

8.2 Airhandler and Building Conditioning Controls

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INTRODUCTION

The main components of HVAC control systems include 1) the various comfort sensors, such as thermostats and humidistats (Section 7.9 in Chapter 7) and pressure sensors (Chapter 5 in Volume 1), 2) the control systems for heat and coolant supply systems, the boilers (Section 8.6), chillers (Sections 8.12 and 8.13), and the cooling towers (8.16 and 8.17), 3) the air and water transportation controls, including the fans and blowers (Section 8.25) and pumping stations (Section 8.34), and 4) the final control elements, including the dampers (Section 7.1), control valves (Chapter 6), and variable-speed drives (Sections 7.10). For more information on the above topics the reader is referred to the noted sections.

This section will concentrate on the control and optimization of the total space conditioning system. This will be approached by first discussing the process being controlled and its various operating modes as the seasons change. Once the “personality” of the process has been described, the control of the various comfort-related variables (temperature, humidity, and air quality) will be discussed. The emphasis will be placed on systems in which air is the final carrier of heat or cooling into the conditioned spaces, although brief mention will also be made of the more traditional, but still used, water-based systems.

In the second half of this section, the emphasis will be on the optimization of the total process by such methods as making the buildings self-heating and by eliminating the chimney effects.

THE AIRHANDLER

The airhandler is the basic unit operation of space conditioning. It is used to keep occupied spaces comfortable (Figure 8.2a) or unoccupied spaces at desired levels of temperature and humidity. In addition to supplying or removing heat or humidity from the conditioned space, the airhandler also provides ventilation and fresh air makeup. Depending on the type of space involved, from 75,000–300,000 BTU/year (19,000–76,000 cal/year) are required to condition 1 ft² (0.092 m²) of office space. Depending on the energy sources used, this corresponds to a yearly operating cost of a few dollars per square foot of floor space.

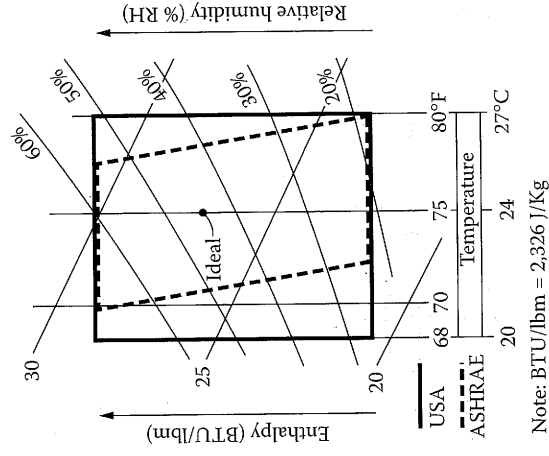


FIG. 8.2a

“Comfort zones” are defined in terms of temperature and humidity.¹

Whereas other unit operations have benefited substantially from the advances in process control, airhandlers have not. Airhandlers today are frequently controlled the same way as they were 20 or 30 years ago. For this reason, airhandler optimization can result in much greater percentages of savings than can the optimization of almost any other unit operation. Optimization can sometimes cut the cost of airhandler operation in half—a savings that can seldom be achieved in any other type of unit operation.

Some of the optimization goals and strategies include the following:

- Let the building heat itself
- Use free cooling or free drying
- Benefit from gap control or zero energy band (ZEB)
- Eliminate chimney effect
- Optimize start-up timing
- Optimize air makeup (CO₂)
- Optimize supply air temperature
- Minimize fan energy use
- Automate the selection of operating modes
- Minimize reheat
- Automate balancing of air distribution

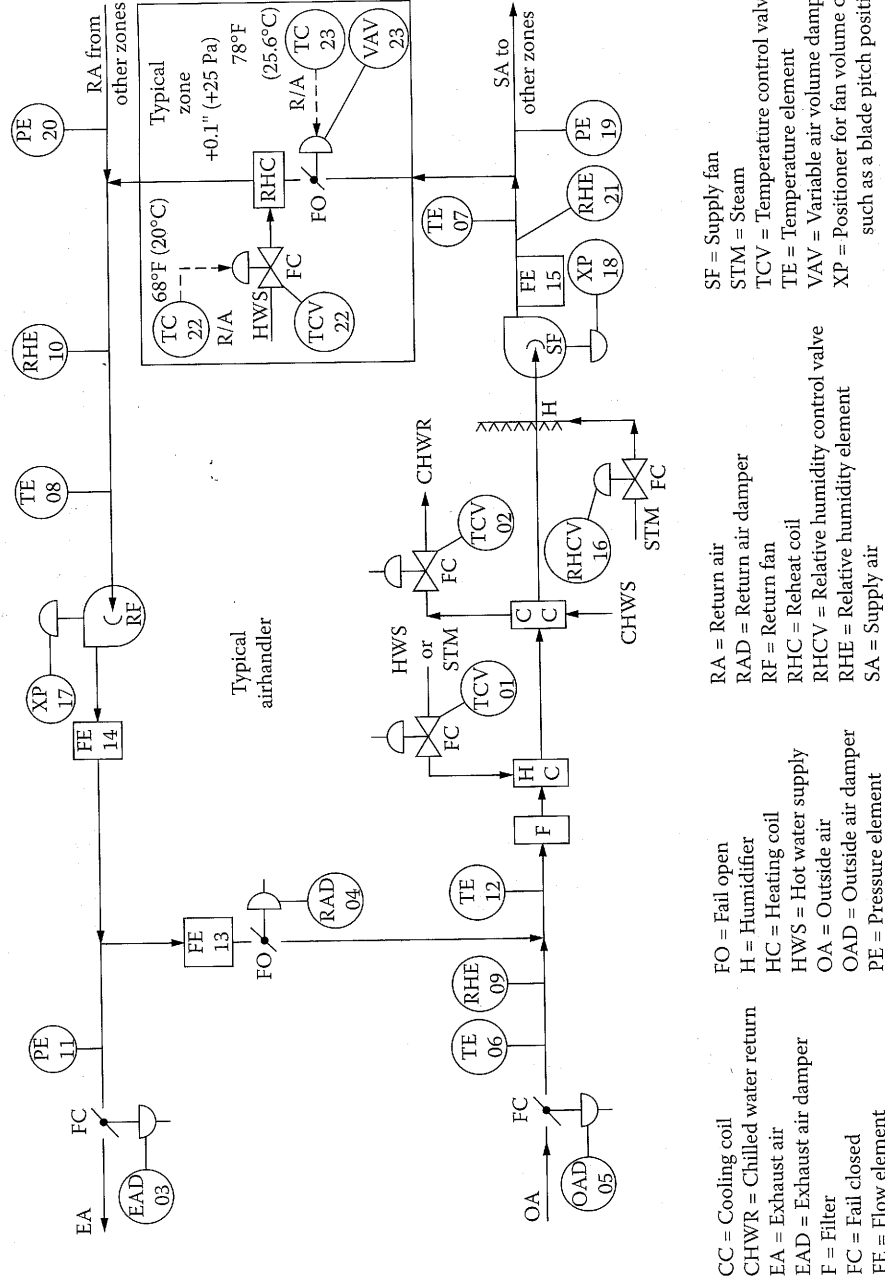


FIG. 8.2b

A typical major airhandler has these components and controls.

Airhandler Components

The purpose of heating, ventilation, and air conditioning (HVAC) controls is to provide comfort in laboratories, cleanrooms, warehouses, offices, and manufacturing spaces. Supply air is the means of providing comfort in the conditioned zone. The air supplied to each zone must provide heating or cooling, raise or lower humidity, and provide air refreshment. To satisfy these requirements, it is necessary to control the temperature, humidity, and fresh-air ratio in the supply air.

Figure 8.2b illustrates the main components of an airhandler. The term *airhandler* refers to the total system, including fans, heat-exchanger coils, dampers, ducts, and instruments. The system operates as follows: Outside air is admitted by the outside air damper (OAD-05) and is then mixed with the return air from the return air damper (RAD-04). The resulting mixed air is filtered (F), heated (HC) or cooled (CC), and humidified (H) or dehumidified (CC) as required. The resulting supply air is then transported to the conditioned zones (groups of offices) by the variable-volume supply fan station. Variable volume means that the air flow rate generated by the fan(s) is variable.

In each zone, the variable air volume damper (VAV-23) determines the amount of air required, and the reheat coil (RHC) adjusts the air temperature as needed. The return air

from the zones is transported by the variable-volume return-air fan station. If the amount of available return air exceeds the demand for it, the excess air is exhausted by the exhaust air damper (EAD-03). The conditioned spaces are typically pressurized to about 0.1 in. H₂O (25 Pa), relative to the barometric pressure on the outside. This pressurization results in some air leakage through the walls and windows, which varies with the quality of construction. Therefore, the air balance around the system is:

$$OA = EA + \text{pressurization loss} \quad 8.2(1)$$

Under "normal" operation, the airhandler operates with about 10% outside air. In the "purge" or "free cooling" modes, RAD is closed, OAD is fully open, and the airhandler operates with 100% outside air.

As can be seen, the HVAC process is rather simple. Its process material is clean air; its utility is water or steam, and its overall system behavior is slow, stable, and forgiving. For precisely these reasons, it is possible to obtain acceptable HVAC performance using inferior-quality instruments that are configured into poorly designed loops. Yet, there is an advantage in applying state-of-the-art process control to the HVAC process, because it can provide a drastic reduction in operating costs, attributable to increased efficiency of operation. Some

of the more efficient control concepts are described in the paragraphs below.

Operating Mode Selection

The correct identification and timing of the various operating modes can contribute to the optimization of the building. The *normal* operating modes include start-up, occupied, night, and purge.

Optimizing the time of *start-up* will guarantee that the minimum required cost is invested in getting the building ready for occupancy. This is done by automatically calculating the amount of heat that needs to be transferred and dividing it according to the capacity of the start-up equipment. A computer-optimized control system will serve to initiate the unoccupied (night) mode of operation; it will also recognize weekends and holidays and, in general, provide a flexible means of time-of-day controls.

The *purge* mode is another convenient tool of optimization. Whenever the outside air is preferred to the return air, the building is automatically purged. In this way, "free cooling" can be obtained on dry summer mornings, or "free heating" can be provided on warm winter afternoons. Purging is the equivalent of opening the windows in a home. In computer-optimized buildings, an added potential is to use the building structure as a means of heat (or coolant) storage. In this case, the purge mode can be automatically initiated during cold nights prior to hot summer days, thereby bringing the building temperature down and storing some free cooling in the building structure.

Summer/Winter Mode Reevaluation Another important mode selection involves switching from summer to winter mode and vice versa. Conventional systems are switched according to the calendar, whereas optimized ones recognize that there are summer-like days in the winter and winter-like hours during summer days. Seasonal mode switching is therefore totally inadequate.

Optimized building operation can be provided only by making the summer/winter selection on an enthalpy basis: If heat needs to be added, it is "winter"; if heat needs to be removed, it is "summer," regardless of the calendar. In those airhandlers that serve a variety of zones, it is essential to first determine if the unit is in a "net" cooling (summer) or "net" heating (winter) mode before the control system can decide if free cooling (or free heating) by outside air can be used to advantage.

