CONTROL SYSTEMS FOR HEATING, VENTILATING, AND AIR CONDITIONING

SIXTH EDITION

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by

Roger W. Haines
Consultant
Laguna Woods, CA

Douglas C. Hittle
Director, Solar Energy Applications Laboratory
Colorado State University
Ft. Collins, CO

Roger W. Haines
Consultant
Laguna Woods, CA

Douglas C Hittle
Director, Solar Energy Applications Laboratory
Colorado State University
Ft. Collins, CO

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README first, please

This book is intended to provide guidance to engineers who are designing controls for heating, ventilating and air-conditioning (HVAC) systems and to those who maintain and trouble shoot these systems.

Since our last edition, the industry has seen a major transition from analog pneumatic control to digital electronic control. New systems are often a mix of digital controllers with pneumatic actuation, especially if big valves or damper actuators are involved. Simultaneously, all electronic systems have piggy backed on the ever expanding networking technology in buildings (and beyond) and offer expanded control options. We have tried to highlight this in a revised chapter 10.

Designers of new systems are referred to chapters 6, 7, and 10.

Another issue of heightend concern is indoor air quality. Chapter 6 discusses how outdoor air control can insure that the requirements of ASHRAE Standard 62 can be met.

Achieving good environmental control in buildings requires a well designed and commisssioned HVAC system. The control system makes or breaks the design. We hope our book will provide useful information to practioners, especially those new to our profession.

1

Control Theory and Terminology

1.1 INTRODUCTION

The purpose of this book is to discuss the design of control systems for heating, ventilating and air conditioning systems. Its intent is to help develop an understanding of controls and control systems, and the air conditioning systems to which they are applied. Out of this you should develop a philosophy of design that will enable you to cope not only with the basic systems discussed here, but with the unusual and special requirements that continue to arise as air conditioning becomes more sophisticated.

The term "heating, ventilating and air conditioning" (HVAC) covers a wide range of equipment, from, for example, a kerosene stove, to the large and sophisticated complex of equipment required for major high-rise building complexes.

Control also may vary from the handwheel adjustment of the kerosene stove wick to the elaborate, computerized system in the World Trade Center.

This book will discuss most of the HVAC systems in use today, together with the methods of control that may be applied to them. The word "together" should be emphasized. The HVAC system, its control system and the building in which they are installed are inseparable parts of a whole. They interact with one another in many ways, so that neglect of any element may cause loss of controllability.

Unfortunately, many systems perform poorly because designers neglected the importance of building controllabe HVAC systems and providing well-engineered control systems. It is possible to design good HVAC systems and controls at a reasonable cost. Designers have a duty to provide the owner with the best possible system within budget limits, not necessarily the cheapest. The cheapest may be the most expensive in the long term, in operating cost and owner dissatisfaction. The best system is one that will provide the required degree of comfort for the application with the least expenditure of energy. That degree of comfort is, of course, a function of the application; that is, we should expect closer control of temperature in a hotel bedroom than in the same hotel's kitchen.

1.2   WHAT IS "CONTROL"?

Although many HVAC control systems appear to be, and are, complicated, the most elaborate system may be reduced to a few fundamental elements. Let us return to that kerosene stove. We are cold, so we strike a match and light the wick (after checking the fuel supply). We turn the wick up high. After a while it gets warm. We sense this and turn the handwheel to lower the wick and get less heat. Or, we may turn it clear down, shutting off the heat entirely.

Here are all the elements of a closed-loop control system. The controlled variable is the air temperature in the room. The process is the stove. The wick is the controlled device. The sensor and controller are represented by the person in the room. You will note that a person is not a really sensitive controller. Nonetheless he/she performs the basic function of the sensor-controller, which is to measure the controlled variable, compare it with a set point (here, the personal sensation of comfort), and adjust the controlled device.

Notice that three elements are necessary for a control system: a sensor, a controller, and a controlled device. The controlled device affects the process in a way that causes a change in the controlled variable. In this

text the process is usually some part of the heating, ventilating, and air conditioning systems.

