CHAPTER 46

DESIGN AND APPLICATION OF CONTROLS

AUTOMATIC control of HVAC systems and equipment usually includes control of temperature, humidity, pressure, and flow rate. Automatic control sequences equipment operation to meet load requirements and provides safe operation of the equipment, using pneumatic, mechanical, electrical, electronic, and direct digital control devices.

This chapter covers (1) control of HVAC elements, (2) control of typical systems, (3) limit control for safe operation, and (4) design of controls for specific HVAC applications. Chapter 15 of the 2001 *ASHRAE Handbook—Fundamentals* covers the basics of control, types of control components, and commissioning of control systems.

CONTROL OF HVAC ELEMENTS

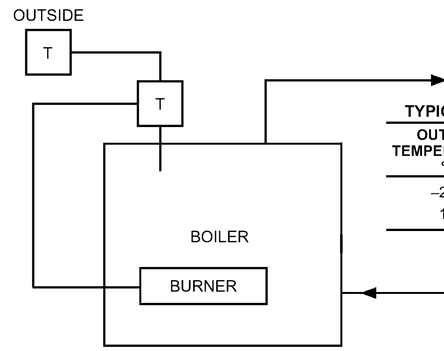
Boiler

Load affects the rate of heat input to a hydronic system. Rate control is accomplished by cycling and modulating the flame and by turning boilers on and off. Flame cycling and modulation are handled by the boiler control package. The control designer decides under what circumstances to add or drop a boiler and at what temperature to control the boiler supply water.

Hot-water distribution control includes temperature control at the hot-water boilers or converter, reset of heating water temperature, and control for multiple zones. Other factors that need to be considered include (1) minimum water flow through the boilers, (2) protection of boilers from temperature shock, and (3) coil freeze protection. If multiple or alternative heating sources (such as condenser heat recovery or solar storage) are used, the control strategy must also include a means of sequencing hot-water sources or selecting 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. Intermittent burner firing usually controls capacity, although fuel input modulation is common in larger systems. In most cases, the boiler is controlled to maintain a constant water temperature, although an outside air thermostat can reset the temperature if the boiler is not used for domestic water heating. Figure 1 contains a typical reset schedule. To minimize condensation of flue gases and boiler damage, water temperature should not be reset below that recommended by the manufacturer, typically 60°C. Larger systems with sufficiently high pump operating costs can use variable-speed pump drives, pump discharge valves with minimum-flow bypass valves, or two-speed drives to reduce secondary pumping capacity to match the load.

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 ranging from 93 to 127°C. The supply water thermostat controls a modulating



TYPICAL RESET SCHEDULE	
OUTSIDE TEMPERATURE, °C	HOT-WATER SERVICE, °C
–20	80
15	60

Fig. 1 Boiler Control

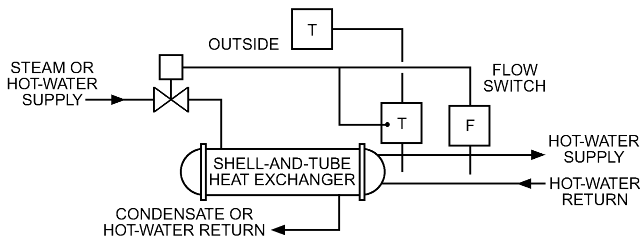


Fig. 2 Steam-to-Water Heat Exchanger Control

two-way valve in the steam (or hot-water) supply line. An outside thermostat usually resets the supply water temperature downward as the load decreases to improve the controllability of heating valves at low load and to reduce piping losses. A flow switch interlock should close the two-way valve when the hot-water pump is not operating. With integrated computer-based control, feedback from zone heating valves can be used to control the starting and stopping of the hot-water pumps. On constant-flow systems, the feedback can be used to reset the hot-water temperature to the lowest temperature that meets zone requirements.

Fan

The most efficient way to change the output of a fan is to change its speed. Because of their simplicity and high efficiency, variable-frequency drives are widely used. Though less efficient, eddy current drives are also an option for electronically controlling fan speed. Other ways of controlling fan ouput include using inlet cones, inlet guide vanes, and discharge or scroll dampers. Axial fans can be controlled by varying the pitch of the blade. Also, dampers and ducting can simply bypass some of the air from the supply side of the fan to the return side (Figure 3). Bypassing does not change the output of the fan, but it can allow the fan to accommodate flow variations in the distribution system without fan instability. The final selection of a control device is determined by efficiency requirements and available funding.

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

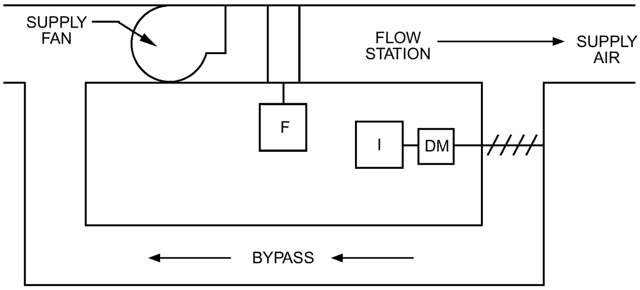


Fig. 3 Fan Bypass Control to Prevent Supply Fan Instability

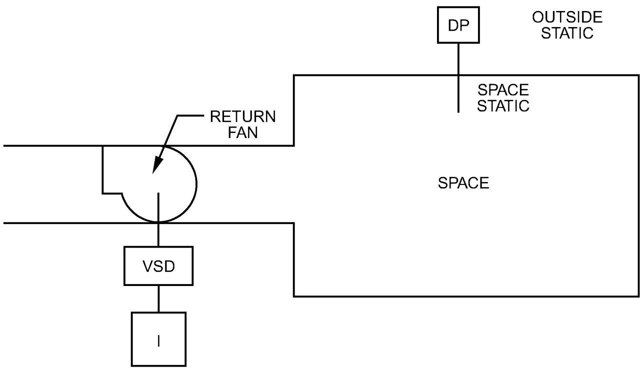


Fig. 4 Direct Building Pressurization Control

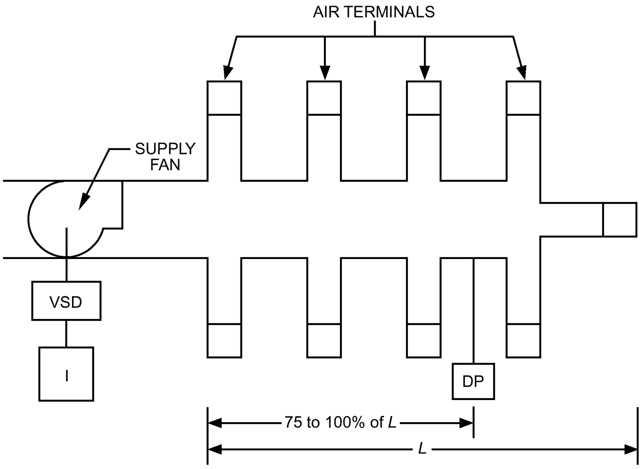


Fig. 5 Duct Static-Pressure Control

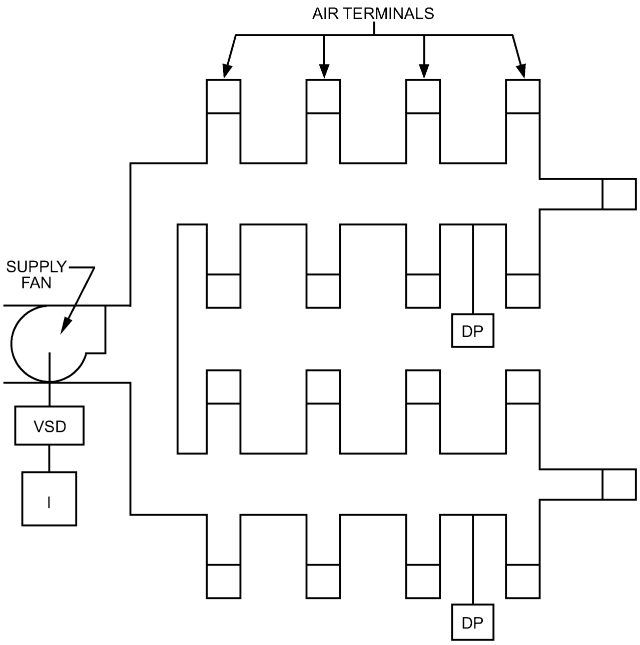


Fig. 6 Multiple Static Sensors

Static-pressure control is required in systems having variable flow rates. To conserve fan energy, the static-pressure controller should be set at the lowest control point permitting proper air distribution at design conditions. The controller requires proportional-plus-integral (PI) control because it eliminates offset while maintaining stability. In proportional-only control, the low proportional gain required to stabilize fan control loops allows static pressure to offset upward as the load decreases, which causes the supply fan to consume more energy.

Differential static-pressure control is used to **pressurize a building** or space relative to adjacent spaces or the outside. Typical applications include clean rooms (positive pressure to prevent infiltration), laboratories (positive or negative, depending on use), and various manufacturing processes, such as spray-painting rooms. The pressure controller usually modulates dampers in the supply duct to maintain the desired pressure as exhaust volumes change. A method for control of the return fan requires measuring the space and outside static pressures (Figure 4). The location for measuring inside static pressure must be selected carefully: away from doors and openings to the outside, away from elevator lobbies, and, when using a sensor, in a large representative area shielded from drafts. The outside location must likewise be selected carefully, typically 3 to 4.5 m above the building and oriented to minimize wind effects from all directions. The amount of minimum outside air varies with building permeability and exhaust fan operation. Control of building pressurization can affect the amount of outside air entering the building.

Duct static-pressure control for variable air volume (VAV) and other terminal systems maintains a static pressure at a measurement point. The most common application for static-pressure control is fan output control in VAV systems. The pressure sensor must be properly placed to maintain optimum pressure throughout the supply duct. 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. If the sensor is located at less than 100% of the distance, the control set point should be adjusted higher to account for the pressure loss between the sensor and the remote terminal (Figure 5). Care must be taken in selecting the reference sensor location. Controller upset from opening and closing doors, elevator shafts, and other sources of air turbulence should also be prevented. The pressure selected provides a minimum static pressure to all air terminal units during all supply fan design conditions.

Multiple static sensors (Figure 6) are required when more than one branch duct runs from the supply fan. The sensor with the highest static requirement controls the fan. Because duct run-outs may vary, a control that uses individual set points for each measurement is preferred.

VAV systems typically incorporate a duct static-pressure control loop to control the supply fan speed output. In a single-duct VAV system, the duct static pressure set point is usually selected by the designer. The sensor should be located in the ductwork where the established set point ensures proper operation of the zone VAV boxes under varying load (supply airflow) conditions. A shortcoming of this approach is that static-pressure control is based on the readings of a single sensor that is assumed to represent the pressure available to all VAV boxes. If the sensor malfunctions or is placed in a location that is not representative, operating problems will result.

