11

Heating, Ventilating, and Air Conditioning Control Systems

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11.1 Introduction

This chapter describes the essentials of control systems for heating, ventilating, and air conditioning (HVAC) of buildings designed for energy conserving operation. Of course, there are other renewable and energy conserving systems that require control. The principles described herein for buildings also apply with appropriate and obvious modification to these other systems. For further reference, the reader is referred to several standard references in the list at the end of this chapter.

HVAC system controls are the information link between varying energy demands on a building's primary and secondary systems and the (usually) approximately uniform demands for indoor environmental conditions. Without a properly functioning control system, the most expensive, most thoroughly designed HVAC system will be a failure. It simply will not control indoor conditions to provide comfort.

The HVAC designer must design a control system that

- Sustains a comfortable building interior environment
- Maintains acceptable indoor air quality

- Is as simple and inexpensive as possible and yet meets HVAC system operation criteria reliably for the system lifetime
- Results in efficient HVAC system operation under all conditions
- Commissions the building, equipment and control systems
- Documents the system operation so that the building staff can successfully operate and maintain the HVAC system

It is a considerable challenge for the HVAC system designer to design a control system that is energy efficient and reliable. Inadequate control system design, inadequate commissioning, and inadequate documentation and training for the building staff often create problems and poor operational control of HVAC systems. This chapter describes the basics of HVAC control and the operational needs for successfully maintained operation. The reader is encouraged to review the following references on the subject: ASHRAE (2002, 2003, 2004, 2005), Haines (1987), Honeywell (1988), Levine (1996), Sauer, Howell, and Coad (2001), Stein and Reynolds (2000), and Tao and Janis (2005).

To achieve proper control based on the control system design, the HVAC system must be designed correctly and then constructed, calibrated and commissioned according to the mechanical and electrical systems drawings. These must include properly sized primary and secondary systems. In addition, air stratification must be avoided, proper provision for control sensors is required, freeze protection is necessary in cold climates, and proper attention must be paid to minimizing energy consumption, subject to reliable operation and occupant comfort.

The principle and final controlled variable in buildings is zone temperature (and to a lesser extent humidity and/or air quality in some buildings). This chapter will therefore focus on methods to control temperature. Supporting the zone temperature control, numerous other control loops exist in buildings within the primary and secondary HVAC systems, including boiler and chiller control, pump and fan control, liquid and air flow control, humidity control, and auxiliary system control (for example, thermal energy storage control). This chapter discusses only automatic control of these subsystems. Honeywell (1988) defines an automatic control system as "a system that reacts to a change or imbalance in the variable it controls by adjusting other variables to restore the system to the desired balance."

Figure 11.1 defines a familiar control problem with feedback. The water level in the tank must be maintained under varying outflow conditions. The float operates a valve that admits water to the tank as the tank is drained. This simple system includes all the elements of a control system:

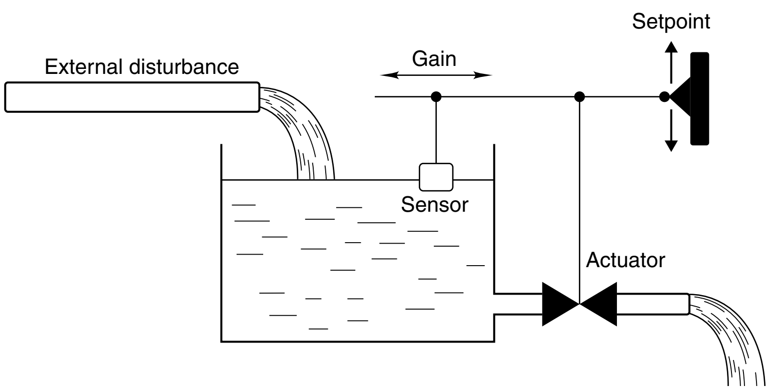

FIGURE 11.1 Simple water level controller. The setpoint is the full water level; the error is the difference between the full level and the actual level.

- Sensor—float; reads the controlled variable, the water level
- Controller—linkage connecting float to valve stem; senses difference between full tank level and operating level and determines needed position of valve stem
- Actuator (controlled device)—internal valve mechanism; sets valve (the final control element) flow in response to level difference sensed by controller
- Controlled system characteristic—water level; this is often termed the controlled variable

This system is called a *closed loop* or *feedback* system because the sensor (float) is directly affected by the action of the controlled device (valve). In an open loop system the sensor operating the controller does not directly sense the action of the controller or actuator. An example would be a method of controlling the valve based on an external parameter such as time of day, which may have an indirect relation to water consumption from the tank.

There are four common methods of control, of which Figure 11.1 shows but one. In the next section, each method will be described in relation to an HVAC system example.

11.2 Modes of Feedback Control

Feedback control systems adjust an output control signal based on feedback. The feedback is used to generate an error signal, which then drives a control element. Figure 11.1 illustrates a basic control system with feedback. Both off–on (two-position) control and analog (variable) control can be used. Numerous methodologies have been developed to implement analog control. These include proportional, proportional-integral (PI), proportional-integral-differential (PID), fuzzy logic, neural networks, and auto-regressive moving average (ARMA) control systems. Proportional and PI control systems are used for most HVAC control applications.

Figure 11.2a shows a steam coil used to heat air in a duct. The simple control system shown includes an air temperature sensor, a controller that compares the sensed temperature to the setpoint, a steam valve controlled by the controller, and the coil itself. This example system will be used as the point of reference when discussing the various control system types. Figure 11.2b is the control diagram corresponding to the physical system shown in Figure 11.2a.

Two-position control applies to an actuator that is either fully open or fully closed. In Figure 11.2a, the valve is a two-position valve if two-position control is used. The position of the steam valve is determined by the value of the coil outlet temperature. Figure 11.3 depicts two-position control of the valve. If the air temperature drops below 95°F, the valve opens and remains open until the air temperature reaches 100°F. The differential is usually adjustable, as is the temperature setting itself. Two-position control is the least expensive method of automatic control and is suitable for control of HVAC systems with large time constants. Examples include residential space and water heating systems. Systems that are fast-reacting should not be controlled using this approach because overshoot and undershoot may be excessive.

Proportional control adjusts the controlled variable in proportion to the difference between the controlled variable and the setpoint. For example, a proportional controller would increase the coil heat rate in Figure 11.2 by 10% if the coil outlet air temperature dropped by an amount equal to 10% of the temperature range specified for the heating to go from off to fully on. Equation 11.1 defines the behavior of a proportional control loop:

$$T = T_{set} + K_p e \tag{11.1}$$

where T is the controller output, T_{set} is the set temperature corresponding to a constant value of controller output when no error exists, K_p is the loop gain that determines the rate or proportion at which the control signal changes in response to the error, and e is the error. In the case of the steam coil, the error is the difference between the air temperature setpoint and the sensed supply air temperature:

$$e = T_{set} - T_{sa} \tag{11.2}$$

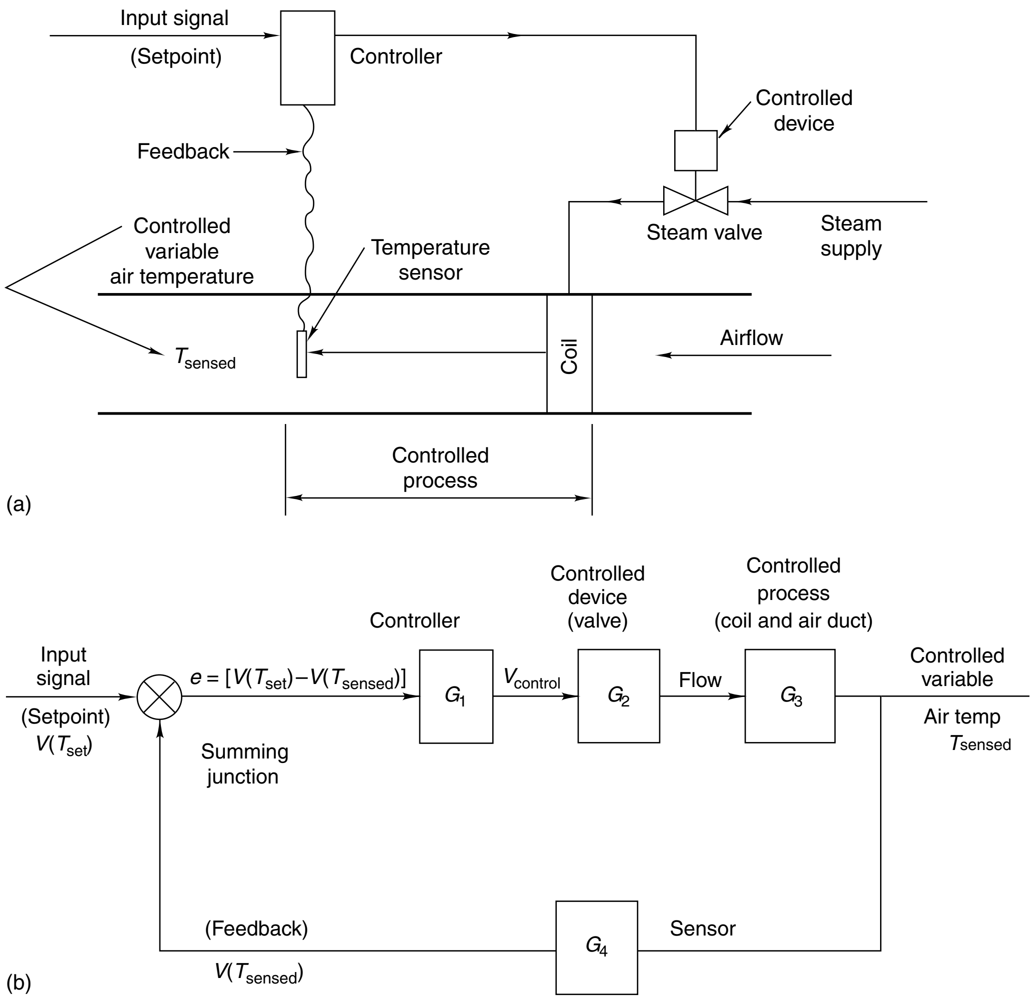

FIGURE 11.2 (a) Simple heating coil control system showing the process (coil and short duct length), controller, controlled device (valve and its actuator) and sensor. The setpoint entered externally is the desired coil outlet temperature. (b) Equivalent control diagram for heating coil. The G's represent functions relating the input to the output of each module. Voltages, *V*, represent both temperatures (setpoint and coil outlet) and the controller output to the valve in electronic control systems.

As coil air-outlet temperature drops farther below the set temperature, error increases, leading to increased control action—an increased steam flow rate. Note that the temperatures in Equation 11.1 and Equation 11.2 are often replaced by voltages or other variables, particularly in electronic controllers.

The throttling range (ΔT_{max}) is the total change in the controlled variable that is required to cause the actuator or controlled device to move between its limits. For example, if the nominal temperature of a zone is 72°F and the heating controller throttling range is 6°F, then the heating control undergoes its full travel between a zone temperature of 69°F and 75°F. This control, whose characteristic is shown in Figure 11.4, is reverse acting; i.e., as temperature (controlled variable) increases, the heating valve position decreases.

The throttling range is inversely proportional to the gain as shown in Figure 11.4. Beyond the throttling range, the system is out of control. In actual hardware, one can set the setpoint and either the

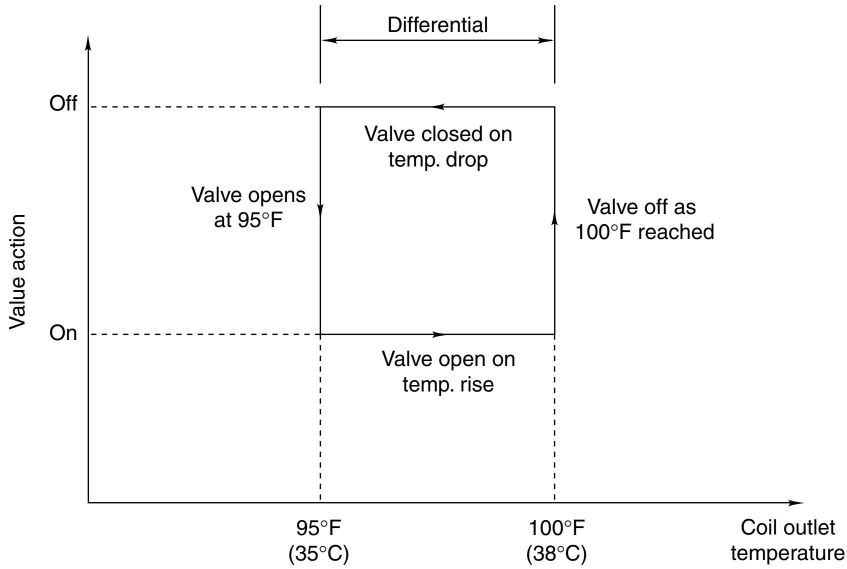

FIGURE 11.3 Two position (on-off) control characteristic.

gain or the throttling range (most common), but not both of the latter. Proportional control by itself is not capable of reducing the error to zero because an error is needed to produce the capacity required for meeting a load, as will be discussed in the following example. This unavoidable value of the error in proportional systems is called the *offset*. It is easy to see from Figure 11.4 that the offset is larger for systems with smaller gains. There is a limit to which one can increase the gain to reduce offset, because high gains can produce control instability.

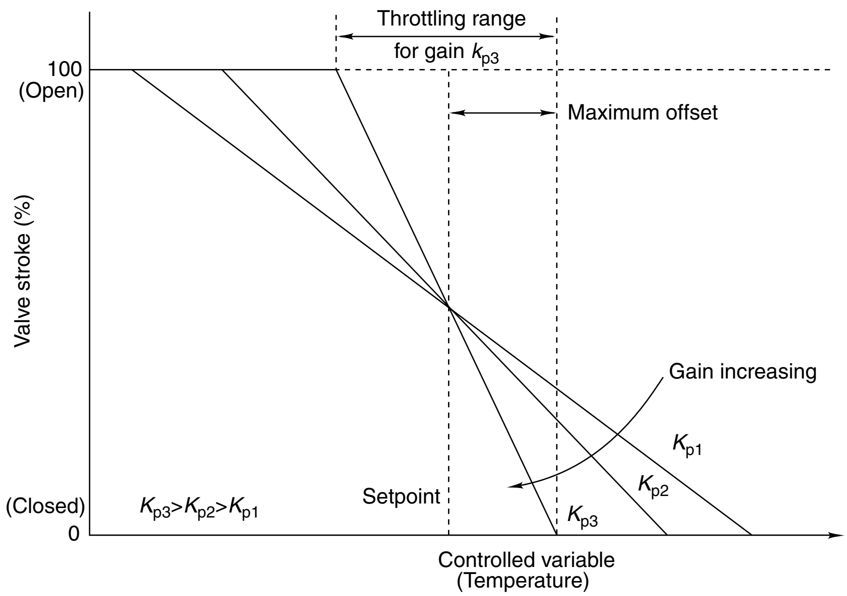

FIGURE 11.4 Proportional control characteristic showing various throttling ranges and the corresponding proportional gains, K_p. This characteristic is typical of a heating coil temperature controller.

Example 11.1 Proportional Gain Calculation

Problem: If the steam heating coil in Figure 11.2a has a heat output that varies from 0 to 20 kW as the outlet air temperature varies from 35 to 45°C in an industrial process, what is the coil gain and what is the throttling range? Find an equation relating the heat rate at any sensed air temperature to the maximum rate in terms of the gain and setpoint.

Solution: Given that $\dot{Q}_{max} = 20$ kW, $\dot{Q}_{min} = 0$ kW, $T_{max} = 45$°C, and $T_{min} = 35$°C for the system in Figure 11.2a, and assuming steady-state operation, the problem is to determine K_p and ΔT_{max}.

The throttling range is the range of the controlled variable (air temperature) over which the controlled system (heating coil) exhibits its full capacity range. The temperature varies from 35 to 45°C; therefore the throttling range is

$$\Delta T_{max} = 45°\text{C} - 35°\text{C} = 10°\text{C} \tag{11.3}$$

The proportional gain is the ratio of the controlled system (coil) output to the throttling range. For this example, the controller output is $\dot{Q}$ and the gain is

$$K_p = \frac{\dot{Q}_{max} - \dot{Q}_{min}}{\Delta T_{max}} = \frac{(20-0)\ \text{kW}}{10\ \text{K}} = 2.0(\text{kW/K}). \tag{11.4}$$

The controller characteristic can be found by inspecting Figure 11.4. It is assumed that the average air temperature (40°C) occurs at the average heat rate (10 kW). The equation of the straight line shown is

$$\dot{Q} = K_p(T_{set} - T_{sensed}) + \frac{\dot{Q}_{max}}{2} = K_p e + \frac{\dot{Q}_{max}}{2}. \tag{11.5}$$

Note that the quantity $(T_{set} - T_{sensed})$ is the error, e, and a nonzero value indicates that the set temperature is not met. However, the proportional control system used here requires presence of an error signal to fully open or fully close the valve.

Inserting the numerical values:

$$\dot{Q} = 2.0\frac{\text{kW}}{\text{K}}(40 - T_{sensed}) + 10\ \text{kW}. \tag{11.6}$$

Comments: In an actual steam-coil control system, it is the steam valve that is controlled directly to indirectly control the heat rate of the coil. This is typical of many HVAC system controls, in that the desired control action is achieved indirectly by controlling another variable that in turn accomplishes the desired result. This is why the controller and controlled device are often shown separately as in Figure 11.2b.

This example illustrates with a simple system how proportional control uses an error signal to generate an offset, and how that offset controls an output quantity. Using a bias value, the error can be set to be zero at one value in the control range. Proportional control requires a nonzero error over the remainder of the control range.