1.3 ELEMENTARY CONTROL SYSTEM

Figure 1-1 illustrates an elementary control system. It shows air flowing through a heating coil in a duct. The sensor measures the temperature of the air downstream of the coil and passes the information to the controller. The controller compares the air temperature with a set point and sends a signal to open or close the hot-water valve (the controlled device) as required to maintain a correspondence between the air temperature and the set point. This is a closed-loop system, in which the change in temperature caused by a change in valve position (and/or load) will be sensed, and additional adjustments will be made as necessary. The air temperature is the controlled variable.
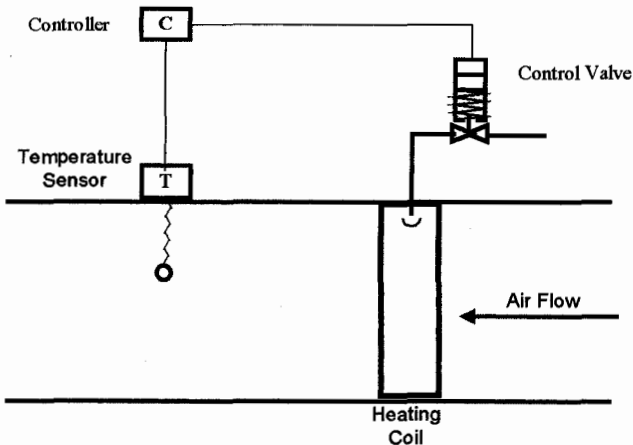
Figure 1-1 Elementary control system

Most control systems fall in the closed-loop classification but open-loop systems are sometimes used. In an open-loop system the sensor is not directly affected by the action of the controlled device. A familiar example of such a system is the electric blanket, where the thermostat senses room and not blanket temperature.

Keep in mind that, despite apparent complexity, all control systems may be reduced to combinations of these essential elements. Most complications occur in an attempt to obtain "better" control; that is, to maintain the controlled variable as close to the set point as possible. It is important to seek a balance of complexity, accurate control, reliability, and cost. The building operator must be able to understand and maintain the HVAC controls -overly complex designs are likely to be "simplified" by all but the best-trained operators.

1.4 PURPOSES OF CONTROL

It usually is thought that the purpose of the automatic control system is to provide control of the temperature and/or the humidity in a space, but these are not the only functions that the system can serve; it also can control the relative pressure between two spaces, a very useful attribute in preventing the spread of contamination. Safety controls prevent the operation of equipment in an unsafe condition and can trigger visual or audible alarms to alert operating personnel to those conditions.

A major function of control is to modulate the capacity of HVAC systems. If systems were always fully loaded, little or no contol would be needed. However, most heating and cooling systems are designed to meet demands under the worst conditions (hottest or coldest climate conditions). These conditions arise only for a short period during the year. Given the margins of safety embedded in design methods and the tendency of engineers to add more safety factors, it is likely that many HVAC systems never see full load conditions. Most of the time the system must operate when heating or cooling requirements are far below the capacity of the heating and cooling equipment. The important function of HVAC control sytems is to control the equipment under part load so that comfort and efficiency are maintained.

1.5   CONTROL ACTION

To satisfy the need for various kinds of control response, several types of control actions are available. They may be broadly classified as follows.

1.5.1   Two-Position or On-Off Action (Figure 1-2)

This simple control has many applications. A familiar example of on-off action is a thermostat starting and stopping a home furnace or air-conditioning system. Any two-position controller needs a *differential* to prevent hunting, or too-rapid cycling. This differential is the difference between the setting at which the controller operates at one position and the setting at which it changes to the other. In a thermostat this is expressed in degrees of temperature. The differential setting of any controller is usually somewhat less than the operating differential of the HVAC system because of the lag of the instrument and the system.

One way to reduce the operating differential is to shorten on or off time artificially in anticipation of system response. A heating thermostat may be provided with a small internal heater that is energized during on periods, thereby giving a false signal to the thermostat. This is called heat anticipation.
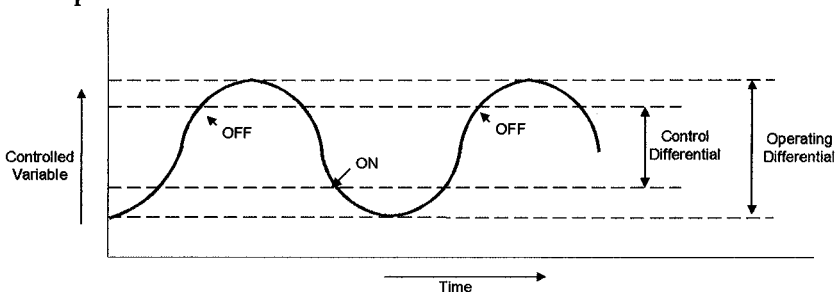
Figure 1-2 Two-position control

