflow should be actively controlled on all systems, regardless of type (VAV, constant-volume, etc.). The most practical way to maintain outside air intake flow rate is to install a permanently mounted airflow measurement device properly selected for use in outside air intakes.

Traditionally, the outside air damper is modulated or the outside air and recirculation dampers are modulated inversely to maintain set point. Unfortunately, most dampers are oversized and are ineffective as control valves, often preventing stable control.

A more reliable approach is to decouple the outside air and recirculation dampers by individually actuating each. Depending on the system, the outside airflow rate is then controlled by sequencing the dampers. An effective sequence sets the outside air damper to a fixed minimum position. The damper is opened if the set point is above the measured airflow rate. Once the outside air damper is fully opened, the recirculation damper begins to close. The return fan (if present) can also be used to maintain outside air set point.

This sequential control strategy improves performance of oversized dampers and allows the use of opposed-blade dampers. If the selected airflow measuring devices can measure low airflows (down to 150 fpm), a single outside air damper can be used on systems with air-side economizer sections, thus simplifying control. Otherwise, a separate minimum outside air damper must be installed.

Dynamic Reset of Intake Rates or Demand-Controlled Ventilation (DCV). Codes and standards address the need to maximize the efficiency of operational energy usage. One area that has received recent attention is minimizing the amount of required intake air when spaces being served have less than maximum occupancy. Other reasons for resetting intake rates include operational changes, environmental changes, and technical compliance.

To effect this type of control and to comply with provisions of codes and standards, the intake rate must be controlled to a specific minimum level for full occupancy and adjusted based on a real-time input that reflects changes in occupancy.

DCV's operational control effectiveness largely depends on the reliability of the individual sensor inputs. The appendix on CO_2-based DCV in the *Standard 62.1 User's Manual* (ASHRAE 2005) illustrates this point.

Because of the direct relationship to occupancy, intake rates may be expressed in flow rate per person for the human or odor component of the intake requirement. Until recently, this relationship was thought to allow several variations of a CO_2 concentration balance formula to be used to calculate the effective occupancy intake requirement based on several prerequisite assumptions during steady-state conditions; research has indicated that this approach may not be accurate or reliable enough for compliance with existing rate-based codes.

The most obvious solution is to use direct counting instruments or systems that provide HVAC controls with a net count of occupants per zone served by or controlled by a specific air-handling unit (AHU). This configuration allows software in the system to dynamically adjust the intake rate at the AHU based on direct feedback of both the demand (changing occupancy level) for control and the ventilation rate (at the intake and supply to the zone).

There are alternatives to direct counting. All of them have serious uncertainties, but none very different from those of more traditional CO_2-based DCV. With these alternatives, ASHRAE *Standard* 62.1's original intention of dynamic reset (i.e., continuously determine intake rate and reset it) can be satisfied based on a known or calculated population.

In recognition of the significance of building-generated contaminants, ASHRAE and international ventilation standards include requirements to reflect both components used to determine minimum ventilation rates: people, and contaminants generated in the space that they occupy. Now, using CO_2 sensing for ventilation control becomes more complicated, because the total requirement has a volume/area component that does not vary with changes in occupancy.

Designers should use caution when implementing demand-based ventilation schemes that reduce outside air without provisions for ensuring that pressurization flow is maintained (i.e., the relationship between the outside air and the exhaust air is maintained). When outside air dew point approaches or exceeds 60°F, it is essential to create a net positive pressurization flow to prevent transport of water and outdoor air contaminants into the building or its envelope. Excessive moisture can result in mold growth, structure damage, and poor indoor air quality. Net positive pressure can dry the building envelope and help ensure proper temperature and humidity control within the building. The concept of net pressure must be understood for building systems. Wind pressure acting on the exposed building surfaces cannot be overcome with the mechanical system without damaging envelope integrity. Therefore, a net positive pressure is the best scenario that can be maintained. As a result, even a properly pressurized building can have exterior walls improperly pressurized for short periods of time. When outside air temperature is less than indoor air dew point, a net neutral pressurization flow should be maintained.

The potential energy savings from minimizing intake rates through demand-controlled ventilation must be weighed against the probability that control errors will overcome and negate the estimated savings, making the risks of underventilation and depressurization the larger consideration.

Air-Side Economizer Cycle. Economizer-cycle control reduces cooling costs when outside air is cool and dry enough to be used as a cooling medium. If outside air is below an economizer changeover temperature limit, typically 65°F (selected to represent outside air with a mean coincident dew point below 60°F), the return, exhaust, and outside air dampers modulate to maintain a mixed-air cooling set point, typically 55 to 60°F (Figure 24). The outside and exhaust dampers would be fully or mostly open and would close, and the return air dampers open, when the supply fan is not operating. When the outside air temperature exceeds the economizer changeover temperature limit set point, the outside air damper closes to a fixed minimum ventilation rate and the exhaust and return air dampers close and open, respectively.

During air-side economizer operation (**free cooling**), or any time air is exhausted to the outside at the air handler, the differential between supply and return airflow rates should be maintained. The difference should equal the pressurization flow plus flow through return and local exhausts. Most free-cooling systems have either a return or relief fan, which should be modulated to maintain the preset supply/return airflow differential.

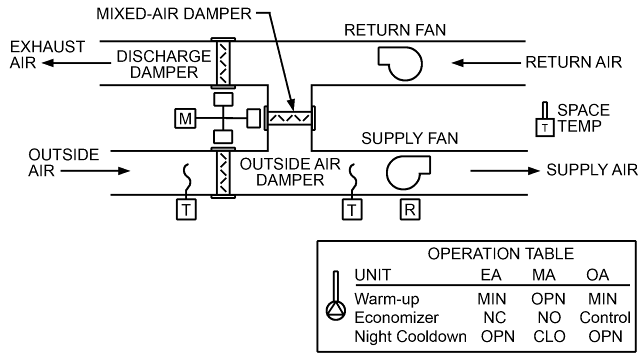

Fig. 24 Economizer Cycle Control

Modulating economizer systems achieve the mixed-air temperature set point by modulating outside air and recirculation dampers in sequence, as described in the section on Minimum Outside Air Control. When a single intake damper is used, the mixed-air temperature set point is maintained by resetting the outside airflow set point. When a minimum/maximum damper arrangement is used, the mixed-air temperature is achieved by directly controlling the dampers. Resetting the outside airflow set point results in more stable temperature control.

The relief air damper should be opened during free cooling, because it is expected that large quantities of supply air will need to leave the building along a planned path of flow and not an unplanned path, such as entry doors standing open. As an added level of security against negative airflow into the relief damper, a differential pressure sensor or bleed airflow sensor could be placed across the relief damper. This measured value would reset the return damper closed or increase the return fan speed to ensure outward flow of relief air.

In enthalpy economizer control, the economizer changeover temperature limit no longer assumes an outdoor humidity but rather also measures relative humidity to compare the calculated enthalpy difference between return and outside air. The economizer cycle is active only when outside conditions have a lower heat content than return air, to further reduce energy costs when latent loads are significant. The interlock function can be based instead on (1) a fixed enthalpy upper limit, (2) a comparison with return air so as not to exceed return air enthalpy, or (3) a combination of enthalpy and high-temperature limits. However, enthalpy-sensing devices require regular maintenance and calibration.

VAV warm-up control during unoccupied periods requires no outside air; typically, outside and exhaust dampers remain closed. The supply fan and return fan airflow are offset to maintain positive or negative duct pressurization.

Where outside conditions allow, night cooldown control (**night purge**) provides 100% outside air for cooling during unoccupied periods. The space is cooled to the space set point, typically 9°F above outside air temperature. Limit controls prevent operation if outside air is above space dry-bulb temperature, if outside dew-point temperature is excessive, or if outside dry-bulb temperature is too cold (typically 50°F or below). When outside air conditions are acceptable and the space requires cooling, the cooldown cycle is the first phase of the optimum start sequence.