An alternative approach to supply fan control in a VAV system uses flow readings from the direct digital control (DDC) zone terminal boxes to integrate zone VAV requirements with supply fan operation. Englander and Norford (1992) suggest that duct static pressure and fan energy can be reduced without sacrificing occupant comfort or adequate ventilation. They compared modified PI and heuristic control algorithms using simulation and demonstrated that either static pressure or fan speed can be regulated directly using a flow error signal from one or more zones. They noted that component modeling limitations constrained their results primarily to a comparison of the control algorithms. The results show that both PI and heuristic control schemes work, but the authors suggest that a hybrid of the two might be ideal.

Supply fan warm-up control for systems having a return fan must prevent the supply fan from delivering more airflow than the return fan maximum capacity during warm-up mode (Figure 7).

Return fan static control from returns having local (zoned) flow control is identical to supply fan static control (Figure 5). Return fan control for VAV systems is required for proper building pressurization and minimum outside air. The return fan is controlled to maintain exhaust and return air plenum pressure. The exhaust air damper is controlled to maintain building static pressure (Figure 8). This ensures that the supply fan does not pull in outside air backward through the exhaust air dampers (Seem et al. 2000).

Airflow tracking uses duct airflow measurements to control the return air fans (Figure 9). Typical sensors, called flow stations, are multiple-point, pitot tube, and averaging. Provisions must be made for exhaust fan switching to maintain pressurization of the building. Warm-up is accomplished by setting the return airflow equal to the supply fan airflow, usually with exhaust fans turned off and limiting supply fan volume to return fan capability. During night cooldown, the return fan operates in the normal mode.

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 to help ensure adequate air distribution and acceptable zone pressurization. In a return fan VAV system, there is significant potential for control system instability because of the interaction of control variables (Avery 1992). In a typical system, these variables might 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. The interaction of these variables and the 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. Kettler (1995) suggests that small errors in sensing total airflow and return flow can cause significant errors in control of the differential flow, making this approach unsatisfactory for minimum outside ventilation control.

Sequencing fans for VAV systems reduces airflow more than other methods and results in greater operating economy and more stable fan operation if airflow reductions are significant. Alternating fans usually provides greater reliability. Centrifugal fans are controlled to keep system disturbances to a minimum when additional fans are started. The added fan is started and slowly brought to capacity while the capacity of the operating fans is simultaneously reduced. The combined output of all fans then equals the output before fan addition.

Vaneaxial fans usually cannot be sequenced in the same manner as centrifugal fans. To avoid stall, the operating fans must be reduced to some minimum level of airflow. Then, additional fans may be started and all fans modulated in parallel to achieve equilibrium.

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.

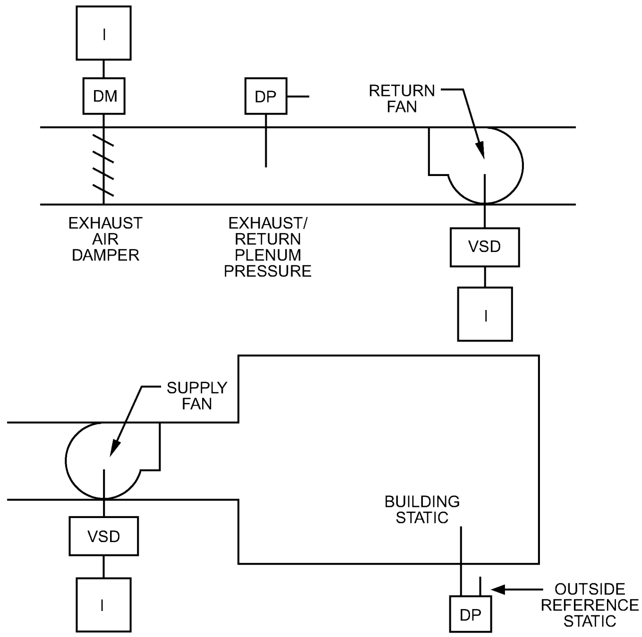


Fig. 8 Duct Static Control of Return Fan

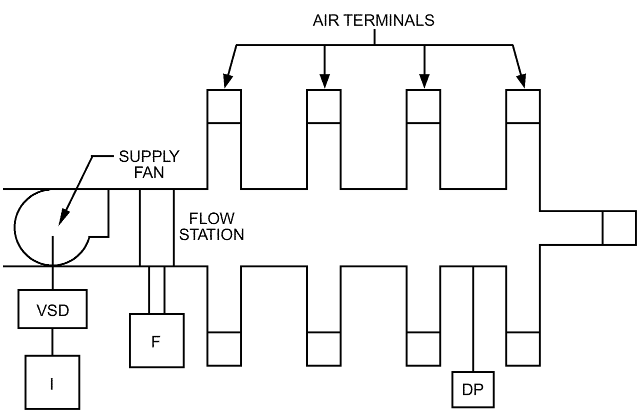


Fig. 7 Supply Fan Warm-Up Control

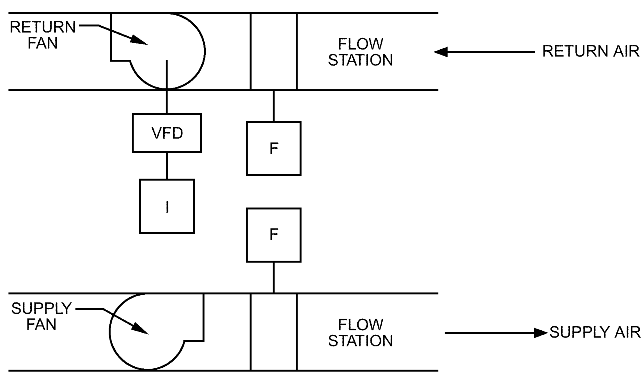


Fig. 9 Airflow Tracking Control

Supply air temperature reset can be used to avoid fan instability by resetting the cooling-coil discharge temperature higher (Figure 10), so that the 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 cooling load and dehumidification requirements can be met.

Cooling Coil

Chilled-water or **brine cooling coils** are controlled by two- or three-way valves (Figure 11). 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 in response to coil air discharge temperature or space temperature.

Direct-expansion (DX) cooling coils are usually controlled by solenoid valves in the refrigerant liquid line (Figure 12). Face and bypass dampers are not recommended because they permit ice to form on the coil when airflow is reduced. Control can be improved by using two or more stages; the solenoid valves are controlled in sequence, and there is a differential of about 1 K between stages (Figure 13). The first stage should be the first coil row on the entering air side; the following rows form the second and succeeding stages. Side-by-side stages tend to generate icing on the stage in use, which causes reduction of airflow and loss of control. Modulating control can be achieved using a variable-suction-pressure controller (Figure 14). This type of control is uncommon but necessary if accurate control of discharge or space temperature is required.

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, depending on the manufacturer's recommendations. On larger towers, two-speed motors or variable-speed drives can reduce fan power consumption at part-load conditions and stabilize condenser water temperature (Figure 15).

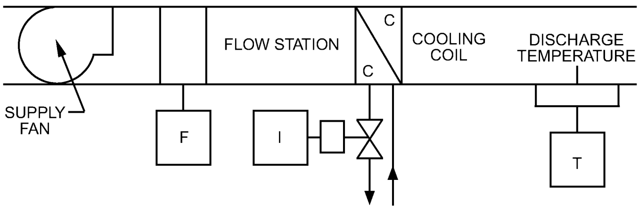


Fig. 10 Coil Reset Control to Prevent Supply Fan Instability

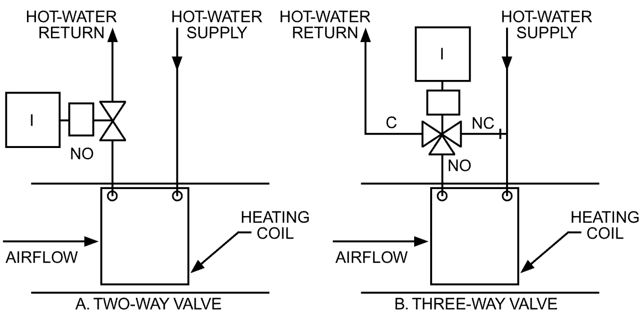


Fig. 11 Control of Hot- and Chilled-Water Coils

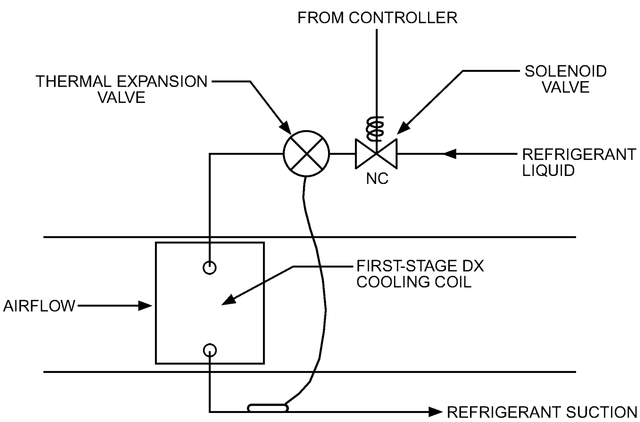


Fig. 12 Direct Expansion—Two-Position Control

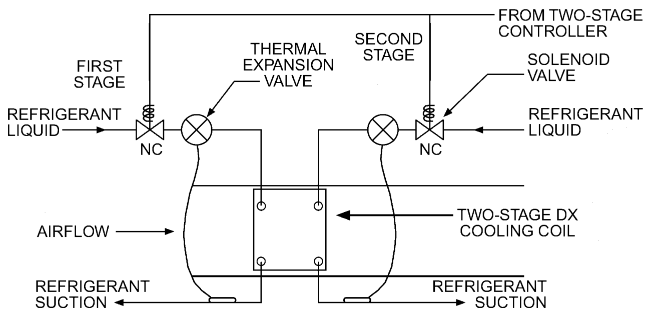


Fig. 13 Two-Stage Direct-Expansion Cooling

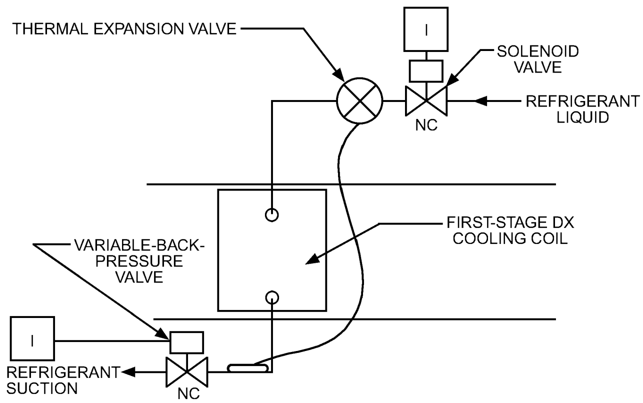


Fig. 14 Modulating Direct-Expansion Cooling

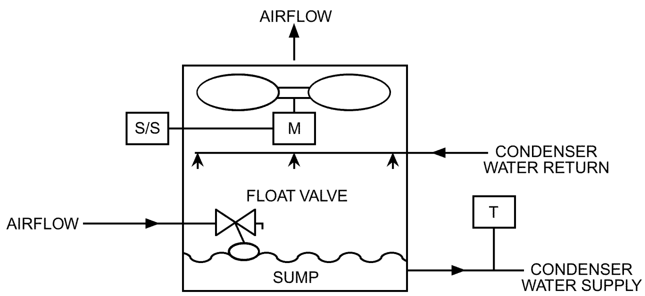


Fig. 15 Cooling Tower

In colder areas that require year-round air conditioning because of high internal loads, cooling towers may require sump heating and/or continuous full flow over the tower to prevent ice formation. In that case, the cooling tower sump thermostat controls a hot-water or steam valve to keep water temperature above freezing.