Real systems also have a time response called a *loop time constant*. This limits proportional control applications to slow response systems, where the throttling range can be set so that the system achieves stability. Typically, slow-responding mechanical systems include pneumatic thermostats for zone control and air handler unit damper control. Fast-acting systems like duct pressure control must be artificially slowed down to be appropriate for proportional control.

Integral control is often added to proportional control to eliminate the offset inherent in proportional-only control. The result—proportional plus integral control—is identified by the acronym PI. Initially, the corrective action produced by a PI controller is the same as for a proportional-only controller. After the initial period, a further adjustment due to the integral term reduces the offset to zero. The rate at which this occurs depends on the time scale of the integration. In equation form, the PI controller is modeled by

$$V = V_0 + K_p e + K_i \int e\mathrm{d}t, \tag{11.7}$$

in which K_i is the integral gain constant. It has units of reciprocal time and is the number of times that the integral term is calculated per unit time. This is also known as the *reset rate; reset control* is an older term used by some to identify integral control.

Today, most PI control implementations use electronic sensors, analog-to-digital converters (A/Ds), and digital logic to implement the PI control. Integral windup must be taken into account when using PI control. Integral windup occurs when the control first starts or comes out of a reset condition. The integral term increases by integrating the error signal, and can have a full output. When the control becomes enabled, a large offset error can occur, driving the output variable to an undesired state. Various methods exist to minimize or eliminate the windup problem.

The integral term in Equation 11.7 has the effect of adding a correction to the output signal V for as long as the error term exists. The continuous offset produced by the proportional-only controller can thereby be reduced to zero because of the integral term. For HVAC systems, the time scale (K_P/K_i) of the integral term is often in the range of 10+ s to 10+ min. PI control is used for fast-acting systems for which accurate control is needed. Examples include mixed air controls, duct static pressure controls, and coil controls. Because the offset is eventually eliminated with PI control, the throttling range can be set rather wide to insure stability under a wider range of conditions than good control would permit with proportional-only control. Therefore, PI control is also used on almost all electronic thermostats.

Derivative control is used to speed up the action of PI control. When derivative control is added to PI control, the result is called *PID control*. The derivative term added to Equation 11.7 generates a correction signal proportional to the time rate of change of error. This term has little effect on a steady proportional system with uniform offset (time derivative is zero) but initially, after a system disturbance, produces a larger correction more rapidly. Equation 11.8 includes the derivative term in the mathematical model of the PID controller

$$V = V_0 + K_p e + K_i \int e\mathrm{d}t + K_d \frac{\mathrm{d}e}{\mathrm{d}t}, \tag{11.8}$$

in which K_d is the derivative gain constant. The time scale (K_d/K_p) of the derivative term is typically in the range of 0.2–15 minutes. Because HVAC systems do not often require rapid control response, the use of PID control is less common than use of PI control. Because a derivative is involved, any noise in the error (i.e., sensor) signal must be avoided to maintain stable control. One effective application of PID control in buildings is in duct static pressure control, a fast-acting subsystem that otherwise has a tendency to be unstable.

Derivative control has limited application in HVAC systems because the PID control loops can easily become unstable. PID loops require correct tuning for each of the three gain constants (Ks) over the performance range that the control loop will need to operate. Another serious limitation centers on the fact that most facility operators lack training and skills in tuning PID control loops.

Figure 11.5 illustrates the loop response for three correctly configured systems when a step function change, or disturbance, occurs. Note that the PI loop achieves the same final control as the PID, only the PI error signal is larger. An improperly configured PID loop can oscillate from the high value to the low value continuously.

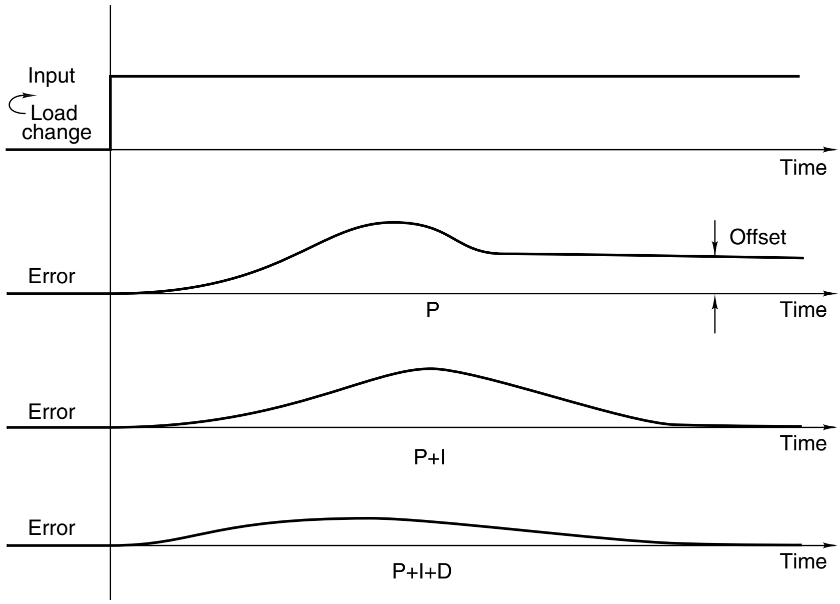

FIGURE 11.5 Performance comparison of P, PI, and PID controllers when subjected to a uniform, input step change.

11.3 Basic Control Hardware

In this section, the various physical components needed to perform the actions required by the control strategies of the previous section are described. Because there are two fundamentally different control approaches—pneumatic and electronic—the following material is so divided. Sensors, controllers, and actuators for principal HVAC applications are described.

11.3.1 Pneumatic Systems

The first widely adopted automatic control systems used compressed air as the signaling transmission medium. Compressed air had the advantages that it could be "metered" through various sensors, and it could power large actuators. The fact that the response of a normal pneumatic loop could take several minutes often worked as an advantage, as well. Pneumatic controls use compressed air (approximately 20 psig in the US) for operation of sensors and actuators. Though most new buildings use electronic controls, many existing buildings use pneumatic controls. This section provides an overview of how these devices operate.

Temperature control and damper control comprise the bulk of pneumatic loop controls. Figure 11.6 shows a method of sensing temperature and producing a control signal. Main supply air, supplied by a compressor, enters a branch line through a restriction. The zone thermostat bleeds out a variable amount of air, depending on the position of the flapper, controlled by the temperature sensor bellows. As more air bleeds out, the branch line pressure (control pressure) drops. This reduction in the total pressure to the control element changes the output of the control element. This control can be forward-acting or reverse-acting. The restrictions typically have hole diameters on the order of a few thousandths of an inch, and consume very little air. Typical pressures in the branch lines range between 3 and 13 psig (20–90 kPa). In simple systems this pressure from a thermostat could operate an actuator such as a control valve for a room heating unit. In this case, the thermostat is both the sensor and the controller—a rather common configuration.

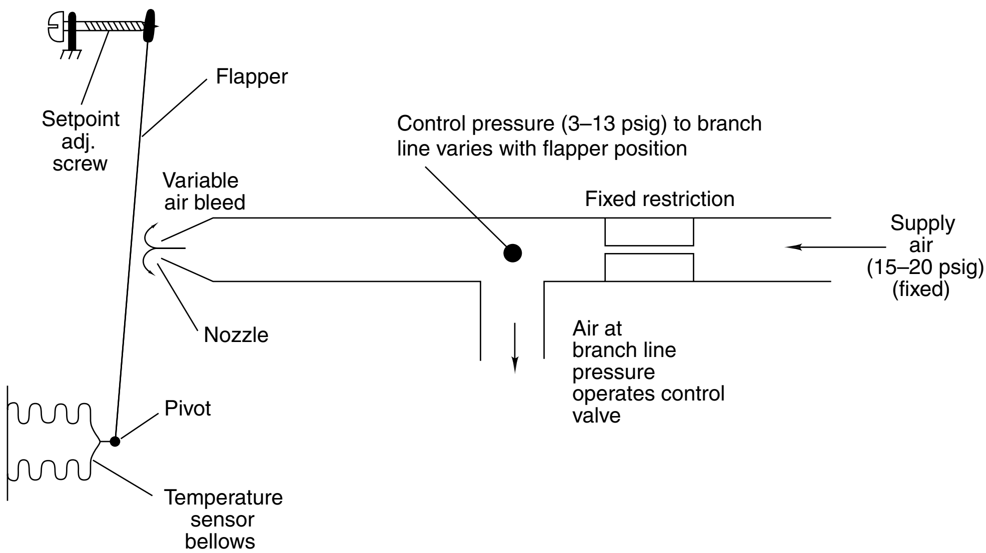

FIGURE 11.6 Drawing of pneumatic thermostat showing adjustment screw used to change temperature setting.

Many other temperature sensor approaches can be used. For example, the bellows shown in Figure 11.6 can be eliminated, and the flapper can be made of a bimetallic strip. As temperature changes, the bimetallic strip changes curvature, opening or closing the flapper/nozzle gap. Another approach uses a remote bulb filled with either liquid or vapor that pushes a rod (or a bellows) against the flapper to control the pressure signal. This device is useful if the sensing element must be located where direct measurement of temperature by a metal strip or bellows is not possible, such as in a water stream or high-velocity ductwork. The bulb and connecting capillary size may vary considerably by application.

Pressure sensors may use either bellows or diaphragms to control branch line pressure. For example, the motion of a diaphragm may replace that of the flapper in Figure 11.6 to control the bleed rate. A bellows similar to that shown in the same figure may be internally pressurized to produce a displacement that can control air bleed rate. A bellows produces significantly greater displacements than a single diaphragm.

Humidity sensors in pneumatic systems are made from materials that change size with moisture content. Nylon or other synthetic hygroscopic fibers that change size significantly (i.e., 1%–2%) with humidity are commonly used. Because the dimensional change is relatively small on an absolute basis, mechanical amplification of the displacement is used. The materials that exhibit the desired property include nylon, hair, and cotton fibers. Human hair exhibits a much more linear response with humidity than nylon; however, because the properties of hair vary with age, nylon has much wider use (Letherman 1981). Humidity sensors for electronic systems are quite different and are discussed in the next section.

An actuator converts pneumatic energy to motion—either linear or rotary. It creates a change in the controlled variable by operating control devices such as dampers or valves. Figure 11.7 shows a pneumatically operated control valve. The valve opening is controlled by the pressure in the diaphragm acting against the spring. The spring is essentially a linear device. Therefore, the motion of the valve stem is essentially linear with air pressure. However, this does not necessarily produce a linear effect on flow, as discussed later. Figure 11.8 shows a pneumatic damper actuator. Linear actuator motion is converted into rotary damper motion by the simple mechanism shown.

Pneumatic controllers produce a branch line (see Figure 11.6) pressure that is appropriate to produce the needed control action for reaching the setpoint. Such controls are manufactured by a number of control firms for specific purposes. Classifications of controllers include the sign of the output (direct- or reverse-acting) produced by an error, by the control action (proportional, PI, or two-position), or by number of inputs or outputs. Figure 11.9 shows the essential elements of a dual-input, single-output

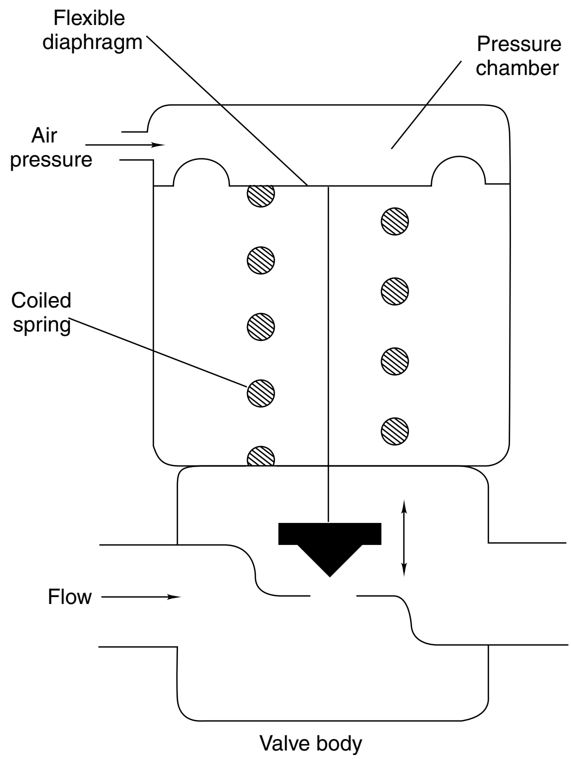

FIGURE 11.7 Pneumatic control valve showing counterforce spring and valve body. Increasing pressure closes the valve.

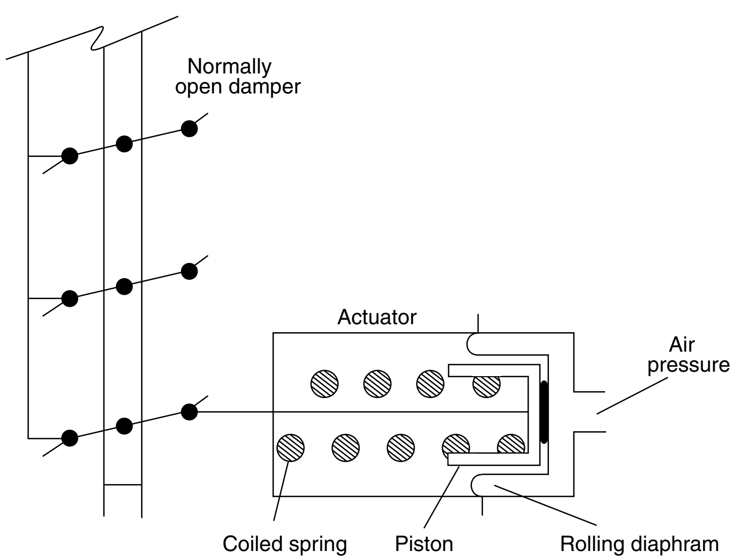

FIGURE 11.8 Pneumatic damper actuator. Increasing pressure closes the parallel blade damper.

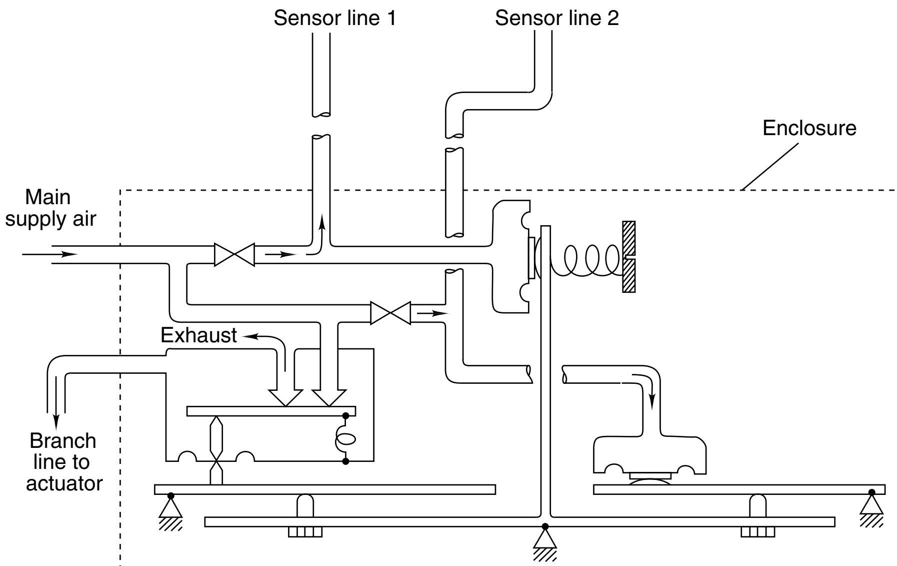

FIGURE 11.9 Example pneumatic controller with two inputs and one control signal output.

controller. The two inputs could be heating system supply temperature and outdoor temperature sensors, used to control the output water temperature setting of a boiler in a building heating system. This is essentially a boiler temperature reset system that reduces heating water temperature with increasing ambient temperature for better system control and reduced energy use.

The air supply for pneumatic systems must produce very clean, oil-free, dry air. A compressor producing 80–100 psig is typical. Compressed air is stored in a tank for use as needed, avoiding continuous operation of the compressor. The air system should be oversized by 50%–100% of estimated nominal consumption. The air is then dried to prevent moisture freezing in cold control lines in air handling units and elsewhere. Dried air should have a dew point of −30°F or less in severe heating climates. In deep cooling climates, the lowest temperature to which the compressed air lines are exposed may be the building cold air supply. Next, the air is filtered to remove water droplets, oil (from the compressor), and any dirt. Finally, the air pressure is reduced in a pressure regulator to the control system operating pressure of approximately 20 psig. Control air piping uses either copper or, in accessible locations, nylon.

11.3.2 Electronic Control Systems

Electronic controls comprise the bulk of the controllers for HVAC systems. Direct digital control systems (DDCs) began to make inroads in the early 1990s and now make up over 80% of all controller sales. Low-end microprocessors now cost under $0.50 each and are thus very economical to apply. Along with the decreased cost, increased functionality can be obtained with DDC controls. BACnet has emerged as the standard communication protocol (ASHRAE 2001), and most control vendors offer a version of the BACnet protocol. In this section, the sensors, actuators and controllers used in modern electronic control systems for buildings are surveyed.