Figure 8.2c illustrates the heat balance evaluation that is required to determine the prevailing overall mode of operation. This type of heat balance calculation, which must be reevaluated every 15 to 30 minutes, can be implemented only through the use of computers.

Emergency Mode In addition to the above operating modes, the airhandler can also be placed in an *emergency* mode, if fire, smoke, freezing temperature, or pressure conditions

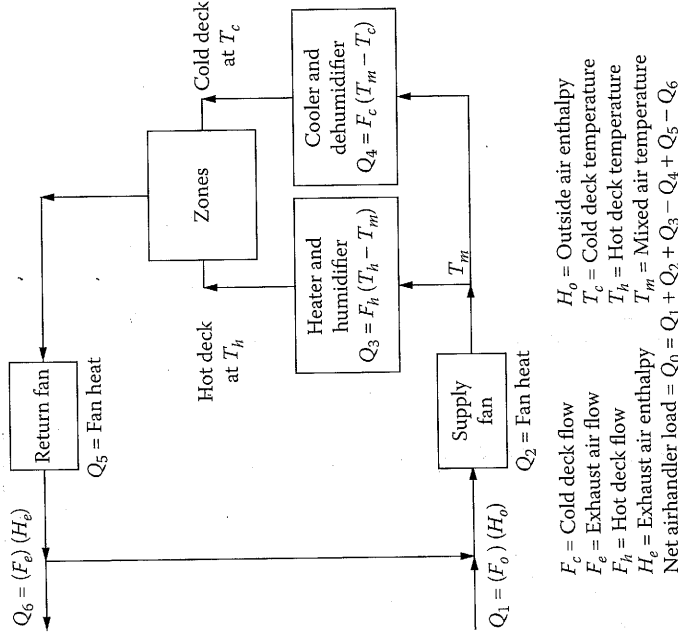


FIG. 8.2c

When the net airhandler load is negative, summer mode is required; when it is positive, winter mode is required.

require it. Table 8.2d lists the status of each fan, damper, and valve in each of the operating modes. In a computer-optimized control system, both the mode selection and the setting of the actuated devices is done automatically.

When a smoke or fire condition is detected by sensors S/F-4 or S/F-8 in Figure 8.2e, the fans stop, the OADs and RADs close, the EAD opens, and an alarm is actuated. The operator can switch the airhandler into its purge mode, so that the fans are started, OAD and EAD are opened, and RAD is closed. If the smoke/fire emergency requires, the fire command panel (FC in Figure 8.2e) can be used by firefighters. From this panel, the fire chief can operate all fans and dampers as needed for safe and orderly evacuation and protection of the building.

In another emergency condition, a freeze-stat switch on one of the water coils is actuated. These switches are usually set at approximately 35°F (1.5°C) and serve to protect from coil damage resulting from freeze-ups. Multistage freeze-stat units might operate as follows:

- At 38°F (3°C): close OAD
- At 36°F (2°C): fully open water valve
- At 35°F (1.5°C): stop fan

If single-stage freeze-stats are used, they will stop the fan, close the OAD, and activate an alarm.

Yet another type of emergency is signaled by excessive pressures in the ductwork on the suction or discharge sides of the fans, resulting from operation against closed dampers

TABLE 8.2d
The Status of Various Actuated Devices during Various Operating Modes

Operating Mode or Emergency Condition	Supply Fan	Return Fan	Outside Air Damper	Exhaust Air Damper	Return Air Damper	Coil Control Valves	Alarm
Off	—	—	C	C	—	C	—
On	On	On	Modulating	Modulating	—	—	—
Warm-up	On	On	C	C	O	O(HC)	—
Cool-down	On	On	C	C	O	O(CC)	—
Night	—	—	Cycled to maintain required nighttime temperature	—	—	—	—
Purge	On	On	O	O	C	Modulating	—
PSH-2	—	Off	—	C	—	—	Yes
PSL-3	—	Off	—	C	—	—	Yes
S/F-4	Off	Off	C	O	C	C	Yes
TSL-5	Off	—	C	—	O	C	Yes
PSL-6	Off	—	C	—	O	—	Yes
PSH-7	Off	—	C	—	O	—	Yes
S/F-8	Off	Off	C	O	C	C	Yes

or from other equipment failures. When this happens, the associated fan is stopped and an alarm is actuated.

Fan Controls

The standard fan controls are shown in Figure 8.2f. Each zone shown in Figure 8.2b is supplied with air through a thermostat-modulated damper, also called a variable air volume box (VAV-23).

The VAV box openings in the various zones determine the total demand for supply air. The pressure in the supply air (SA) distribution header is controlled by PIC-19, which

modulates the supply air fan station to match the demand (Figure 8.2f). When the PIC-19 output has increased the fan capacity to its maximum, PSH-19 actuates and starts an additional fan. Inversely, as the demand for supply air drops, FSL-15 will stop one fan unit whenever the load can be met by fewer fans than the number in operation. The important point to remember is that in cycling fan stations, fan units are started on pressure and are stopped on flow control. The operating cost of such a fan station is 20–40% lower than if constant-volume fans with conventional controls were used (Figure 8.2g).

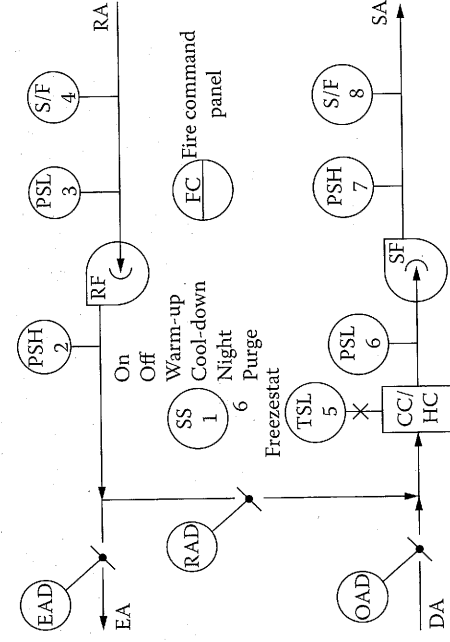


FIG. 8.2e

The safety and operating mode selection instruments used on an airhandler. Most abbreviations used on this figure have already been defined in connection with Figure 8.2b; S/F = smoke and fire detector, SS = selector switch, FC = fire command panel.

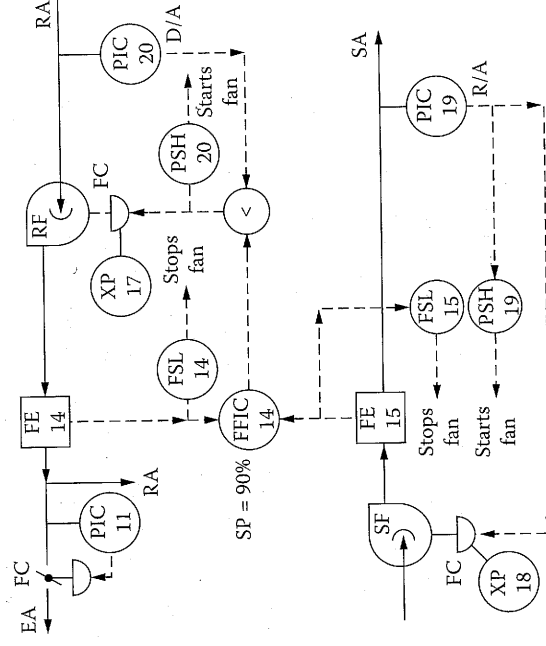


FIG. 8.2f

Variable-volume fan controls operate as shown here.

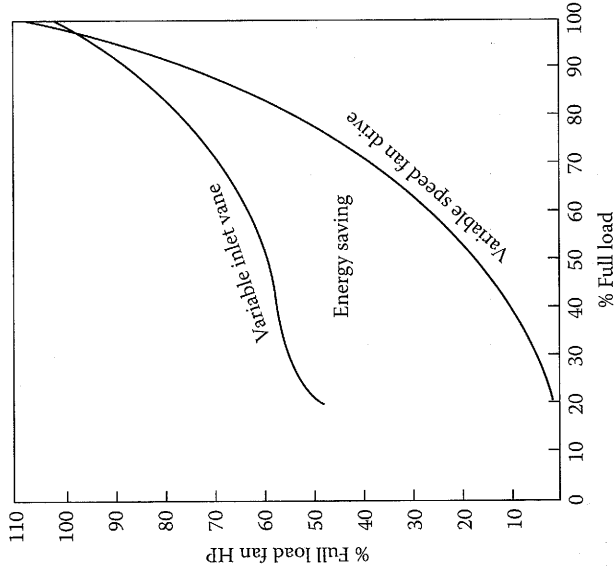


FIG. 8.2g

Using variable-speed fans can save significant amounts of energy. (Courtesy of Dana Corp.)

Because the conditioned zones are pressurized slightly, some of the conditioned air will leak into the atmosphere, creating pressurization loss. Being able to control the pressurization loss is one of the advantages of the control system described in Figure 8.2f. The flow ratio controller FFIC-14 is set at 90%, meaning that the return-air fan station is modulated to return 90% of the air supplied to the zones. Therefore, pressurization loss is controlled at 10%, which corresponds to the minimum fresh-air makeup requirement, resulting in a minimum-cost operation.

Because the conditioned zones represent a fairly large capacity, a change in supply air flow will not immediately result in a need for a corresponding change in the return air flow. Thus, PIC-20 (Figure 8.2f) is included in the system to prevent the flow-ratio controller from increasing the return air flow rate faster than required. This dynamic balancing eliminates cycling and protects against collapsing the ductwork under excessive vacuum. Closure of the exhaust-air damper by PIC-11 indicates that the control system is properly tuned and balanced and is operating at maximum efficiency. Under such conditions, the outside air admitted into the airhandler exactly matches the pressurization loss, and no return air is exhausted.

To maximize the benefits of such an efficient configuration, the dampers must be of tight shut-off design. When exposed to a pressure difference of 4 in. H_2O (996 Pa), a closed conventional damper will leak at a rate of approximately 50 cfm/ft^2 [$15.2 (m^3/min)/m^2$]. In the HVAC industry, a 5 cfm/ft^2 [$1.52 (m^3/min)/m^2$] leakage rate is considered to represent a tight shut-off design. Actually, it is cost-effective to install tight shut-off dampers with leakage rates of less than 0.5 cfm/ft^2

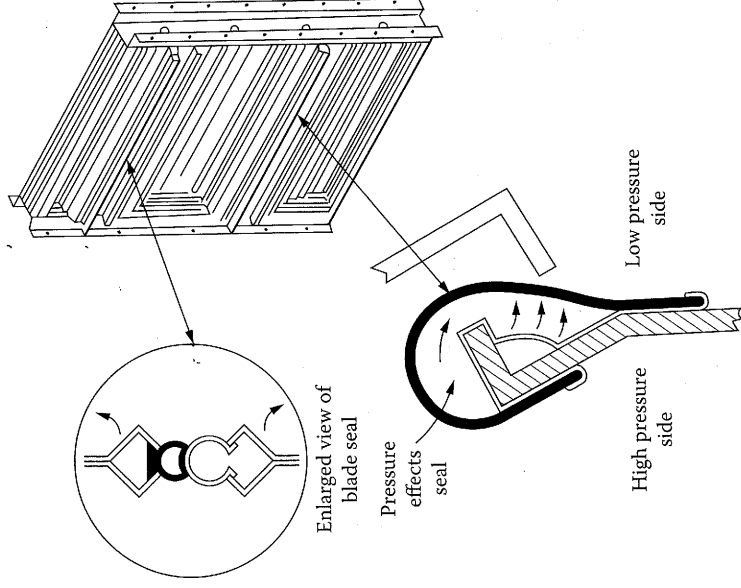


FIG. 8.2h

Low-leakage damper designs increase the efficiency of HVAC systems.