Constant-Volume (CV) Systems

In a constant-volume system, supply and return fan airflow rates are manually set to meet the maximum airflow requirements for thermal load and ventilation. The air-handling unit's mixed-air dampers, cooling coil, and heating coil (where applicable) are controlled by supply air temperature. The supply air temperature set point is set to satisfy the zone with the maximum load, and, to improve energy efficiency, can be reset from the zone with the greatest load. Reheat coils on constant-volume terminal boxes are controlled by individual space thermostats to establish the final space temperatures. If warm-up mode is used, the supply air set point is adjusted upward to the desired value.

The mixed-air dampers are sequenced with the cooling coil with or without overlap, depending on the design strategy. The mixed-air damper control can be enabled by a dry-bulb- or enthalpy-based economizer or a demand-controlled ventilation scheme (see the Variable-Air-Volume section for details). Balancing dampers may also be incorporated to ensure the required airflow for various positions of the mixed-air dampers.

During unoccupied periods when the fan must run for morning warm-up or night cooldown cycle, mixed-air dampers should be positioned to 100% return air to help achieve set point and to conserve energy. At the end of morning warm-up and the beginning of occupancy, when the space reaches set point, the unit switches to occupied mode. If the space reaches the warm-up set point before occupancy, the outside air damper should remain closed to minimize the additional heating energy needed. Morning warm-up occurs only once in a day. In the warm-up mode, with heating supplied by the air handling unit, terminal and perimeter heating units may be disabled as appropriate to use the lowest-cost energy source.

When the fan runs for night cooldown, the mixed-air dampers are positioned to 100% return air unless positioned by an economizer. In this case, the cooldown mode uses the economizer as the first stage of cooling and mechanical cooling as next stage(s) of cooling. When the space reaches the cooldown set point, the unit switches to occupied mode. If the space reaches the cooldown set point before occupancy, the system remains in occupied mode to prevent redundant cooldown operation. Night cooldown occurs only once in a day.

Changeover/Bypass Zoning Systems

Changeover/bypass zoning systems, often referred to as variable-volume/variable-temperature systems, are typically used to convert a single-zone constant-volume air-handling unit into multiple variable-volume temperature-controlled zones (Figure 25). The unit fan usually operates continuously in occupied mode, and cycles as required to maintain setback/setup zone temperatures in unoccupied mode. The unit remains a constant-volume unit and, as zones close off airflow to a particular zone, a static pressure sensor in the supply duct typically modulates a damper to bypass air back to the return duct to keep airflow across the unit's coils relatively constant. A controller on the unit turns on stages of heat or cooling based on zone thermostat requirements. Zone control is typically pressure-dependent, and the zone controllers modulate a damper between maximum and minimum position based on the zone's temperature requirements and supply air temperature. Pressure-independent control is an option on some systems; the zone controller measures airflow to the zone and resets the air volume to the zone between maximum and minimum airflow limits, based on zone temperature requirements and supply air temperature. Digital controllers on the zone dampers "vote" for heating or cooling based on zone temperature and deviation from set point. When the vote favors heating, the unit controller stages heating on and broadcasts to the zone controllers that warm air is coming down the duct. Any zone requiring heat opens its damper, and any zone requiring cooling closes its damper to minimum position or flow. Similarly, when the vote is for cooling, the unit turns on stages of cooling and broadcasts to the zones that cool air is coming down the duct. If no zone requires heat or cooling, the supply fan remains on in occupied mode, but either keeps the heating and cooling off or cycles heating and cooling as required to maintain a supply temperature approximately equal to the zone temperature ±5°F.

These systems typically serve fewer than 15 zones per air-handling unit, and 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.

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 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 40°F dew point on the coil surface. As indicated in Figure 26, this results in a relative humidity of about 30% at a space temperature of 75°F, which is adequate for most commercial applications. When lower humidity is needed, a chemical dehumidifier is required.

Dehumidification can be achieved in several ways. One is to override control of the cooling coil. The temperature of the coil is lowered until sufficient moisture is removed from the supply air to maintain the humidity set point. When maximum relative humidity control is required, a space or return air humidistat is provided in addition to the space thermostat. To limit maximum humidity, a control function selects the higher of the output signals from the two devices and controls the cooling coil valve accordingly. A reheat coil may be required to maintain the space temperature if moisture removal results in too low a supply air temperature (Figure 27). If humidification is also provided, this cycle is sometimes called a

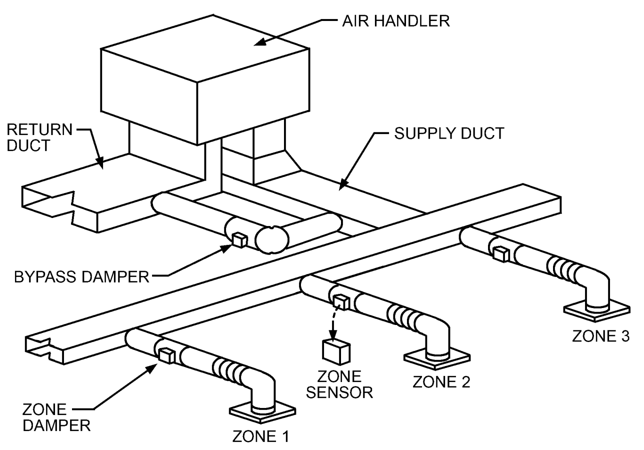

Fig. 25 Changeover/Bypass Zoning System

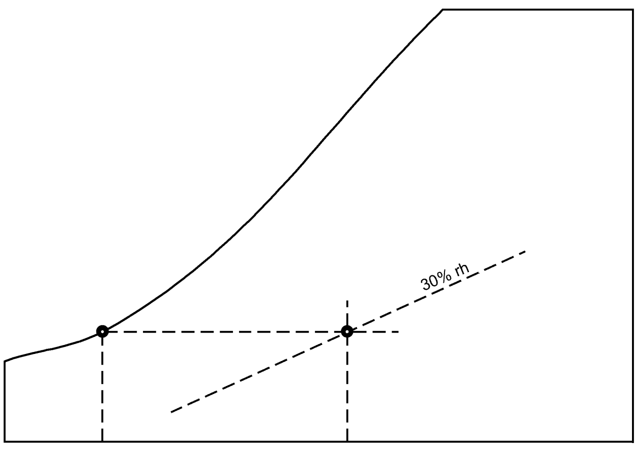

Fig. 26 Psychrometric Chart: Cooling and Dehumidifying, Practical Low Limit

constant-temperature, constant-humidity cycle. Although simple cooling by refrigeration maintains an upper limit to space humidity, without additional equipment it does not control humidity.

Sprayed-coil dehumidifiers (Figure 28) have been used for dehumidification. Space relative humidity ranging from 35 to 55% at 75°F 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.

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 regenerate the material. The adsorption also generates heat (Figure 29). Figure 30 shows a typical control.

Humidification can be achieved by adding moisture to supply air, using evaporative pans (usually heated), steam jets, 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 31). 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.

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.