Direct digital control (DDC) enhances the previous analog-only electronic system with digital features. Modern DDC systems use analog sensors (converted to digital signals within a computer) along with digital computer programs to control HVAC systems. The output of this microprocessor-based system can be used to control electronic, electrical, or pneumatic actuators or a combination. DDC systems have the advantage of reliability and flexibility that others do not. For example, it is easier to accurately set control constants in computer Software than by making adjustments at a control panel with a

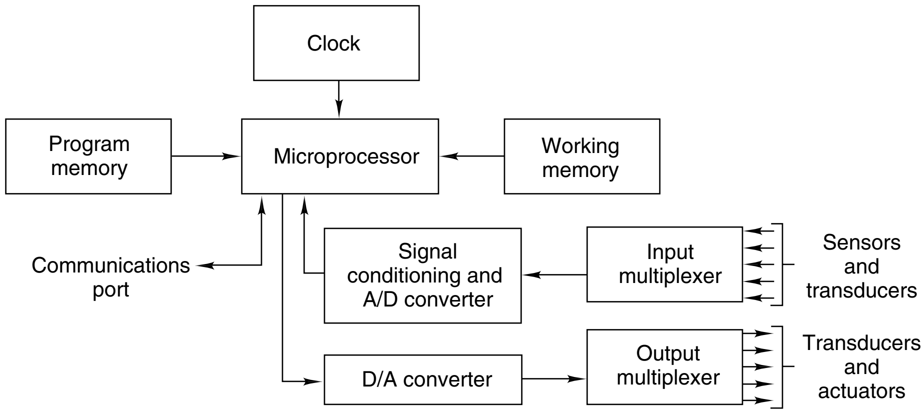

FIGURE 11.10 Block diagram of DDC controller.

screwdriver. DDC systems offer the option of operating energy management systems (EMS) and HVAC diagnostic, knowledge-based systems because the sensor data used for control is very similar to that used in EMSs. Pneumatic systems do not offer this ability. Figure 11.10 shows a schematic diagram of a DDC controller. The entire control system must include sensors and actuators not shown in this controller-only drawing.

Temperature measurements for DDC applications are made by three principal methods:

- Thermocouples
- Resistance temperature detectors (RTDs)
- Thermistors

Each has its advantages for particular applications. Thermocouples consist of two dissimilar metals chosen to produce a measurable voltage at the temperature of interest. The voltage output is low (in the millivolt range) but a well-established function of the junction temperature. Except for flame temperature measurements, thermocouples produce voltages too small to be useful in most HVAC applications (for example, a type-J thermocouple produces only 5.3 mV at 100°C).

RTDs use small, responsive sensing sections constructed from metals whose resistance-temperature characteristic is well established and reproducible. To first order,

$$R = R_0(1 + kT), \tag{11.9}$$

where R is the resistance (Ω), R_0 is the resistance at the reference temperature of 0°C (Ω), k is the temperature coefficient of resistance (°C^{-1}), and T is the RTD temperature (°C). This equation is easy to invert to find the temperature as a function of resistance. Although complex higher-order expressions exist, their use is not needed for HVAC applications.

Two common materials for RTDs are platinum and Balco (a 40%-nickel, 60%-iron alloy). The nominal values of k, respectively, are 3.85×10^{-3} and 4.1×10^{-3} °C^{-1}.

Modern electronics measure current and voltage and then determine the resistance using Ohm's law. The measurement causes power dissipation in the RTD element, raising the temperature and creating an error in the measurement. This Joule self-heating can be minimized by reducing the power dissipated in the RTD. Raising the resistance of the RTD helps reduce self-heating, but the most effective approach requires pulsing the current and making the measurement in a few milliseconds. Because one measurement per second will generally satisfy the most demanding HVAC control loop, the power dissipation can be reduced by a factor of 100 or more. Modern digital controls can easily handle the calculations necessary to implement piecewise linearization and other curve-fitting methods to improve the accuracy of the RTD measurements. In addition, lead wire resistance can cause lack of accuracy for the class of platinum RTDs whose nominal resistance is only 100 Ω, because the lead resistance of 1–2 Ω is not negligible by comparison to that of the sensor itself.

Thermistors are semiconductors that exhibit a standard exponential dependence for resistance versus temperature given by

$$R = Ae^{(B/T)}. \tag{11.10}$$

A is related to the nominal value of resistance at the reference temperature (77°F) and is on the order of several thousands of ohms. The exponential coefficient B (a weak function of temperature) is on the order of 5400–7200 R (3000–4000 K). The nonlinearity inherent in a thermistor can be reduced by connecting a properly selected fixed resistor in parallel with it. The resulting linearity is desirable from a control system design viewpoint. Thermistors can have a problem with long-term drift and aging; the designer and control manufacturer should consult on the most stable thermistor design for HVAC applications. Some manufacturers provide linearized thermistors that combine both positive and negative resistive dependence on temperature to yield a more linear response function.

Humidity measurements are needed for control of enthalpy economizers, or may also be needed to control special environments as such as clean rooms, hospitals and areas housing computers. Relative humidity, dew point, and humidity ratio are all indicators of the moisture content of air. An electrical, capacitance-based approach using a polymer with interdigitated electrodes has become the most common sensor type. The polymer material absorbs moisture and changes the dielectric constant of the material, changing the capacitance of the sensor. The capacitance of the sensor forms part of a resonant circuit so that when the capacitance changes, the resonant frequency changes. This frequency can then be correlated to the relative humidity and provide reproducible readings, if not saturated by excessive exposure to high humidity levels (Huang 1991). The response times of tens of seconds easily satisfy most HVAC application requirements. These humidity sensors need frequent calibration, generally yearly. If a sensor becomes saturated or has condensation on the surface, it becomes uncalibrated and exhibits an offset from its calibration curve. Older technologies used ionic salts on gold grids. These expensive sensors failed frequently.

Pressure measurements are made by electronic devices that depend on a change of resistance or capacitance with imposed pressure. Figure 11.11 shows a cross-sectional drawing of each. In the resistance type, stretching of the membrane lengthens the resistive element, thereby increasing resistance. This resistor is an element in a Wheatstone bridge; the resulting bridge voltage imbalance is linearly related to the imposed pressure. The capacitative-type unit has a capacitance between a fixed and a flexible metal that decreases with pressure. The capacitance change is amplified by a local amplifier that produces an output signal proportional to pressure. Pressure sensors can burst from overpressure or a water-hammer effect. Installation must carefully follow the manufacturer's requirements.

DDC systems require flow measurements to determine the energy flow for air and water delivery systems. Pitot tubes (or arrays of tubes) and other flow measurement devices can be used to measure either air or liquid flow in HVAC systems. Air flow measurements allow for proper flow in variable air volume (VAV) system control, building pressurization control and outside air control. Water flow measurements enable chiller and boiler control, and monitoring of various water loops used in the HVAC system. Some controls require only the knowledge of flow being present. Open–closed sensors fill this need and typically have a paddle that makes a switch connection in the presence of flow. These types of switches can also be used to detect "end of range," i.e., fully open or closed for dampers and other mechanical control elements.

Temperature, humidity and pressure transmitters are often used in HVAC systems. They amplify signals produced by the basic devices described in the preceding paragraphs and produce an electrical signal over a standard range, thereby permitting standardization of this aspect of DDC systems. The standard ranges are

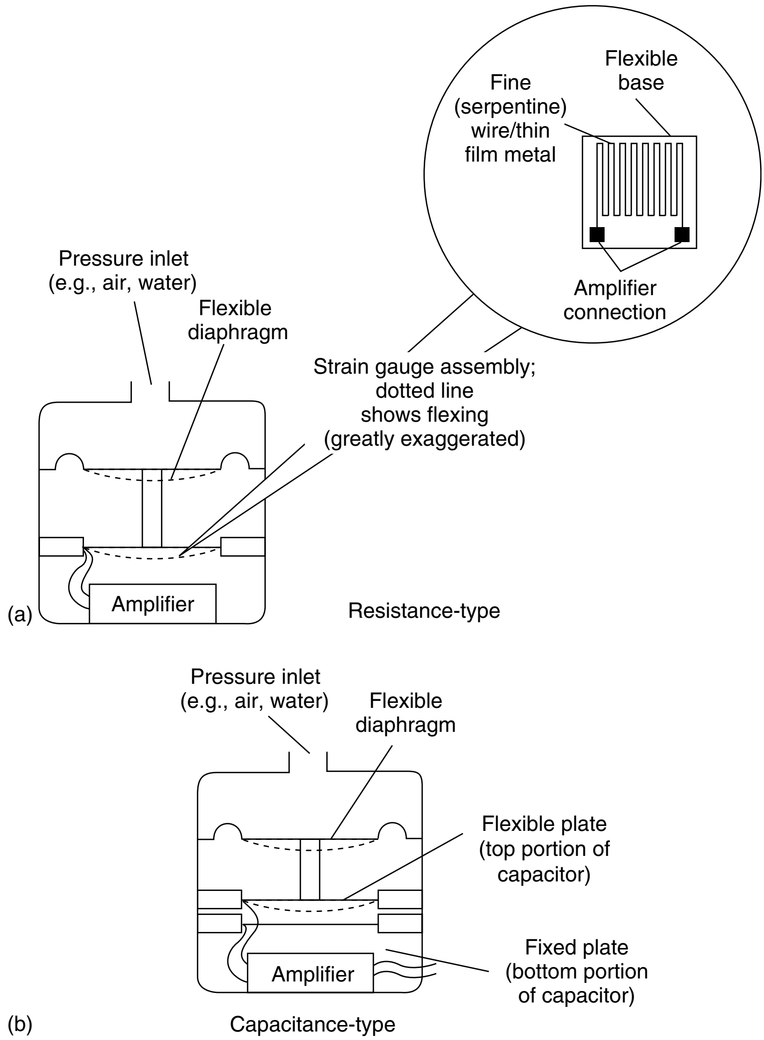

FIGURE 11.11 Resistance and capacitance type pressure sensors.

Current: 4–20 ma (DC)

Voltage: 0–10 volts (DC)

Although the majority of transmitters produce such signals, the noted values are not universally used.

Figure 11.10 shows the elements of a DDC controller. The heart of the controller is a microprocessor that can be programmed in either a standard or system-specific language. Control algorithms (linear or not), sensor calibrations, output signal shaping, and historical data archiving can be programmed as the user requires. A number of firms have constructed controllers on standard personal computer platforms. It is beyond the scope of this chapter to describe the details of programming HVAC controllers because each manufacturer uses a different approach. The essence of any DDC system, however, is the same as shown in the figure. Honeywell (1988) discusses DDC systems and their programming in more detail. Actuators for electronic control systems include

- Motors—operate valves, dampers
- Variable speed controls—pump, fan, chiller drives

- Relays and motor starters—operate other mechanical or electrical equipment (pumps, fans, chillers, compressors), electrical heating equipment
- Transducers—for example, convert electrical signal to pneumatic (EP transducer)
- Visual displays—not actuators in the usual sense, but used to inform system operator of control and HVAC system function

Pneumatic and DDC systems have their own advantages and disadvantages. Pneumatic systems possess increasing disadvantages of cost, hard to find replacements, requiring an air compressor with clean oil-free air, sensor drift, and imprecise control. The retained advantages include explosion-proof operation and a fail-soft degradation of performance. DDC systems have emerged and have taken the lead over pneumatic controls for HVAC systems because of the ability to integrate the control system into a large energy management and control system (EMCS), the accuracy of the control, and the ability to diagnose problems remotely. Systems based on either technology require maintenance and skilled operators.

11.4 Basic Control System Design Considerations

This section discusses selected topics in control system design, including control system zoning, valve and damper selection, and control logic diagrams. The following section shows several HVAC system control design concepts. Bauman (1998) may be consulted for additional information.

The ultimate purpose of an HVAC control system is to control zone temperature (and secondarily air motion and humidity) to conditions that assure maximum comfort and productivity of the occupants. From a controls viewpoint, the HVAC system is assumed to be able to provide comfort conditions if controlled properly. A zone is any portion of a building having loads that differ in magnitude and timing sufficiently from those of other areas, such that separate portions of the secondary HVAC system and control system are needed to maintain comfort.

Having specified the zones, the designer must select the location for the thermostat (and other sensors, if used). Thermostat signals are either passed to the central controller or used locally to control the amount and temperature of conditioned air or coil water introduced into a zone. The air is conditioned either locally (e.g., by a unit ventilator or baseboard heater) or centrally (e.g., by the heating and cooling coils in the central air handler). In either case, a flow control actuator is controlled by the thermostat signal. In addition, airflow itself may be controlled in response to zone information in VAV systems. Except for variable speed drives used in variable volume air or liquid systems, flow is controlled by valves or dampers. The design selection of valves and dampers is discussed next.

11.4.1 Steam and Liquid Flow Control

The flow through valves such as that shown in Figure 11.7 is controlled by valve stem position, which determines the flow area. The variable flow resistance offered by valves depends on their design. The flow characteristic may or may not be linear with position. Figure 11.12 shows flow characteristics of the three most common valve types. Note that the plotted characteristics apply only for constant valve pressure drop. The characteristics shown are idealizations of actual valves. Commercially available valves will resemble, but not necessarily exactly match, the curves shown.

The linear valve has a proportional relation between volumetric flow, $\dot{V}$, and valve stem position, z.

$$\dot{V} = kz. \tag{11.11}$$

The flow in equal percentage valves increases by the same fractional amount for each increment of opening. In other words, if the valve is opened from 20 to 30% of full travel, the flow will increase by the same percentage as if the travel had increased from 80 to 90% of its full travel. However, the absolute volumetric flow increase for the latter case is much greater than for the former. The equal percentage

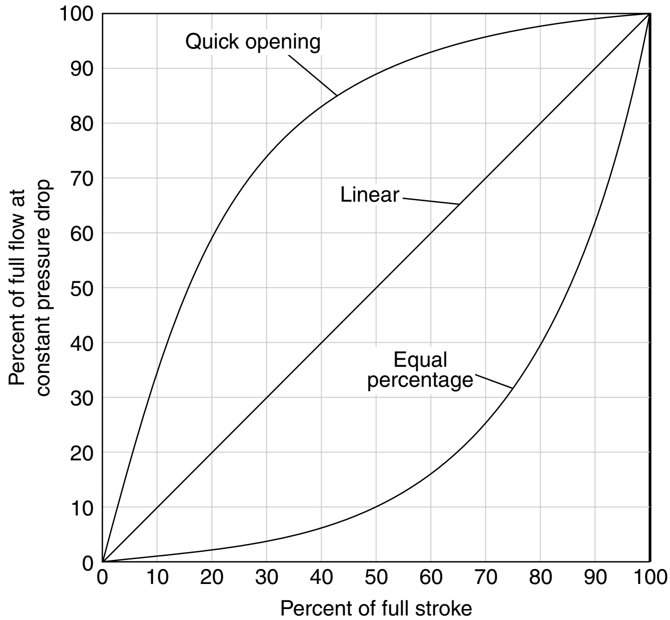

FIGURE 11.12 Quick opening, linear and equal percentage valve characteristics.

valve flow characteristic is given by

$$\dot{V} = K\mathrm{e}^{(kz)}, \tag{11.12}$$

in which k and K are proportionality constants for a specific valve. Quick-opening valves do not provide good flow control, but are used when rapid action is required with little stem movement for on/off control.

Example 11.2 Equal Percentage Valve

Problem: A valve at 30% of travel has a flow of 4 gal/min. If the valve opens another 10% and the flow increases by 50% to 6 gal/min, what are the constants in Equation 11.12? What will be the flow at 50% of full travel? (See Figure 11.12.)

Assumptions: Pressure drop across the valve remains constant.

Solution: The problem is to determine k, K, $\dot{V}_{50}$. Equation 11.12 can be evaluated at the two flow conditions. If the results are divided by each other, then

$$\frac{\dot{V}_2}{\dot{V}_1} = \frac{6}{4} = \mathrm{e}^{k(z_2 - z_1)} = \mathrm{e}^{k(0.4-0.3)}. \tag{11.13}$$

In this expression, the travel, z, is expressed as a fraction of the total travel and is dimensionless. Solving this equation for k gives the result

$$k = 4.05 \text{ (no units)}$$

From the known flow at 30% travel, the second constant, K, can be determined:

$$K = \frac{4 \text{ gal/min}}{\mathrm{e}^{4.05 \times 0.3}} = 1.19 \text{ gal/min}. \tag{11.14}$$

Finally, the flow is given by

$$\dot{V} = 1.19e^{4.05z}. \tag{11.15}$$

At 50% travel, the flow can be found from

$$\dot{V}_{50} = 1.19e^{4.05\times 0.5} = 9.0 \text{ gal/min}. \tag{11.16}$$

Comments: This result can be checked because the valve is an equal percentage valve. At 50% travel, the valve has moved 10% beyond its 40% setting, at which the flow was 6 gal/min. Another 10% stem movement will result in another 50% flow increase from 6 gal/min to 9 gal/min, confirming the solution.

The plotted characteristics of all three valve types assume constant pressure drop across the valve. In an actual system, the pressure drop across a valve will not remain constant, but if the valve is to maintain its control characteristics, the pressure drop across it must be the majority of the entire loop pressure drop. If the valve is designed to have a full open-pressure drop equal to that of the balance of the loop, good flow control will exist. This introduces the concept of valve authority, defined as valve pressure drop as a fraction of total system pressure drop:

$$A \equiv \frac{\Delta p_{\text{v,open}}}{(\Delta p_{\text{v,open}} + \Delta p_{\text{system}})}. \tag{11.17}$$

For proper control, the full-open valve authority should be at least 0.5. If the authority is 0.5 or more, control valves will have installed characteristics not much different from those shown in Figure 11.12. If not, the valve characteristic will be distorted upward because the majority of the system pressure drop will be dissipated across the valve.

Valves are further classified by the number of connections or ports. Figure 11.13 shows sections of typical two-way and three-way valves. Two-port valves control flow through coils or other HVAC equipment by varying valve flow resistance as a result of flow area changes. As shown, the flow must oppose the closing of the valve. If not, near closure the valve would slam shut or oscillate, both of which cause excessive wear and noise. The three-way valve shown in the figure is configured in the diverting mode. That is, one stream is split into two, depending on the valve opening. The three-way valve shown is double seated (single-seated three-way valves are also available); it is therefore easier to close than a single-seated valve, but tight shutoff is not possible.

Three-way valves can also be used as mixing valves. In this application two streams enter the valve, and one leaves. Because their internal design is different, mixing and diverting valves cannot be used interchangeably, to ensure that they can each seat properly. Particular attention is needed by the installer to be sure that connections are made properly; arrows cast in the valve body show the proper flow direction. Figure 11.14 shows an example of three-way valves for both mixing and diverting applications.