1.5.2 Floating Action (Figure 1-3)

This term refers to a controlled device that modulates toward the on or off position any time appropriate contacts are closed but stops when the contacts open.   For example, one set of contacts on a two-position thermostat might close when a room is too cold causing an electric motor to begin to open a steam valve on a heating coil.   As the room temperature rises, the contacts open, stopping the motor in its new position.   If the room temperature continues to rise, another set of contacts might close causing the motor to turn in the other direction, closing the steam valve. Note that the controller must have a dead spot or neutral zone in which neither set of contacts is closed. This allows the device to float in a partly open position. For good operation this system requires a rapid response in the controlled variable.   Otherwise it will not stop at an intermediate point.
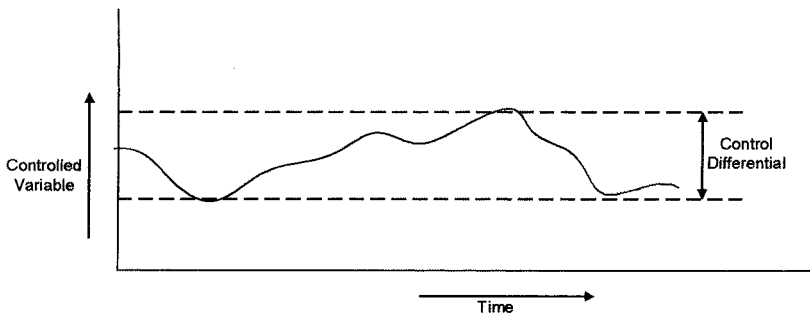
Figure 1-3. Floating Action

1.5.3 Modulating Control (Figure 1-4)

Modulating means that the output of the controller can vary infinitely over the range of the controller. In this situation the controlled device will seek a position corresponding the controller output. Here are some terms encountered in discussing modulating control:
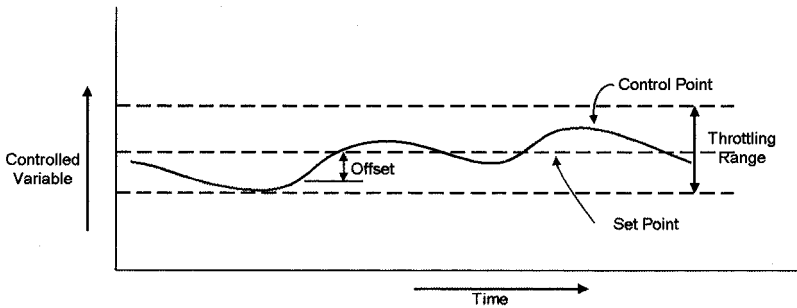
Figure 1-4 Modulating control

• <u>Throttling range</u> is the amount of change in the controlled variable required to run the controlled device from one extreme to the other.

• <u>Set point</u> is the controller setting and is the desired value of the controlled variable.

• <u>Control point</u> is the actual value of the controlled variable. If the control point lies within the throttling range of the controller, it is said to be <u>in control</u>. When it exceeds the throttling range it is said to be <u>out of control</u>.

• Offset or <u>error</u> is the difference between the set point and the control point. This is sometimes called <u>drift</u>, <u>droop</u>, or <u>deviation</u>. The amount of offset theoretically possible is determined by the throttling range, but this value may be exceeded in out-of-control situations.

1.5.4 Control Modes-Proportional

There are three control modes encountered in modulating control. The first and simplest of these is proportional control. This is the control mode used in most pneumatic and older electric systems for HVAC. The mathematical expression for proportional control is:

$$O = A + K_p e \qquad\qquad\qquad Eq.1\text{-}1$$

where :

   O = controller output

   A = a constant equal to the value of the controller output with no
      error

   e = the error, equal to the difference between the set point and the
      measured value of the controlled variable

   K_p = proportional gain constant

This means that the output of a proportional controller is equal to a constant plus the error multiplied by the proportional gain (a constant).