[$0.15 (m^3/min)/m^2$], because the resulting savings over the life of the buildings will be much greater than the increase in initial investment for better dampers (Figure 8.2h).

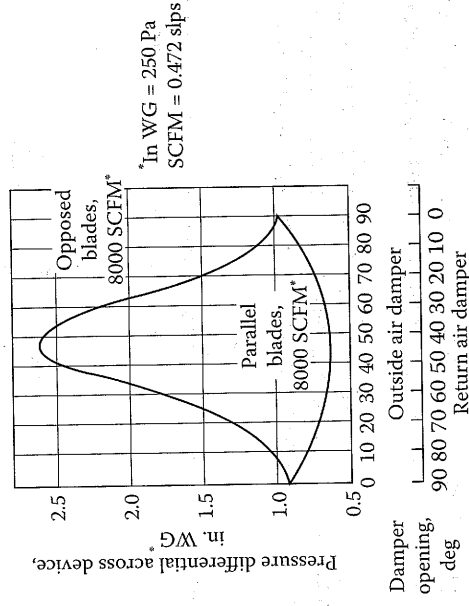
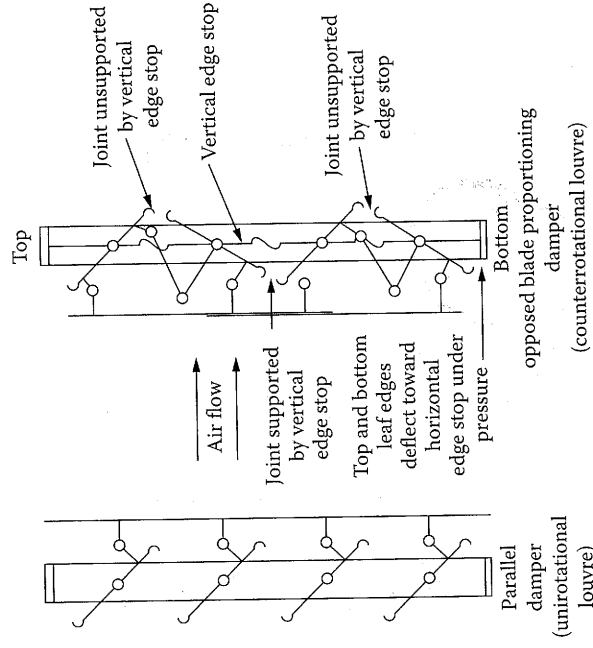
Dampers and Pressure Distribution

In order for dampers to give good control, a fair amount of pressure drop should be assigned to them. They should be sized for a ΔP of about 10% of the total system drop. On the other hand, excessive damper drops should also be avoided, because they will increase the operating costs of the fans. A good sizing basis for outside and return air fans is to size them for 1500 fpm (457 m/min) velocity at maximum flow.

In locations where two air streams are mixed, such as when outside and return airs are ratioed (RAD-04 and OAD-05 in Figure 8.2b), it is important that the damper ΔP be relatively constant as the ratio is varied. Figure 8.2i shows that parallel blade dampers give a superior performance in this service.

Figure 8.2j illustrates the pressure levels in the various portions of typical airhandlers. It can be seen that the kind of pressure drops that would be required by opposed blade dampers (Figure 8.2i) are simply not available. Therefore, if such dampers were installed, the airhandler would be starved for air (the dampers could not pass the design flow) whenever the ratio was near 50:50 (percent).

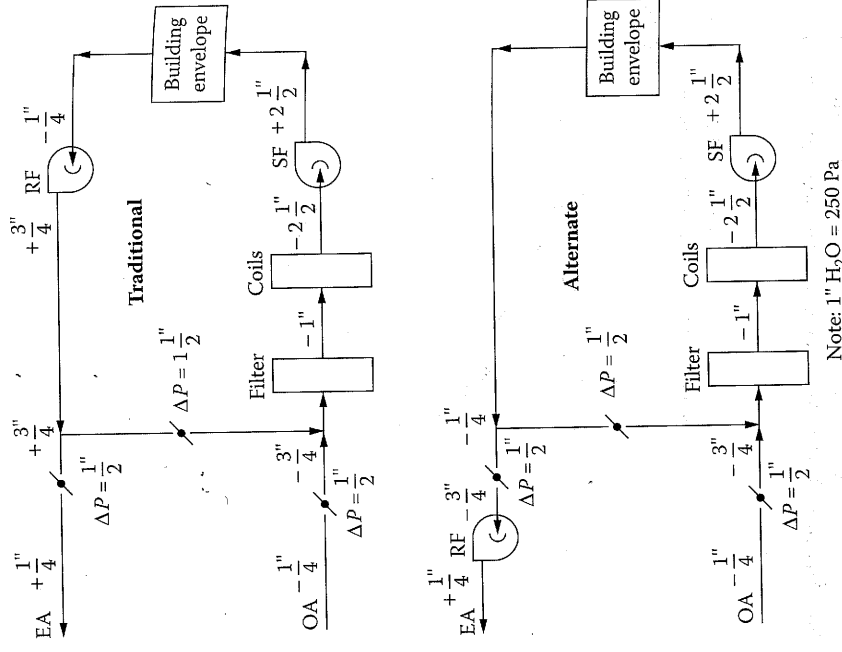
Figure 8.2j also shows that in traditional airhandlers more fan energy is used than necessary. This is because the return air fan is sized to generate the pressure needed to exhaust the

**FIG. 8.2i**

When outside and return air dampers are throttled to vary their ratio at constant total flow, the required pressure drop varies with damper design (see upper portion of figure). The lower portion of this figure shows the results of American Warming and Ventilating Co. tests (per AMCA Standard 500) of pressure drops across parallel-blade and opposed-blade outside and return air damper sets.²

air from the building. A consequence of this is that the pressure drop of $1\frac{1}{2}$ in. H_2O (375 Pa) across the return air damper is three times greater than what is necessary ($\frac{1}{2}$ in. H_2O , or 125 Pa).

The alternate system shown in the lower portion of Figure 8.2j eliminates this waste of fan energy. Here, only the supply fan (SF) operates continuously, which reduces the pressure drop across the return air damper to $\frac{1}{2}$ in. H_2O (125 Pa). The return fan (RF) is started only when air needs to be relieved, and its speed is varied to adjust the amount of air to be exhausted. Relocating RF also removes its heat input, which, in the traditional system, represents an added load on the cooling coil.

**FIG. 8.2j**

Damper pressure drops and the typical pressure levels in the various segments of air-handlers.

Temperature Controls

Space temperatures are controlled by thermostats. The traditional thermostat is a proportional-only controller (see also Section 2.2). The pressure of the output signal from a pneumatic "stat" is a near straight-line function of the measurement, described by the following relationship:

$$O = K_c(M - M_o) + O_o \quad 8.2(2)$$

where

O = output signal

K_c = proportional sensitivity (K_c can be fixed or adjustable, depending on the design)

M = measurement (temperature)

M_o = "normal" value of measurement, corresponding to the center of the throttling range

O_o = "normal" value of the output signal, corresponding to the center of the throttling range of the control valve (or damper)

Another term used to describe the sensitivity of thermostats is *throttling range*. As shown in Figure 8.2k, this term refers to the amount of temperature change that is required to change the thermostat output from its minimum to its

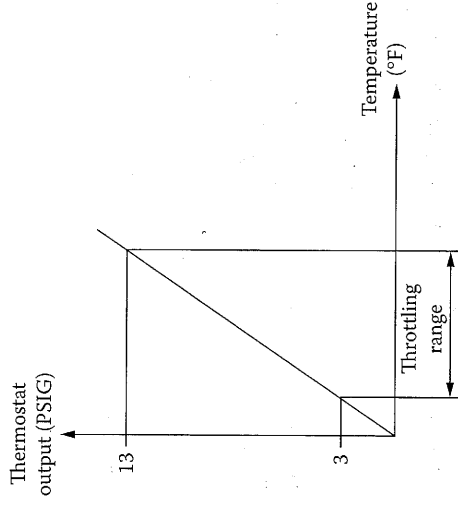


FIG. 8.2k

Throttling range can be defined as the temperature change required to change the thermostat output from its minimum to its maximum value.

maximum value, such as from 3 to 13 PSIG (21 to 90 kPa). The throttling range is usually adjustable from 2 to 10°F (1 to 5°C).

One important point to remember is that thermostats do not have set points, in the sense of having a predetermined temperature to which they would seek to return the controlled space. (Integral action must be added in order for a controller to be able to return the measured variable to a set point after a load change.) M_o does not represent a set point; it only identifies the space temperature that will cause the cooling damper in Figure 8.2l to be 50% open. This can be called a “normal” condition, because relative to this point the thermostat can both increase and decrease the cooling air flow rate as space temperature changes.

If the cooling load doubles, the damper will need to be fully open, which cannot take place until the controlled space temperature has risen to 73°F (23°C). As long as the cooling load remains that high, the space temperature must also stay up at the 73°F (23°C) value. Similarly, the only way this thermostat can reduce the opening of the cooling damper below 50% is to first allow the space temperature to drop below 72°F (23°C). Thus, thermostats have throttling ranges, not set points. If a throttling range is narrow enough, this gives the appearance that the controller is keeping the variable near the set point, when in fact the narrow range only allows the variable to drift within limits.