Valve flow capacity is denoted in the industry by the dimensional flow coefficient, C_v, defined by

$$\dot{V}(\text{gal/min}) = C_v[\Delta p(\text{psi})]^{0.5}. \tag{11.18}$$

Δp is the pressure drop across the fully open valve, so C_v is specified as the flow rate of 60°F water that will pass through the fully open valve if a pressure difference of 1.0 psi is imposed across the valve. If SI units (m^3/s and Pa) are used, the numerical value of C_v is 17% larger than in USCS units. After the designer has determined a value of C_v, manufacturer's tables can be consulted to select a valve for the known pipe size. If a fluid other than water is to be controlled, the C_v found from Equation 11.18 should be multiplied by the square root of the fluid's specific gravity.

Steam valves are sized using a similar dimensional expression

$$\dot{m}(\text{lb/h}) = 63.5C_v[\Delta p(\text{psi})/v(\text{ft.}^3/\text{lb})]^{0.5}, \tag{11.19}$$

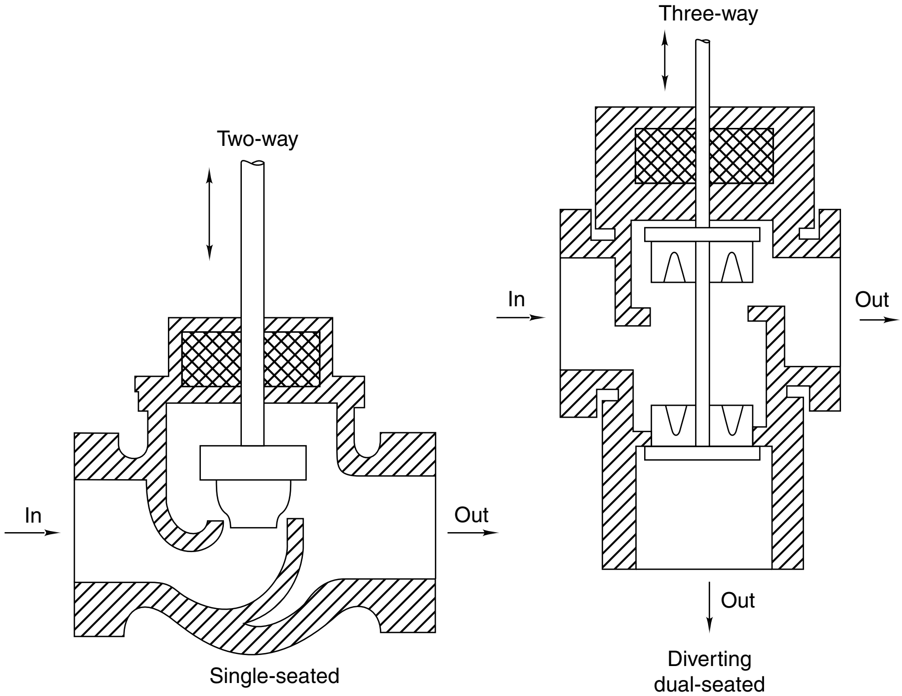

FIGURE 11.13 Cross-sectional drawings of direct-acting, single-seated, two-way valve and dual-seated, three-way, diverting valve.

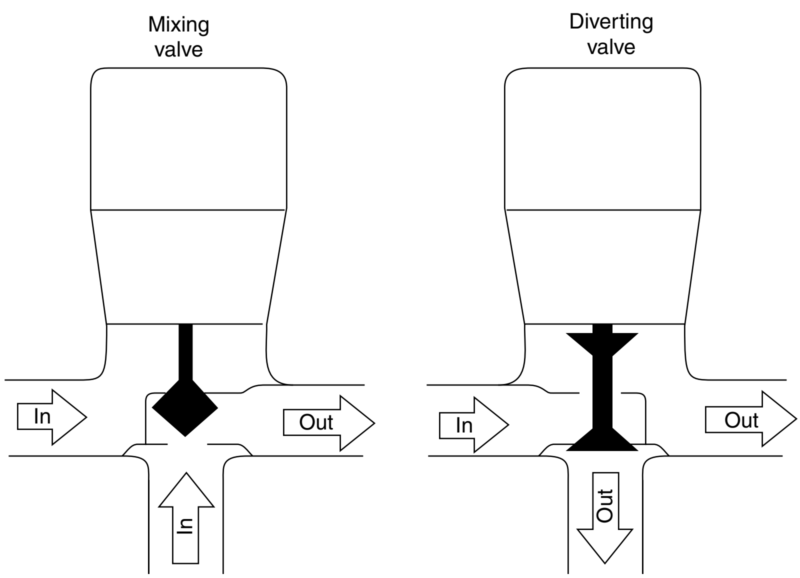

FIGURE 11.14 Three-way mixing and diverting valves. Note the significant difference in internal construction. Mixing valves are more commonly used.

in which v is the steam specific volume. If the steam is highly superheated, multiply C_v found from Equation 11.19 by 1.07 for every 100°F of superheat. For wet steam, multiply C_v by the square root of the steam quality. Honeywell (1988) recommends that the pressure drop across the valve to be used in the equation be 80% of the difference between steam supply and return pressures (subject to the sonic flow limitation discussed below). Table 11.1 can be used for preliminary selection of control valves for either steam or water.

The type of valve (linear or not) for a specific application must be selected so that the controlled system is as nearly linear as possible. Control valves are very commonly used to control the heat transfer rate in coils. For a linear system, the combined characteristic of the actuator, valve, and coil should be linear. This will require quite different valves for hot water and steam control, for example.

Figure 11.15 shows the part load performance of a hot water coil used for air heating; at 10% of full flow the heat rate is 50% of its peak value. The heat rate in a cross-flow heat exchanger increases roughly in exponential fashion with flow rate—a highly nonlinear characteristic. This heating coil nonlinearity follows from the longer water residence time in a coil at reduced flow, and the relatively large temperature difference between air being heated and the water heating it.

However, if one were to control the flow through this heating coil by an equal percentage valve (positive exponential increase of flow with valve position), the combined valve plus coil characteristic would be roughly linear. Referring to Figure 11.15, 50% of stem travel corresponds to 10% flow. The third graph in the figure is the combined characteristic. This near-linear subsystem is much easier to control than if a linear valve were used with the highly nonlinear coil. Hence the rule: use equal percentage valves for heating coil control.

Linear, two-port valves are to be used for steam flow control to coils because the transfer of heat by steam condensation is a linear, constant temperature process—the more steam supplied, the greater the heat rate, in exact proportion. Note that this is a completely different coil flow characteristic than for hot-water coils. However, steam is a compressible fluid and the sonic velocity sets the flow limit for a given valve opening when the pressure drop across the valve is more than 60% of the steam supply line absolute pressure. As a result, the pressure drop to be used in Equation 11.19 is the *smaller* of (1) 50% of the absolute stream pressure upstream of the valve or (2) 80% of the difference between the steam supply and return line pressures. The 80% rule gives good valve modulation in the subsonic flow regime (Honeywell 1988).

Chilled water control valves should also be linear because the performance of chilled water coils (smaller air–water temperature difference than in hot-water coils) is more similar to steam coils than to hot-water coils.

Either two- or three-way valves can be used to control flow at part load through heating and cooling coils as shown in Figure 11.16. The control valve can be controlled from either coil outlet water or air temperature. Two- or three-way valves achieve the same local result at the coil when used for part load control. However, the designer must consider effects on the balance of the secondary system when selecting the valve type.

In essence, the two-way valve flow control method results in variable flow (tracking variable loads) with constant coil water temperature change, whereas the three-way valve approach results in roughly constant secondary loop flow rate, but smaller coil water temperature change (beyond the local coil loop itself). In large systems, a primary/secondary design with two-way valves is preferred, unless the primary equipment can handle the range of flow variation that will result without a secondary loop. Because chillers and boilers require that flow remain within a restricted range, the energy and cost savings that could accrue due to the two-way valve, variable volume system are difficult to achieve in small systems unless a two-pump, primary/secondary loop approach is employed. If this dual-loop approach is not used, the three-way valve method is required to maintain required boiler or chiller flow.

The location of the three-way valve at a coil must also be considered by the designer. Figure 11.16b shows the valve used downstream of the coil in a mixing, bypass mode. If a balancing valve is installed in the bypass line, and set to have the same pressure drop as the coil, the local coil loop will have the same pressure drop for both full and zero coil flow. However, at the valve mid-flow position, overall flow resistance is less,

TABLE 11.1 Quick sizing Chart for Control Valves

	Steam Capacity, lb/h												
	Vacuum Return Systems[a]			Atmospheric Return Systems			Water Capacity, gal/min						
	2-psi Supply Press.	5-psi Supply Press.	10-psi Supply Press.	2-psi Supply Press.	5-psi Supply Press.	10-psi Supply Press.	Differential Pressure, psig						
Cv	3.2-psi Press. Drop[b]	5.6-psi Press. Drop[b]	9.6-psi Press. Drop[b]	1.6-psi Press. Drop[b]	4.0-psi Press. Drop[b]	8.0-psi Press. Drop[b]	2	4	6	8	10	15	20
0.33	7.7	11.0	16.0	5.4	9.3	14.6	0.41	0.66	0.81	0.93	1.04	1.27	1.47
0.63	14.6	20.9	30.5	10.4	17.7	27.8	0.89	1.26	1.54	1.78	1.99	2.4	2.81
0.73	17.0	24.3	35.4	12	20.5	32.2	1.0	1.46	1.78	2.06	2.3	2.8	3.25
1.0	23.0	33.2	48.5	16.4	28	44	1.4	2.0	2.44	2.82	3.16	3.9	4.46
1.6	37.09	53.1	77.6	26.8	45	70.6	2.25	3.2	3.9	4.51	5.06	6.2	7.13
2.5	58.25	82.9	121.2	41.9	70.25	110.25	3.53	5.0	6.1	7.05	7.9	9.68	11.15
3.0	69.9	99.5	145.5	50.2	84.3	132.3	4.23	6.0	7.32	8.46	9.48	11.61	13.38
4.0	93.2	132.2	194.0	67	112.4	177.4	5.6	8.0	9.76	11.28	12.6	15.5	17.87
5.0	116.2	165.2	242.5	82.7	140.5	220.5	7.1	10.0	12.2	14.1	15.8	19.4	22.3
6.0	139	200	291.0	99	168	265	8.5	12.0	14.6	16.92	18.9	23.2	27.0
6.3	146	209	311.5	104	177	278	8.9	12.6	15.4	17.78	19.9	24.4	28.1
7.0	162	233	339.5	115	196	309	9.9	14.0	17.1	19.74	22.1	27.1	31
8.0	186.5	264.4	388.0	131.2	224.8	352.8	11.3	16.0	19.5	22.56	25.3	31.6	35.7
10.0	232	332	485.0	164	281	441	14.1	20	24.4	28.2	31.6	38.7	44.6
11.0	256	366	533.5	181	309	486	15.5	22	27	31.02	34.4	42.5	49
13.0	303	434	630.5	213.7	365.3	373.3	18.3	27	31.7	36.7	41.1	50.3	58
14.0	326	465	679.0	232	393	617	19.7	28	34	39	44	54	62
15.0	349.3	497.6	727.5	246	421.5	661.5	21.1	30	36.6	42.3	47.4	58	66.9
16.0	370.9	531	776.0	268	450	706	22.5	32	39	45.1	50.6	62	71.3
18.0	419	597	873.0	301	505	794	25	36	44	51	57	70	80
20.0	466	664	970.0	335	562	882	28	40	49	56	63	77	89
23.0	541	763	1,115	385	646	1,014	32	46	56	65	73	89	103
25.0	582.5	829	1,212	419	702.5	1,102.5	35.3	50	61	70.5	79	96.8	111.5
27.0	628.2	896	1,309	452.5	758.7	1,190.7	38.1	54	65.9	76.1	85.3	104.5	120.4
30.0	699	995	1,455	502	843	1,323	42.3	60	73.2	84.6	94.8	116.1	133.8
38.0	885	1,257	1,833	636	1,069	1,676	53	76	93	107	120	147	169
40.0	932	1,322	1,940	670	1,124	1,764	56	80	97.6	112.8	126	155	178.7
50.0	1,162	1,652	2,425	827	1,405	2,205	71	100	122	141	158	194	223
56.0	1,305	1,851	2,716	938	1,574	2,469	79	112	137	158	177	217	250

63.0	1,460	2,090	3,056	1,043	1,770	2,778	89	126	154	178	199	244	281
75.0	1,748	2,481	3,637	1,230	2,107	3,307	106	150	183	212	237	290	335
80.0	1,865	2,644	3,880	1,312	2,248	3,528	113	160	195	225.6	253	316	357
90.0	2,096	2,980	4,365	1,476	2,529	3,969	127	180	220	254	284	348	401
97.0	2,229	3,204	4,703	1,590	2,725	4,277	137	196	231	274	307	375	432
100.0	2,330	3,319	4,850	1,640	2,816	4,410	141	200	244	282	316	387	446
105.0	2,442	3,481	5,092	1,722	2,950	4,630	148	210	256	296	332	406	468
130.0	3,030	4,340	6,305	2,137	3,653	5,733	183	270	317	367	411	503	580
150.0	3,493	4,976	7,275	2,460	4,215	6,615	211	300	366	423	474	280	699
160.0	3,709	5,310	7,760	2,680	4,500	7,060	225	320	390	451	560	620	713
170.0	3,960	5,642	8,245	2,788	4,777	7,497	240	340	415	479	537	658	758
190.0	4,450	6,310	9,215	3,116	5,339	8,379	268	360	464	536	600	735	847
244.0	5,670	7,930	11,834	4,001	6,856	10,760	344	488	595	688	771	944	1,088
250.0	5,825	8,290	12,125	4,190	7,025	11,025	353	500	610	705	790	968	1,115
270.0	6,282	8,960	13,095	4,525	7,587	11,907	381	540	659	761	853	1,045	1,204
300.0	6,990	9,950	14,550	5,025	8,430	13,230	423	600	732	846	948	1,161	1,338
350.0	8,160	11,590	16,975	5,860	9,835	15,435	494	700	854	987	1,106	1,355	1,561
360.0	8,380	11,910	17,460	6,030	10,116	15,876	508	720	878	1,015	1,137	1,393	1,606
430.0	10,010	14,225	20,855	7,200	12,083	18,963	606	860	1,049	1,213	1,359	1,664	1,918
480.0	11,180	15,860	23,280	8,045	13,408	21,168	677	960	1,171	1,353	1,517	1,858	2,141
640.0	14,910	21,180	31,040	10,496	17,984	28,224	902	1,280	1,561	1,805	2,022	2,477	2,854
760.0	17,70	25,120	36,860	12,464	21,356	33,516	1,071	1,520	1,854	2,143	2,401	2,941	3,390
1,000.0	23,300	33,190	48,500	16,400	28,160	44,100	1,410	2,000	2,440	2,820	3,160	3,870	4,460
1,200.0	27,150	39,790	58,200	19,680	33,720	52,920	1,692	2,400	2,928	3,384	2,792	4,644	5,352
1,440.0	33,290	47,160	69,840	23,616	40,464	63,504	2,030	2,880	3,514	4,061	4,550	5,573	6,422

[a] Assuming a 4-in. through 8-in. vacuum.
[b] Pressure drop across fully open valve taking 80% of the pressure difference between supply and return main pressures.
Source: From Honeywell, Inc., *Engineering Manual of Automatic Control*, Honey well, Inc., Minneapolis, MN, 1988.

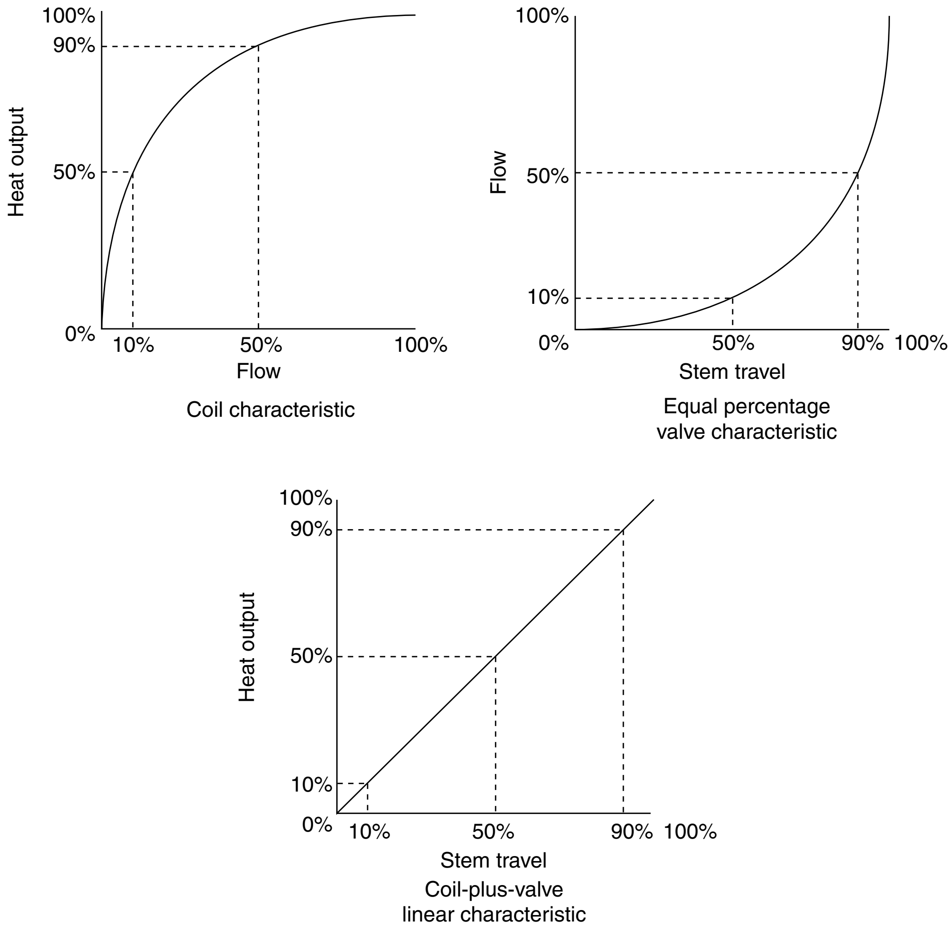

FIGURE 11.15 Heating coil, equal percentage valve, and combined coil + valve linear characteristic.

because two parallel paths are involved, and the total loop flow increases to 25% more than that at either extreme.