Economizer Cycle

Economizer cycle control reduces cooling costs when outside conditions are suitable, that is, when outside air is cool enough to be used as a cooling medium. If outside air is below a high-temperature limit, typically 18°C, the return, exhaust, and outside air dampers modulate to maintain a ventilation cooling set point, typically 13 to 16°C (Figure 16). The outside and exhaust dampers close and the return air dampers open when the supply fan is not operating. When the outside air temperature exceeds the high-temperature limit set point, the outside air damper is closed to a fixed minimum and the exhaust and return air dampers close and open, respectively.

In enthalpy economizer control, the high-temperature limit interlock system of the economizer cycle is replaced to further reduce energy costs when latent loads are significant. The interlock function (Figure 16) 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 (Figure 17). The supply fan and return fan airflow offset to maintain positive or negative duct pressurization.

Night cooldown control (night purge) provides 100% outside air for cooling during unoccupied periods (Figure 18). The space is cooled to the space set point, typically 5 K 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 10°C or below. The night cooldown cycle is initiated before sunrise, when overnight outside temperatures are usually the coolest. When outside air conditions are acceptable and the space requires cooling, the cooldown cycle is the first phase of the optimum start sequence.

Heating Coil

Heating coils that are not subject to freezing can be controlled by simple two- or three-way modulating valves (see Figure 11). Steam-distributing coils are required to ensure proper steam coil control. The valve is controlled by coil discharge air temperature or by space temperature, depending on the HVAC system. 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 will revert back to the 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 mixing may allow freezing air to impinge on the coil.

Steam preheat coils should have two-position valves and vacuum breakers to prevent a build-up of condensate in the coil. The valve should be fully open when outside air (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 19). The bypass damper should be sized

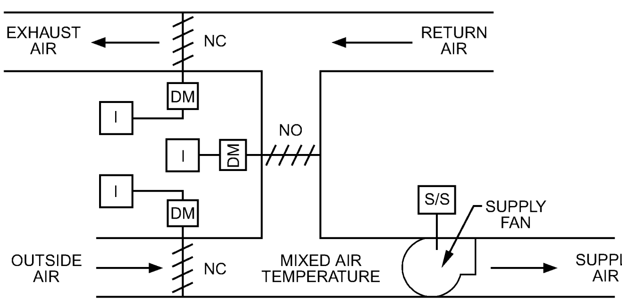


Fig. 16 Economizer Cycle Control

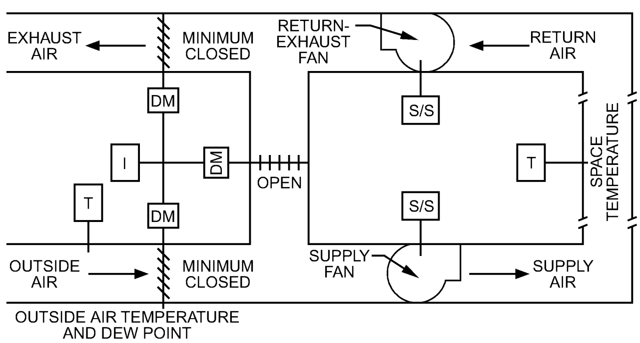


Fig. 17 Warm-Up Control

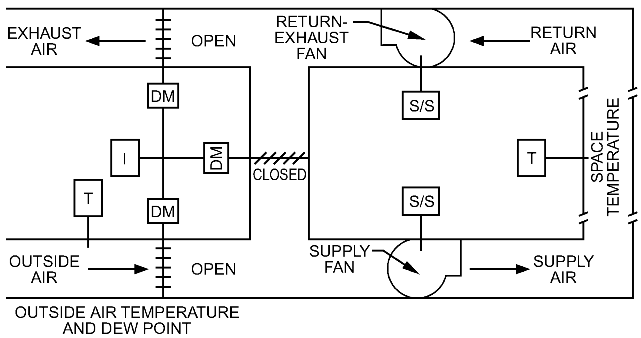


Fig. 18 Night Cooldown Control

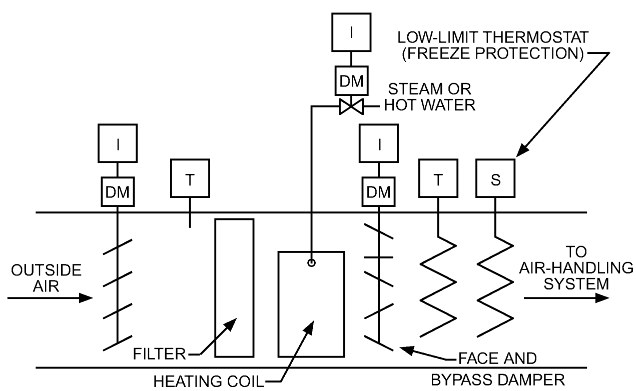


Fig. 19 Preheat with Face and Bypass Dampers

to provide the same pressure drop at full bypass airflow as the combination of face damper and coil does at full airflow.

Hot-water coils must maintain a minimum water velocity in the tubes (on the order of 0.9 m/s) to prevent freezing. A two-position valve combined with face and bypass dampers can be used (Figure 19), but a coil pump is more common. There are many coil pump piping schemes; the most common are shown in Figures 20 and 21A and B. 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.

Figure 20 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 with either variable flow through the primary loop if a two-way valve is used, or constant flow through the primary loop if a three-way valve is used (shown dashed in the figure).

In Figure 21A the coil pump is in series with the primary pumps when the three-way valve is open to the coil. Flow through the coil will be relatively constant but can rise and fall depending on the differential pressure available from primary pumps. Despite the use of the three-way valve, flow through the primary circuit is variable, not constant. The coil pump will act as a booster pump when the valve is open to the coil, potentially offsetting some losses in the primary circuit as well as handling the coil circuit pressure drop, depending on the design of the coil pump. However, this can also reduce flow through other coils served by the primary system that do not nave booster pumps.

Figure 21B 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 even if either the primary pump or coil pump fails. This design results in flow varying 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 through the coil (when the valve is wide open). Unlike the options in the previous two figures, the primary pump must be sized for the pressure drop of 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 will significantly exceed the design primary water flow. The entering water temperature will be 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.

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 power required by 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 is proportional; therefore, face and bypass dampers are not used. The use of electric heating coils is restricted in some areas because of energy consumption. Code compliance should be checked before application. A control system with a solid-state controller and safety controls is shown in Figure 22.

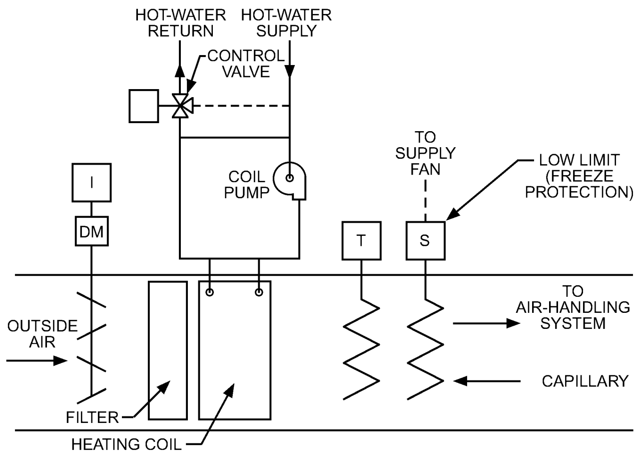


Fig. 20 Coil Pump Piped Primary/Secondary

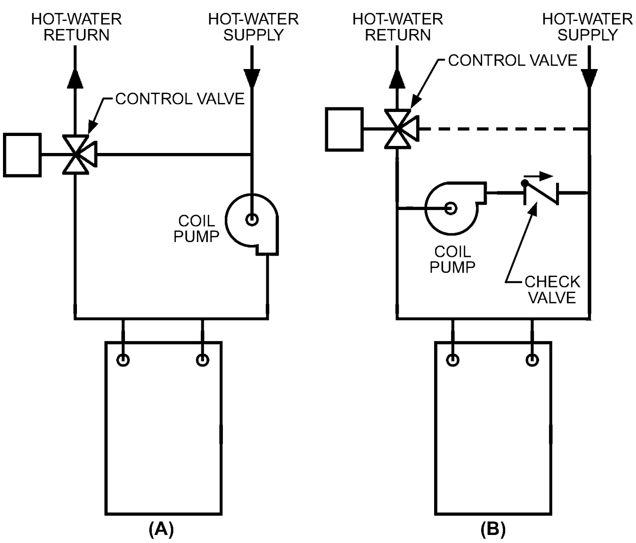


Fig. 21 Coil Pump (A) Piped in Series and (B) Piped in Parallel

Heat Pump

A heat pump is a refrigeration device in which the evaporator is used for cooling, and heat that is normally rejected from the condenser is used for heating. Many conventional means used to control refrigeration equipment can be used to control heat pump heating

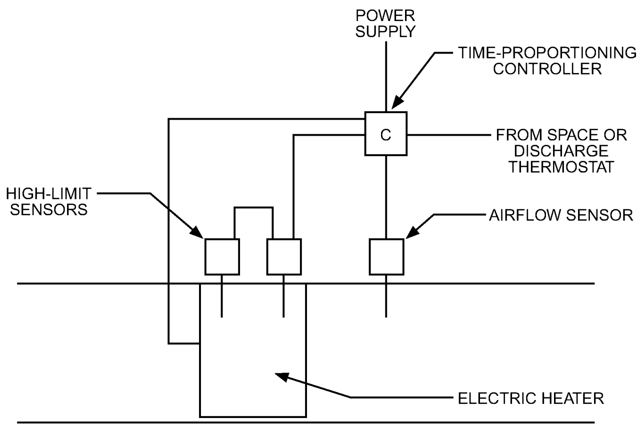


Fig. 22 Electric Heat: Solid-State Controller

and cooling cycles. Chapters 8 and 45 of the 2000 *ASHRAE Handbook—HVAC Systems and Equipment* include details.

Humidity Control

Humidity control is based 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 freezing point of the coolant. If water condensing out of the airstream freezes on the coil surface, airflow is restricted and, in severe cases, may be shut off. The practical limit is about 5°C dew point on the coil surface. As indicated in Figure 23, this results in a relative humidity of about 30% at a space temperature of 24°C, 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 the 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 the moisture removal process results in too low a supply air temperature (Figure 24). If humidification is also provided, this cycle is sometimes called a 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 25) have been used for dehumidification. Space relative humidity ranging from 35 to 55% at 24°C can be obtained with this equipment; however, the cost 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 process also generates heat (Figure 26). Figure 27 shows a typical control.