The proportional gain is related inversely to the throttling range. For example, in a pneumatic temperature controller the output will range from 3 to 13 psi (10 psi range). If the throttling range is 10 ^{O}F then the gain will be the ratio of 10 over 10 (1.0), meaning that the controller output signal will change 1 psi for every degree of error. If the throttling range is reduced to 4 ^{O}F the gain will increase to 10 over 4, or 2.5 psi per degree. Increasing the gain will make the controller more responsive, but too high a gain may make the system unstable, causing it to oscillate continuously or hunt around the set point. Decreasing the gain will improve stability but increase the possible error and decrease response and sensitivity.

Figure 1-5 is a graphic illustration of proportional control. This shows the control point-the actual value of the controlled variable-plotted over time. If the HVAC system is started after a prolonged shutdown, the variable will be out of control. It will, then, be driven so rapidly toward the set point that it will cross the set point value before the system can respond. The return swing will again cross the set point value, and so forth. If the system is stable, it will settle out after a few cycles. If the system is unstable, it will continue to oscillate indefinitely (Figure 1-6).

P Only

CONTROLLED
VARIABLE

SET POINT

OFFSET

CONTROL POINT

0

TIME
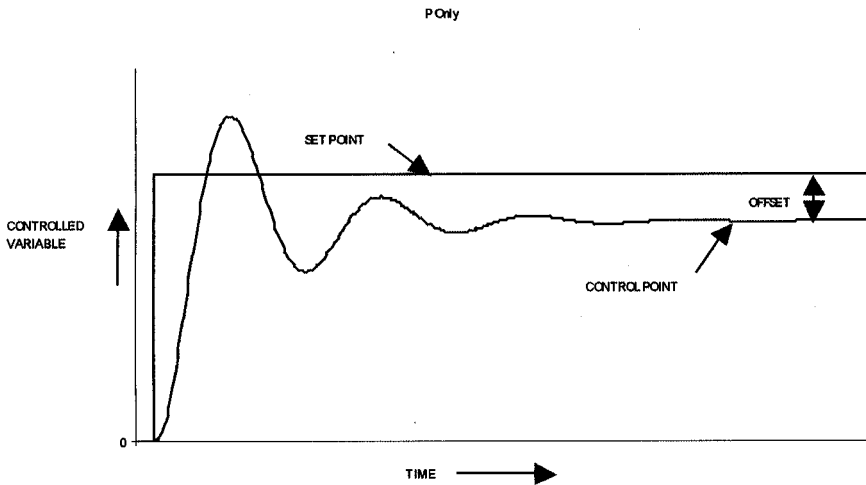
Figure 1-5. Proportional control--stable

There usually will be an offset with proportional control because the error needed to generate the controller output will produce only enough capacity to match the load on the system. The offset will be greater with low values of gain and at light or heavy load conditions. It will also be affected by system gains. The existence of the continuous offset affects system accuracy, comfort, and energy consumption. This will be discussed further in later chapters.

With most controllers, gain adjustment requires only a screwdriver. With computer-controlled systems, gain is a number in the software. (See Chapter 10.)

1.5.5   Control Modes-Proportional plus Integral

This designation is proportional plus integral, usually abbreviated PI. Mathematically, another term is added to the control equation:
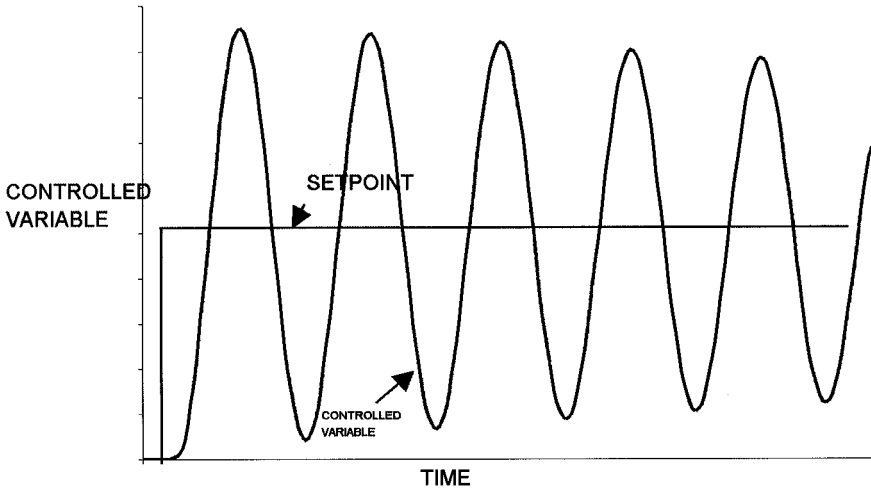
Figure 1-6 Proportional control—on the verge of being unstable

$$O = A + K_p e + K_i \int edt \qquad\qquad Eq. \ 1-2$$

where:

K_i = integral gain constant.