Alternatively, the three-way valve can also be used in a diverting mode, as shown in Figure 11.16c. In this arrangement, essentially the same considerations apply as for the mixing arrangement discussed earlier.[1] However, if a circulator (small pump) is inserted, as shown in Figure 11.16d, the direction of flow in the branch line changes and a mixing valve is used. The reason that pumped coils are used is that control is improved. With constant coil flow, the highly nonlinear coil characteristic shown in Figure 11.15 is reduced because the residence time of hot water in the coil is constant, independent of load. However, this arrangement appears to the external secondary loop the same as a two-way valve. As load is decreased, flow into the local coil loop also decreases. Therefore, the uniform secondary loop flow normally associated with three-way valves is not present unless the optional bypass is used.

[1]A little-known disadvantage of three-way valve control has to do with the conduction of heat from a closed valve to a coil. For example, the constant flow of hot water through two ports of a closed three-way heating coil control valve keeps the valve body hot. Conduction from the closed, hot valve mounted close to a coil can cause sufficient air heating to actually decrease the expected cooling rate of a downstream cooling coil during the cooling season. Three-way valves have a second practical problem; installers often connect three-way valves incorrectly, given the choice of three pipe connections and three pipes to be connected. Both of these problems are avoided by using two-way valves.

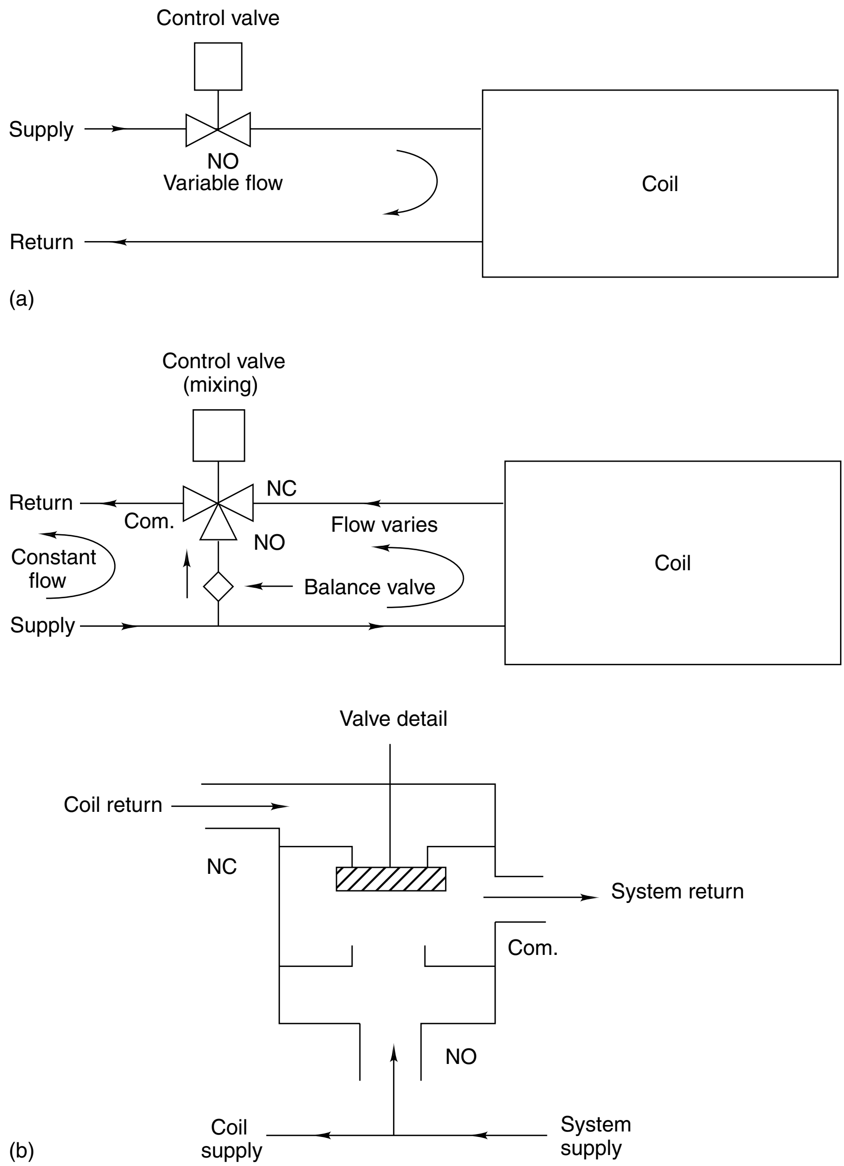

FIGURE 11.16 Various control valve piping arrangements (a) two-way valve; (b) three-way mixing valve; (c) three-way diverting valve; (d) pumped coil with three-way mixing valve.

For HVAC systems requiring precise control, high-quality control valves are required. The best controllers and valves are of "industrial quality." The additional cost for these valves compared to conventional building hardware results in more accurate control and longer lifetime.

11.4.2 Air Flow Control

Dampers are used to control airflow in secondary HVAC air systems in buildings. In this section, the characteristics of dampers used for flow control in systems where constant speed fans are involved are discussed. Figure 11.17 shows cross sections of the two common types of dampers used in commercial buildings. Parallel blade dampers use blades that all rotate in the same direction. They are most often

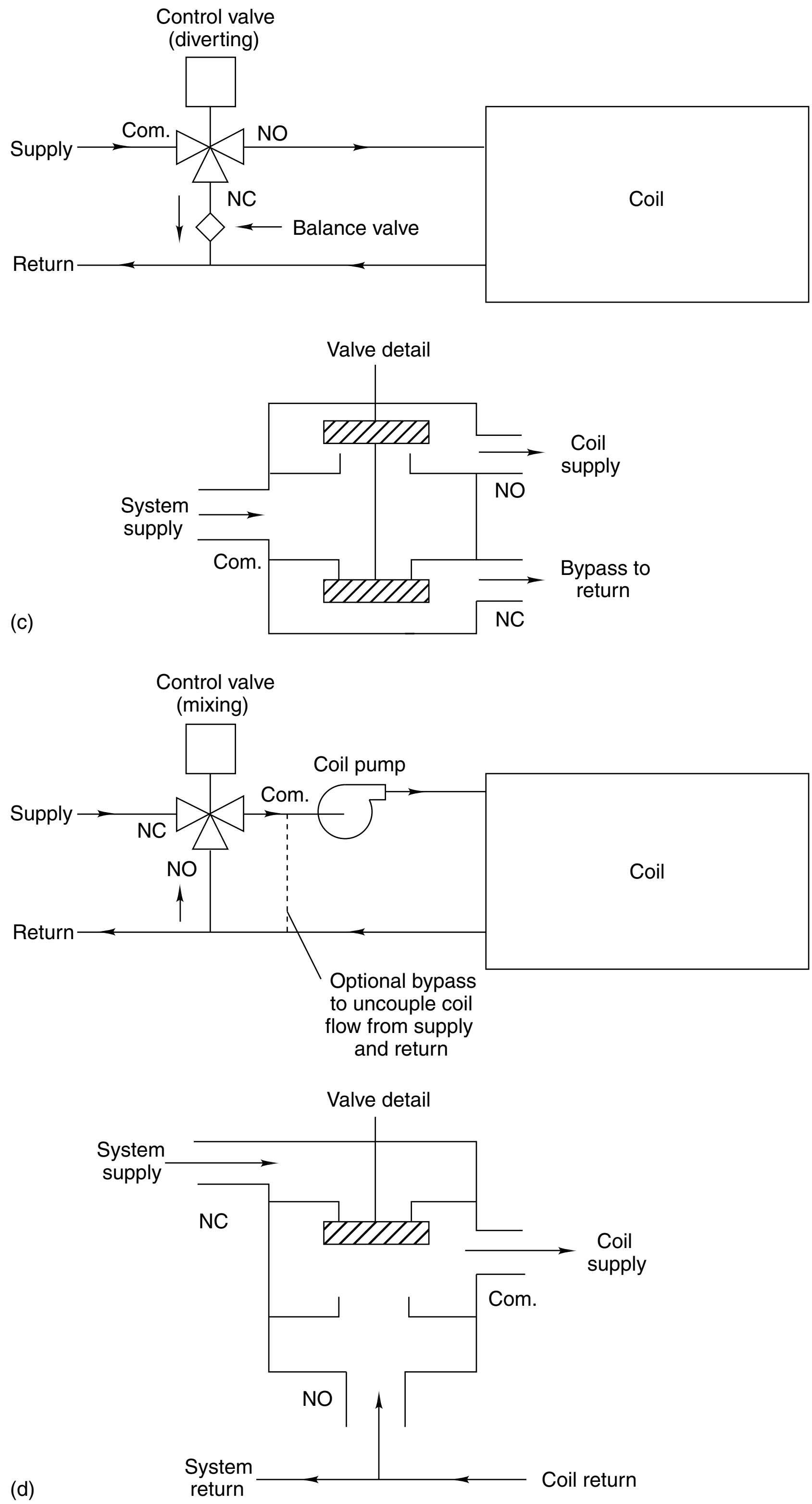

FIGURE 11.16 (*continued*)

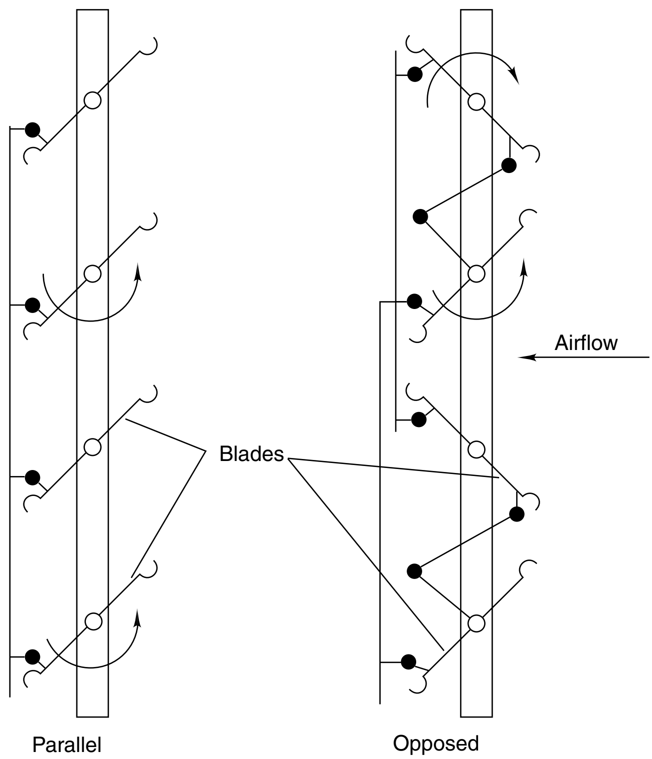

FIGURE 11.17 Diagram of parallel and opposed blade dampers.

applied to two position locations—open or closed. Use for flow control is not recommended. The blade rotation changes airflow direction, a characteristic that can be useful when airstreams at different temperatures are to be effectively blended.

Opposed blade dampers have adjacent counter-rotating blades. Airflow direction is not changed with this design, but pressure drops are higher than for parallel blading. Opposed blade dampers are preferred for flow control. Figure 11.18 shows the flow characteristics of these dampers to be closer to the desired linear behavior. The parameter α on the curves is the ratio of system pressure drop to fully open damper pressure drop.

A common application of dampers controlling the flow of outside air uses two sets in a face and bypass configuration as shown in Figure 11.19. For full heating, all air is passed through the coil and the bypass dampers are closed. If no heating is needed in mild weather, the coil is bypassed (for minimum flow resistance and fan power cost, flow through fully open face and bypass dampers can be used if the preheat coil water flow is shut off). Between these extremes, flow is split between the two paths. The face and bypass dampers are sized so that the pressure drop in full bypass mode (damper pressure drop only) and full heating mode (coil plus damper pressure drop) is the same.

11.5 Example HVAC Control Systems

Several widely used control configurations for specific tasks are described in this section. These have been selected from the hundreds of control system configurations that have been used for buildings. The goal of this section is to illustrate how control components described above are assembled into systems, and what design considerations are involved. For a complete overview of HVAC control system configurations see Honeywell (1988), Grimm and Rosaler (1990), Sauer, Howell, and Coad (2001), ASHRAE

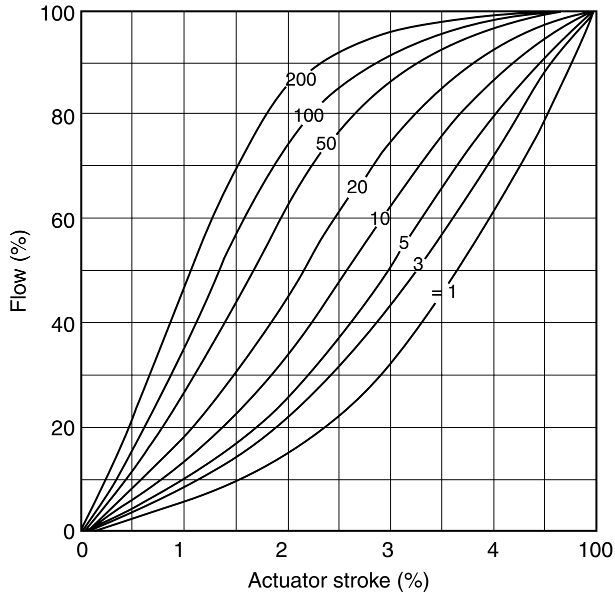

FIGURE 11.18 Flow characteristics of opposed blade dampers. The parameter α is the ratio of system resistance (not including the damper) to damper resistance. An approximately linear damper characteristic is achieved if this ratio is about 10 for opposed blade dampers.

(2002, 2003, 2004), and Tao and Janis (2005). The illustrative systems in this section are drawn in part from the first of these references.

In this section, seven control systems in common use will be discussed. Each system will be described using a schematic diagram, and its operation and key features will be discussed in the accompanying text.

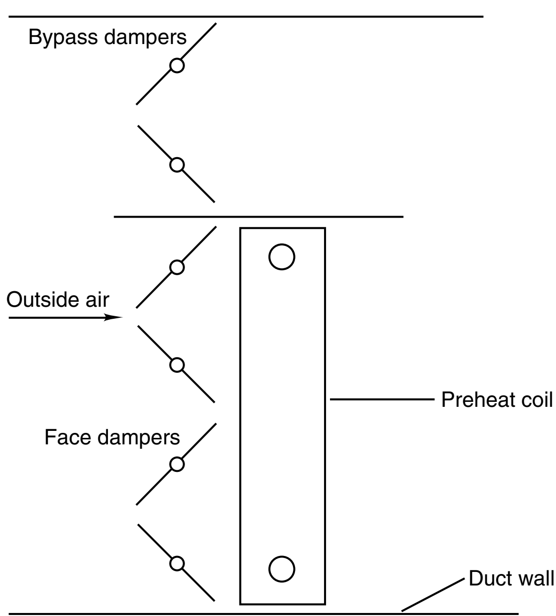

FIGURE 11.19 Face and bypass dampers used for preheating coil control.

11.5.1 Outside Air Control

Figure 11.20 shows a system for controlling outside and exhaust air from a central air handling unit equipped for economizer cooling when available. In this and the following diagrams, the following symbols are used:

C—cooling coil
DA—discharge air (supply air from fan)
DX—direct-expansion coil
E—damper controller
EA—exhaust air
H—heating coil
LT—low-temperature limit sensor or switch, must sense the lowest temperature in the air volume being controlled
M—motor or actuator (for damper or valve), variable speed drive
MA—mixed air
NC—normally closed
NO—normally open
OA—outside air
PI—proportional plus integral controller
R—relay
RA—return air
S—switch
SP—static pressure sensor used in VAV systems

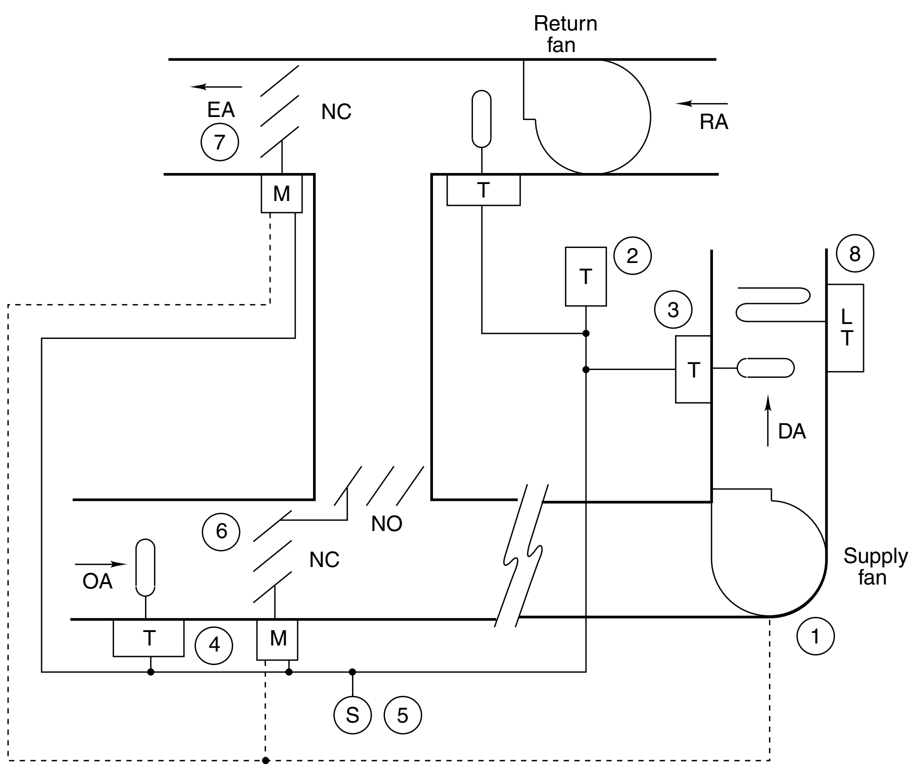

FIGURE 11.20 Outside-air-control system with economizer capability.