Special-Purpose Thermostats *Day-night, set-back, or dual room thermostats* will operate at different “normal” temperature values for day and night. They are provided with both a “day” and a “night” setting dial, and the change from day to night operation can be made automatic for a group of thermostats. The pneumatic day-night thermostat uses a two-pressure air supply system, the two pressures often being 13 and 17 PSIG (89.6 and 177 kPa) or 15 and 20 PSIG (103.35 and 137.8 kPa). Changing the pressure at a central point from

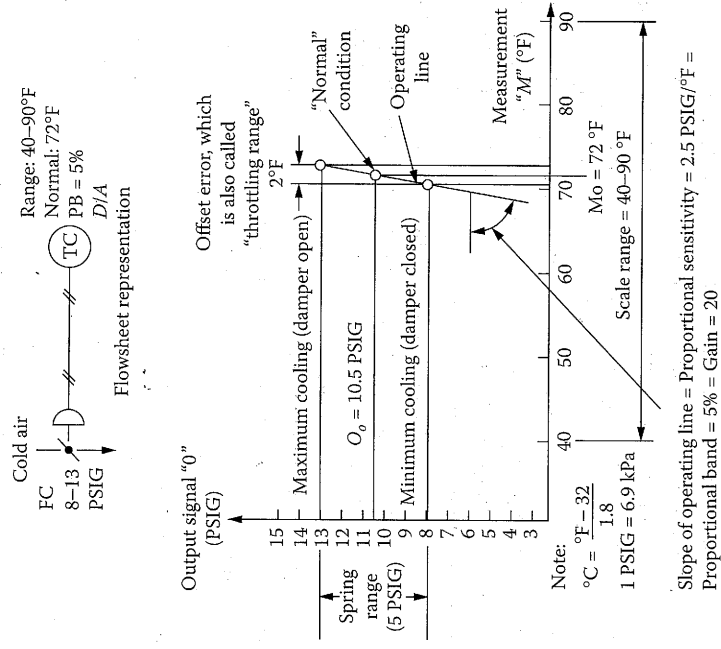


FIG. 8.2l

A fixed proportional band thermostat has a fixed throttling range and no setpoint.

one value to the other actuates switching devices in the thermostat and indexes them from day to night or vice versa.

Supply air mains are often divided into two or more circuits so that switching can be accomplished in various areas of the building at different times. For example, a school building may have separate circuits for classrooms, offices and administrative areas, the auditorium, and the gymnasium and locker rooms. In some of the electric designs, dedicated clocks and switches are built into each thermostat.

The *heating-cooling* or *summer-winter thermostat* can have its action reversed and, if desired, can have its set point changed by means of indexing. This thermostat is used to actuate controlled devices, such as valves or dampers, that regulate a heating source at one time and a cooling source at another. It is often manually indexed in groups by a switch, or automatically by a thermostat that senses the temperature of the water supply, the outdoor temperature, or another suitable variable.

In the heating-cooling design, there are frequently two bimetallic elements, one being direct acting for the heating mode, the other being reverse acting for the cooling mode. The mode is switched automatically in response to a change in the air supply pressure, much as the day-night thermostats operate.

The *limited control range thermostat* usually limits the room temperature in the heating season to a maximum of 75°F (24°C), even if the occupant of the room has set the

thermostat beyond these limits. This is done internally, without placing a physical stop on the setting knob.

A *slave or submaster thermostat* has its set point raised or lowered over a predetermined range, in accordance with variations in the output from a master controller. The master controller can be a thermostat, manual switch, pressure controller, or similar device. For example, a master thermostat measuring outdoor air temperature can be used to adjust a submaster thermostat controlling the water temperature in a heating system. Master-submaster combinations are sometimes designated as single-cascade action. When action is accomplished by a single thermostat having more than one measuring element, it is referred to as *compensated control*.

Multistage thermostats are designed to operate two or more final control elements in sequence.

A *wet-bulb thermostat* is often used for humidity control, as the difference between wet- and dry-bulb temperature is an indication of moisture content. A wick or other means for keeping the bulb wet and rapid air motion to ensure a true wet-bulb measurement are essential.

A *dew-point thermostat* is a device designed to control humidity on the basis of dew point temperatures.

A *smart thermostat* is usually a microprocessor-based unit with RTD-type or transistorized solid-state sensor. It is usually provided with its own dedicated memory and intelligence, and it can also be equipped with a communication link (over a shared data bus) to a central computer. Such units can minimize building operating costs by combining time-of-day controls with intelligent comfort gap selection and maximized self-heating.

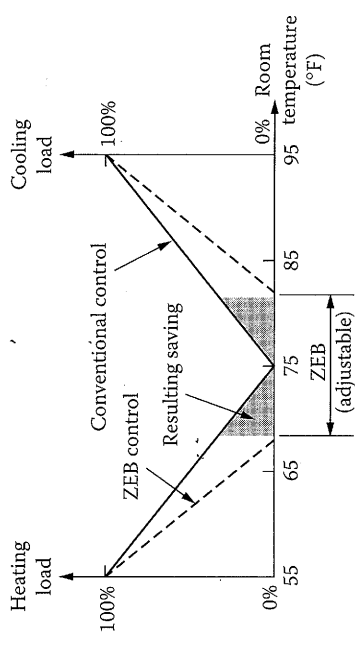


FIG. 8.2m

Zero energy band (ZEB) control is designed to save energy by not using any when the room is comfortable.

Zero Energy Band Control A recent addition to the available thermostat choices is the zero energy band design. The idea behind ZEB control is to conserve energy by not using any when the room is comfortable. As illustrated by Figure 8.2m, the conventional thermostat wastes energy by continuing to use it when the area's temperature is already comfortable. The "comfort gap" or "ZEB" on the thermostat is adjustable and can be varied to match the nature of the particular space.

ZEB control can be accomplished in one of two ways. The single set point and single output approach is illustrated on the left side of Figure 8.2n. Here the cooling valve fails closed and is shown to have an 8–11 PSIG (55–76 kPa) spring range, while the heating valve is selected to fail open and

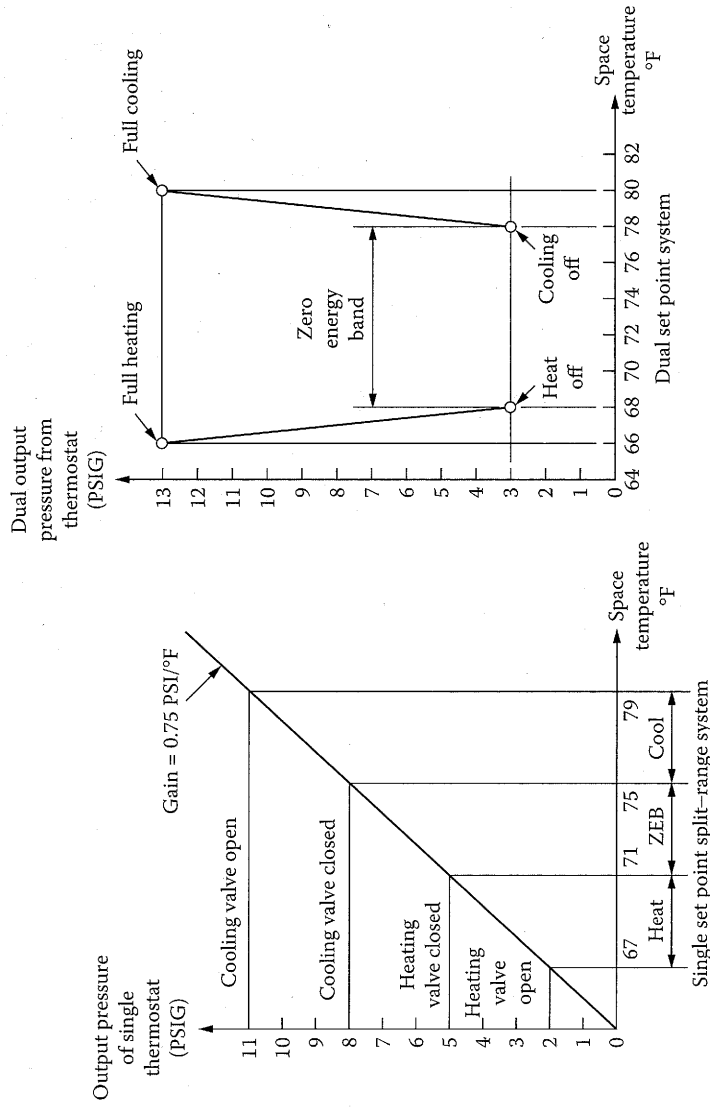


FIG. 8.2n

ZEB control schemes include the single setpoint split range approach (shown on left) and the dual set point approach (shown on right).

has a 2–5 PSIG (14–34 kPa) range. Therefore, between 5 and 8 PSIG (34 and 55 kPa), both valves are closed; no pay energy is expended while the thermostat output is within this range. The throttling range is usually adjustable from 5 to 25°F (3 to 13°C). Thus, if the ZEB is 30% of the throttling range, it can be varied from a gap size of 1.5°F (0.85°C) to 7.5°F (4.2°C) by changing the throttling range (or gain).

Although the split-range approach is a little less expensive than the dual set-point scheme (shown on the right of Figure 8.2n), it is also less flexible and more restrictive. The two basic limitations of the split-range approach are:

- 1) The gap width can be adjusted only by also changing the thermostat gain; maximum gap width is limited by the minimum gain setting of the unit.
- 2) The heating valve must fail open, which is undesirable in terms of energy conservation.

These limitations are removed when a dual set point, dual output thermostat is used. Here both valves can fail closed, and the bandwidth and the thermostat gain are independently adjustable. The gains of the heating and cooling thermostats are also independently adjustable. In Figure 8.2n, the heating thermostat is reverse-acting and the cooling thermostat is direct-acting.

The most recent advances in thermostat technology are the microprocessor-based units. These are programmable devices with memory capability. They can be monitored and reset by central computers, using telephone wires or other communication links. Microprocessor-based units can be supplied by continuously recharged backup batteries and accurate room temperature sensors. They can also operate without a host computer (in stand-alone mode). In this case, the user manually programs the thermostat to maintain various room temperatures as a function of the time of day and other considerations.

Gap Control and the Self-Heating Building

The winter and summer enthalpy settings of a building are illustrated in Figure 8.2o. The concept of gap control is simple: When comfort level in a zone is somewhere between acceptable limits, the use of “pay energy” is no longer justified. Allowing the zones to float between limits instead of maintaining them at arbitrarily fixed conditions can substantially reduce the operating cost of the building.

The savings come from two sources. First, there is a direct trade-off between the selected acceptable limits of discomfort and the yearly total of required degree days of heating and cooling. Second, there is the added side benefit that the building becomes self-heating during winter conditions. This occurs because during winter conditions, gap control automatically transfers the heat generated in the inside of the building to the perimeter areas, where heating is needed. This can result in long periods of building operation without the use of any pay energy.

Gap control can be looked upon as an override mode of control that is superimposed on the operation of the individ-

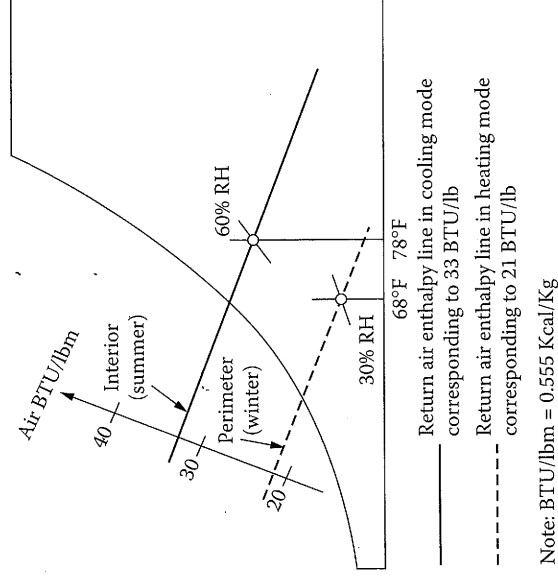


Fig. 8.2o

The building can be made self-heating, because the return air can transport about 10 BTU/lbm to the perimeter spaces, which in the winter, because of the windows, do require heat.

ual zone thermostats. As such, it can be implemented by all levels of automation, but the flexibility and ease of adjustment of the computerized systems make them superior in those applications in which the gap limits of the various zones are likely to change frequently.