The added term means that the output of the controller is now affected by the error signal integrated over time and multiplied by the integral gain constant. Note that the sign of the error may be positive or negative; therefore, the integral term may be plus or minus. The effect of this term is that the controller output will continue to change as long as any error persists, and the control offset will be eliminated (Figure 1-7).

Although PI mode has long been used in the process control industry, it is somewhat new to HVAC. Some pneumatic controllers and most electronic controllers use PI mode. Computers can be programmed for any mode.
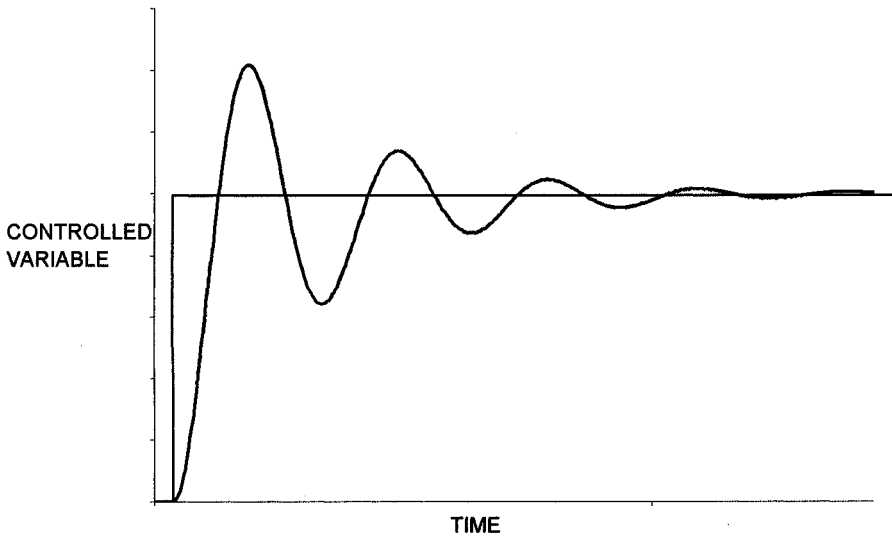
Figure 1-7. Proportional plus integral control

1.5.6 Control Modes-Derivative

For derivative control mode, still another term is added to the control equation:

$$O = A + K_p e + K_i \int edt + K_d \frac{de}{dt} \qquad\qquad Eq. 1 - 3$$

where :

K_d = derivative gain constant

The derivative term provides additional controller output related to the rate of change of the controlled variable. A rapid rate of change in the error will increase the absolute value of the derivative term. A small rate of change will decrease the value.

Derivative control is used to reduce overshoot when a rapid response is desired (requiring a high proportional gain). In most HVAC systems, derivative control adds unneeded complexity. For most control loops, a

smooth transition by the controlled variable to the set point is desired. A properly adjusted PI controller can achieve this response without the need for derivative control action.

1.6   ENERGY SOURCES FOR CONTROL SYSTEMS

Control systems may be pneumatic, electric, electronic, fluidic, hydraulic, self-contained,. or combinations of these. Chapters 2, 3, 4 and 5 will discuss the control elements in some detail. A brief description here will serve to introduce these chapters.

1.6.1 Pneumatic Systems

Pneumatic systems usually use low pressure (less than 20 psig) compressed air to transmit sensor and control signals. Changes in output pressure from the controller will cause a corresponding position change at the controlled device.

1.6.2 Electric Systems

So-called electric systems provide control by starting and stopping the flow of electricity or varying the voltage and current using a rheostat or bridge circuits. These systems usually operate with alternating current at line voltage.

1.6.3 Electronic Systems

These systems use direct current at low voltages (24 V or less) and currents for sensing and transmission, with amplification by electronic

circuits or servo-mechanisms as required for operation of controlled devices.