T—temperature sensor; must be located to read the average temperature representative of the air volume being controlled

This system is able to provide the minimum outside air during occupied periods; to use outdoor air for cooling when appropriate, by means of a temperature-based economizer cycle; and to operate fans and dampers under all conditions. The numbering system used in the figure indicates the sequence of events as the air handling system begins operation after an off period:

1. The fan control system turns on when the fan is turned on. This may be by a clock signal or a low- or high-temperature space condition.
2. The space temperature signal determines whether the space is above or below the setpoint. If above, the economizer feature will be activated if the OA temperature is below the upper limit for economizer operation, and will control the outdoor and mixed air dampers. If below, the outside air damper is set to its minimum position.
3. The discharge air PI controller controls both sets of dampers (OA/RA and EA) to provide the desired mixed air temperature.
4. When the outdoor temperature rises above the upper limit for economizer operation, the outdoor air damper is returned to its minimum setting.
5. Switch S is used to set the minimum setting on outside and exhaust air dampers manually. This is ordinarily done only once, during building commissioning and flow testing.
6. When the supply fan is off, the outdoor air damper returns to its NC position and the return air damper returns to its NO position.
7. When the supply fan is off, the exhaust damper also returns to its NC position.
8. Low temperature sensed in the duct will initiate a freeze-protect cycle. This may be as simple as turning on the supply fan to circulate warmer room air. Of course, the OA and EA dampers remain tightly closed during this operation.

11.5.2 Heating Control

If the minimum air setting is large in the preceding system, the amount of outdoor air admitted in cold climates may require preheating. Figure 11.21 shows a preheating system using face and bypass dampers. (A similar arrangement is used for direct-expansion [DX] cooling coils.) The equipment shown is installed upstream of the fan in Figure 11.20. This system operates as follows:

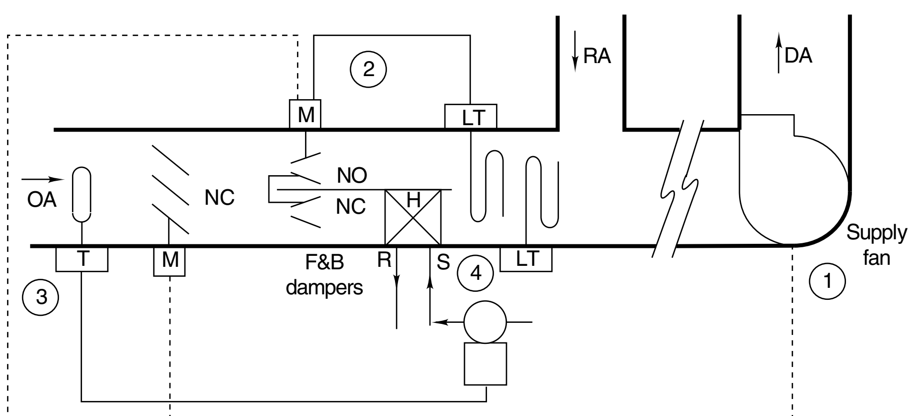

FIGURE 11.21 Preheat control system. Counter flow of air and hot water in the preheat coil results in the highest heat transfer rate.

1. The preheat subsystem control is activated when the supply fan is turned on.
2. The preheat PI controller senses temperature leaving the preheat section. It operates the face and bypass dampers to control the exit air temperature between 45 and 50°F.
3. The outdoor air sensor and associated controller controls the water valve at the preheat coil. The valve may be either a modulating valve (better control) or an on–off valve (less costly).
4. The low-temperature sensors (LTs) activate coil freeze protection measures, including closing dampers and turning off the supply fan.

Note that the preheat coil (as well as all other coils in this section) is connected so that the hot water (or steam) flows counter to the direction of airflow. Counter flow provides a higher heating rate for a given coil than does parallel flow. Mixing of heated and cold bypass air must occur upstream of the control sensors. Stratification can be reduced by using sheet metal air blenders or by propeller fans in the ducting. The preheat coil should be located in the bottom of the duct. Steam preheat coils must have adequately sized traps and vacuum breakers to avoid condensate buildup that could lead to coil freezing at light loads.

The face and bypass damper approach enables air to be heated to the required system supply temperature without endangering the heating coil. (If a coil were to be as large as the duct—no bypass area—it could freeze when the hot water control valve cycles open and closed to maintain discharge temperature.) The designer should consider pumping the preheat coil as shown in Figure 11.19d to maintain water velocity above the 3 ft./s needed to avoid freezing. If glycol is used in the system, the pump is not necessary, but heat transfer will be reduced.

During winter in heating climates, heat must be added to the mixed air stream to heat the outside air portion of mixed air to an acceptable discharge temperature. Figure 11.22 shows a common heating subsystem controller used with central air handlers. (It is assumed that the mixed air temperature is kept above freezing by action of the preheat coil, if needed.) This system has the added feature that coil discharge temperature is adjusted for ambient temperature because the amount of heat needed decreases with increasing outside temperature. This feature, called *coil discharge reset*, provides better control and can reduce energy consumption. The system operates as follows:

1. During operation the discharge air sensor and PI controller controls the hot water valve.
2. The outside air sensor and controller resets the setpoint of the discharge air PI controller up as ambient temperature drops.

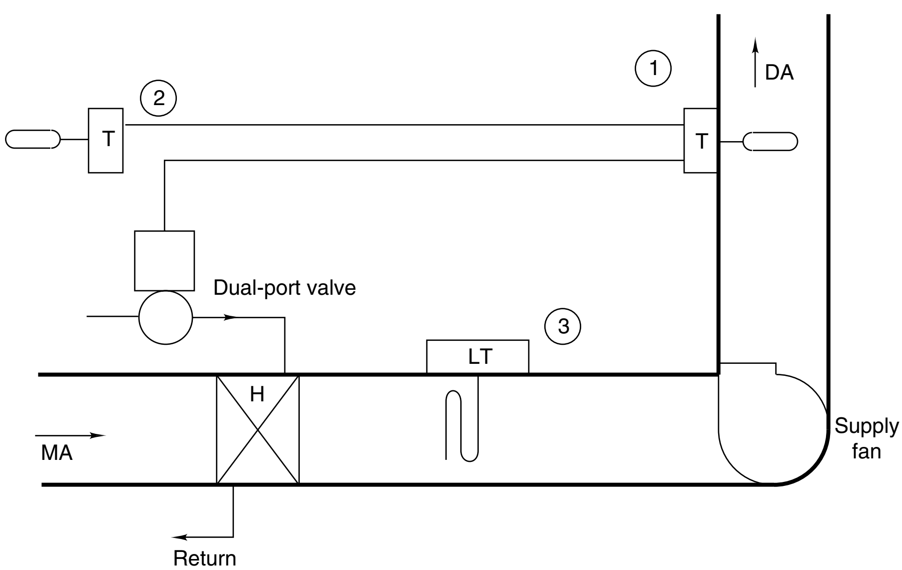

FIGURE 11.22 Heating-coil control subsystem using two-way valve and optional reset sensor.

3. Under sensed low-temperature conditions, freeze-protection measures are initiated as discussed earlier.

Reheating at zones in VAV or other systems uses a system similar to that just discussed. However, boiler water temperature is reset and no freeze protection is normally included. The air temperature sensor is the zone thermostat for VAV reheat, not a duct temperature sensor.

11.5.3 Cooling Control

Figure 11.23 shows the components in a cooling coil control system for a single-zone system. Control is similar to that for the heating coil discussed above, except that the zone thermostat (not a duct temperature sensor) controls the coil. If the system were a central system serving several zones, a duct sensor would be used. Chilled water supplied to the coil partially bypasses and partially flows through the coil, depending on the coil load. The use of three- and two-way valves for coil control has been discussed in detail previously. The valve NC connection is used as shown so that valve failure will not block secondary loop flow.

Figure 11.24 shows another common cooling coil control system. In this case the coil is a direct-expansion (DX) refrigerant coil and the controlled medium is refrigerant flow. DX coils are used when precise temperature control is not required because the coil outlet temperature drop is large whenever refrigerant is released into the coil because refrigerant flow is not modulated; it is most commonly either on or off. The control system sequences as follows:

1. The coil control system is energized when the supply fan is turned on.
2. The zone thermostat opens the two-position refrigerant valve for temperatures above the setpoint and closes it in the opposite condition.
3. At the same time, the compressor is energized or de-energized. The compressor has its own internal controls for oil control and pumpdown.
4. When the supply fan is off, the refrigerant solenoid valve returns to its NC position and the compressor relay to its NO position.

At light loads, bypass rates are high and ice may build up on coils. Therefore, control is poor at light loads with this system.

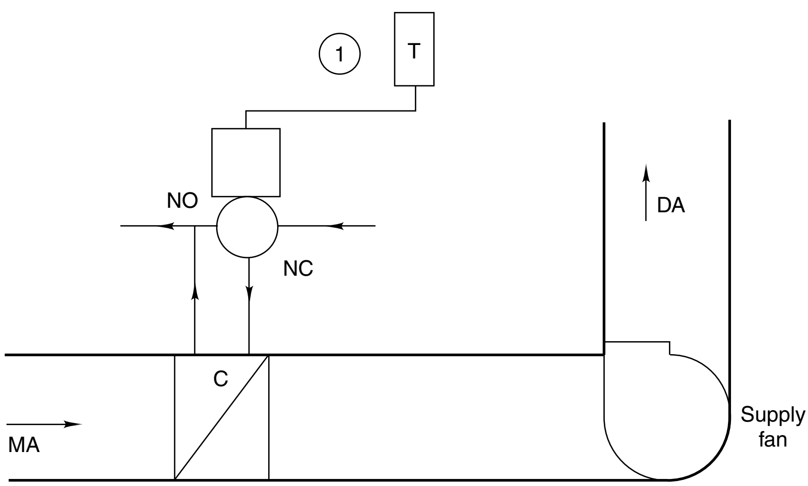

FIGURE 11.23 Cooling-coil control subsystem using three-way diverting valve.

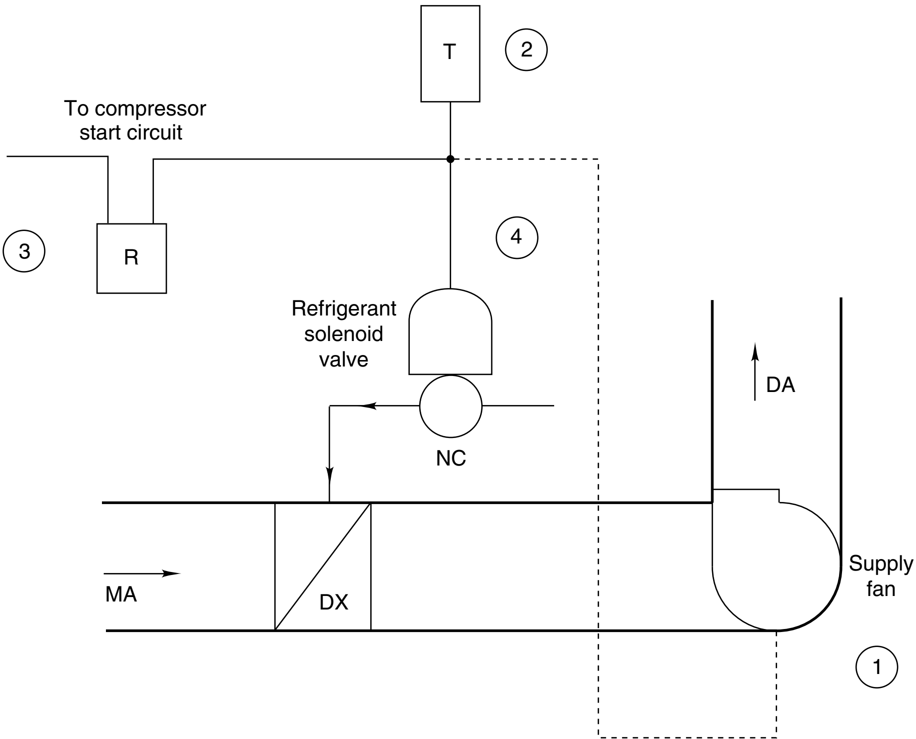

FIGURE 11.24 DX cooling coil control subsystem (on-off control).

11.5.4 Complete Systems

The preceding five example systems are actually control subsystems that must be integrated into a single control system for the HVAC system's primary and secondary systems. In the remainder of this section, two complete HVAC control systems widely used in commercial buildings will be briefly described. The first is a constant volume system, and the second is a VAV system.

Figure 11.25 shows a constant volume, central air-handling system equipped with supply and return fans, heating and cooling coils, and economizer for a single-zone application. If the system were to be used for multiple zones, the zone thermostat shown would be replaced by a discharge air temperature sensor. This constant volume system operates as follows:

1. When the fan is energized, the control system is activated.
2. The minimum outside air setting is set (usually only once, during commissioning, as described above).
3. The OA temperature sensor supplies a signal to the damper controller.
4. The RA temperature sensor supplies a signal to the damper controller.
5. The damper controller positions the dampers to use outdoor or return air, depending on which is cooler.
6. The mixed-air low-temperature controller controls the outside air dampers to avoid excessively low-temperature air from entering the coils. If a preheating system were included, this sensor would control it.
7. The space temperature sensor resets the coil discharge air PI controller.
8. The discharge air controller controls the
   a. Heating coil valve
   b. Outdoor air damper

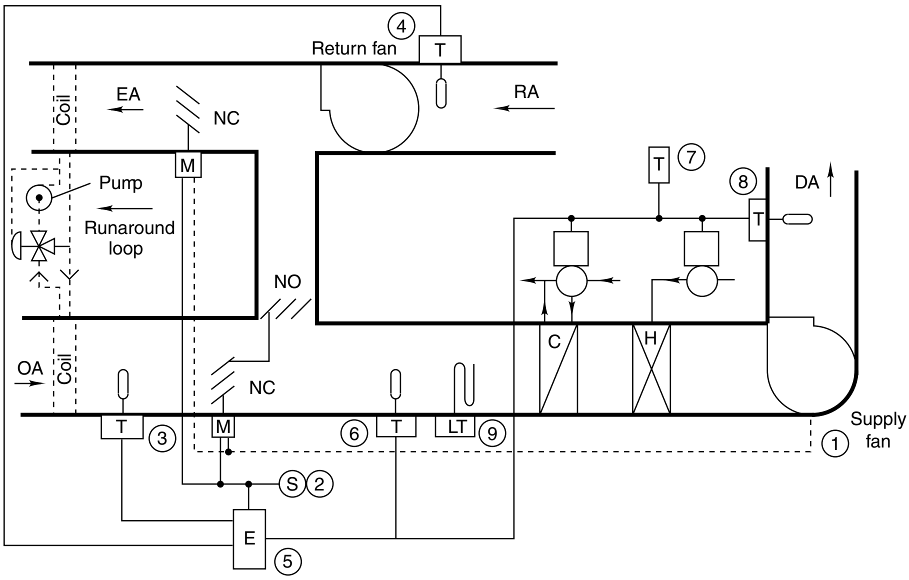

FIGURE 11.25 Control for a complete, constant volume HVAC system. Optional runaround heat recovery system is shown to left in dashed lines.

   c. Exhaust air damper
   d. Return air damper
   e. Cooling coil valve (after the economizer cycle upper limit is reached)
9. The low-temperature sensor initiates freeze protection measures as described previously.

A method for reclaiming either heating or cooling energy is shown by dashed lines on the left side of Figure 11.25. This so-called "runaround" system extracts energy from exhaust air and uses it to precondition outside air. For example, the heating season exhaust air may be at 75°F, while outdoor air is at 10°F. The upper coil in the figure extracts heat from the 75°F exhaust and transfers it through the lower coil to the 10°F intake air. To avoid icing of the air intake coil, the three-way valve controls this coil's liquid inlet temperature to a temperature above freezing. In heating climates, the liquid loop should also be freeze protected with a glycol solution. Heat reclaiming systems of this type can also be effective in the cooling season, when outdoor temperatures are well above indoor temperatures.

A VAV system has additional control features including a motor speed control (or inlet vanes in some older systems) and a duct static pressure control. Figure 11.26 shows a VAV system serving both perimeter and interior zones. It is assumed that the core zones always require cooling during the occupied period. The system shown has a number of options, and does not include every feature present in all VAV systems. However, it is representative of VAV design practice. The sequence of operation during the heating season is as follows:

1. When the fan is energized, the control system is activated. Prior to activation, during unoccupied periods the perimeter zone baseboard heating is under control of room thermostats.
2. Return and supply fan interlocks are used to prevent pressure imbalances in the supply air ductwork.
3. The mixed air sensor controls the outdoor air dampers or preheat coil (not shown) to provide proper coil air inlet temperature. The dampers will typically be at their minimum position at about 40°F.
4. The damper minimum position controls the minimum outdoor airflow.