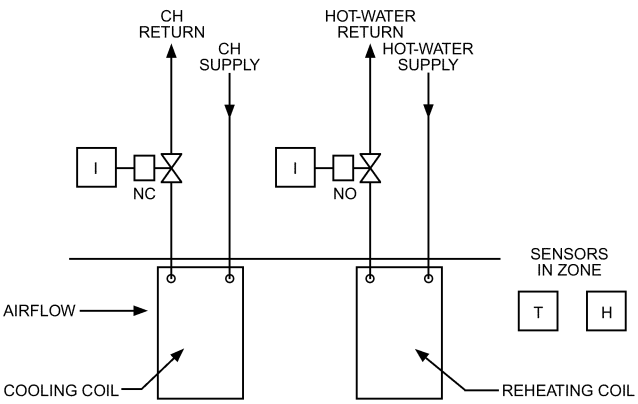

Fig. 27 Cooling and Dehumidifying with Reheat

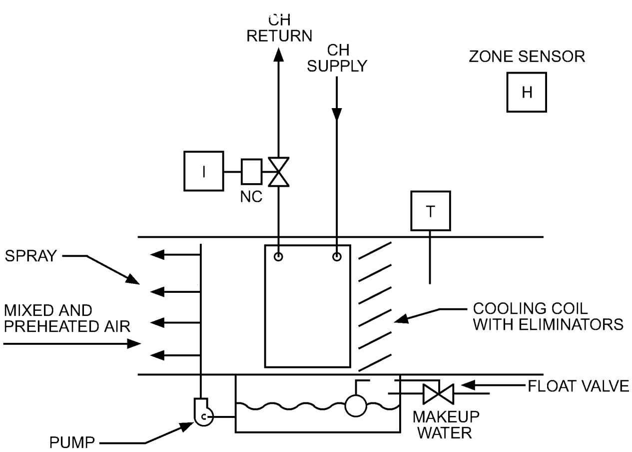

Fig. 28 Sprayed-Coil Dehumidifier

Single-Duct Constant-Volume. Reheat terminals use a single constant-volume fan system that serves multiple zones (Figure 32). All delivered 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) is reset as required to maintain the space condition. Because these systems consume more energy than VAV systems, they are generally limited to applications with fixed ventilation needs, such as hospitals and special processes or laboratories.

No fan control is required 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 allow the supply temperature to respond to demand from the greatest cooling load, thus conserving energy.

Single-Duct Variable-Volume. A **throttling VAV terminal** has an inlet damper that controls the flow of supply air (Figure 33). For spaces requiring heating, a reheat coil can be installed in the discharge. A common control sequence for this system (particularly with pneumatic controls) is the single maximum scheme depicted in

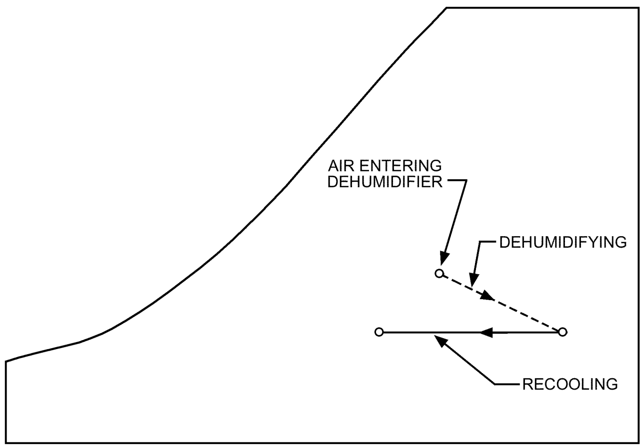

Fig. 29 Psychrometric Chart: Chemical Dehumidification

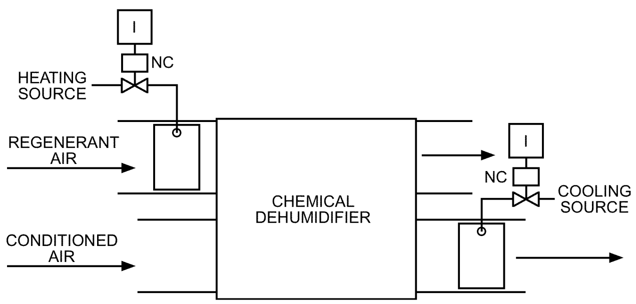

Fig. 30 Chemical Dehumidifier

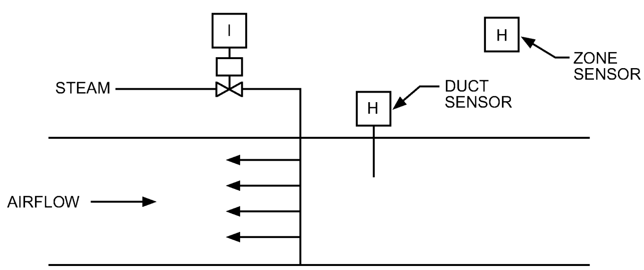

Fig. 31 Steam Jet Humidifier

Figure 33. In this sequence, as temperature in the space drops below the set point, the damper begins to close and reduce airflow to the space. When airflow reaches the minimum limit, the valve on the reheat coil begins to open.

One disadvantage of this sequence is that the minimum flow set point must be high enough to meet the design heating load at a supply air temperature that is low enough to prevent stratification (e.g., less than 90°F). Therefore, the minimum flow set point typically must be 30 to 50% of the maximum flow set point. This wastes a great deal of reheat and fan energy, particularly for zones that are very conservatively sized.

A much more energy-efficient sequence is the dual maximum sequence in Figure 34. 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. Then the supply air temperature is reset from minimum (e.g., 55°F) to maximum (e.g., 90°F), 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. One of the advantages of the dual maximum sequence is that the minimum flow set point is not limited by stratification (as described for the single maximum) and can be set as low as 10 to 20% of the maximum flow, depending on ventilation requirements and the lowest nonzero controllable flow. Thus, the dual maximum sequence greatly reduces wasted reheat and fan energy, and should be used whenever DDC controls are used.

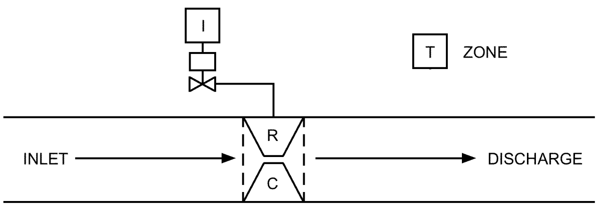

Fig. 32 Single-Duct Constant-Volume Zone Reheat

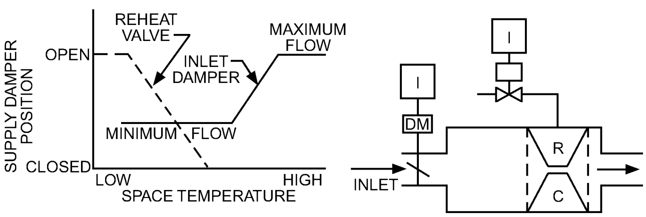

Fig. 33 Throttling VAV Terminal Unit: Single Minimum Control Sequence

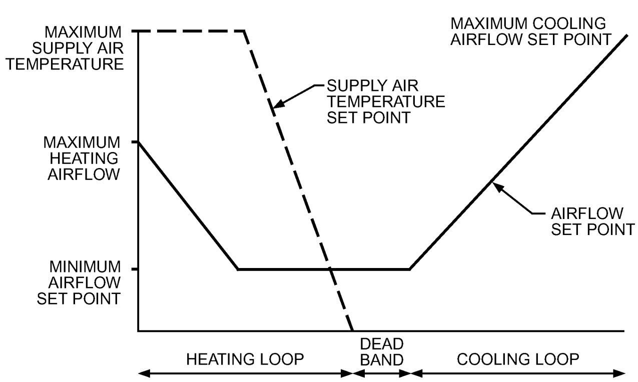

Fig. 34 Throttling VAV Terminal Unit: Dual Minimum Control Sequence

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 35). 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.

A **bypass VAV terminal** has a damper that diverts part of the supply air into the return plenum (Figure 36). Control of the diverting damper is based on output of the space temperature sensor. When temperature in the space drops below the set point, the bypass damper begins to open, routing some supply air to the plenum, which reduces the amount of supply air entering the space. When the bypass is fully open, the reheat coil's control valve opens as required to maintain space temperature. A manual balancing damper in the bypass is adjusted to match the resistance in the discharge duct. In this way, the supply of air from the primary system remains at a constant volume. The maximum airflow through the bypass must be restricted to maintain minimum airflow into the space. Although airflow to the space is reduced, the fan's total airflow remains constant, so fan power and associated energy cost are not reduced. These terminals can be added to a single-zone constant-volume system to provide zoning without the energy penalty of a conventional reheat system. However, if a return or exhaust fan is not used and a majority of the terminals go to bypass, the return plenum may become positive in relation to the space, forcing return air back into the space.