1.6.4 Digital Systems

Digital systems consist of electronic sensors connected via electronics to a digital computer. Control algorithms are implemented in software. It is common in larger systems to use a digital controller on an air-handler with transducers to permit valves and damper to be actuated pneumatically. Even if the air handling unit is controlled digitally, room thermostats and controller may still be pneumatic.

1.6.5 Fluidic Systems

Fluidic systems use the dynamics of air jets rather than static pressure signals as the control signal generating mechanism. Poor reliability caused this approach to be discontinued, but many fluidic systems were installed and some may still be in service.

1.6.6 Hydraulic Systems

These are similar in principle to pneumatic systems but use a liquid rather than air. Hydraulic actuators have been used only rarely in older systems and are not used in new systems.

1.6.7 Self-Contained System

This type of system incorporates sensor, controller and controlled device in a single package. No external power or other connection is required. Energy needed to adjust the controlled device is provided by the reaction of the sensor with the controlled variable. An example of this type of control is a self contained radiator control valve. Refrigerant pressure within the valve changes with local temperature

and acts against an adjustable spring to modulate the steam or water valve.

1.7 MEASUREMENT

All the control actions described above depend first on the measurement of a controlled variable. Accurate and rapid measurement is a serious challange to engineers in the control industry. Although it is essential for proper control, it is very difficult to obtain an accurate and instantaneous reading, especially if the property being measured is fluctuating or changing very rapidly.

Control system designers must be aware of the practical limitations of the accuracy and the response of available sensing devices.

Thermostats will be affected by the presence or the absence of air motion (drafts), the temperature of the surfaces on which they are mounted (if greatly different from the air temperature), the mass of the sensing element and the presence of radiant effects from windows or hot surfaces. For a residence or an office a variation of one or two degrees Fahrenheit on either side of the set point may be acceptable. For a standards laboratory (Chapter 9) a variation of $\pm$ 0.50 degrees may be unacceptable.

A pressure sensor that is located at a point of turbulence (such as a turn or change of pipe size) in the fluid can never provide accurate or consistent readings. For this purpose a long straight run of duct or pipe generally is required. Straightening vanes can be used where long straightaways are not possible.

Several books have been written on sensors and measuring problems. (See in the bibliography, for example, Considine, 1957 and Haines, 1961) It is not the purpose of this book to go into such detail; but we hope you will study carefully the location of all sensors and their relationship to the rest of the system.

1.8 SYMBOLS AND ABBREVIATIONS

A list of the control symbols and abbreviations is given in the appendix. Most of these are typical for the HVAC control industry but it should be noted that there is no industry standard. The nearest thing to such a standard is the recommended symbol list in the ASHRAE Handbook.

1.9 PSYCHROMETRICS

Many discussions of control systems use psychrometric chart examples, and it is assumed that the reader has a basic knowledge of psychrometrics. For those readers who lack this background, a short chapter on the use of psychrometric charts is included (Chapter 11).

1.10 RELATIONSHIPS

Throughout this book it will be evident that HVAC control systems do not exist by and for themselves. There is a symbiotic relationship among the building (or space), the HVAC system, and the control system. Often, when the environment is not properly controlled, operators or occupants blame the control system. But, many times, the real culprit is the HVAC system or the building.

The building must be properly designed to allow the degree of environmental control required. That is, to take an extreme case, a warehouse cannot be used as a clean room.

The HVAC system must be designed to provide the degree of environmental control required. For example, a grossly oversized heating coil with an oversized valve will be very difficult to control because the slightest valve movement will produce large changes in the

coil discharge temperature. In this case the system gain (the ratio of discharge temperature change to change in valve position) is so high that adequate control is almost impossible. Only when the building and HVAC system are properly designed for the required service can the control designer provide a control system that will produce the desired level of control. Thus, the custom of calling the control designer after the building and HVAC system have been designed is not conducive to a satisfactory solution. All the elements need to be considered together. The control designer must have a thorough understanding of these potential problems and be involved from the beginning of the design process.

1.11 SUMMARY

This chapter has discussed the elements of a control system, the basic types of control action, and the energy sources commonly used for controls. Many control systems use combinations of these energy types. An interface between unlike types of energy is provided by relays or transducers. The next chapters discuss the various types of control elements in detail. The chapters following the element descriptions will consider how these elements may be combined to produce control systems of varying degrees of complexity to perform specific functions.