Figure 8.2a illustrates the concept of comfort envelopes. Any combination of temperature and humidity conditions within such envelopes is considered to be comfortable. Therefore, as long as the space conditions fall within this envelope, there is no need to spend money or energy to change those conditions. This comfort gap is also referred to as zero energy band, meaning that if the space is within this band, no pay energy of any type will be used. This concept is very cost-effective.

When a zone of the airhandler in Figure 8.1b is within the comfort gap, its reheat coil is turned off and its VAV box is closed to the minimum flow required for air refreshment. When all the zones are inside the ZEB, the HW, CHW, and STM supplies to the airhandler are all closed and the fan is operated at minimum flow. When all other airhandlers are also within the ZEB, the pumping stations, chillers, cooling towers, and HW generators are also turned off.

With larger buildings that have interior spaces that are heating even in the winter, ZEB control can make the building self-heating. Optimized control systems in operation today are transferring the interior heat to the perimeter without requiring any pay heat until the outside temperature drops below 10 to 20°F (2.3 to 6.8°C). In regions in which winter temperature does not drop below 10°F (–2.3°C), ZEB control can eliminate the need for pay heat altogether. In regions farther north, ZEB control can lower the yearly heating fuel bill by 30–50%.

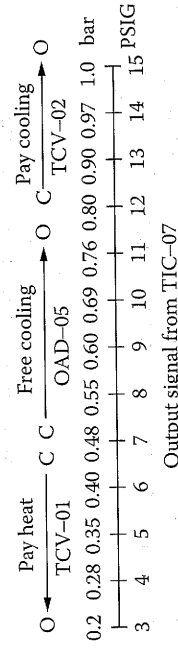
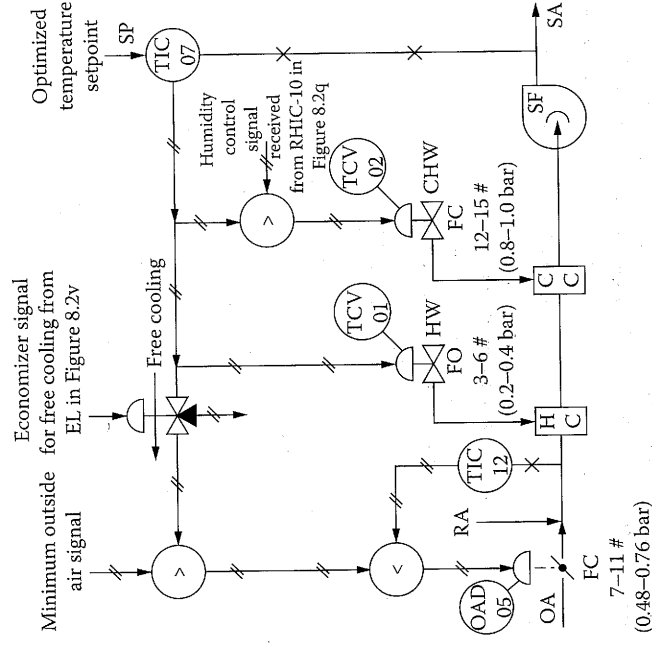


Fig. 8.2p

Illustration of a fully coordinated, pneumatic, split-range temperature control system. Such controls can reduce the yearly operating costs by more than 10%.⁸

Supply Air Temperature Control A substantial source of inefficiency in conventional HVAC control systems is the uncoordinated arrangement of temperature controllers. Two or three separate temperature control loops in series are not uncommon. For example, one of these uncoordinated controllers may be used to control the mixed air temperature, another to maintain supply (SA) temperature, and a third to control the zone-reheat coil. Such practice can result in simultaneous heating and cooling and, therefore, in unnecessary waste. Using a fully coordinated split-range temperature control system, such as that shown in Figure 8.2p, will reduce yearly operating costs by more than 10%.

In this control system, the SA temperature set point (set by the temperature controller, TIC-07) is continuously modulated to follow the load. The methods of finding the correct set point will be discussed under Optimizing Strategies. The loop automatically controls all heating or cooling modes. When the TIC-07 output signal is low—3–6 PSIG (20.7–41.3 kPa)—heating is done by TCV-01. As the output signal reaches 6 PSIG (41.3 kPa), heating is terminated; if free cooling is available, it is initiated at 7 PSIG (48.2 kPa). When the output signal reaches 11 PSIG (75.8 kPa)—the point at which OAD-05 is fully open—the cooling potential represented by free cooling is exhausted, and at 12 PSIG (82.7 kPa), “pay cooling” is

started by opening TCV-02. In such split-range systems, the possibility of simultaneous heating and cooling is eliminated. Also eliminated are interactions and cycling.

Figure 8.2p also shows some important overrides. TIC-12, for example, limits the allowable opening of OAD-05, so that the mixed-air temperature will never be allowed to drop to the freezing point and permit freeze-up of the water coils.

The minimum outdoor air requirement signal guarantees that the outside air flow will not be allowed to drop below this limit.

The economizer signal allows the output signal of TIC-07 to open OAD-05 only when “free cooling” is available. (A potential for free cooling exists when the enthalpy of the outdoor air is below that of the return air.)

Finally, the humidity controls will override the TIC-07 signal to TCV-02 when the need for dehumidification requires that the supply air temperature be lowered below the set point of TIC-07.

Humidity Controls

Humidity in the zones is controlled according to the moisture content of the combined return air (see Figure 8.2q). The

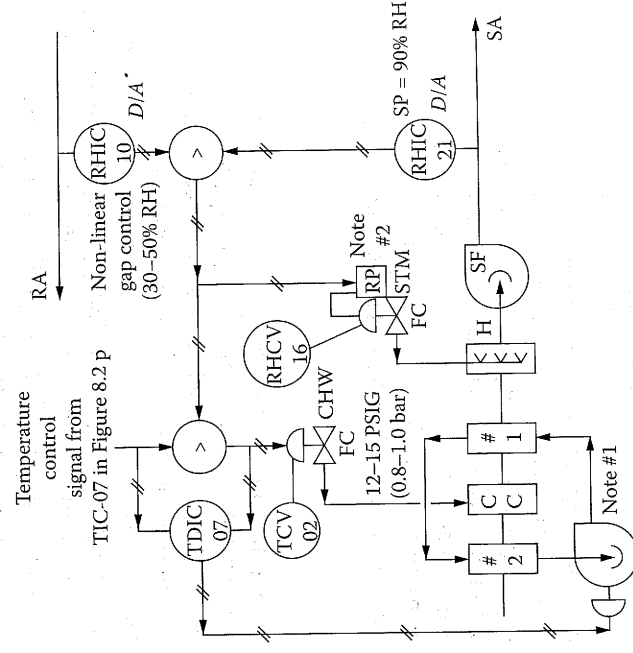


Fig. 8.2q

Humidity is controlled in the combined return air. Note 1: When the need for dehumidification (in the summer) overcools the supply air and therefore increases the need for reheat at the zones, this pump-around economizer loop is started. TDIC-07 will control the pump to “pump around” only as much heat as is needed. Note 2: This reversing positioner functions as follows:

Input from RHIC-10	Output to RHCV-16
3 PSIG = 0%	100% (open)
9 PSIG = 50%	0% (closed)

process controlled by RHIC-10 is slow and contains large dead-time and transport-lag elements. In other words, a change in the SA humidity will not be detected by RHIC-10 until some minutes later. During the winter, it is possible for RHIC-10 to demand more and more humidification. To prevent possible saturation of the supply air, the RHIC-10 output signal is limited by RHIC-21. In this way, the moisture content of the supply air is never allowed to exceed 90% RH.

For best operating efficiency, a nonlinear controller with a neutral band is used at RHIC-10. This neutral band can be set to a range of humidity levels—say, between 30% and 50% RH. If the RA is within these limits, the output of RHIC-10 is at 50%, and neither humidification nor dehumidification is demanded. This arrangement can lower the cost of humidity control during the spring and fall by approximately 20%.

The same controller (RHIC-10) controls both humidification (through the relative humidity control valve, RHCV-16) and dehumidification (through the temperature control valve, TCV-02) on a split-range basis. As the output signal increases, the humidifier valve closes, between 3 and 9 PSIG (20.7 and 62 kPa). At 9 PSIG (62 kPa), RHCV-16 closes and remains so, as the output signal increases to 12 PSIG (82.7 kPa). At this condition, TCV-02 starts to open.

Dehumidification is accomplished by cooling through TCV-02. This chilled water valve is controlled by humidity (RHIC-10) or temperature (TIC-07). The controller that requires more cooling will be the one allowed to throttle TCV-02.

Subcooling the air to remove moisture can substantially increase operating costs if this energy is not recovered. The dual penalty incurred for overcooling for dehumidification purposes is the high chilled water cost and the possible need for reheat at the zone level. The savings from a pump-around economizer can eliminate 80% of this waste. In this loop, whenever TDIC-07 detects that the chilled water valve (TCV-02) is open more than would be necessary to satisfy TIC-07, the pump-around economizer is started. This loop in coil #1 reheats the dehumidified supply air, using the heat that the pump-around loop removed from the outside air in coil #2 before it entered the main cooling coil.

In this way, the chilled-water demand is reduced in the cooling coil (TCV-02), and the need for reheating at the zones is eliminated. Although Figure 8.2q shows a modulating controller setting the speed of a circulating pump, it is also possible to use a constant-speed pump operated by a gap switch.

Outdoor Air Controls

Outdoor air is admitted to satisfy the requirements for fresh air or to provide free cooling. Both control loops are shown in Figure 8.2i.

The minimum requirement for fresh outdoor air while the building is occupied is usually 10% of the airhandler's capacity. In most advanced control systems, this value is not controlled as a fixed percentage but as a function of the number

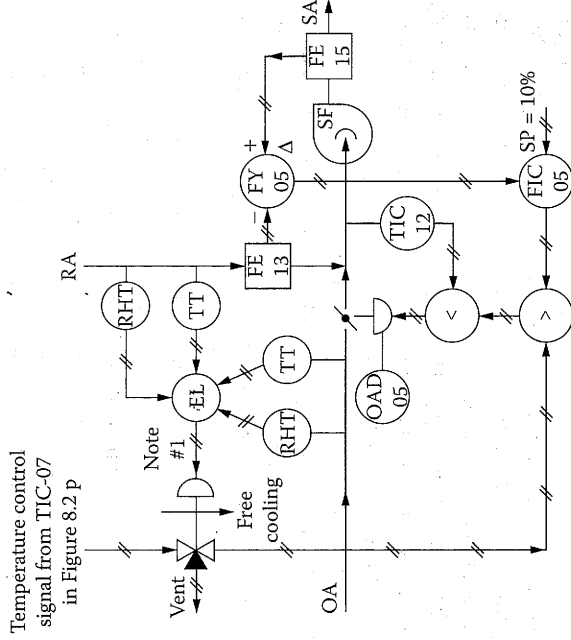


FIG. 8.2r

Outside air control loops provide fresh air or free cooling. Note: The enthalpy logic unit (EL) compares the enthalpies of the outside and return air and vents its output signal if free cooling is available. Therefore, the economizer cycle is initiated whenever $H_{oa} < H_{ra}$.

of people in the building or of the air's carbon dioxide content. In most conventional systems, the minimum outdoor air is provided by keeping 10% of the area of the outdoor air damper always open when the building is occupied.