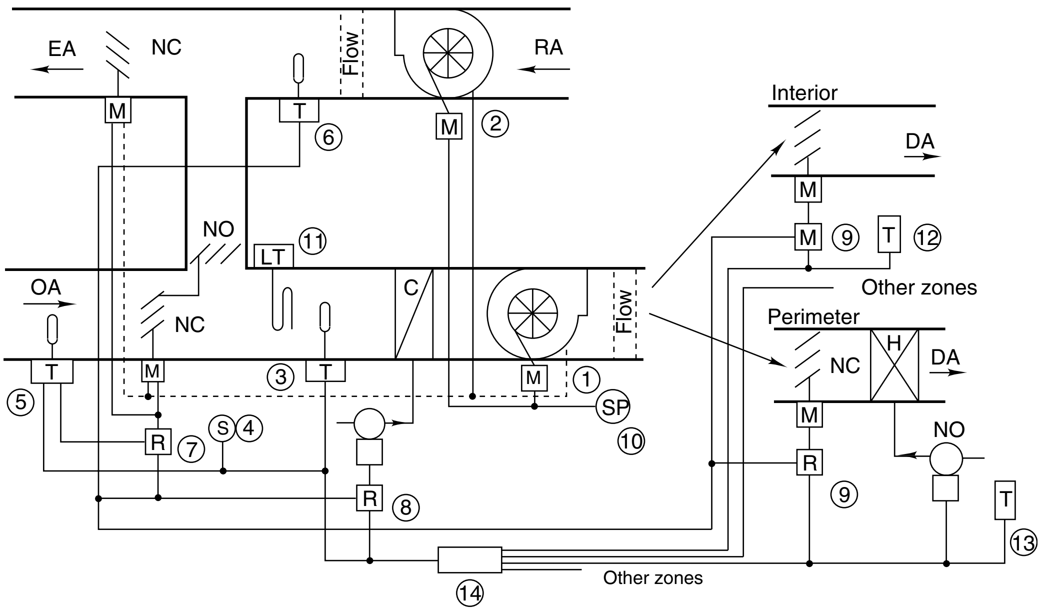

FIGURE 11.26 Control for complete, VAV system. Optional supply and return flow stations shown with dashed lines.

5. As the upper limit for economizer operation is reached, the OA dampers are returned to their minimum position.
6. The return air temperature is used to control the morning warmup cycle after night setback (option present only if night setback is used).
7. The outdoor air damper is not permitted to open during morning warmup, by action of the relay shown.
8. Likewise, the cooling coil valve is de-energized (NC) during morning warmup.
9. All VAV box dampers are moved full open during morning warmup by action of the relay override. This minimizes warmup time. Perimeter zone coils and baseboard units are under control of the local thermostat.
10. During operating periods, the PI static pressure controller controls both supply and return fan speeds (or inlet vane positions) to maintain approximately 1.0 in. WG of static pressure at the pressure sensor location (or, optionally, to maintain building pressure). An additional pressure sensor (not shown) at the supply fan outlet will shut down the fan if fire dampers or other dampers should close completely and block airflow. This sensor overrides the duct static pressure sensor shown.
11. The low-temperature sensor initiates freeze-protection measures.
12. At each zone, room thermostats control VAV boxes (and fans, if present); as zone temperature rises the boxes open more.
13. At each perimeter zone room, thermostats close VAV dampers to their minimum settings and activate zone heat (coil or perimeter baseboard) as zone temperature falls.
14. The controller, using temperature information for all zones (or at least for enough zones to represent the characteristics of all zones), modulates outdoor air dampers (during economizer operation) and the cooling control valve (above the economizer cycle cutoff), to provide air sufficiently cooled to maintain acceptable zone humidity and meet the load of the warmest zone.

The duct static pressure controller is critical to the proper operation of VAV systems. The static pressure controller must be of PI design because a proportional-only controller would permit duct pressure to drift upward as cooling loads drop due to the unavoidable offset in P-type controllers. In addition, the control system should position inlet vanes (if present) closed during fan shutdown, to avoid overloading on restart.

Return fan control is best achieved in VAV systems by an actual flow measurement in supply and return ducts as shown by dashed lines in the figure. The return airflow rate is the supply rate less local exhausts (fume hoods, toilets, etc.) and exfiltration needed to pressurize the building.

VAV boxes are controlled locally, assuming that adequate duct static pressure exists in the supply duct and that supply air is at an adequate temperature to meet the load (this is the function of the controller described in item 11.14). Figure 11.27 shows a local control system used with a series-type, fan-powered VAV box. This particular system delivers a constant flow rate to the zone by action of the airflow controller, to assure proper zone air distribution. Primary air varies with cooling load, as shown in the lower part of the figure. Optional reheating is provided by the coil shown.

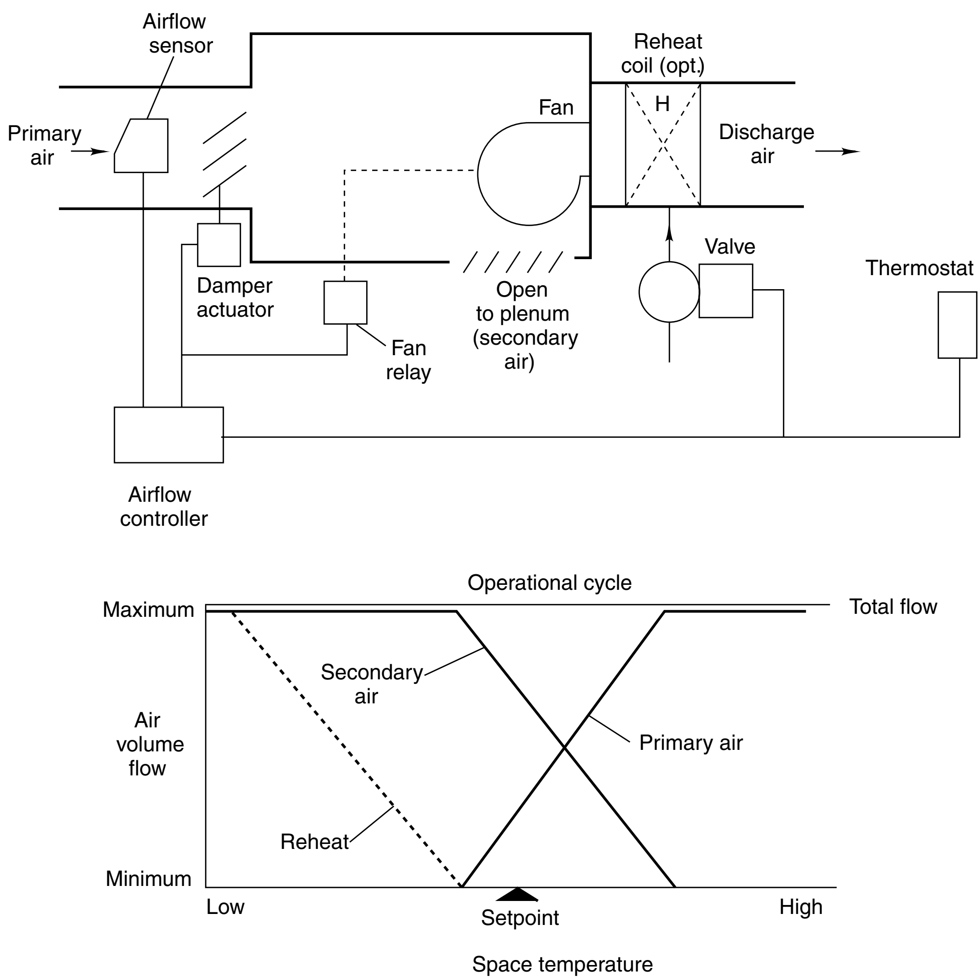

FIGURE 11.27 Series type, fan powered VAV box control subsystem and primary flow characteristic. The total box flow is constant at the level identified as “maximum” in the figure. The difference between primary and total air flow is secondary air recirculated through the return air grille. Optional reheat coil requires air flow shown by dashed line.

11.5.5 Other Systems

This section has not covered the control of central plant equipment such as chillers and boilers. Most primary system equipment controls are furnished with the equipment and as such do not offer much flexibility to the designer. However, Braun et al. (1989) have shown that considerable energy savings can be made by properly sequencing cooling tower stages on chiller plants, and by properly sequencing chillers themselves in multiple chiller plants.

Fire and smoke control are important for life safety in large buildings. The design of smoke control systems is controlled by national codes. The principal goal is to eliminate smoke from the zones where it is present, while keeping adjacent zones pressurized to prevent smoke infiltration. Some components of space conditioning systems (e.g., fans) can be used for smoke control, but HVAC systems are generally not designed to be smoke control systems.

Electrical systems are primarily the responsibility of the electrical engineer on a design team. However, HVAC engineers must make sure that the electrical design accommodates the HVAC control system. Interfaces between the two occur where the HVAC controls activate motors on fans or chiller compressors, pumps, electrical boilers, or other electrical equipment.

In addition to electrical specifications, the HVAC engineer often conveys electrical control logic using a ladder diagram. An example, for the control of the supply and return fans in a central system, is shown in Figure 11.28. The electrical control system, shown at the bottom, operates on low voltage (24 or 48 VAC) from the control transformer shown. The supply fan is started manually by closing the “start” switch. This activates the motor starter coil labeled 1M, thereby closing the three contacts labeled 1M in the supply fan circuit. The fourth 1M contact (in parallel with the start switch) holds the starter closed after the start button is released.

The hand-off auto switch is typical, and allows both automatic and manual operation of the return fan. When switched to the “hand” position, the fan starts. In the “auto” position, the fans will operate only when the adjacent contacts 3M are closed. Either of these actions activates the relay coil 2M, which in turn closes the three 2M contacts in the return fan motor starter. When either fan produces actual airflow, a flow switch is closed in the ducting, thereby completing the circuit to the pilot lamps L. The fan motors are protected by fuses and thermal overload heaters. If motor current draw is excessive, the heaters shown in the figure produce sufficient heat to open the normally closed thermal overload contacts.

FIGURE 11.28 The Brooke Army Medical Center (BAMC) in San Antonio, Texas.

This example ladder diagram is primarily illustrative, and is not typical of an actual design. In a fully automatic system, both fans would be controlled by 3M contacts actuated by the HVAC control system. In a fully manual system, the return fan would be activated by a fifth 1M contact, not by the 3M automatic control system.

11.6 Commissioning and Operation of Control Systems

This chapter emphasizes the importance of making sound decisions in the design of HVAC control systems. It is also extremely important that the control system be commissioned and used properly. The design process requires many assumptions about the building and its use. The designer must be sure that the systems will provide comfort under extreme conditions, and the sequence of design decisions and construction decisions often leads to systems that are substantially oversized. Operation at loads far below design conditions is generally much less efficient than at larger loads. Normal control practice can be a major contributor to this inefficiency. For example, it is quite common to see variable volume air handler systems operating at minimum flow as constant volume systems almost all the time, due to design flows that are sometimes twice as large as the maximum flow used in the building.

Thus, it is very important that following construction, the control system and the rest of the HVAC system be commissioned. This process (ASHRAE 2005) normally seeks to ensure that the control system operates according to design intent. This is really a minimum requirement to be sure that the system functions as designed. However, after construction, the control system setup can be modified to meet the loads actually present in the building, and to fit the way the building is actually being used, rather than basing these decisions on the design assumptions. If the VAV system is designed for more flow than is required, minimum flow settings of the terminal boxes can be reduced below the design value, ensuring that the system will operate in the VAV mode most of the time. Numerous other adjustments may be made as well. Such adjustments, commonly made during the version of commissioning known as *Continuous Commissioning*®[2](CC®), can frequently reduce the overall building energy use by 10% or more (Liu, Claridge, and Turner 2002). If the process is applied to an older building where control practices have drifted away from design intent and undetected component failures have further eroded system efficiency, energy savings often exceed 20% (Claridge et al. 2004).

11.6.1 Control Commissioning Case Study

A case study in which this process was applied to a major Army hospital facility located in San Antonio, Texas is reported in Zhu et al. (2000a, 2000b, 2000c). The Brooke Army Medical Center (BAMC) was a relatively new facility when the CC® process was begun. The facility was operated for the Army by a third-party company, and it was operated in accordance with the original design intent.

BAMC is a large, multifunctional medical facility with a total floor area of 1,349,707 $ft.^2$. The complex includes all the normal inpatient facilities, as well as outpatient and research areas. The complex is equipped with a central energy plant, which has four 1,200 ton water-cooled electric chillers. Four primary pumps (75 hp each) are used to pump water through the chillers. Two secondary pumps (200 hp each) equipped with VFDs supply chilled water from the plant to the building entrance. Fourteen chilled water risers equipped with 28 pumps totaling 557 hp are used to pump chilled water to all of the AHUs and small fan coil units. All of the chilled water riser pumps are equipped with VFDs. There are four natural gas-fired steam boilers in this plant. The maximum output of each boiler is 20 MMBtu/h. Steam is supplied to each building, where heating water was generated, at 125 psi (prior to commissioning).

There are 90 major AHUS serving the whole complex, with a total fan power of 2570 hp. VFDs are installed on 65 AHUs while the others are constant volume systems. There are 2,700 terminal boxes in the

[2]Continuous Commissioning and CC are registered trademarks of the Texas Engineering Experiment Station.

complex, of which 27% are dual duct variable volume (DDVAV) boxes, 71% are dual duct constant volume (DDCV) boxes, and 2% are single duct variable volume (SDVAV) boxes.

The HVAC systems (chillers, boilers, AHUs, pumps, terminal boxes and room conditions) are controlled by a DDC control system. Individual controller-field panels are used for the AHUs and water loops located in the mechanical rooms. The control program and parameters can be changed by either the central computers or the field panels.

11.6.1.1 Design Conditions

The design control program was being fully utilized by the EMCS. It included the following features:

1. Hot deck reset control for AHUS
2. Cold deck reset during unoccupied periods for some units
3. Static pressure reset between high and low limits for VAV units
4. Hot water supply temperature control with reset schedule
5. VFD control of chilled water pumps with ΔP setpoint (no reset schedule)
6. Terminal box level control and monitoring

It was also determined that the facility was being well maintained by the facility operator, in accordance with the original design intent. The building is considered energy efficient for a large hospital complex.

The commissioning activities were performed at the terminal box level, AHU level, loop level and central plant level. Several different types of improved operation measures and energy solutions were implemented in different HVAC systems, due to the actual function and usage of the areas and rooms. Each measure will be discussed briefly, starting with the air-handling units.

11.6.1.2 Optimization of AHU Operation

EMCS trending, complemented by site measurements and use of short-term data loggers, found that many supply fans operated above 90% of full speed most of the time. Static pressures were much higher than needed. Wide room temperature swings due to AHU shutoff led to hot and cold complaints in some areas. Through field measurements and analysis, the following possible means of improving the operation of the two AHUs were identified.

- Improve zone air balancing and determine new static pressure setpoints for VFDs
- Optimize the cold deck temperature setpoints with reset schedules
- Optimize the hot deck temperature reset schedules
- Improve control of outside air intake and relief dampers during unoccupied periods to reduce ventilation during these periods
- Optimize time schedule for fans to improve room conditions
- Improve the preheat temperature setpoint to avoid unnecessary preheating

Implementation of these measures improved comfort and reduced heating, cooling, and electric use.

11.6.1.3 Optimization at the Terminal Box Level

Field measurements showed that many VAV boxes had minimum flow settings that were higher than necessary, and that some boxes were unable to supply adequate hot air due to specific control sequences. New control logic was developed that increased hot air capacity by 30% on average, in the full heating mode, and reduced simultaneous heating and cooling. During unoccupied periods, minimum flow settings on VAV boxes were reduced to zero and flow settings were reduced in constant volume boxes.

During commissioning, it was found that some terminal boxes could not provide the required airflow either before or after the control program modification. Specific problems were identified in about 200 boxes, with most being high flow resistance due to kinked flex ducts.

11.6.1.4 Water Loop Optimization

There are 14 chilled water risers, equipped with 28 pumps, which provide chilled water to the entire complex. During the commissioning assessment phase, the following were observed:

- All the riser pumps were equipped with VFDs, which were running at 70%–100% of full speed.
- All of the manual balancing valves on the risers were only 30%–60% open.
- The ΔP sensor for each riser was located 10–20 ft. from the far-end coil of the AHU on the top floor.
- Differential pressure setpoints for each riser ranged from 13 to 26 psi.
- There was no control valve on the return loop.
- Although most of the cold deck temperatures were holding well, there were 13 AHUs whose cooling coils were 100% open, but which could not maintain cold deck temperature setpoints.

Because the risers are equipped with VFDs, traditional manual balancing techniques are not appropriate. All the risers were rebalanced by initially opening all of the manual balancing valves. The actual pressure requirements were measured for each riser, and it was determined that the ΔP for each riser could be reduced significantly. Pumping power requirements were reduced by more than 40%.

11.6.1.5 Central Plant Measures

1. *Boiler System*: Steam pressure was reduced from 125 psi to110 psi, and one boiler operated instead of two during summer and swing seasons.
2. *Chilled Water Loop*: Before the commissioning, the blending valve separating the primary and secondary loops at the plant was 100% open. The primary and secondary pumps were both running. The manual valves were partially open for the secondary loop, although the secondary loop pumps are equipped with VFDs. After the commissioning assessment and investigations, the following were implemented:

- Open the manual valves for the secondary loop
- Close the blending stations
- Shut down the secondary loop pumps

As a result, the primary loop pumps provide required chilled water flow and pressure to the building entrance for most of the year, and the secondary pumps stay offline most of the time. The operator drops the online chiller numbers according to the load conditions, and the minimum chilled water flow can be maintained to the chillers. At the same time, the chiller efficiency is increased.

11.6.1.6 Results

For the fourteen-month period following initial CC® implementation, measured savings were nearly $410,000, or approximately $30,000/month, for a reduction in both electricity and gas use of about 10%. The contracted cost to meter, monitor, commission, and provide a year's follow-up services was less than $350,000. This cost does not include any time for the facilities operating staff who repaired kinked flex ducts, replaced failed sensors, implemented some of the controls and subroutines, and participated in the commissioning process.