A **series fan-powered VAV terminal unit** has an integral fan that supplies a constant volume of air to the space (Figure 37). In addition to enhancing air distribution in the space, a reheat coil can be added to maintain a minimum temperature in the space when the

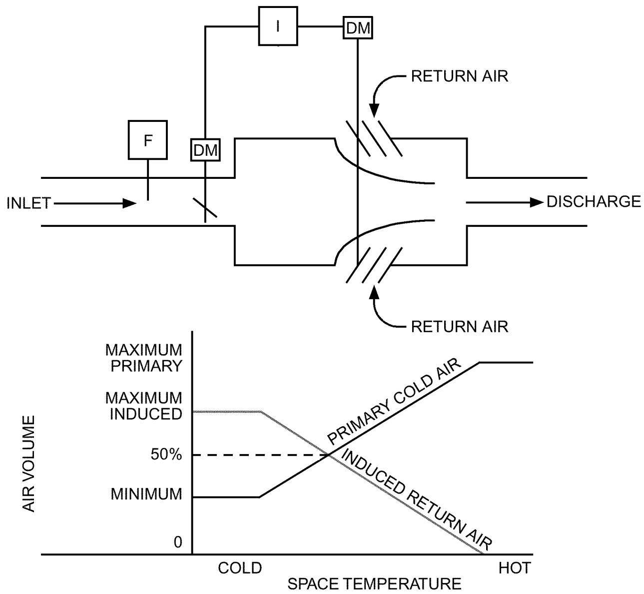

Fig. 35 Induction VAV Terminal Unit

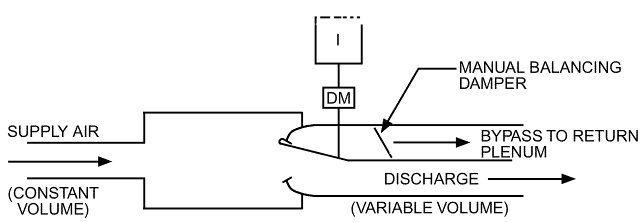

Fig. 36 Bypass VAV Terminal Unit

primary system is off. When the space is occupied, the fan runs constantly to provide a constant volume of 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 set point, the supply air damper begins to close and the fan draws more air from the return plenum. Units serving the perimeter area of a building can include a reheat coil. Then, when supply air reaches its minimum volume, the valve to the reheat coil begins to open.

A **plenum** or **parallel fan terminal** has a fan that pulls air from the return plenum and mixes it with the supply air (Figure 38). A reheat coil may be placed in the discharge to the space or in the return plenum opening. The fan provides a minimum airflow to the space. Total airflow to the space is the sum of the fan output and supply air quantity. When space temperature drops below the 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, the reheat coil valve starts to open. When the space is unoccupied and requires heating, the supply air damper is closed, the fan turns on, and the reheat coil valve modulates to maintain the unoccupied set point.

Dual-Duct Constant-Volume. A **mixing box terminal** generally is used where large amounts of ventilation at a low static pressure are required. The hot- and cold-duct dampers are linked to operate in reverse directions. A space thermostat positions the mixing dampers through a damper actuator to mix warm and cool supply air. Discharge air volume depends on the static pressure in each supply duct at that location. Static pressures in the supply ducts vary because of the varying airflow in each duct (Figure 39).

Dual-duct constant-volume mixing box terminals are typically used in high-static-pressure applications where airflow quantity to each space is critical. The units are the same as those described in the previous paragraph, except that they include either an integral mechanical constant-volume regulator or an airflow constant-volume control furnished by the unit manufacturer (Figure 40).

Dual-Duct Variable-Volume. Mixing box terminals have inlet dampers on the heating and cooling supply ducts. These dampers are linked to operate in reverse directions, and require a single control actuator. Also, a sensor in the discharge monitors total airflow. The space thermostat controls inlet mixing dampers directly, and the airflow controller controls the volume damper. The space thermostat resets the airflow controller from maximum to minimum flow as the thermal load on the conditioned area changes. Figure 41 shows the control schematic and damper operation. Note that, in part of the control range, heating and cooling supply air mix.

Variable, constant-volume (zero-energy-band) dual-duct terminal units (Figure 42) have inlet dampers (with individual damper actuators and airflow controllers) on the cooling and heating supply ducts and no total airflow volume damper. The zero-energy-band (ZEB) space thermostat resets the airflow controller set points in sequence as the space load changes. The airflow controllers maintain adjustable minimum flows for ventilation (with no overlap of damper operations) in the ZEB when neither heating nor cooling is required.

Heat Pump. A heat pump is a refrigeration device that uses the evaporator to cool a warm space, as is typical of a window air conditioner, but uses heat that is normally rejected from the condenser