This method is inaccurate, because a constant damper opening does not result in a constant air flow. This flow varies with fan load, because changes in load will change the fan's suction pressure and will therefore alter the ΔP across the damper. This conventional design results in waste of air-conditioning energy at high loads and insufficient air refreshment at low loads.

The control system depicted in Figure 8.2r reduces operating costs while maintaining a constant minimum rate of air refreshment, which is unaffected by fan loading. Direct measurement of outdoor airflow is usually not possible because of space limitations. For this reason, Figure 8.2r shows the outdoor air flow as being determined as the difference between FE-15 and FE-13. FIC-05 controls the required minimum outdoor airflow by throttling OAD-05.

CO₂-Based Ventilation In conventional installations the amount of outdoor air admitted is usually based on one of the following criteria:

- 0.1–0.25 cfm/ft² (30–76 lpm/m²) of floor area
- 10–25% of total air supply rates
- About 5 cfm (25 lps) volumetric rate per person

These criteria all originated at a time when energy conservation was no serious consideration; therefore, their aim

was to provide simple, easily enforceable rules that will guarantee that the outdoor air intake always exceeds the required minimum. Today the goal of such systems is just the opposite: It is to make sure that air quality is guaranteed at minimum cost. As the floor does not need oxygen—only people do—some of the above rules make little sense.

There is a direct relationship between savings in building operating costs and reduction in outdoor air admitted into the building. According to one study in the United States,⁴ infiltration of outdoor air accounted for 55% of the total heating load and 42% of the total cooling load. Another survey⁵ showed that 75% of fuel oil consumed in New York City schools was devoted to heating ventilated air. Because building conditioning accounts for nearly 20% of all the energy consumed in the U.S.,⁶ optimized admission of outdoor air can make a major contribution to reducing our national energy budget. This goal can be well served by CO₂-based ventilation controls.

The purpose of ventilation is not to meet some arbitrary criteria, but to maintain a certain air quality in the conditioned space. Smoke, odors, and other air contaminant parameters can all be correlated to the CO₂ content of the return air.⁷ This then becomes a powerful tool of optimization, because the amount of outdoor air required for ventilation purposes can be determined on the basis of CO₂ measurement, and the time of admitting this air can be selected so that the air addition will also be energy efficient. With this technique, health and energy considerations will no longer be in conflict, but will complement each other.

CO₂-based ventilation controls can easily be integrated with the economizer cycle and can be implemented by use of conventional or computerized control systems. Because the rate of CO₂ generation by a sedentary adult is 0.75 cfm (27 lph), control by CO₂ concentration will automatically reflect the level of building occupancy.⁸ Energy savings of 40% have been reported⁹ by converting conventional ventilation systems to intermittent CO₂-based operation.

Economizer Cycles The full use of free cooling can reduce the yearly air-conditioning load by more than 10%. The enthalpy logic unit (EL) in Figure 8.2r will allow the temperature-controller signal (TIC-07 in Figure 8.2p) to operate the outdoor air damper whenever free cooling is available. This economizer cycle is therefore activated whenever the enthalpy of the outdoor air is below that of the return air.

Free cooling can also be used to advantage while the building is unoccupied. Purging the building with cool outdoor air during the early morning results in cooling capacity being stored in the building structure, reducing the daytime cooling load.

The conventional economizers—such as the one shown in Figure 8.2r—are rather limited devices for two reasons. First, they determine the enthalpy of the outdoor and return air streams, using somewhat inaccurate sensors. Secondly, although they consider the free cooling potential of the outside

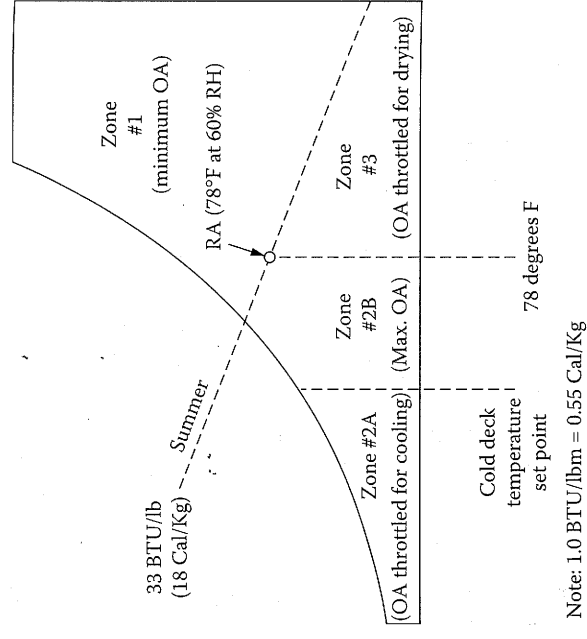


FIG. 8.2s

Free cooling and drying can often be obtained in the summer, depending on the zone where the outside air falls relative to the return air.

air, they disregard all the other possibilities of using the outdoor air to advantage.

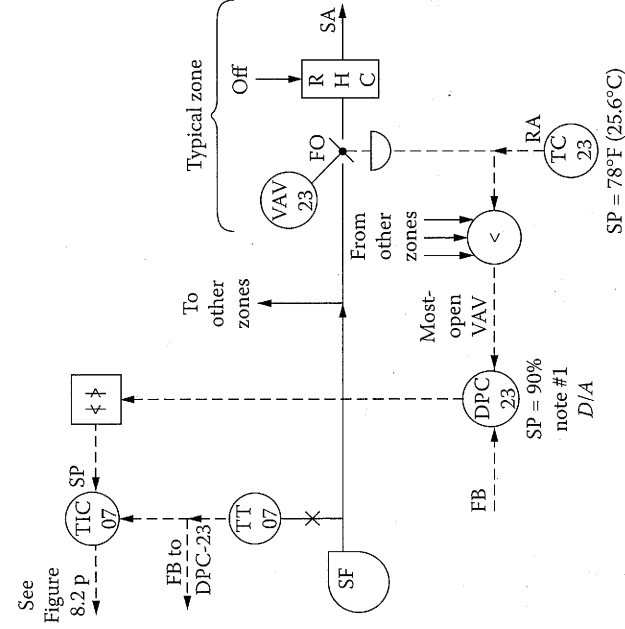
Advanced, microprocessor-based economizers overcome both of these limitations. They use accurate sensors and the psychrometric chart to evaluate all potential uses of outside air, not only free cooling. Figure 8.2s illustrates the various zones of operation, based on the relative conditions of the outside and return airs.

If the enthalpy of the outside air falls in zone 1 (that is, if its BTU content exceeds 33), no free cooling is available; therefore, the use of outside air should be minimized in the summer. In the winter or fall, it is possible that the enthalpy of the outside air on sunny afternoons will exceed the return air enthalpy, which in the winter is about 21 BTU/lbm (11.6 cal/kg). Under such conditions, “free heating” can be obtained by admitting the outside air in zone 1.

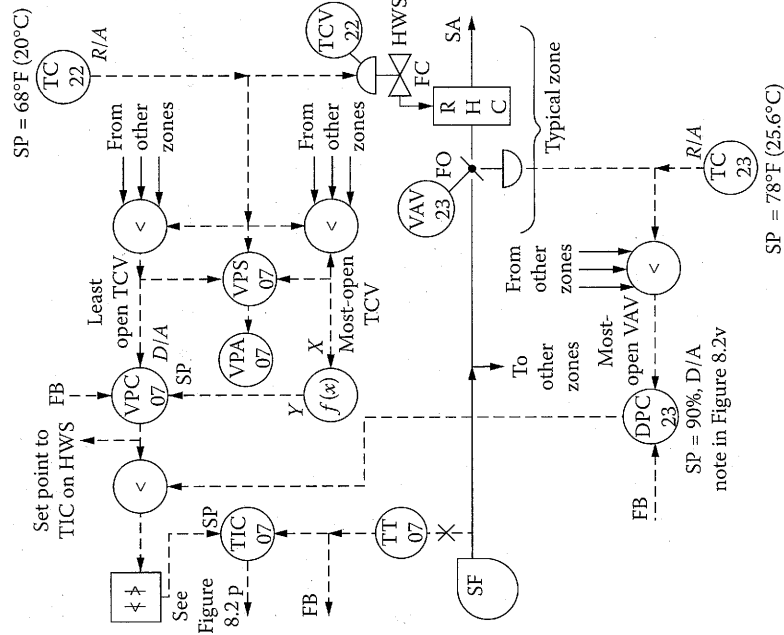
If the condition of OA corresponds to zone 2 (BTU < 33 and temperature < 78°F), free cooling is available. If the condition of OA corresponds to zone 3 (BTU < 33 and temperature > 78°F), free dehumidifying (latent cooling) is available.

When there are both cold and hot air ducts in the building (dual duct system), the control system in zone 2 will function differently depending on whether the outside air temperature is above or below the cold deck temperature. If it is above that temperature (zone 2B), maximum (100%) outside air can be used; if it is below that temperature (zone 2A), the use of outside air needs to be modulated or time-proportioned.

Therefore, in zone 2A, where the outside air is cooler than the cold deck temperature, free cooling is available, but only some of the total potential can be used. The OA damper

**FIG. 8.2v**

This method of air supply temperature optimization in summer (cooling mode) should be used if fan operating costs are less than cooling costs. Note: The damper position controller (DPC) has integral action only, with its setting being 10 times the integral time of TIC-07. External feedback is provided to eliminate reset wind-up.

**FIG. 8.2w**

This control system optimizes the water and air temperatures in both summer and winter. See also Figures 8.2u and 8.2v.

means of reducing operating costs in buildings. The savings can amount to more than 30% during the transitional seasons.

When the temperature in one of the zones reaches 78°F (26°C), the air supply temperature set point will be lowered by DPC-23 and the air-side controls will be automatically switched to cooling (as depicted in Figure 8.2v). If, at the same time, some other zone temperatures drop below 68°F (20°C), requiring heating, their heat demand will have to be met by the heat input of the zone-reheat coils only. This mode of operation is highly inefficient because of the simultaneous cooling and reheating of the air.

Fortunately, this combination of conditions is highly unlikely, because under proper design practices, the zones served by the same airhandler should display similar load characteristics. The advantage of the control loop in Figure 8.2w is that it can automatically handle any load or load combination, including this unlikely, extreme case.