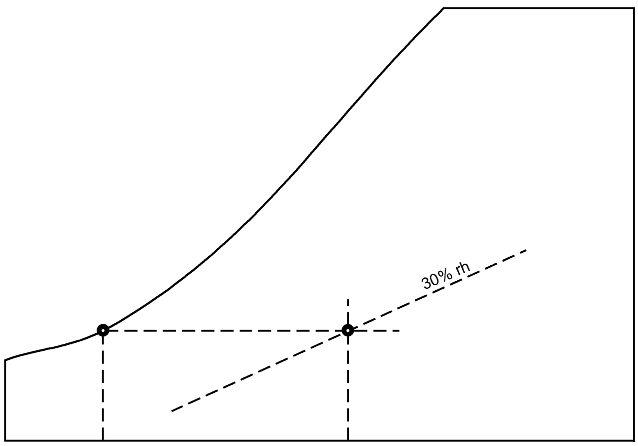


Fig. 23 Psychrometric Chart: Cooling and Dehumidifying—Practical Low Limit

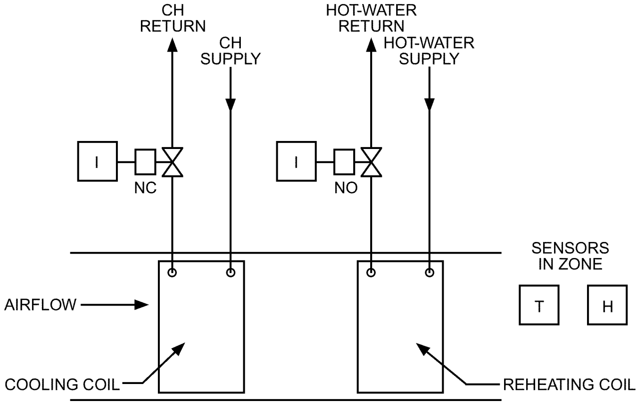


Fig. 24 Cooling and Dehumidifying with Reheat

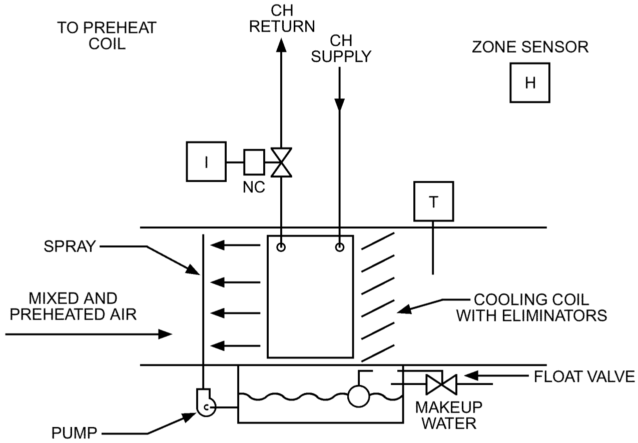


Fig. 25 Sprayed-Coil Dehumidifier

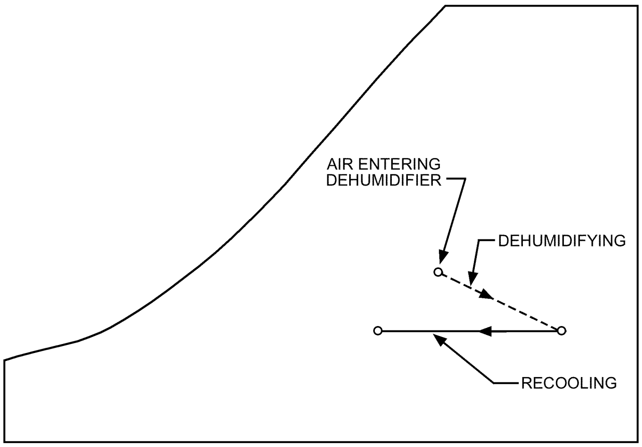


Fig. 26 Psychrometric Chart: Chemical Dehumidification

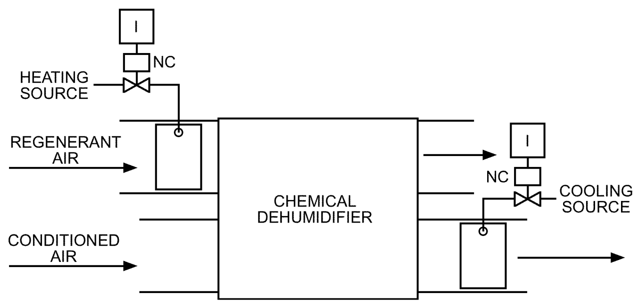


Fig. 27 Chemical Dehumidifier

Humidification can be achieved by adding moisture to supply air. Evaporative pans (usually heated), steam jets, and atomizing spray tubes are all used for space humidification. 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 28). With proper use and control, humidifiers can achieve high space humidity, although they more often maintain design minimum humidity during the heating season.

Outside Air Control

Fixed minimum outside air control provides ventilation air, space pressurization (exfiltration), and makeup air for exhaust fans. For systems without return fans, the outside air damper is interlocked to remain open only when the supply fan operates (Figure 29). The minimum outside air damper should open quickly when the fan turns on, to prevent excessive negative duct pressurization. In some applications, the fan on-off switch opens the outside air damper before the fan is started. The rate of outside airflow is determined by the damper opening and by the pressure difference between the mixed air plenum and the outside air plenums.

For systems with return fans, two variations of fixed minimum outside air control are used. Minimum outside airflow is determined by the pressure drop across the outside damper at minimum position, and an injection fan (Figure 30) or airflow station is installed in the minimum outside air section (Figure 31). If the outside air supplied is greater than the difference between the supply and return fan airflows, a variation of economizer cycle control is used (Figure 31).

In systems using 100% outside air, all air goes to the fan and no air is returned (Figure 32). The outside air damper is interlocked and usually opens before the fan starts.

Radiant Cooling and Heating

Radiation can be used either alone or to supplement another heater. 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 return air to the unit or on a wall at occupant level. Return air control is generally less accurate and results in wider space-temperature fluctuations. 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 only 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 radiation 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

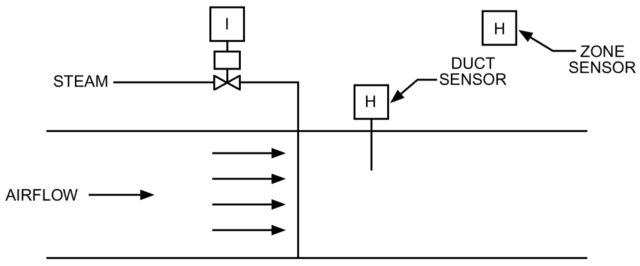


Fig. 28 Steam Jet Humidifier

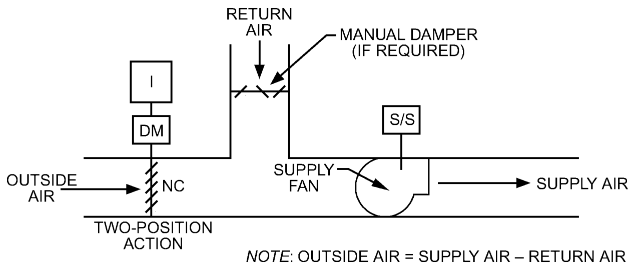


Fig. 29 Fixed Minimum Outside Air Control Without Return Fans

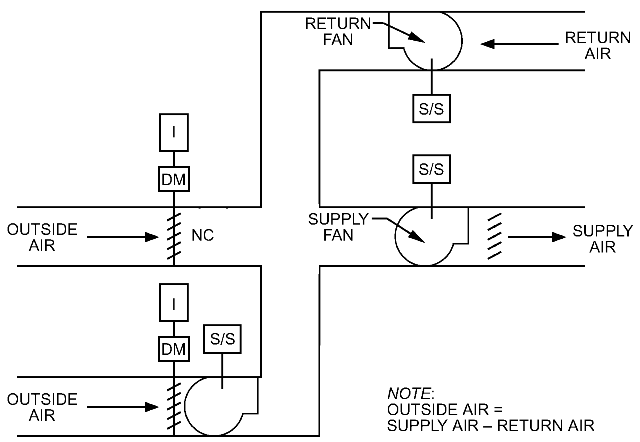


Fig. 30 Fixed Minimum Outside Air Control With Return Fans

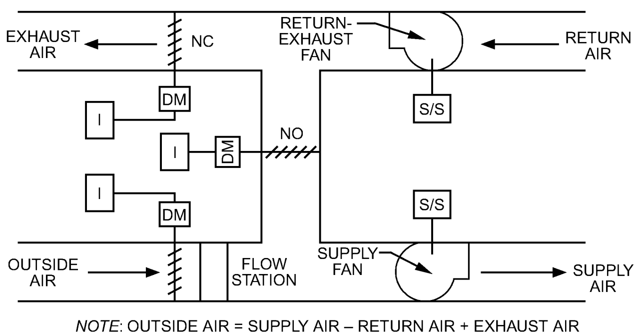


Fig. 31 Fixed Minimum Outside Air Control With Return-Exhaust Fans

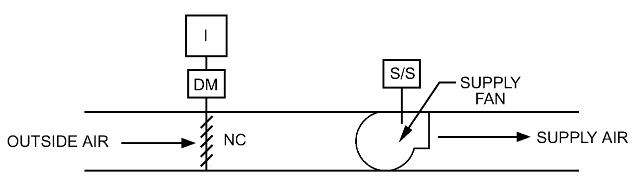


Fig. 32 100% Outside Air Control

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.

Radiant panels for both heating and cooling require controls similar to those described for the four-pipe heating/cooling single-coil 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.

Terminal Units

A system is considered to be variable-volume if airflow to the space varies. The airflow from the fan may be constant for some terminal units.

Single-Duct Constant-Volume. Reheat terminals use a single constant-volume fan system that serves multiple zones (Figure 33). 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, 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 a damper in the inlet that controls the flow of supply air (Figure 34). For spaces requiring heating, a reheat coil can be installed in the discharge. As the temperature in the space drops below the set point, the damper begins to close and reduce the flow of air to the space. When the airflow reaches the minimum limit, the valve on the reheat coil begins to open.

Single-duct VAV systems, which supply warm air to all zones when heating is required and cool air to all zones when cooling is required, have limited application and are used where heating is required only for morning warm-up. They should not be used if some zones require heating at the same time that others require cooling. These systems, like single-duct cooling-only systems, are generally controlled during occupancy.