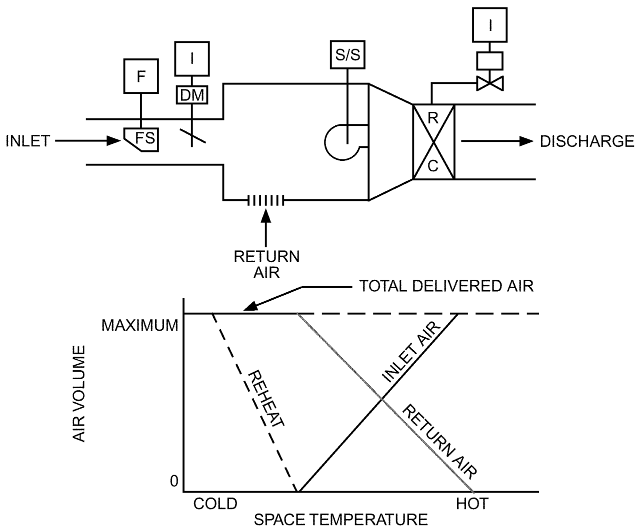

Fig. 37 Series Fan-Powered VAV Terminal Unit

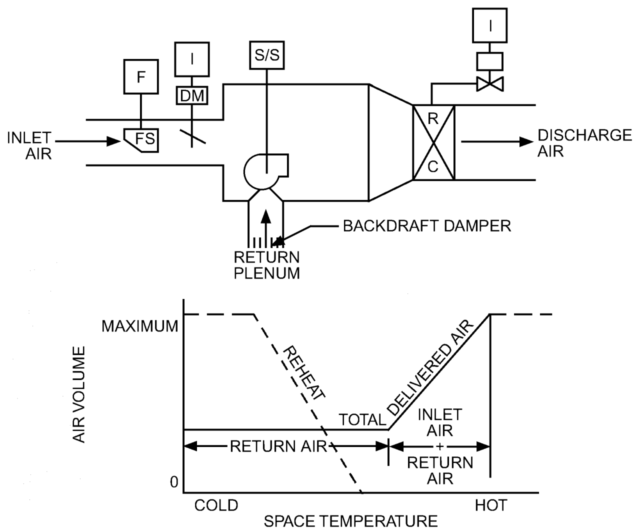

Fig. 38 Plenum or Parallel Fan Terminal Unit

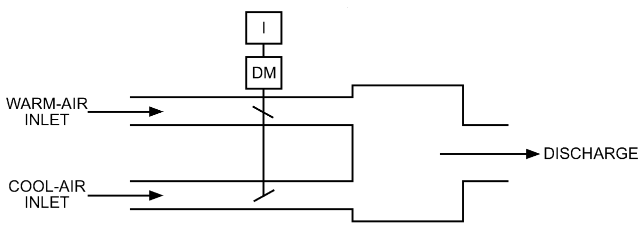

Fig. 39 Dual-Duct Mixing Box Terminal Unit

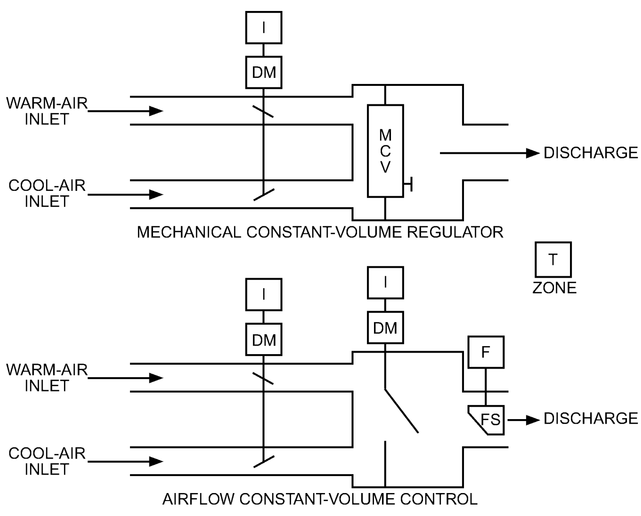

Fig. 40 Dual-Duct Pressure-Independent Constant-Volume Mixing Box Terminal Units

for heating. The air outside is cooled, and heat taken from the outside is "pumped" into the space. Many conventional means to control refrigeration equipment can be used to control heat pump heating and cooling cycles; see Chapters 8 and 45 of the 2004 *ASHRAE Handbook—HVAC Systems and Equipment* for details.

Air Handling

Generally, air is distributed by either single-duct or dual-duct systems. Airflow to each zone is controlled by a terminal box. A special case of a dual-duct system is a multizone unit with terminal boxes incorporated into the air handler. In this case, the hot and cold ducts are referred to as the **hot** and **cold decks**. One special multizone configuration has three decks, which may be used as an alternative (Figure 43). Zone dampers in this unit operate with sequenced dampers to either mix hot supply air with bypass air when the cold-deck damper is closed or mix cold supply air with bypass air when the hot-deck damper is closed.

A single-zone system (Figure 44) uses a constant-volume air handler (usually factory-packaged). No fan speed control is required, because fan volume and duct static pressure are set by design and component selection. Single-zone systems do not require terminal boxes, because zone temperature can be maintained by modulating the heating and cooling control valves in sequence.

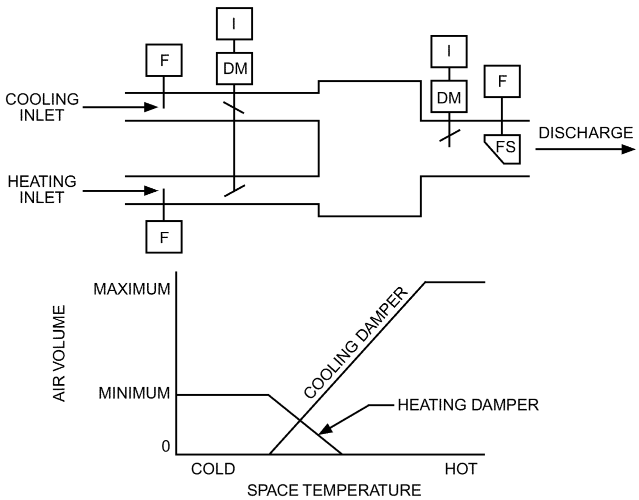

Fig. 41 Pressure-Independent Dual-Duct VAV Terminal Unit

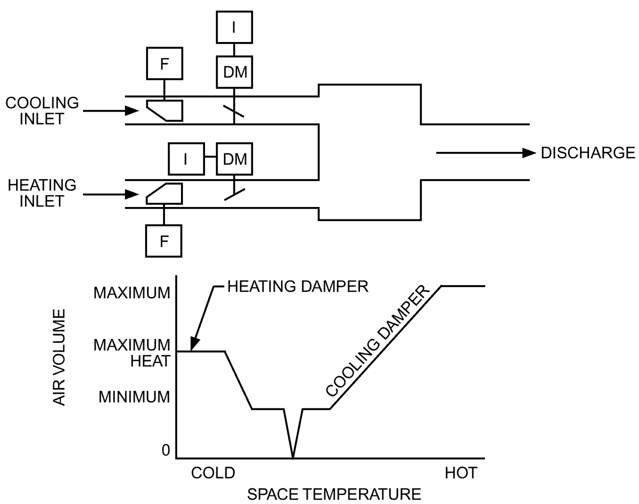

Fig. 42 Variable, Constant-Volume (ZEB) Dual-Duct Terminal Unit

During warm-up, as determined by a time clock or manual switch, a constant heating supply air temperature is maintained. During warm-up and unoccupied cycles, outside air dampers should be closed.

A **unit ventilator** is designed to heat, ventilate, and cool a space by introducing up to 100% outside air. Optionally, it can cool and dehumidify with a cooling coil (either chilled-water or direct-expansion). Heating can be by hot water, steam, or electric resistance. Control of these coils can be by valves or by face-and-bypass dampers. Consequently, controls applied to unit ventilators are many and varied. The four most commonly used control schemes are Cycle I, Cycle II, Cycle III, and Cycle W.

Cycle I Control. Except during the warm-up stage, Cycle I (Figure 45) supplies 100% outside air at all times. During warm-up, the heating valve is open, the outside air (OA) damper is closed, and the return air (RA) damper is open. As temperature rises into the operating range of the space thermostat, the OA damper opens fully, and the RA damper closes. The heating valve is positioned to maintain space temperature. The airstream thermostat can override space thermostat action on the heating valve to prevent discharge air from dropping below a minimum temperature. Figure 45 shows positions of the heating valve and ventilation dampers in relation to space temperature.

Cycle II Control. During the heating stage, Cycle II (Figure 45) supplies a set minimum quantity of outside air. Outside air is gradually increased as required for cooling. During warm-up, the heating valve is open, the OA damper is closed, and the 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 45 shows the relative positions of the heating valve and ventilation dampers with respect to space temperature.

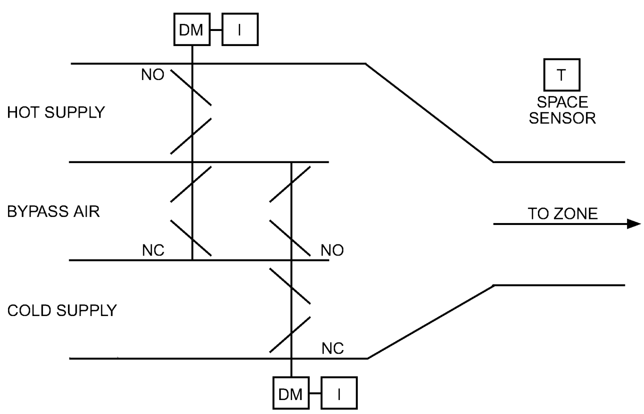

Fig. 43 Zone Mixing Dampers: Three-Deck Multizone System

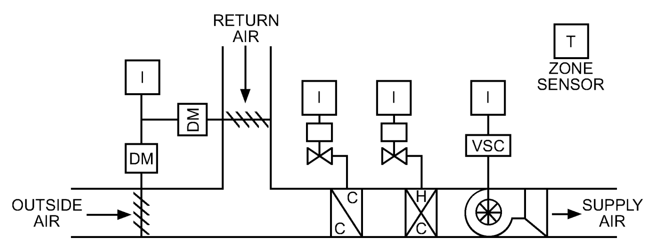

Fig. 44 Single-Zone Fan System

Cycle III Control. During heating, ventilating, and cooling stages, Cycle III (Figure 46) supplies a variable amount of outside air as required to maintain the air entering the heating coil at a fixed temperature (typically 55°F). When heat is not required, this air is used for cooling. During warm-up, the heating valve is open, the OA air damper is closed, and the RA damper is open. As the space temperature rises into the operating range of the space thermostat, ventilation dampers control the air entering the heating coil at the set temperature. Space temperature is controlled by positioning the heating valve as required. Figure 46 shows the relative positions of the heating valve and ventilation dampers with respect to space temperature.