AutoBalancing of Buildings

In computerized building control systems, the optimization potentials are greater than those that have been discussed up to this point. When all zone conditions are detected and controlled by the computer, it can optimize not only the normal operation but also the start-up of the building.

The optimization of airhandler fans is directed at two goals simultaneously. The first goal is to find the optimum value for the set point of the supply air pressure controller (PIC-19 in Figure 8.2f). Generally, the supply air pressure is at an optimum value when it is at the lowest possible value, while all loads are satisfied. As the supply pressure is lowered, the fan operating cost is reduced, but with lowered supply pressures the VAV boxes serving the individual zones (VAV-23 in Figure 8.2b) will have to open up so that the airflow to the zones will not be reduced. Therefore, the optimum setting for PIC-19 is that pressure at which the most open VAV box is nearly 100% open, while all other VAV boxes are less than 100% open.

The second goal of optimization is to automatically rebalance the air distribution in the building as the load changes. If the VAV boxes (VAV-23 in Figure 8.2b) are not pressure independent (are not able to maintain constant air flow when the supply pressure changes), manual rebalancing is required every time the load distribution changes. Naturally this is a very labor-demanding and inefficient operation. The optimization strategy described below serves the multiple purposes of automatic rebalancing and finding the optimum set points for the supply air pressure and temperature.

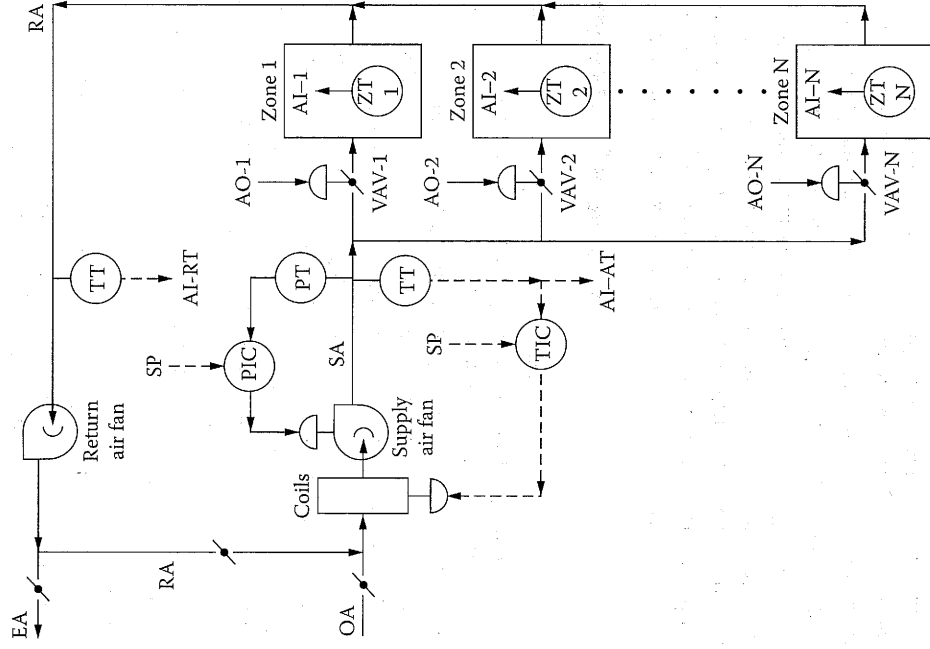
Figure 8.2x illustrates an airhandler that is serving several zones. The abbreviations used in that figure and in the algorithm tables that follow are listed below:

AI-1 to AI-N: Analog inputs (zone temperatures)

AI-AT: Analog input (air supply temperatures)

AI-RT: Analog input (return air temperature)

AO-1 to AO-N: Analog output (zone VAV opening)

**FIG. 8.2x**

Air-handler optimization and auto-balancing can be handled efficiently by computer.

AT: Supply air temperature

EA: Exhaust air

OA: Outside air

PIC: Pressure controller (supply air)

PT: Pressure transmitter

RA: Return air

RT: Return air temperature

SA: Supply air

SP: Set point

TH: Upper limit of comfort zone (Figure 8.2a) (maximum allowable zone temperature)

TIC: Temperature controller (air supply)

TL: Lower limit of comfort zone (Figure 8.2a) (minimum allowable zone temperature)

VAV-1 to VAV-N: Variable air volume boxes

ZT-1 to ZT-N: Zone temperatures

ZT5-1 to ZT5-N: ZT 5 min after start-up

ZT10-1 to ZT10-N: ZT 10 minutes after start-up

XMIN: Minimum VAV opening required for ventilation

XSET-1 to XSET-N: Initial VAV opening after "start-up"

XMAX-1 to XMAX-N: Maximum limit on VAV opening during normal operation

Start-Up Algorithm

All VAV boxes are set to their minimum openings required for ventilation purposes (XMIN), such as 25%. Therefore, at the time of start-up, AO-1 through AO-N are all set for 25%. PIC is set to mid-scale; therefore, the start-up value of its SP = 50%. TIC is set for $(TL + 25)^\circ\text{F}$ in the heating mode and for $(TH - 25)^\circ\text{F}$ in the cooling mode.

After 5 min of operation, the zone temperatures are detected (ZT5-1, ZT5-2, and so on), and after 10 min of operation they are detected again (ZT10-1, ZT10-2, and so on). At the end of the first 10 min of operation, the supply air temperature is also measured as AT10 and the return air temperature as RT10.

Once the above readings are obtained, they are entered into a table such as Table 8.2y, which serves as the basis for determining the required start-up openings of each of the VAV boxes (XSET-1, XSET-2, and so on). The purpose of this table is to select the initial opening for each VAV box in a logical manner. Therefore, if the zone temperature after 10 minutes of operation is already within 5°F of reaching the comfort zone, the VAV box can be left at its minimum opening.

If comfort is not yet within 5°F , a higher opening is needed. The initial VAV opening is increased on the basis of the zone's performance during the previous 5 minutes. The larger the temperature change experienced by the zone during the previous 5 minutes, the sooner it will reach the comfort zone and therefore the smaller the opening that is required. By this logic, the VAV boxes on those zones that are furthest from comfort and that are moving most slowly toward comfort will be given the highest openings.

Normal Algorithm for VAV Throttling

The initial VAV opening for each zone (XSET), which is determined by the methods above, is then used as the maximum limit on the VAV opening (XMAX) during the first 5 minutes of normal operation. The value of XMAX is reevaluated every 5 minutes, as shown in Table 8.2z. The logic here is to increase the maximum limit on VAV opening (XMAX) to any zone in which the VAV has been open on its maximum limit for 5 minutes. Similarly, this logic will lower the XMAX limit if the VAV damper was at its XMIN during the previous 5 minutes. If a VAV damper has been throttled somewhere in between these two limits (XMAX and XMIN), its limit will not be altered. The change increment of 10% shown in Table 8.2z is adjustable for maximum flexibility.

The algorithm described above and illustrated in Table 8.2z guarantees that changes in load distribution will not result in starving some zones; the building will be automatically rebalanced in an orderly manner. The value of XMAX from Table 8.2z and the permanent values of XMIN, determined by ventilation requirements, are used to reevaluate the individual VAV openings every 2 minutes, as described by the algorithm in Table 8.2aa.

TABLE 8.2y
Algorithm to Determine Start-Up Openings of Individual VAV Boxes (XSET)

Input Conditions		Output	
Operating Mode	Approach between Zone and Supply Temperatures	Amount of Temperature Change During Last 5 Minutes of Start-Up	Initial Value of XSET to be Used for AO-1 to AO-N (%)
Heating (AT > RT)	(TL - ZT10) > 5°F	(ZT10 - ZT5) < 0.5°F	100
		(ZT10 - ZT5) 0.5-1°F	90
		(ZT10 - ZT5) 1-1.5°F	80
		(ZT10 - ZT5) 1.5-2°F	70
		(ZT10 - ZT5) 2-3°F	60
		(ZT10 - ZT5) 3-4°F	50
Cooling (AT < RT)	(ZT10 - TH) > 5°F	(ZT10 - ZT5) 4-5°F	40
		(ZT10 - ZT5) > 5°F	30
		Disregard	25
		(ZT5 - ZT10) < 0.5°F	100
		(ZT5 - ZT10) 0.5-1°F	90
		(ZT5 - ZT10) 1-1.5°F	80
Cooling (AT < RT)	(ZT10 - TH) > 5°F	(ZT5 - ZT10) 1.5-2°F	70
		(ZT5 - ZT10) 2-3°F	60
		(ZT5 - ZT10) 3-4°F	50
		(ZT5 - ZT10) 4-5°F	40
		(ZT5 - ZT10) > 5°F	30
		Disregard	25

Note: °C = (F - 32)/1.8.

The main optimizing and auto-balancing feature of this algorithm is that whenever a zone is inside the comfort gap, its VAV opening is reduced to XMIN. This reduces the load on the fans and also provides more air to the zones experiencing the highest loads.

Optimization of Air Supply Pressure and Temperature

Optimization means that the load is met at minimum cost. The cost of operating an airhandler is the sum of the cost of

air transportation and conditioning. These two cost factors tend to change in opposite directions; minimizing the cost of one will increase the cost of the other. Therefore, it is important to monitor both the transportation and the conditioning costs continuously and to minimize the larger one when optimizing the system. Computerized control systems allow these costs to be readily calculated on the basis of utility costs and quantities.

TABLE 8.2aa
Algorithm to Determine Analog Outputs, Setting the Openings of VAV Boxes

Input Conditions		Output	
Operating Mode	Control Criteria	Required VAV Opening: AO-1 to AO-N is to be Equal	
Heating (AT > ZT)	ZT < (TL - 1)	XMAX	
	(TL - 1) < ZT < (TL + 1)	No change	
	ZT > (TL + 1)	XMIN	
Cooling (AT < ZT)	ZT > (TH + 1)	XMAX	
	(TH - 1) < ZT < (TH + 1)	No change	
	ZT < (TH - 1)	XMIN	

TABLE 8.2z
Reevaluation of Value of XMAX

Input Conditions		Output	
Has VAV been Continuously Open to its XMAX During Last 5 Minutes?	Has VAV been Continuously Throttled to its XMIN During the Last 5 Minutes?	Incremental Change in Value of XMAX at the End of 5-Minute Period	
Yes	Yes	Leave XMAX = XMIN	
No	No	Increase by 10%	
	Yes	Decrease by 10%	
	No	Leave as is	

TABLE 8.2bb

Optimization of Supply Air Pressure and Temperature, When Fan Costs Exceed Conditioning Costs (Frequency = 5 min)

VAV Status	Airhandler Mode	Is TIC SP at Its Limit?	Incremental Ramp Adjustment in the Set Points of	
			TIC	PIC
None at 100% for 15 minutes continuously	Heating (AT > RT)	Yes, at max.	-2°F	N.C.* (at min.)
		No	-1°F	N.C. (at min.)
	Cooling (AT < RT)	Yes, at min.	+2°F	N.C. (at min.)
		No	+1°F	N.C. (at min.)
Not more than one at 100% for more than 30 minutes continuously	Heating (AT > RT)	Yes, at max.	N.C. (at max.)	N.C.
		No	N.C.	N.C.
	Cooling (AT < RT)	Yes, at min.	N.C. (at max.)	N.C.
		No	N.C.	N.C.
More than one at 100% for more than 30 minutes continuously	Heating (AT > RT)	Yes, at max.	(at max.)	+0.25 in. H ₂ O
		No	+1°F	N.C.
	Cooling (AT < RT)	Yes, at min.	(at min.)	+0.25 in. H ₂ O
		No	-1°F	N.C.
More than one at 100% for more than 60 minutes continuously	Heating (AT > RT)	Yes, at max.	(at max.)	+0.5 in. H ₂ O
		No	+2°F	N.C.
	Cooling (AT < RT)	Yes, at min.	(at min.)	+0.5 in. H ₂ O
		No	-2°F	N.C.