2

Pneumatic Control Devices

2.1 INTRODUCTION

Chapter 1 dealt with the fundamentals of control circuits. This chapter and the three following will consider various types of control devices: pneumatic, electric, electronic, and fluidic. Succeeding chapters will discuss the use of these devices in control systems.

No attempt will be made here to provide an exhaustive catalog of control instruments. Rather, some basic principles of operation will be considered. With this as background, it will then be possible for the reader to evaluate a catalog description of a control device with a reasonable degree of understanding.

For simplicity, the chapters that follow classify control devices by energy type: pneumatic, electric, electronic, and fluidic. This presentation is followed by a discussion of flow control devices (valves and dampers), as these devices are essentially independent of the actuator.

Schematic diagrams are used, which avoid details that are peculiar to individual manufacturer. Details are available from manufacturers' service manuals.

2.2 PNEUMATIC CONTROL DEVICES

Pneumatic controls are powered by compressed air, usually 15 to 20 psig pressure although higher pressures are occasionally used for

operating very large valves or dampers. Pneumatic devices are inherently modulating, as air pressure can be modulated with infinite variation over the control range. Because of their simplicity and low cost, pneumatic controls are frequently found on commercial and industrial installations where more than eight or ten devices are used. If only a few control components are needed, electronic or electric controls may be less costly than pneumatic controls because an air compressor and pneumatic piping are not required.

Available pneumatic devices include sensors, controllers, actuators, relays and transducers, which are described in the paragraphs that follow. The principles discussed form the basis of most manufacturers' designs. Of course, each has variations.

2.2.1 Definitions

Except for the first two the following definitions are peculiar to pneumatic devices.

- Direct-acting: A controller is direct-acting when an increase in the level of the sensor signal (temperature, pressure, etc.) results in an increase in the level of the controller output (in a pneumatic system this would be an increase in output air pressure).

- Reverse-acting is the opposite of direct-acting; that is, an increase in the level of the sensor signal results in a decrease in the level of the controller output.

- SCFM: standard cubic feet per minute. This refers to air at standard atmospheric pressure of 14.7 psia and a temperature of 70 °F. For ease of comparison most air compressors are rated in SCFM.

- psia: pounds per square inch, absolute pressure.

- psig: pounds per square inch, gage.

- SCIM: standard cubic inches per minute. This is similar to SCFM, but SCIM is usually used to describe pneumatic device air consumption. One SCFM equals 1728 (12^3) SCIM.

- Manufacturers, in their literature, will often use the terms sensitivity or proportional band. These terms are synonymous with "gain."

Other terms such as bleed, non-bleed, and submaster, are defined with the device description.

2.2.2 Bleed-Type Controllers

The bleed-type sensor or controller (Figure 2-1) is one of the simplest pneumatic devices. A restricted air supply flows both to a nozzle and to the output. Because only a small amount of air can flow through the restrictor, the pressure in the nozzle and output lines will vary with the force that the flapper exerts in covering the nozzle. In steady operation, this force will be balanced by the force created by the pressure in the nozzle. If the pressure times the area of the nozzle exceeds the force exerted by the flapper, the flapper will be pushed away from the nozzle, allowing more air to leak out and causing the pressure in the nozzle and output lines to drop until a force balance is achieved. If the flapper force exceeds the nozzle force, the flapper closes against the nozzle, and pressure builds until a force balance is reached.
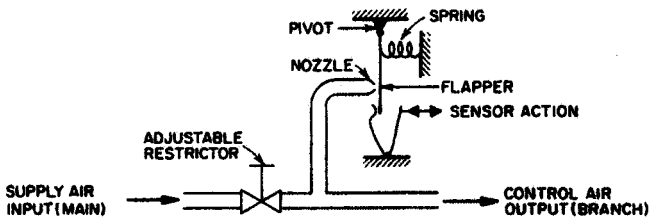
Figure 2-1. Bleed-type controller.

Notice that whenever a bleed-type sensor or controller is in steady operation, some air is leaking out of the nozzle (if it were not leaking,

pressure would be building until leakage began to occur); hence the designation as a bleed-type controller.

Bleed-type controllers and sensors may be direct- or reverse-acting, depending on the sensor linkage. They are inherently proportional. The output signal may go directly to an actuator, or it may be used as input to a relay-controller, or pilot positioner to be described later.