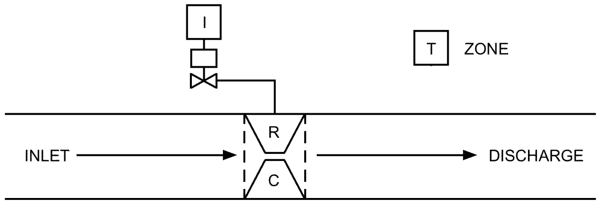


Fig. 33 Single-Duct Constant-Volume Zone Reheat

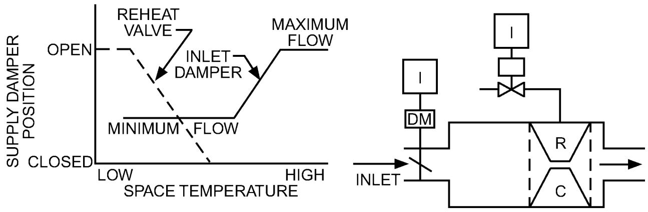


Fig. 34 Throttling VAV Terminal Unit

An **induction VAV terminal** controls the space temperature by reducing the supply airflow to the space and by inducing return air from the plenum space 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 the 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 the output of the space temperature sensor. When the temperature in the space drops below the set point, the bypass damper begins to open, routing some of the supply air to the plenum, which reduces the amount of supply air entering the space. When the bypass is fully open, the control valve for the reheat coil opens as required to maintain the 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 total airflow of the fan remains constant, so the 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

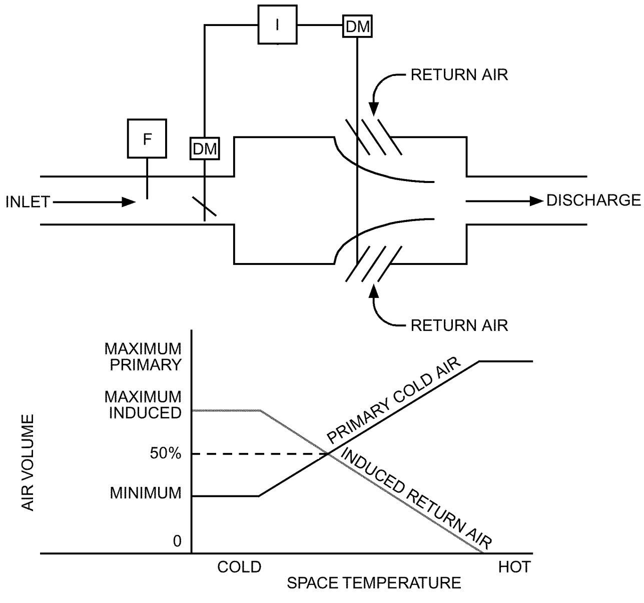


Fig. 35 Induction VAV Terminal Unit

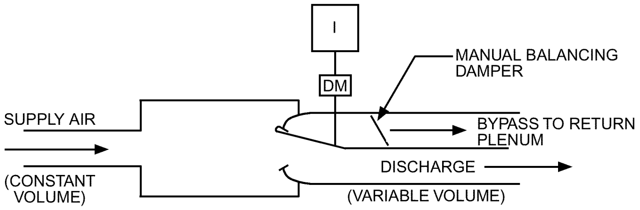


Fig. 36 Bypass VAV Terminal Unit

be added to maintain a minimum temperature in the space when the 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 the reduced supply air. As the 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 the supply air reaches its minimum level, 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 level of airflow to the space. Total airflow to the space is the sum of the fan output and the supply air quantity. When the 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 has reached 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.

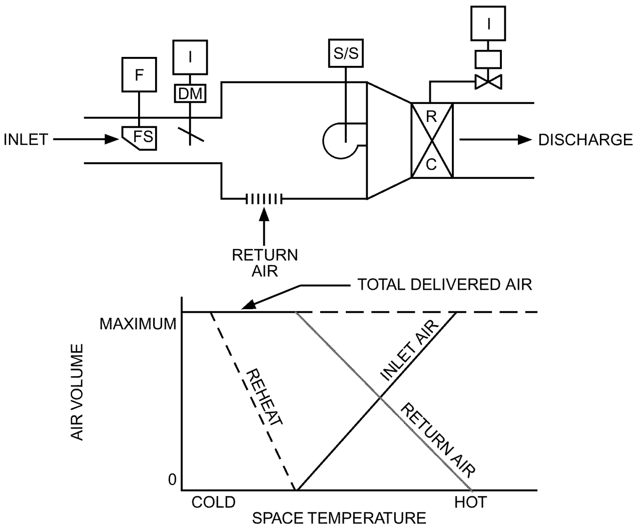


Fig. 37 Series Fan-Powered VAV Terminal Unit

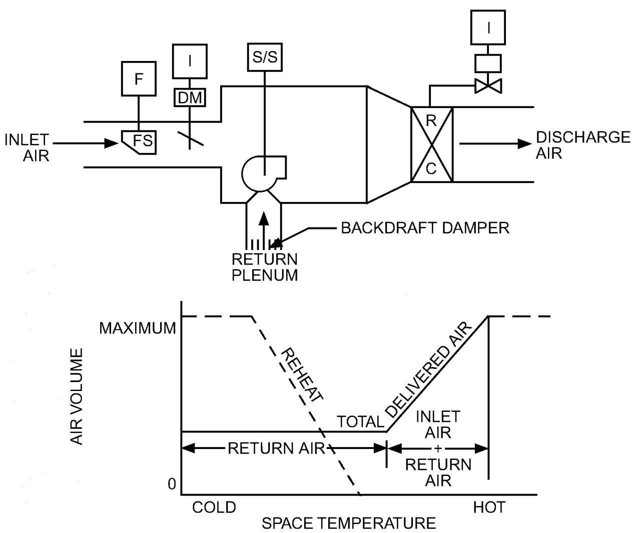


Fig. 38 Plenum or Parallel Fan Terminal Unit

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-duct damper and cold-duct damper 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. The 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 the 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 interlinked to operate in opposite 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 a portion 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

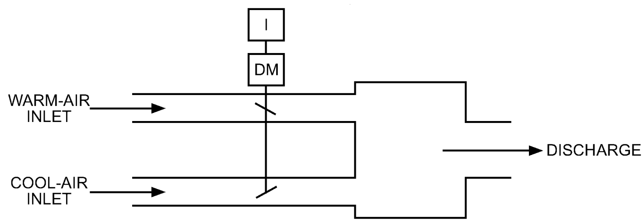


Fig. 39 Dual-Duct Mixing Box Terminal Unit

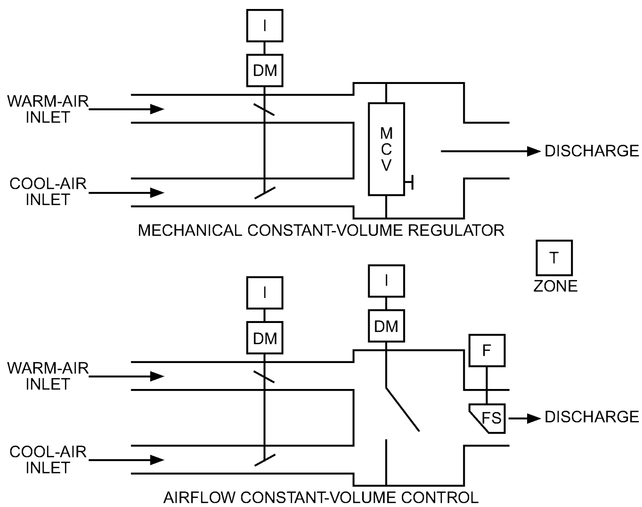


Fig. 40 Dual-Duct Pressure-Independent Constant-Volume Mixing Box Terminal Units

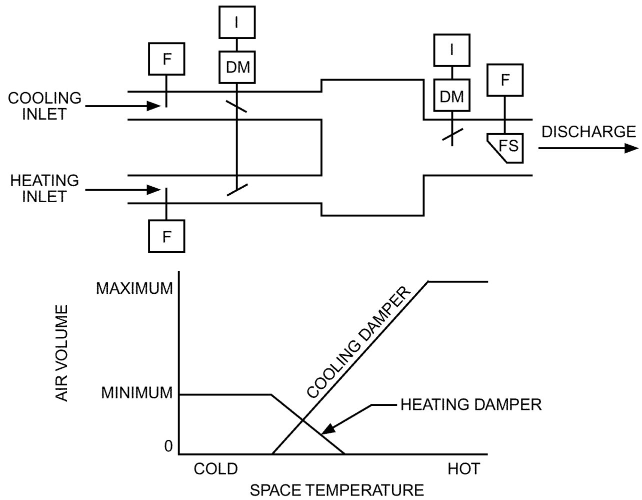


Fig. 41 Pressure-Independent Dual-Duct VAV Terminal Unit

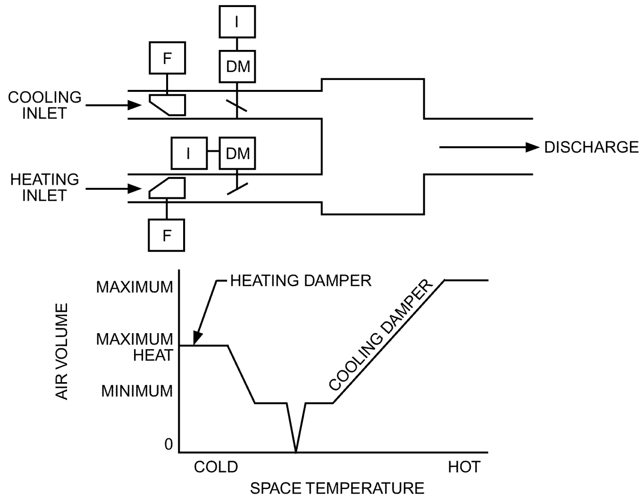


Fig. 42 Variable, Constant-Volume (ZEB) Dual-Duct Terminal Unit

maintain adjustable minimum flows for ventilation (with no overlap of damper operations) in the ZEB when neither heating nor cooling is required.

CONTROL OF SYSTEMS

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 in a multizone system (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 the design and selection of components. Single-zone systems do not require terminal boxes because the zone temperature can be

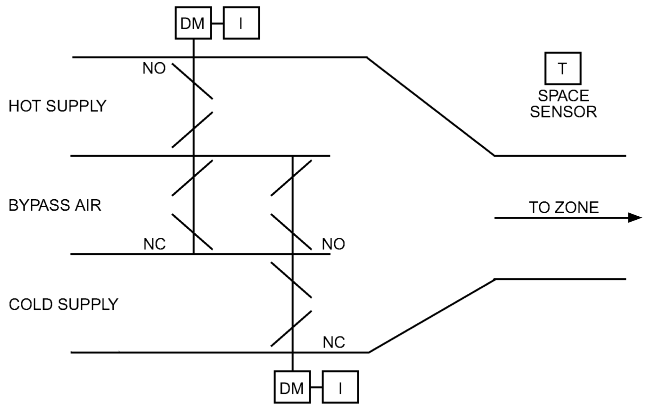


Fig. 43 Zone Mixing Dampers—Three-Deck Multizone System

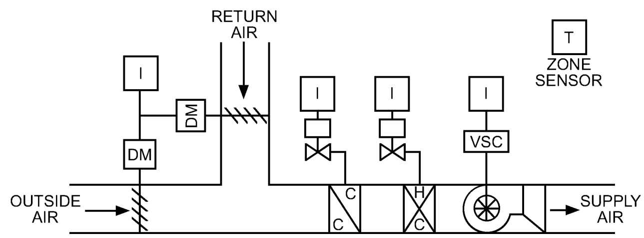


Fig. 44 Single-Zone Fan System

maintained by modulating in sequence the heating and cooling control valves.