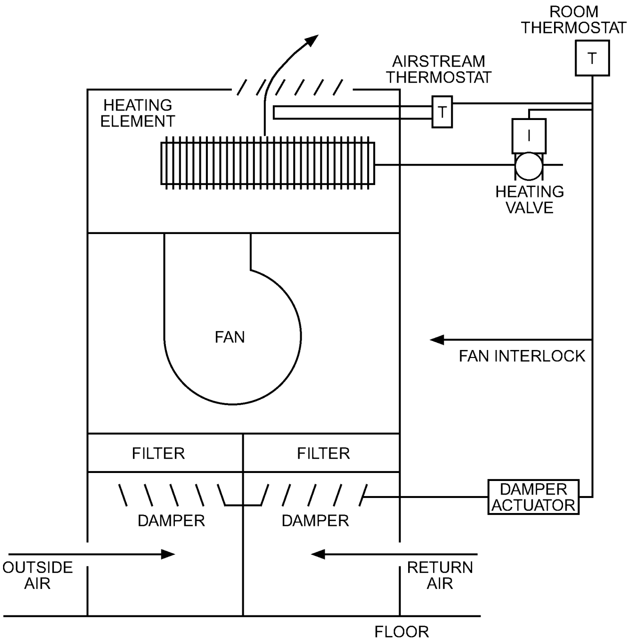

Fig. 45 Cycles I, II, and W Control Arrangements

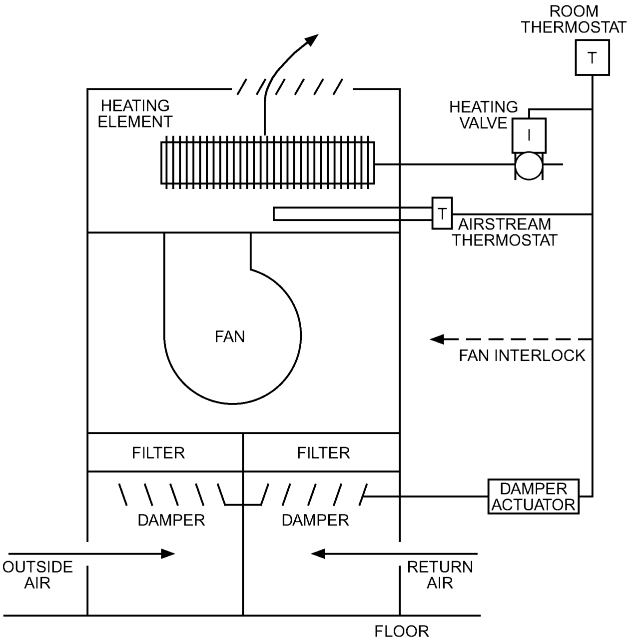

Fig. 46 Cycle III Control Arrangement

Day/night thermostats are frequently used with any of these control schemes to maintain a lower space temperature during unoccupied periods by cycling the fan with the outside air damper closed. Another common option is a low-temperature limit control placed next to the heating coil, to turn off the unit when near-freezing temperatures are sensed.

Cycle W Control. Cycle W is similar to Cycle II, except that the heating valve is controlled by the room thermostat and the dampers are controlled by the low-limit thermostat (Figures 45 and 47).

Makeup air units (Figure 48) replace air exhausted from the building through exfiltration or by laboratory or industrial processes. Makeup air must be at or near space conditions to minimize uncomfortable air currents. 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 outside air damper is fully open as proven by an end switch. The two-position outside air damper remains closed when the makeup fan is not in operation. The outside air limit control opens the preheat coil valve when outside air temperature drops to the point where the air requires heating to raise it to the desired supply air temperature. A capillary element thermostat 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.

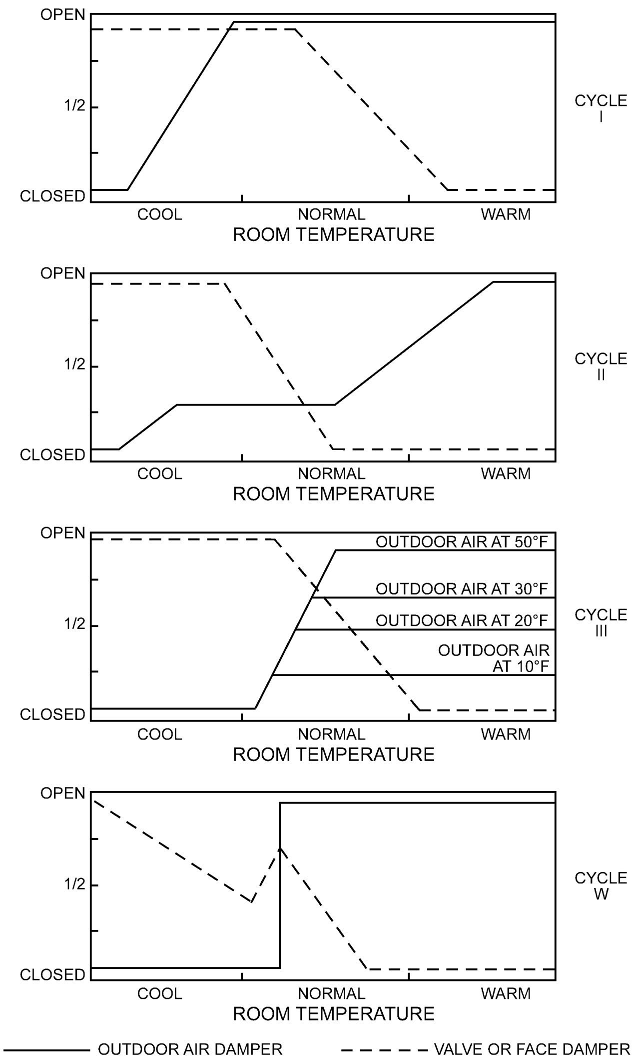

Fig. 47 Valve and Damper Positions with Respect to Room Temperature

Multizone, Single-Duct System

This system (Figure 49) is most effective where all zones are in either heating or cooling mode at the same time. When necessary and where energy regulations allow it, reheat can be used to accommodate systems with simultaneous heating and cooling loads.

Multizone, Dual-Duct Systems

A **single supply fan system** uses a single fan to supply separate heating and cooling ducts (Figure 50). Terminal mixing boxes are used to control the zone temperature. 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.

Control of a return air fan is similar to that described in the section on Fans, in the paragraph on Return Fan Static Control. Flow stations are usually located in each supply duct, and a signal corresponding to the sum of the two airflows is transmitted to the RA fan volume controller to establish the set point of the return fan controller.

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 controller set point may be reset from the greatest representative demand zone: based on zone temperature, the hot deck may be reset from the zone with the greatest heating demand, and the cold deck from the zone with the greatest cooling demand.

Control based on the zone requiring the most heating or cooling increases operation economy because it reduces the energy delivered at less-than-maximum load conditions. However, the expected economy is lost if air quantity to a zone is too low, a zone's temperature set point is set to an extreme value, a zone sensor is placed so that it senses spot loads (from coffeepots, the sun, copiers, etc.), a sensor is located in an unoccupied zone, or a zone sensor malfunctions. In these cases, a weighted average of zone signals can recover the benefit at the expense of some comfort in specific zones.

Ventilation dampers (outside, return, and exhaust air) are controlled for cooling, with outside air as the first stage of cooling in sequence with the cooling coil from the cold-deck discharge temperature controller. Control is similar to that in single-duct systems.