11.6.2 Commissioning Existing Buildings

The savings achieved from commissioning HVAC systems in older buildings are even larger. In addition to the opportunities for improving efficiency similar to those in new buildings, opportunities come from:

- Control changes that have been made to "solve" problems, often resulting in lower operating efficiency

- Component failures that compromise efficiency without compromising comfort
- Deferred maintenance that lowers efficiency

Mills et al. (2004 and 2005) surveyed 150 existing buildings that had been commissioned and found median energy cost savings of 15%, with savings in one-fourth of the buildings exceeding 29%. Over 60% of the problems corrected were control changes, and another 20% were related to faulty components that prevented proper control. This suggests that relatively few control systems actually achieve the efficiency they are capable of providing.

11.7 Advanced Control System Design Topics: Neural Networks

Neural networks offer considerable opportunity to improve the control possible in standard PID systems. This section provides a short introduction to this novel approach to control.

11.7.1 Neural Network Introduction

An artificial neural network is a massively parallel, dynamic system of interconnected, interacting parts based on some aspects of the brain. Neural networks are considered to be intuitive because they learn by example rather than by following programmed rules. The ability to "learn" is one of the key aspects of neural networks. A neural network consists of several layers of neurons that are connected to each other. A *connection* is a unique information transport link from one sending to one receiving neuron. The structure of part of an NN is schematically shown in Figure 11.29. Any number of input, output, and "hidden layer" neurons can be used (only one hidden layer is shown). One of the challenges of this technology is to construct a net with sufficient complexity to learn accurately without imposing a burden of excessive computational time.

The neuron is the fundamental building block of a network. A set of inputs is applied to each. Each element of the input set is multiplied by a weight, indicated by the W in the figure, and the products are summed at the neuron. The symbol for the summation of weighted inputs is termed $INPUT$ and must be calculated for each neuron in the network. In equation form this process for one neuron is

$$INPUT = \sum_i O_i W_i + B \tag{11.20}$$

where O_i are inputs to a neuron, i.e., outputs of the previous layer, W_i are weights, and B is the bias. After $INPUT$ is calculated, an activation function, F, is applied to modify it, thereby producing the neuron's output as described shortly.

Artificial networks have been trained by a wide variety of methods (McClelland and Rumelhart 1988). Back-propagation is one systematic method for training multilayer neural networks. The weights of a net are initiated with small random numbers. The objective of training the network is to adjust the weights iteratively so that application of a set of inputs produces the desired set of outputs matching a training data set. Usually a network is trained with a data set that consists of many input–output pairs; these data are called a *training set*. Training the net using back-propagation requires the following steps:

1. Select a training pair from the training set and apply the input vector to the network input layer.
2. Calculate the output of the network, OUT_i.
3. Calculate the error, $ERROR_i$, the network output, and the desired output (the target vector from the training pair).
4. Adjust the weights of the network in a way that minimizes the error.
5. Repeat steps 1 through 4 for each vector in the training set until the error for the entire set is lower than the user specified, preset training tolerance

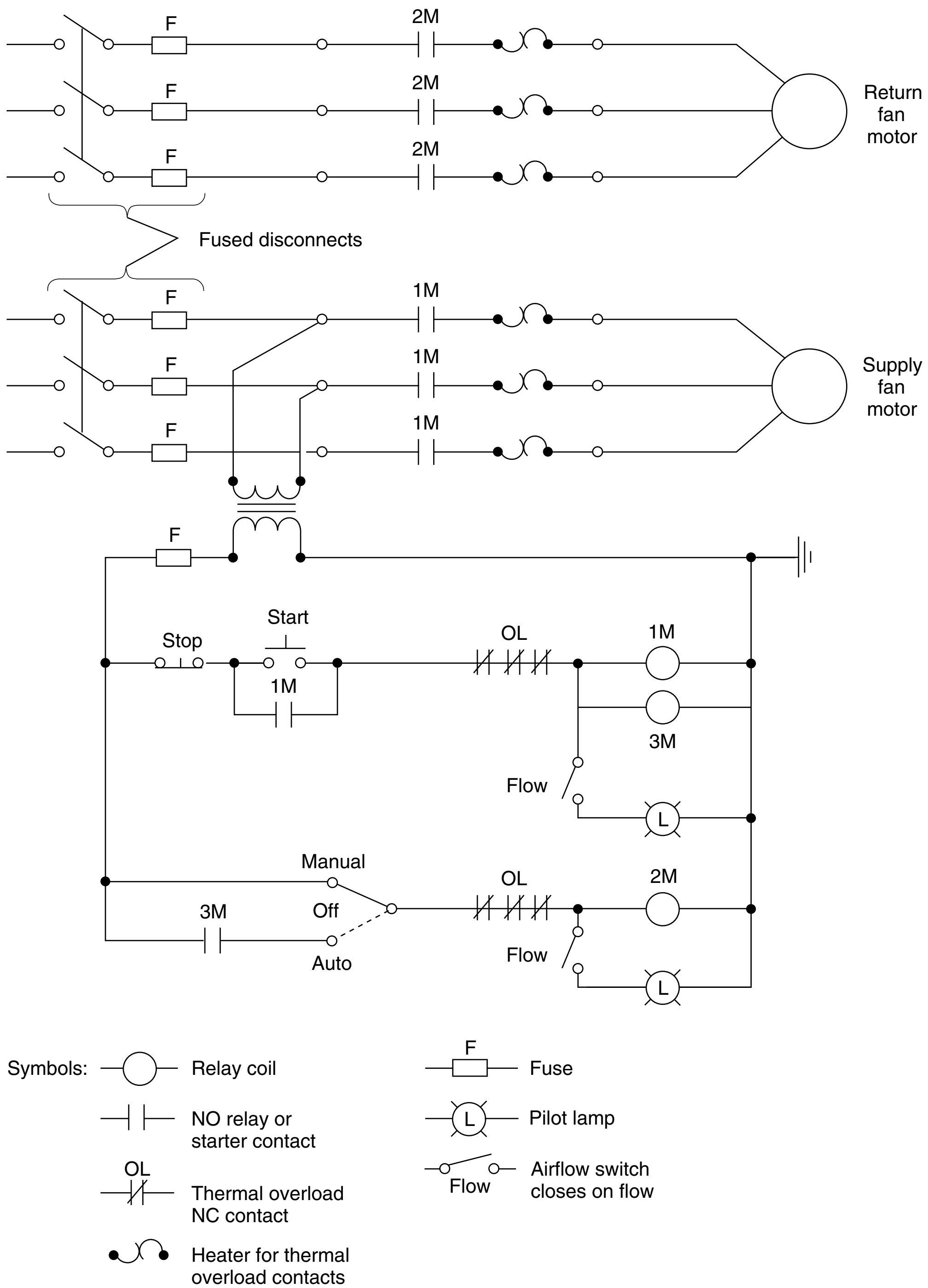

FIGURE 11.29 Ladder diagram for supply and return fan control. Hand-off auto switch permits manual or automatic control of the return fan.

Steps 1 and 2 are the "forward pass." The following expression describes the calculation process in which an activation function, *F*, is applied to the weighted sum of inputs, *INPUT*, as follows.

$$OUT = F(INPUT) = F\left(\sum_i O_i W_i + B\right), \tag{11.21}$$

where *F* is the activation function and *B* is the bias of each neuron.

The activation function used for this work was selected to be

$$F(INPUT) = \frac{1}{1 + e^{-INPUT}}. \tag{11.22}$$

This is referred to as a *sigmoid function* and is shown in Figure 11.30. It has a value of 0.0 when *INPUT* is a large negative number and a value of 1.0 for large and positive *INPUT*, making a smooth transition between these limiting values. The bias, *B*, is the activation threshold for each neuron. The bias avoids the tendency of a sigmoid function to get "stuck" in the saturated, limiting value area.

Steps 3 and 4 comprise the "reverse pass" in which the delta rule is used as follows: for each neuron in the output layer, the previous weight $W(n)$ is adjusted to a new value $W(n+1)$ to reduce the error by the following rule:

$$W(n+1) = W(n) + (\eta\delta)OUT, \tag{11.23}$$

where $W(n)$ is the previous value of a weight, $W(n+1)$ is the weight after adjusting, η is the training rate coefficient. δ is calculated from

$$\delta = \left(\frac{\partial INPUT}{\partial OUT}\right)(TARGET - OUT) = OUT(1 - OUT)(TARGET - OUT), \tag{11.24}$$

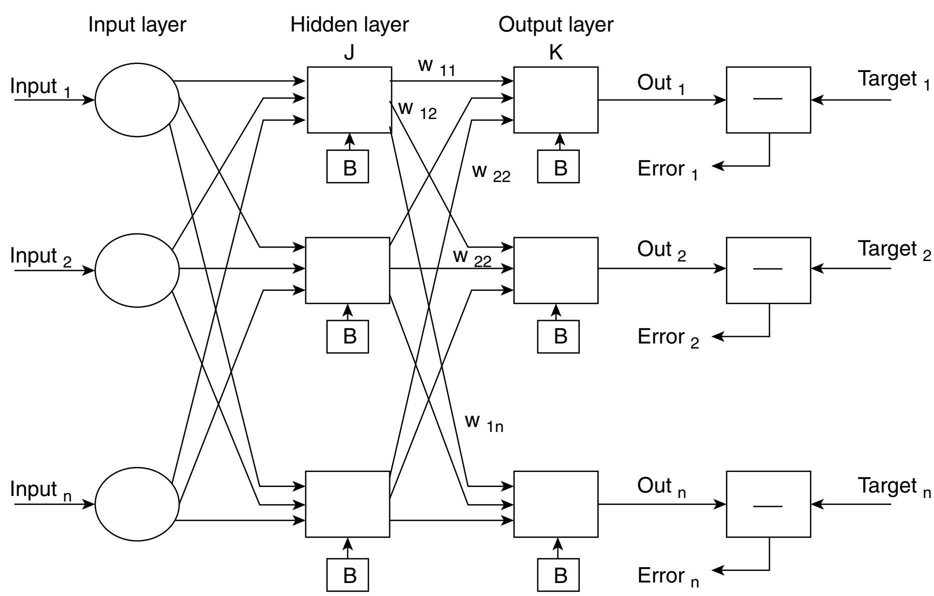

FIGURE 11.30 Schematic diagram of a neural network showing input layer, hidden layers, and output along with target training values. Hidden and output layers consist of connected neurons; the input layer does not contain neurons.

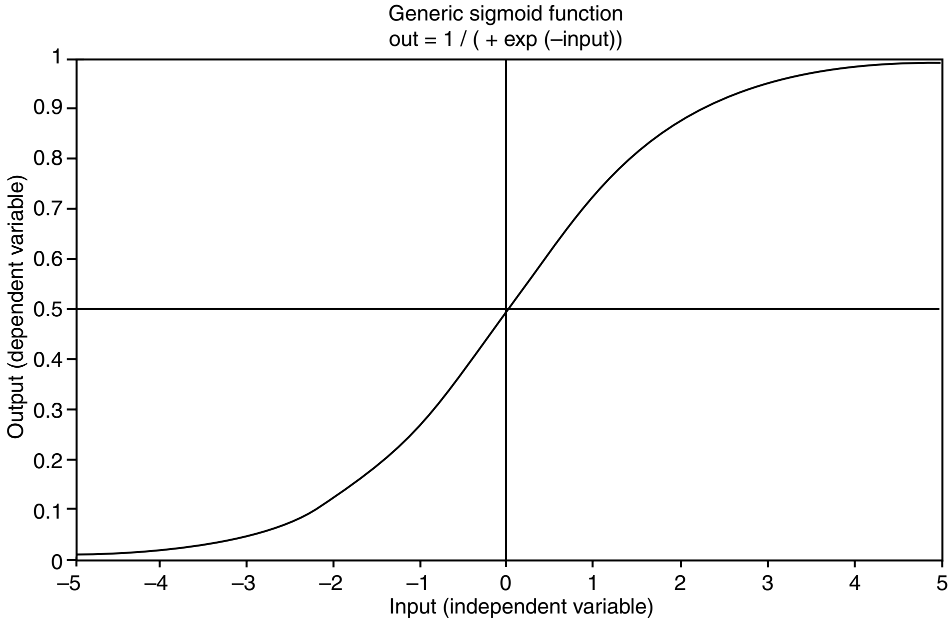

FIGURE 11.31 Sigmoid function used to process the weighted sum of network inputs.

in which the derivative has been calculated from Equation 11.21 and Equation 11.22, and *TARGET* (see Figure 11.29) is the training-set target value. This method of correcting weights bases the magnitude of the correction on the error itself.

Of course, hidden layers have no target vector; therefore, back-propagation trains these layers by propagating the output error back through the network layer by layer, adjusting weights at each layer. The delta rule adjustment δ is calculated from

$$\delta_j = OUT(1-OUT)\sum(\delta_{j+1}W_{j+1}) \tag{11.25}$$

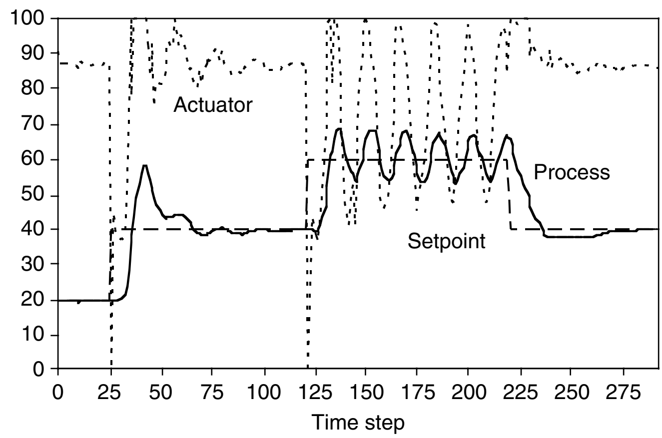

FIGURE 11.32 PID controller response to step changes in coil load. Proportional gain of 2.0. (From Curtiss, P. S., Kreider, J. F., and Brandemuehl, M. J., *ASHRAE Transactions*, 99 (1) 1993.)

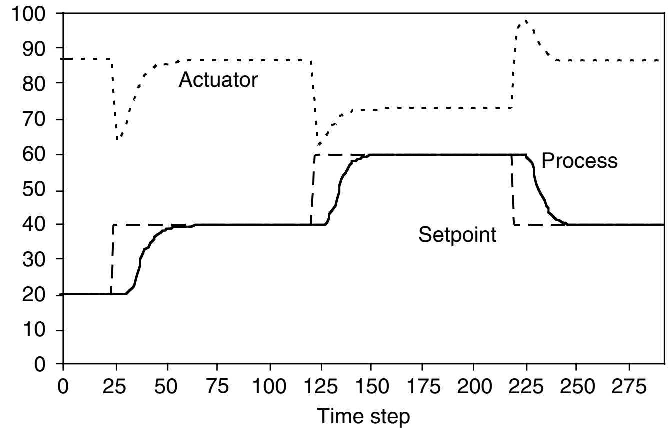

FIGURE 11.33 NN controller with learning rate of 1.0 and window of 15 time steps. (From Curtiss, P. S., Kreider, J. F., and Brandemuehl, M. J., *ASHRAE Transactions*, 99 (1) 1993.)

where δ_j and δ_{j+1} belong to the jth and $(j+1)$th hidden layers, respectively (being numbered with increasing values from left to right in Figure 11.29). This overall method of adjusting weights belongs to the general class of steepest descent algorithms. The weights and biases after training contain meaningful system information; before training, the initial, random biases and random weights have no physical meaning.

11.7.2 Commercial Building Adaptive Control Example

A proof of concept experiment in which neural networks (NNs) were used for both local and global control of a commercial building HVAC system was conducted in the JCEM laboratory in which full-scale, repeatable testing of multizone HVAC systems can be done. Data collected in the laboratory were used to train NNs for both the components and the full systems involved (Curtiss, Brandemuehl, and Kreider 1993; Curtiss, Kreider, and Brandemuehl 1993). Any neural network-based controller will be useful only if it can perform better than a conventional PID controller. Figure 11.31 and Figure 11.32 show typical results for the PID and NN control of a heating coil. The difficulty that the PID controller experienced is due to the highly nonlinear nature of the heating coil. A PID controller tuned at one level of load is unable to control acceptably at another, while the NN controller does not have this difficulty. With the NN controller, excellent control is demonstrated—minimal overshoot and quick response to the setpoint changes.

In an affiliated study, Curtiss, Brandemuehl, and Kreider (1993) showed that NNs offered a method for global control of HVAC systems as well. The goal of such controls could be to reduce energy consumption as much as possible, while meeting comfort conditions as a constraint. Energy savings of over 15% were achieved by the NN method vs. standard PID control (Figure 11.33).

11.8 Summary

This chapter has introduced the important features of properly designed control systems for HVAC applications. Sensors, actuators, and control methods have been described. Methods for determining control system characteristics, either analytically or empirically, have been discussed.

The following rules (adapted from ASHRAE 1987) should be followed to ensure that the control system is as energy efficient as possible. Neural networks offer one method for achieving energy efficient control.

1. Operate HVAC equipment only when the building is occupied or when heat is needed to prevent freezing.
2. Consider the efficacy of night setback vis à vis building mass. Massive buildings may not benefit from night setback due to the overcapacity needed for the morning pickup load.
3. Do not supply heating and cooling simultaneously. Do not supply humidification and dehumidification at the same time.
4. Reset heating and cooling air or water temperature to provide only the heating or cooling needed.
5. Use the most economical source of energy first, the most costly last.
6. Minimize the use of outdoor air during the deep heating and cooling seasons, subject to ventilation requirements.
7. Consider the use of "dead-band" or "zero-energy" thermostats.
8. Establish control settings for stable operation to avoid system wear and to achieve proper comfort.
9. Commission the control system and HVAC system for optimum efficiency based on actual building conditions and use.

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