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 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 47 shows the 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 the 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

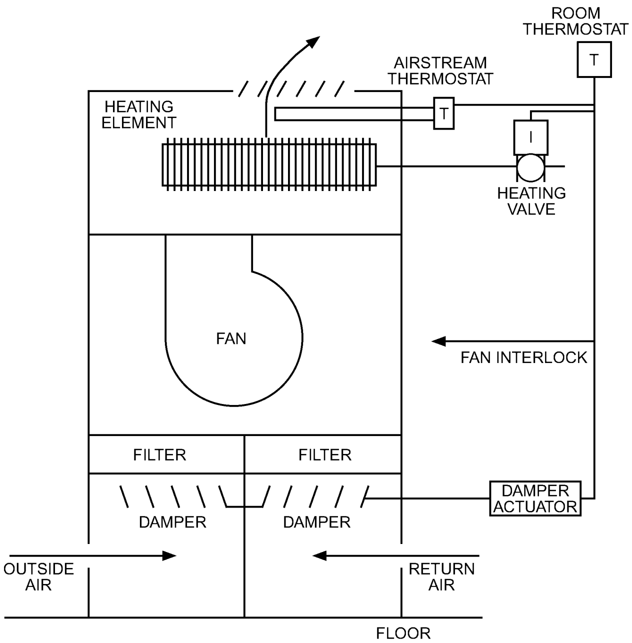


Fig. 45 Cycles I, II, and W Control Arrangements

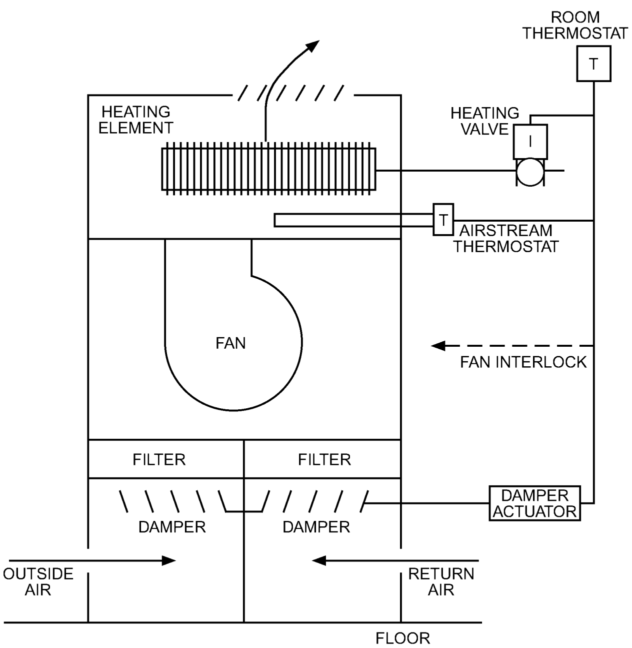


Fig. 46 Cycle III Control Arrangement

from dropping below a minimum temperature. Figure 47 shows the relative positions of the heating valve and ventilation dampers with respect to space temperature.

Cycle III Control. During the 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 fixed temperature (typically 13°C). 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

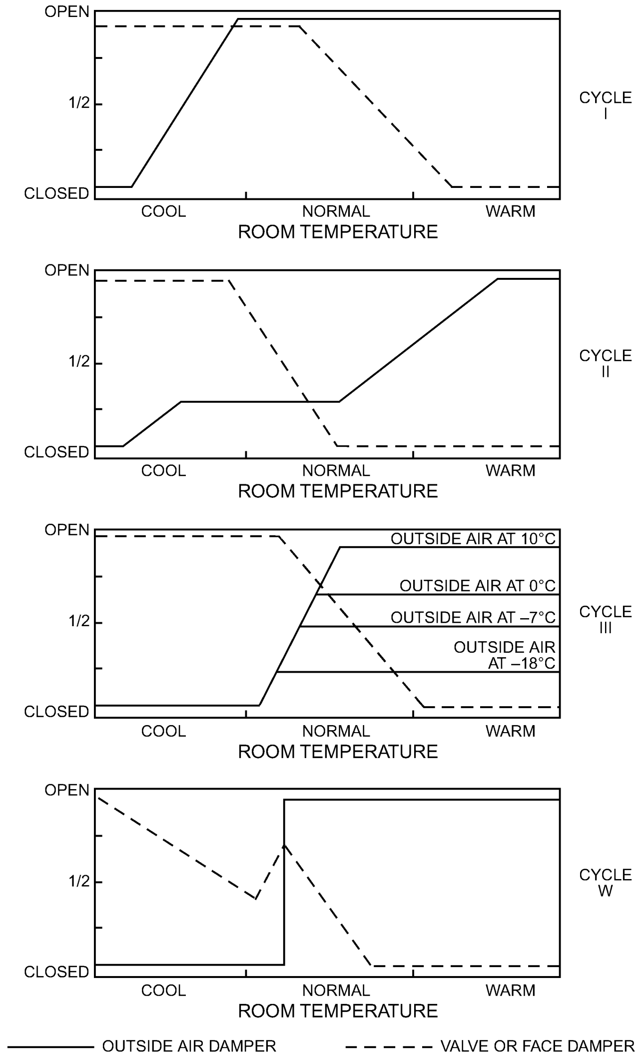


Fig. 47 Valve and Damper Positions with Respect to Room Temperature

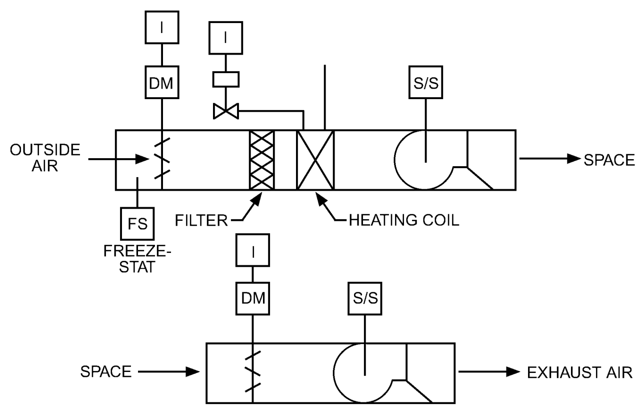


Fig. 48 Makeup Air Unit

heating valve as required. Figure 47 shows the relative positions of the heating valve and ventilation dampers with respect to space temperature.

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 freezestat placed next to the heating coil that shuts the unit off when near-freezing temperatures are sensed.

Cycle W Control. Cycle W is similar to Cycle II, except the heating valve is controlled by the room thermostat and the dampers are controlled by the low-limit thermostat (Figures 45 and Figure 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 freeze protection if air temperature approaches freezing at any spot along the sensing element.

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, temperature in a space is set to an extreme value, a zone sensor is placed so that it senses spot loads (from coffee pots, the sun, copiers, etc.), a sensor