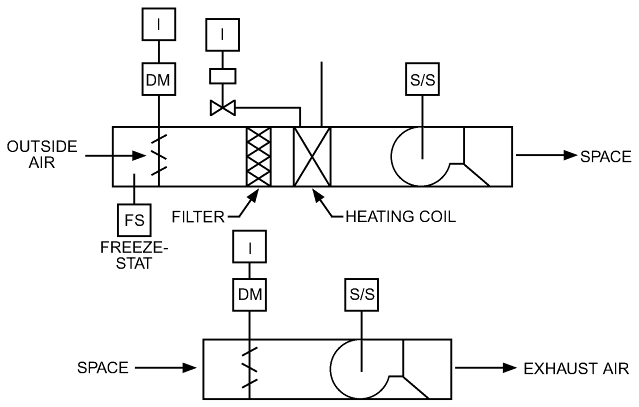

Fig. 48 Makeup Air Unit

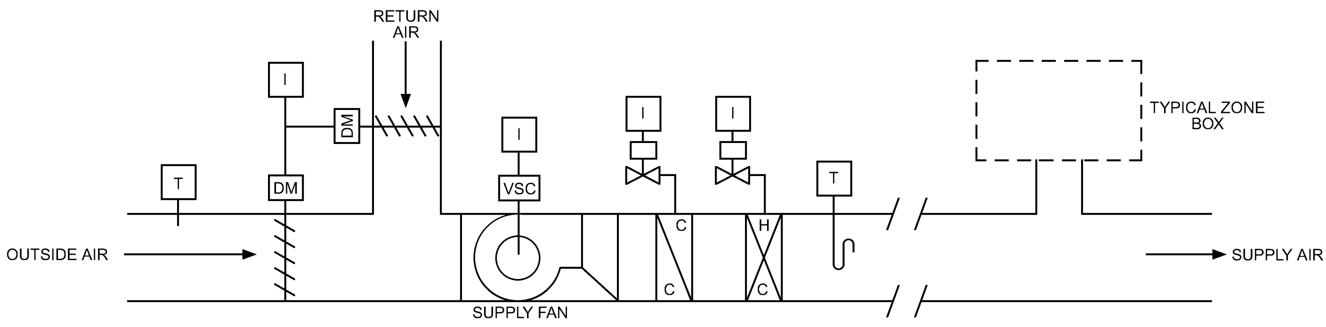

Fig. 49 Multizone Single-Duct System

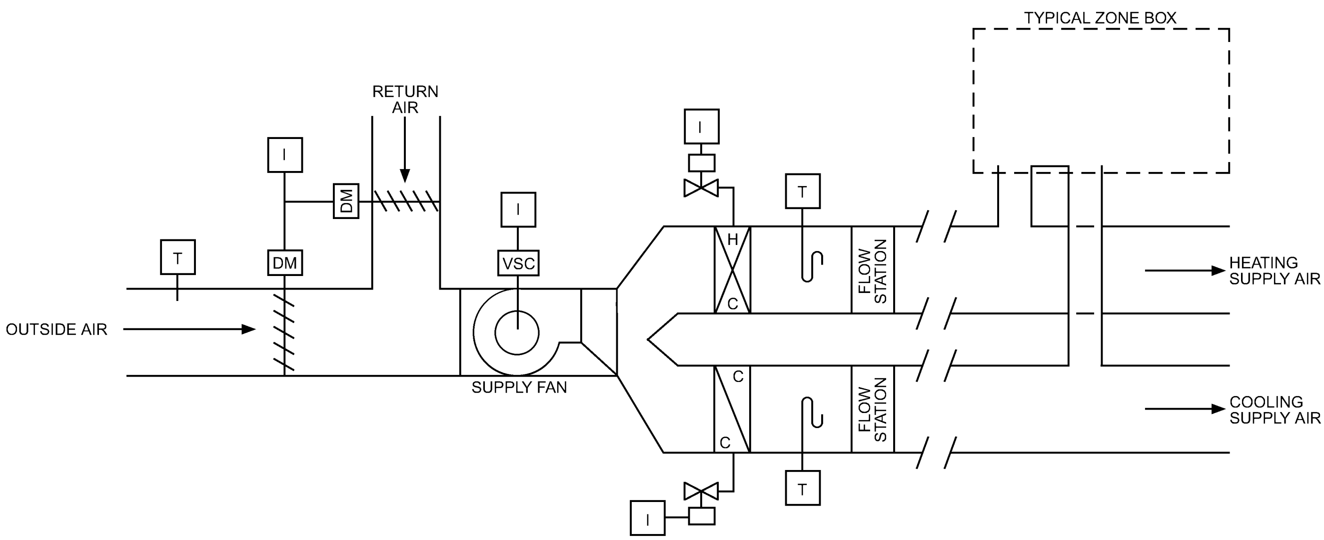

Fig. 50 Dual-Duct Single-Supply Fan System

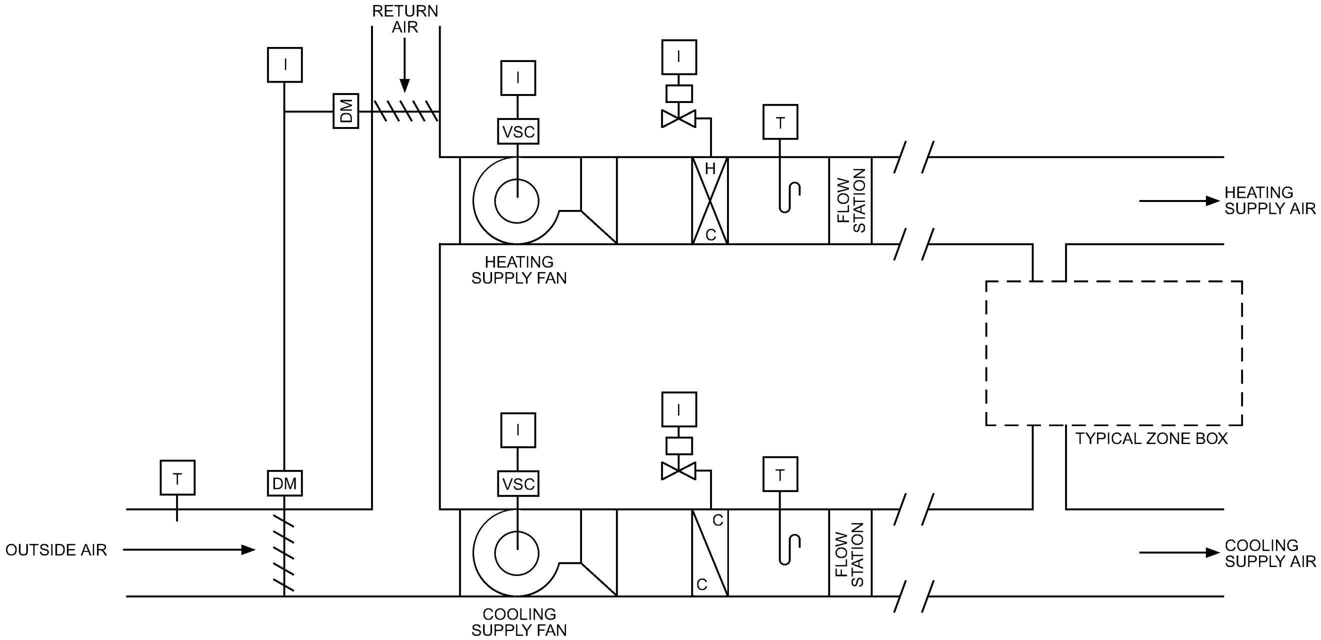

Fig. 51 Dual-Supply Fan System

A more accurate OA flow-measuring system can replace the minimum positioning switch.

Dual-supply fan systems (Figure 51) use separate supply fans for the heating and cooling ducts. Static-pressure control is similar to that for VAV dual-duct single-supply fan systems, except that each supply fan has its own static pressure sensor and control. If the system has a return air fan, volume control is similar to that described in the section on Fans, in the paragraph on Return Fan Static Control. Temperature, ventilation, and humidity control are similar to those for VAV dual-duct single-supply fan systems.