*N.C. = No change is made at the end of that 5-minute period.

For example, if the transportation cost exceeds the conditioning cost, the optimization goal is to minimize fan operation. This is achieved by conditioning the space with as little air as possible. The quantity of air transported can be minimized if each pound of air is made to transport more conditioning energy; that is, if each pound of air carries more cooling or heating BTUs. Therefore, when the goal is to minimize fan costs, the air supply pressure is held as low as possible, and the air supply temperature is maximized in the winter and minimized in the summer. Fan costs tend to exceed conditioning costs when the loads are low, such as in the spring or fall, or when the economizer cycle is used to provide free cooling.

Table 8.2bb describes the algorithm used to achieve this goal. When none of the VAV boxes (Figure 8.2x) are fully open, indicating that all loads are well satisfied, the air pressure (PIC set point) is kept at a minimum, and the air temperature (TIC set point) is lowered in the winter and raised in the summer. When more than one VAV boxes are fully open, the air supply temperature is increased in the winter (lowered in the summer). When its limit is reached, the algorithm will start raising the PIC set point.

Table 8.2cc describes the algorithm used when the conditioning costs are higher than the fan operating costs. This is likely to be the case when the loads are high, such as in the

summer or the winter. Under such conditions, the supply pressure is maximized before the supply air temperature is increased in the winter or lowered in the summer. When none of the VAV boxes in Figure 8.2x are fully open, the PIC set point is lowered, while the TIC set point is at or near minimum in the winter (maximum in summer). When more than one VAV box is fully open, the PIC set point is increased to its maximum setting. When that is reached, the supply temperature starts to be increased in the winter (decreased in the summer).

The algorithms described above provide the dual advantages of automatic balancing and minimum operating cost. They eliminate the need for manual labor or for the use of pressure-independent VAV boxes, while reducing operating cost by about 30%. They also provide the flexibility of assigning different comfort envelopes (different TL and TH values) to each zone. Thereby, as occupancy or use changes, the comfort zone assigned to the particular space can be changed automatically.

ELIMINATION OF CHIMNEY EFFECTS

In high-rise buildings, the natural draft resulting from the chimney effect tends to pull in ambient air at near ground elevation and to discharge it at the top of the building.

TABLE 8.2cc*Optimization of Supply Air Pressure and Temperature, When Conditioning Costs Exceed Fan Costs (Frequency = 5 min)*

VAV Status	Airhandler Mode	Is PIC Set Point at Its Maximum?	Incremental Ramp Adjustment in the Set Points of	
			TIC	PIC
None at 100% for 15 minutes continuously	Heating (AT > RT)	Yes	-1°F	-0.5 in. H ₂ O
		No	N.C.* (at min.)	-0.25 in. H ₂ O
	Cooling (AT < RT)	Yes	+1°F	-0.5 in. H ₂ O
		No	N.C. (at max.)	-0.25 in. H ₂ O
Not more than one at 100% for more than 30 minutes continuously	Heating (AT > RT)	Yes	N.C.	N.C. (at max.)
		No	N.C.	N.C.
	Cooling (AT < RT)	Yes	N.C.	N.C. (at max.)
		No	N.C.	N.C.
More than one at 100% for more than 30 minutes continuously	Heating (AT > RT)	Yes	+1°F	(at max.)
		No	N.C.	+0.25 in. H ₂ O
	Cooling (AT < RT)	Yes	-1°F	(at max.)
		No	N.C.	+0.25 in. H ₂ O
More than one at 100% for more than 60 minutes continuously	Heating (AT > RT)	Yes	+2°F	(at max.)
		No	N.C.	+0.5 in. H ₂ O
	Cooling (AT < RT)	Yes	-2°F	(at max.)
		No	N.C.	+0.5 in. H ₂ O

* N.C. = No change.

Although eliminating the chimney effect can lower the operating cost by approximately 10%, few systems with this capability are yet in operation.

Figure 8.2dd shows the required pressure controls. The key element of this control system is the reference riser, which allows all pressure controllers in the building to be referenced to the barometric pressure of the outside atmosphere at a selected elevation. Using this pressure reference allows all zones to the operated at 0.1 in. H₂O (25 Pa) above that reference pressure (PC-7) and permits this constant pressure to be maintained at both ends of all elevator shafts (PC-8 and -9).

If the space pressure is the same on the various floors of a high-rise building, there will be no pressure gradient to motivate the vertical movement of the air, and as a consequence, the chimney effect will have been eliminated. A side benefit of this control strategy is the elimination of all drafts or air movements between zones, which also minimizes the dust content of the air. Another benefit is the capability of adjusting the "pressurization loss" of the building by varying the setting of PC-7, -8, and -9.

Besides reducing operating costs, the use of pressure-controlled elevator shafts increases comfort because drafts and the associated noise are eliminated.

Figure 8.2dd also shows the use of cascaded fan controls. The set points of the cascade slaves (PC-2 and PC-5) are programmed so that the air pressure at the fan is adjusted as

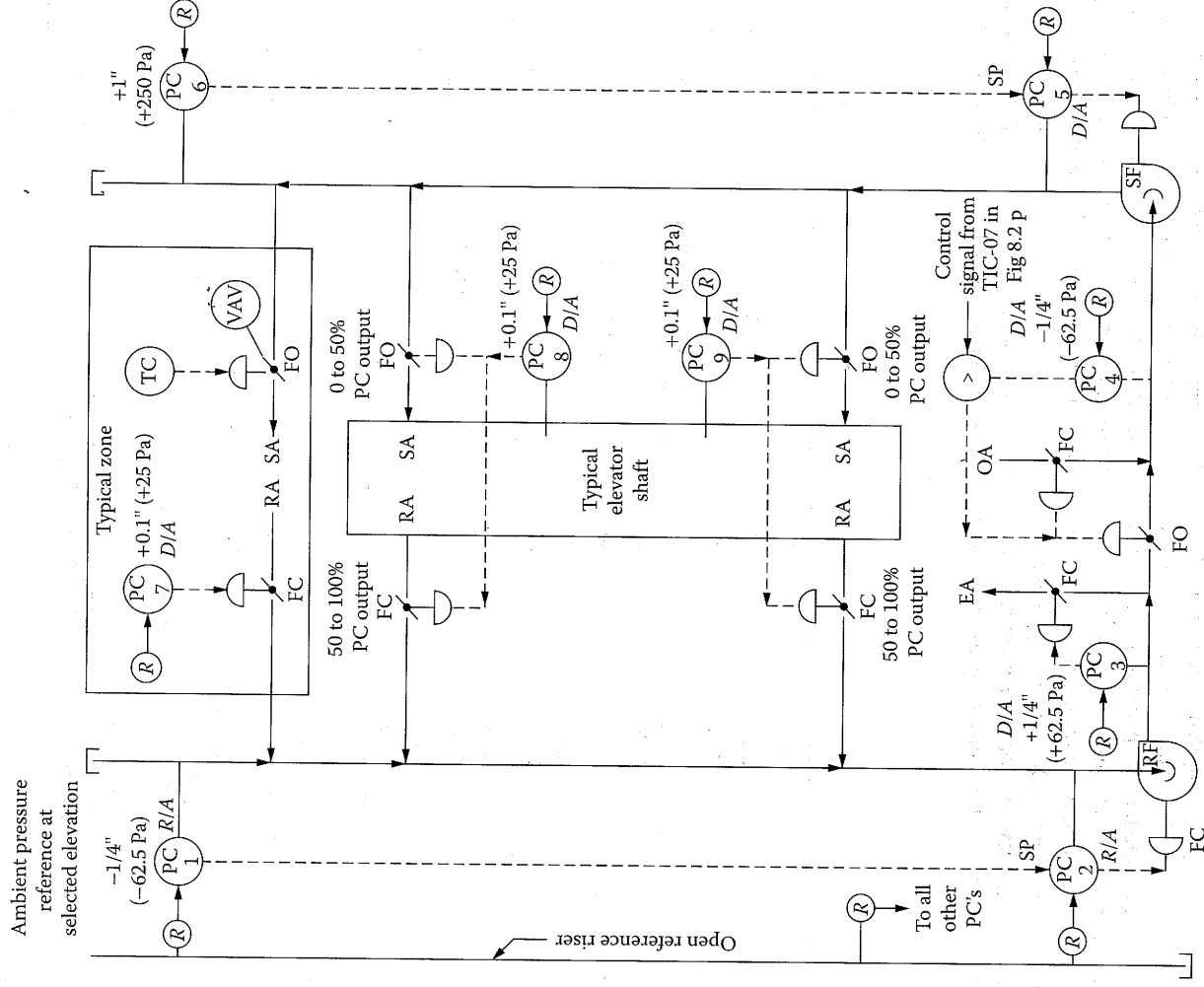
the square of flow and the pressure at the end of the distribution headers (cascade masters PC-1 and PC-6) remains constant. This control approach results in the most efficient operation of variable-air volume fans.

If the building is maintained at a constant pressure that equals the pressure at ground elevation plus 0.1 in. H₂O, this will result in higher pressure differentials on the higher floors, as the barometric pressure on the outside drops. Therefore, air losses due to out-leakage and pressure differentials on the windows will both rise. If the pressure reference is taken at some elevation above ground level, these effects will be reduced on the upper floors, but on the lower floors the windows will be under positive pressure from the outside and air infiltration will be experienced.

CONCLUSIONS

The airhandler is just one of the industrial unit operations. The process of air conditioning is similar to all other industrial processes. Fully exploiting state-of-the-art instrumentation and control results in dramatic improvements. There are few other processes in which the use of optimization and of instrumentation know-how alone can halve the operating cost of a process.

The control and optimization strategies described in this chapter can be implemented by pneumatic or electronic

**FIG. 8.2ddd**

Chimney effects in high-rise buildings can be eliminated by using the proper pressure controls.

instruments and can be controlled by analog or digital systems. The type of hardware used in optimization is less important than the understanding of the process and of the control concepts that are to be implemented. The main advantage of digital and computerized systems is their flexibility and convenience in making changes, without the need to modify equipment or wiring.

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