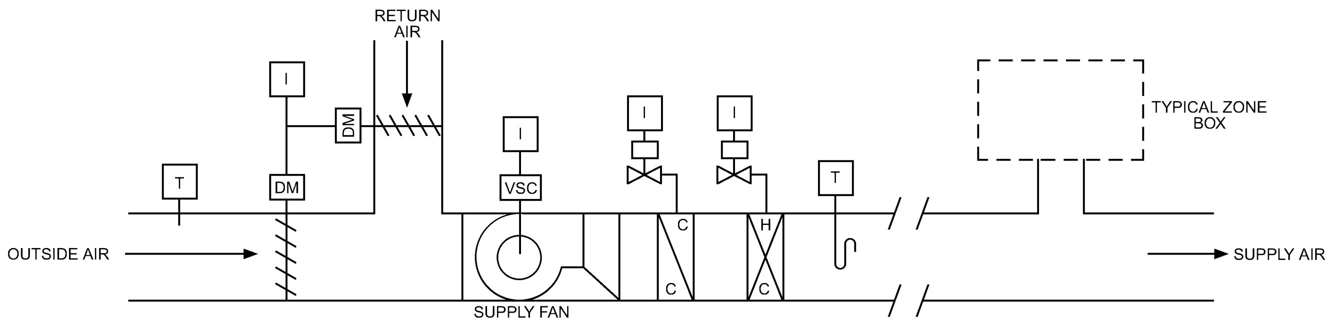


Fig. 49 Multizone Single-Duct System

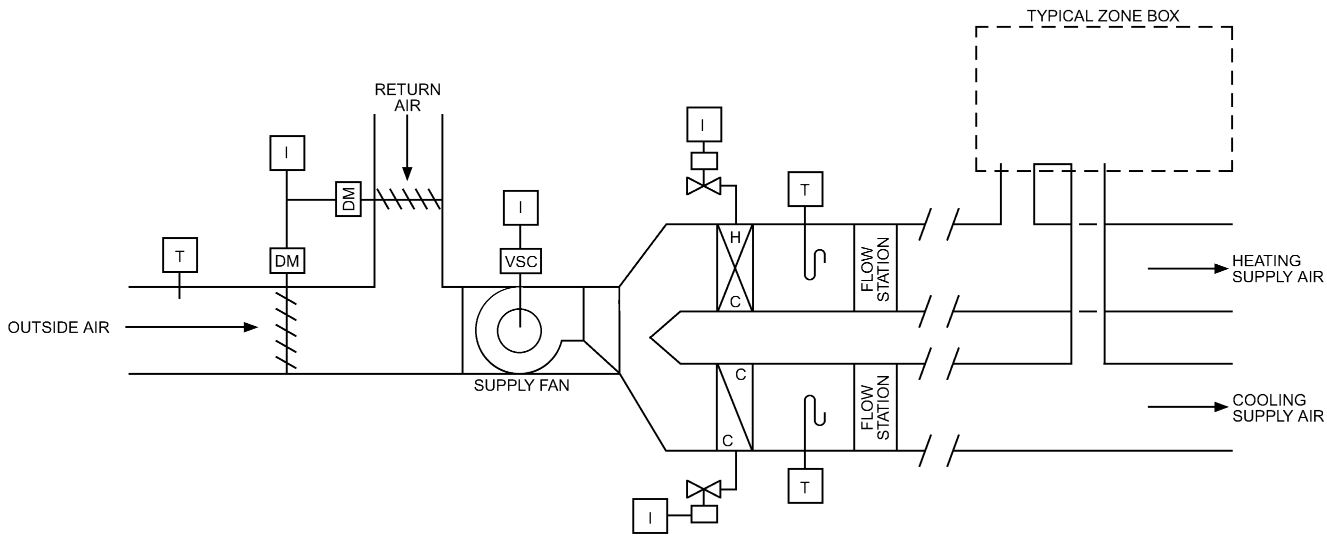


Fig. 50 Dual-Duct Single-Supply Fan System

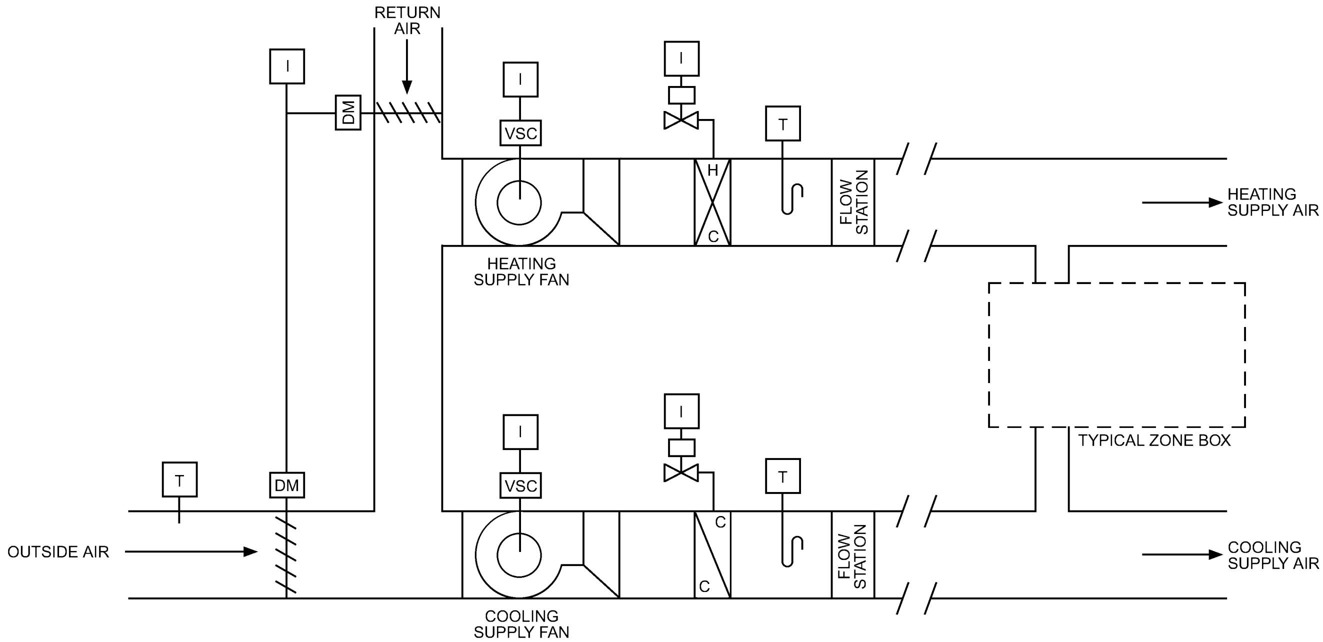


Fig. 51 Dual-Supply Fan System

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. 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.

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 designed on a custom basis. Chapter 43 of the 2002 *ASHRAE Handbook—Refrigeration* describes various chillers (e.g., centrifugal, reciprocating, screw, and scroll). Chapter 11 of the 2000 *ASHRAE Handbook—HVAC Systems and Equipment* covers variations in piping configurations (e.g., series and parallel chilled-water flow) and some associated control concepts.

Chiller plants are generally one of two types: variable flow (Figures 52 and 53) or constant flow (Figure 54). The figures show a parallel-flow piping configuration. Control of the remote load determines which type should be used. Throttling coil valves vary the flow in response to the load and a temperature differential that tends to remain near the design temperature differential. 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.

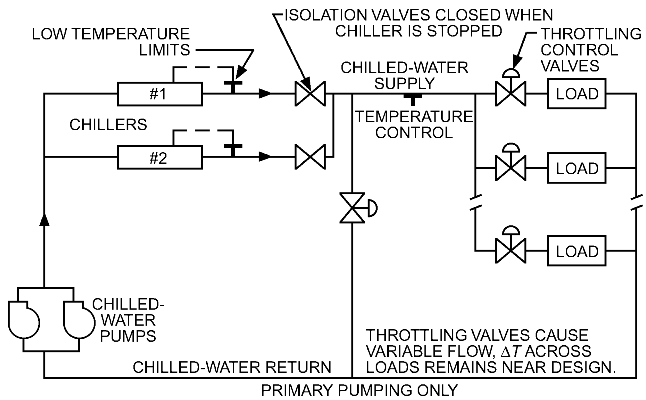


Fig. 52 Variable-Flow Chilled-Water System

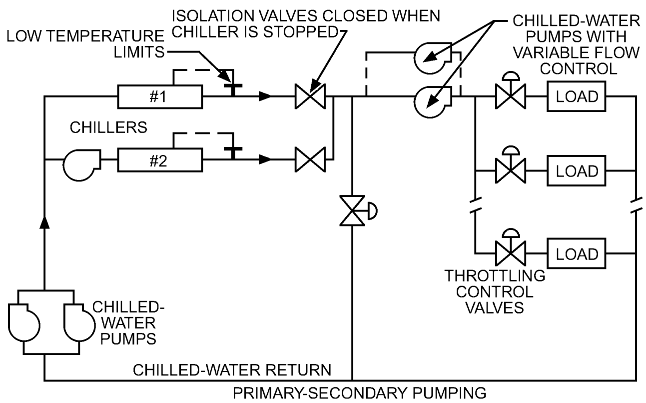


Fig. 53 Variable-Flow Chilled-Water System

The constant-flow system (Figure 54) is only "constant flow" under each combination of chillers on line; a major upset occurs whenever a chiller is added or dropped. The load reset function ensures that the zone with the largest load is satisfied, whereas supply or return water control treats average zone load.

Refrigerant Pressure Optimization

Chiller efficiency is a function of the percent of full load on the chiller and the difference in refrigerant pressure between the condenser and the evaporator. In practice, the pressure is represented by condenser water exit temperature minus chilled-water supply temperature. To reduce the refrigerant pressure, the chilled-water supply temperature must be increased and/or the condenser water temperature decreased. An energy saving of about 3% is obtained for each 1 K reduction in condenser water temperature supplied to a chiller.

The following methods are used to reduce refrigerant pressure:

1. Use chilled-water load reset to raise the supply set point as load decreases (use manufacturer's recommendations). Figure 55 shows the basic function of this method. Varying degrees of sophistication are available, including computer control. Note that the additional pump and fan power of VAV air handlers must be considered in calculating net energy savings.
2. Lower the condenser temperature to the lowest safe temperature (use manufacturer's recommendations) by keeping the cooling tower bypass valve closed, operating at full condenser water pump capacity, and maintaining water temperature within the design cooling tower approach (see Chapter 36 in the 2000 *ASHRAE Handbook—HVAC Systems and Equipment*).

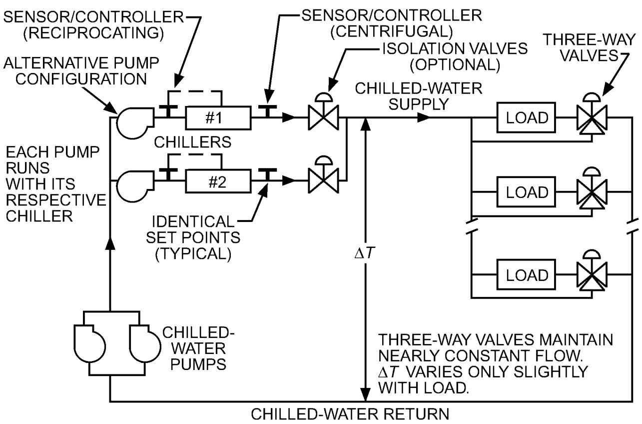


Fig. 54 Constant-Flow Chilled-Water System

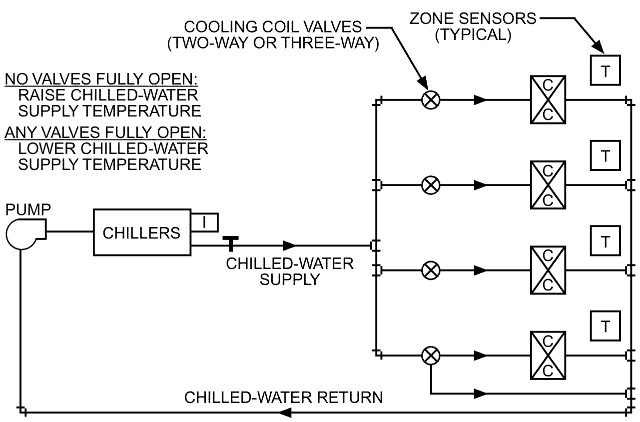


Fig. 55 Chilled-Water Load Reset

Operation Optimization

Multiple-chiller plants should be operated at the most efficient point on the part-load curve. Figure 56 shows a typical part-load curve for a centrifugal chiller operated at design conditions. Figure 57 shows similar curves at different pressure-limiting conditions. Figure 58 indicates one way to determine a point at which a chiller should be added or dropped in a two-unit plant. In general, the part-load curves are plotted for all combinations of chillers; then, the break-even point between *n* and *n* + 1 chillers can be determined.

Daily start-up of the chiller plant should be optimized to **minimize run time** based on start-up time of the air-handling units. Chillers are generally started at the same time as the first fan system.