WATER HEATING

A basic constant-volume hydronic system is shown in Figure 52. In a primary-secondary system, a variable-speed drive could be added to the secondary pump motor and the three-way valves could be changed to two-way valves to save energy.

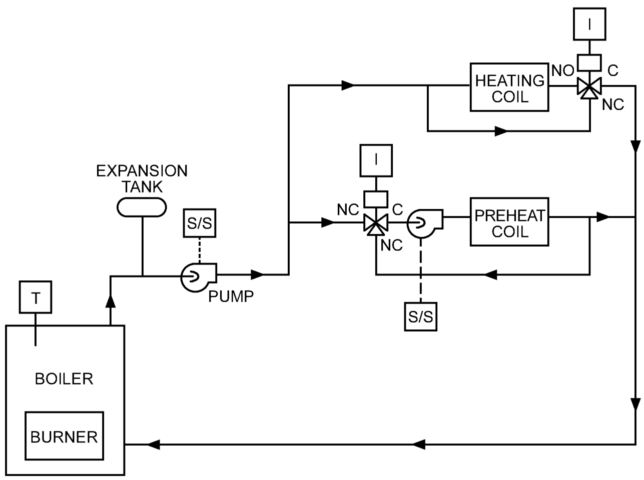

Fig. 52 Load and Zone Control in Constant-Volume System

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 10,000 ft.

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 a 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.

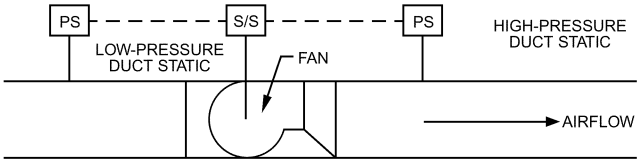

Fig. 53 High-Limit Static Pressure Controller

Duct Static Pressure Limit Control

In addition to the remote static-pressure controllers, a high- and low-limit static pressure controller should be placed at the fan discharge and inlet (Figure 53) to turn the fan off or limit discharge static pressure in the event of excessive duct pressure (e.g., when a fire or smoke damper closes between the fan and the remote sensor). Supply fan static pressure control devices such as inlet guide vanes and variable-speed drives should be interlocked to move to the minimum-flow or closed position when the fan is not running; this precaution prevents fan overload or damage to ductwork on start-up.

When selecting automatic controls, the type needed to control high or low limits (safety) must be considered. This control may be inoperative for days, months, or years, but then must operate immediately to prevent serious damage to equipment or property. Separate operating and limit controls are always recommended, even for the same functions.

Duct static limit control prevents excessive duct pressures, usually at the discharge of the supply fan. Two variations are used: (1) the fan shutdown type, which is a safety high-limit control switch that turns fans off; and (2) the controlling high-limit speed control duct pressure reset type (Figure 53), which is used in systems that have zone fire dampers. When the zone fire damper closes, and the duct static pressure sensor is downstream of the zone, duct pressure falls from lack of air, causing the duct static control to increase fan speed; however, the controlling high limit overrides and stop the fan.

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 a central control point to diagnose malfunctions that might result in damage from delay, and to reduce 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 outside air temperature.

Distributed processing using microprocessors has augmented computer use at many locations other than the central control point. The local loop controller can be a DDC instead of a pneumatic or electric thermostat, and some energy management functions may be performed by a DDC.

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.

Extraordinary Incidents

Building owners and design engineers are sometimes interested in applying the building automation system (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).

Many references are available to guide an engineer or building owner in organizing a comprehensive plan, including ASHRAE (2003b), FEMA (2003), and NIOSH (2002). Also see Chapter 58 for more information on this topic.

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, and (4) a description of each control function.

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.

Building and System Subdivision

The following factors must be considered in the building and mechanical system subdivision:

- Varying heating and cooling loads: ability to heat or cool any area of a building at any time
- Occupancy schedules and flexibility to meet needs without undue initial and/or operating costs
- Fire and smoke control and possibly compartmentation that matches the air-handling layout and operation

Energy-Efficient Controls

The U.S. Green Building Council's (USGBC) Leadership in Energy Efficient Design (LEED) program for new construction and existing buildings provides guidance on designing and building commercial, institutional, and government buildings to produce quantifiable benefits for occupants, owners, and the environment.

During design and construction, LEED provides many opportunities for using a building automation system (BAS). LEED awards points, based on design criteria, to determine at which level a building is certified: platinum, gold, silver, or certified. (For specifics, see http://www.usgbc.org.) Many of these points depend on the sophistication, flexibility, and power of the BAS (which can provide up to 40% of the required points for a certification level), not only for comfort and energy efficiency, but also for sustainability verification for the facility's lifetime.

Although many publications address green and LEED design for buildings and equipment selection, **green controls** is a relatively new term and is still evolving. For the most up-to-date information, consult the latest publications in the field.

Applications. It is recommended that the LEED design team choose the controls consultant/contractor and to have them on board 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). Under LEED, 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 outside air (OA) without overventilating
- 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 LEED. 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 in **measurement and verification (M&V)**. Typical operation sequences, such as optimizing outside air, chiller efficiency, and multiple sensing points for providing thermal comfort, are, however, still used, and are critical for maintaining sustainability.

Perhaps the most important aspect of green controls 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 maintains its design energy efficiency.

CONTROL PRINCIPLES FOR ENERGY CONSERVATION

Temperature and Ventilation Control. VAV systems are typically designed to supply constant-temperature air at all times. To conserve central plant energy, the temperature of supply air can be raised in response to demand from the zone with the greatest load (load analyzer control). However, because more cool air must then be supplied to match a given load, the mechanical cooling energy saved may be offset by an increase in fan energy. Equipment operating efficiency should be studied closely before implementing temperature reset in cooling-only VAV systems.

Outside, return, and exhaust air ventilation dampers are controlled by the discharge air temperature controller to provide free cooling as the first stage in the cooling sequence. When outside air temperature rises to the point that it can no longer be used for cooling, an outside air limit (economizer) control overrides the discharge controller and moves ventilation dampers to the minimum ventilation position. An enthalpy control system can replace outside air limit control in some climates.

After the general needs of a building 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. The 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 between 55 and 60°F to prevent freezing. Start morning warm-up as late as possible to achieve design internal temperature by occupancy time, considering residual space temperature, outside temperature, and equipment capacity (optimum start control). Under most conditions, 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.
- *Sequence heating and cooling*. Do not supply heating and cooling simultaneously unless it is requested for humidity control. Central fan systems should use cool outside air 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).
- *Supply heating and cooling from the most efficient source*. Use free or low-cost energy sources first, then higher-cost sources as necessary.
- *Apply outside air control*. When on minimum outside air, use no less than that recommended by ASHRAE *Standard* 62.1. In areas where it is cost-effective, use enthalpy rather than dry-bulb temperature to determine whether outside or return air is the most energy-efficient air source for the cooling mode.

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) have the ability to 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 outside 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.

Many U.S. jurisdictions no longer permit constant-volume systems that reheat cold air or that mix heated and cooled air. Such systems should be avoided. If selected, 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. Avoid reheat. 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 5000 ft^2 because 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 an 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 outside wall locations. Mount thermostats at generally accessible heights according to the Americans with Disabilities Act (ADA) (USDOJ 1994) (usually 48 in.) 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 injury. 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% outside air for natural cooling, however, a forced-flow sampling chamber should be provided 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** 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.

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 process must have a plan that defines the commissioning process and is developed in increasing detail as the project progresses. Phases include plan, design, construction, and acceptance. The most useful tool used to challenge (simulate changes to) systems operation is the control system itself. A DDC system provides the added convenience of central execution of test steps, and the ability to record responses. Commissioning the DDC system must occur before it can be used to validate the HVAC systems. Commissioning DDC systems is discussed in ASHRAE GPC 11. Commissioning HVAC systems is recommended for construction of new buildings, and should be repeated periodically in existing buildings. See Chapter 42 and ASHRAE *Guidelines* 0 and 1 for more information.

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