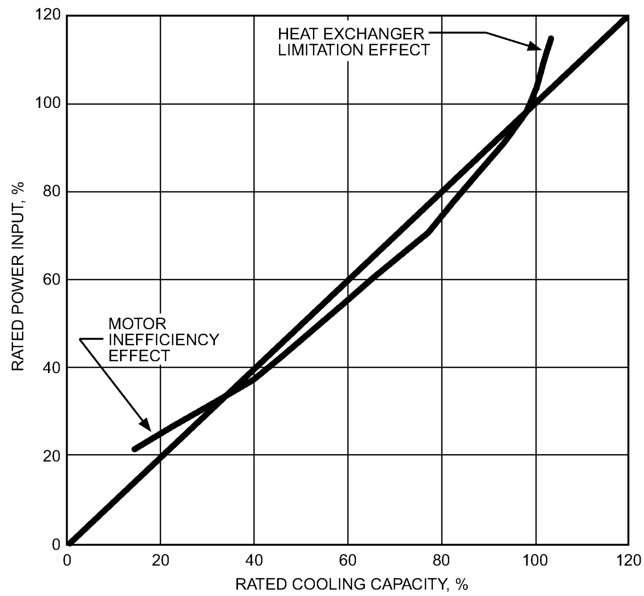


Fig. 56 Chiller Part-Load Characteristics at Design Refrigerant Pressure

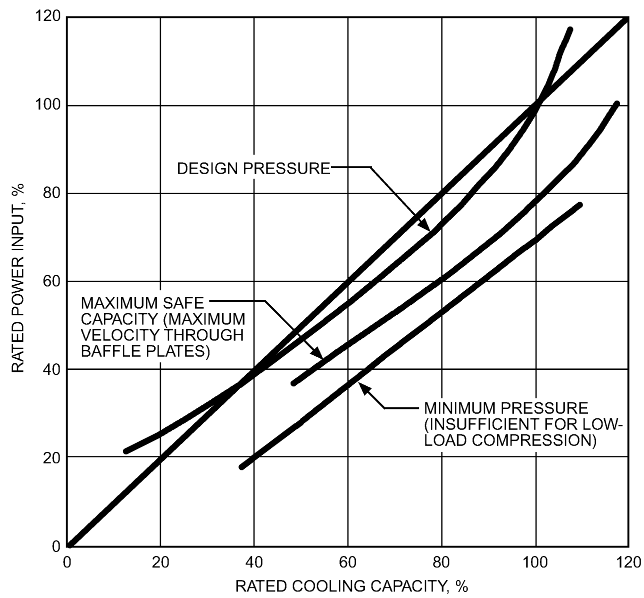


Fig. 57 Chiller Part-Load Characteristics with Variable Pressure

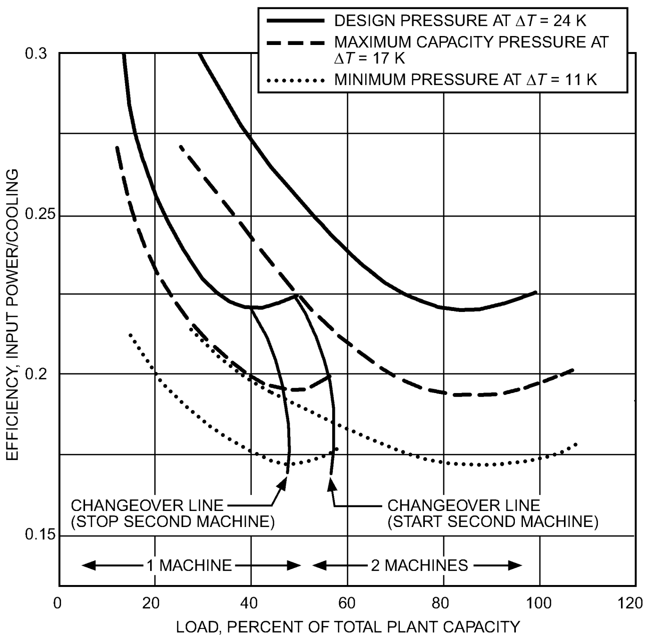


Fig. 58 Multiple-Chiller Operation Changeover Point—Two Equal-Sized Chillers

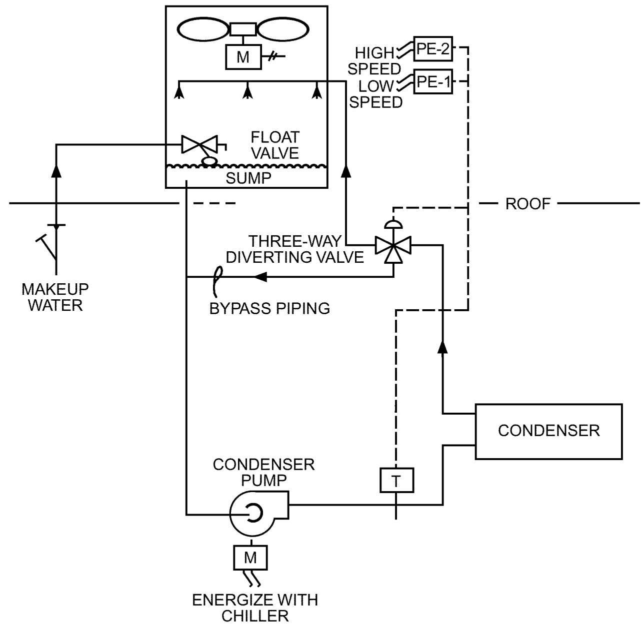


Fig. 59 Condenser Water Temperature Control

Chillers may be started early if the water distribution loop has great thermal mass; they may be started later if outside air can provide cooling to fan systems at start-up.

The **condenser water circuit** and control arrangement for the central plant are shown in Figure 59. The control system designer works with liquid chiller control when the equipment is integrated into the central chiller plant. Typically, cooling tower, chiller pump, and condenser pump control must be considered if the overall plant is to be stable and energy-efficient.

With centrifugal chillers, condenser supply water temperature is allowed to float as long as the temperature remains above a low limit. The manufacturer should specify the minimum entering condenser water temperature required for satisfactory performance of the particular chiller. The control schematic in Figure 59 works as follows: for a condenser supply temperature (e.g., above a set point of 24°C), the valve is open to the tower, the bypass valve is closed, and the tower fan or fans are operating. As water temperature decreases (e.g., to 18°C), tower fan speed can be reduced to low-speed operation if a variable-speed drive or two-speed motor is used. On a further decrease in condenser water supply temperature, the tower fan or fans stop and the bypass valve begins to open or close to maintain the acceptable minimum water temperature.

Water Heating

A basic constant-volume hydronic system is shown in Figure 60. 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.

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

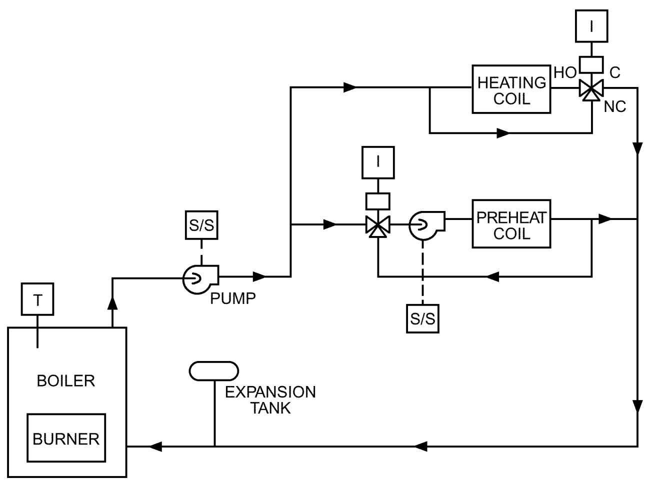


Fig. 60 Load and Zone Control in Constant-Volume System

variations in altitude. The effect can be substantial; for example, barometric pressure decreases by nearly one-third as altitude increases from sea level to 3000 m.

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

Explosive Atmospheres

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

When using electric control, the control case and contacts can be surrounded with an explosionproof case, permitting only the

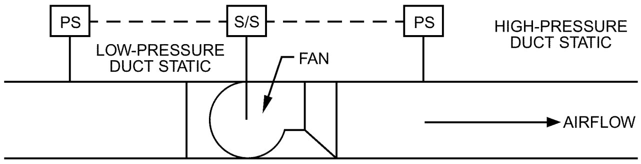


Fig. 61 High-Limit Static Pressure Controller

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. The latter 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.

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 61) 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 that turns the fans off; and (2) the controlling high-limit type (Figure 61), which is used in systems having zone fire dampers. When the zone fire damper closes, and the duct static pressure sensor is downstream of the zone, duct pressure rises, causing the duct static control to increase fan modulation; however, the controlling high limit will override.

Steam or hot-water exchangers tend to be self-regulating, unlike electrical resistance heat transfer devices. For example, if airflow through a steam or hot-water coil stops, coil surfaces approach the temperature of the entering steam or hot water, but cannot exceed it. Convection or radiation losses from the steam or hot water to the surrounding area take place, so the coil is not usually damaged. Electric coils and heaters, on the other hand, can be damaged 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 in case a fire damper closes or some duct lining blocks the air passage. Limit thermostats should also be installed to turn off the heaters when temperatures exceed safe operating levels.

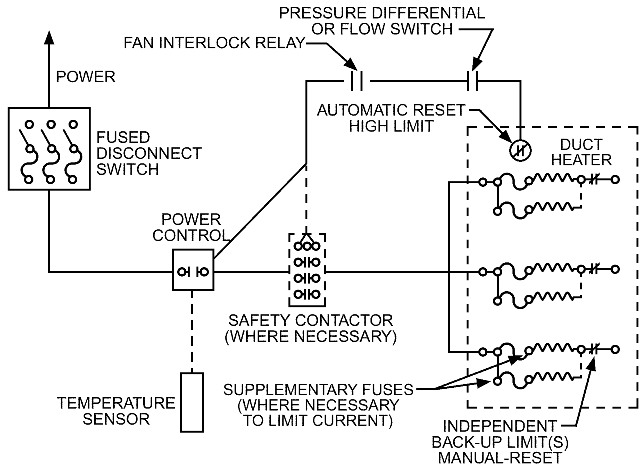


Fig. 62 Duct Heater Control

Duct Heaters

The current in individual elements of electric duct heaters is normally limited to a maximum safe value established by the *National Electrical Code* or local codes. Two safety devices in addition to the airflow interlock device are usually applied to duct heaters (Figure 62). 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 backup high-limit safety device is generally set independently to interrupt all current to the heater in case 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.

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. A local loop control 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.

Distributed processing using microprocessors has augmented computer use at many locations other than the central control point. The local loop controller can be a direct digital controller (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.

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 the 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:

- Heating and cooling loads as they vary—the ability to heat or cool any area of a building at any time
- Occupancy schedules and the 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

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:

1. *Run equipment only when needed.* Schedule HVAC unit operation for occupied periods. Run heat at night only to maintain internal temperature between 13 and 16°C to prevent freezing. Start morning warm-up as late as possible to achieve design internal temperature by occupancy time, considering residual space temperature, 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 the end of occupancy.
2. *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.
3. *Provide only the heating or cooling actually needed.* Reset the supply temperature of hot and cold air (or water).
4. *Supply heating and cooling from the most efficient source.* Use free or low-cost energy sources first, then higher-cost sources as necessary.
5. *Apply outside air control.* When on minimum outside air, use no less than that recommended by ASHRAE *Standard* 62. 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. The type of system and the 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 systems and single-coil systems) 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 of 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 jurisdictions in the United States 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 the reheat function 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 permit 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. Reheat should be avoided. 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 is the control, the smaller are 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 500 m^2 because the difficulty of obtaining good distribution and of finding a representative location for the 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 1220 mm) and in locations where they will not be affected by heat from sources such as direct sun rays, wall pipes or ducts, convectors, or direct air currents from diffusers or equipment (e.g., copy machines, coffee makers, or refrigerators). Air circulation should be ample and unimpeded by furniture or other obstructions, and the thermostat should be protected against mechanical injury. Thermostats located 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. The sensing element 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 the 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.

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 the cooperation of various parties, including the owner, design engineers, and the contractors and the subcontractors. A commissioning authority (the designated person, company, or agent who implements the overall commissioning process) generally leads the commissioning process of a project. Each commissioning process must have a commissioning plan that defines the commissioning process and is developed in increasing detail as the project progresses. Commissioning phases include plan, design, construction, and acceptance. Commissioning of HVAC systems is recommended for construction of new buildings as well as existing buildings. See Chapter 42 and ASHRAE *Guideline* 1 for more information.

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