2.2.3 Relay-Type Controllers

Relay-type controllers may be either non-bleed or pilot-operated bleed-type. The non-bleed controller uses air only when exhausting or filling the line to the controlled device.

Figure 2-2 is a schematic diagram of a non-bleed controller. A positive movement from the sensor (increase in temperature or pressure) will cause an inward movement of the diaphragm and a downward movement on the right end of the lever, which raises the other end of the lever and allows the supply air valve to open. This increases the output pressure and the pressure in the valve chamber. As the chamber pressure increases, it acts on the diaphragm to offset the pressure from the sensor, so that when equilibrium is attained, the supply valve closes and the balanced pressure becomes the output to the controlled device.
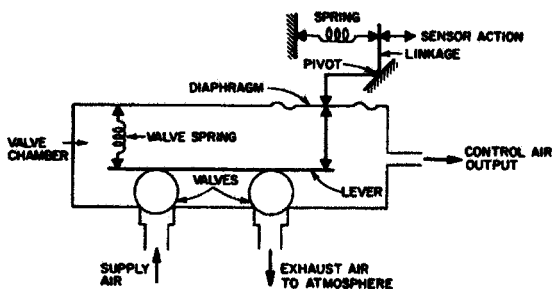
Figure 2-2. Non-bleed controller.

A further positive sensor movement will cause a rebalancing at some higher output pressure. A negative sensor movement will decrease the

pressure on the right end of the lever, allowing the exhaust valve to open until the chamber and sensor pressures are again in balance. A simple reversal of the linkage will change the controller to reverse-acting.

The bleed-type piloted controller uses a reduced-airflow bleed-type pilot circuit combined with an amplifying non-bleed relay to produce a sensitive, fast-acting control device. The controller can be adjusted to produce a large change in output for a small change in pilot pressure, and can be provided with negative feedback for proportional action or positive feedback for two-position action.

The proportional arrangement is as shown in Figure 2-3. The orifice plate is provided to restrict the flow of air to the pilot chamber. The control port may be partially or completely restricted by the flapper valve, which is operated by the sensor.
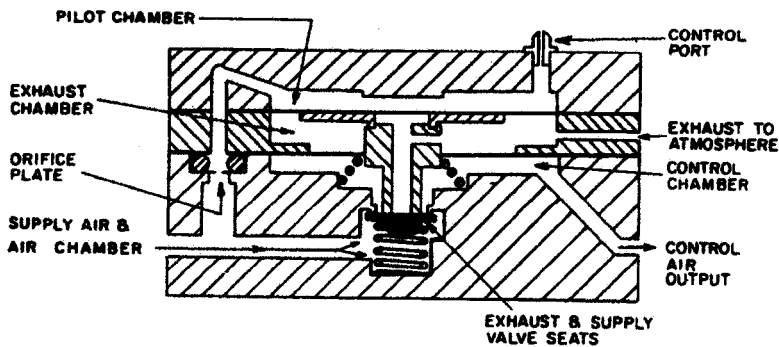
Figure 2-3. Proportional relay controller, pilot-bleed type. (Courtesy Johnson Service Company).

In Figure 2-4 the various operating conditions are shown. In Figure 2-4(A) the control port is open, the pilot chamber pressure is essentially zero, the exhaust port is open, and output is zero. In Figure 2-4(B) the flapper valve has been partially closed, and the pressure begins building up. At some initial pressure, usually 3 psig, the spring allows the exhaust seat to close. In Figure 2-4(C), the pressure continues to increase, pushing the pilot diaphragm down and opening the supply

valve. Air flows into the control chamber and to the output line. As the pressure in the control chamber increases, the pilot pressure is opposed (Figure 2-4(D)) (negative feedback) until the pressures balance and the supply valve closes.

Controllers of this type are provided with a broad sensitivity (gain) adjustment. For example, the gain of a temperature controller may be adjusted from as little as 1 psi per degree to as much as 10 psi per degree. (The latter probably would result in an unstable control loop.)

The mounting of the sensing element determines whether the controller is direct- or reverse-acting. Some devices may be changed in the field; others are fixed and cannot be changed.

A similar two-position controller is shown in Figure 2-5. Notice the important differences in valve and spring arrangement.

Figure 2-6 shows the operation: In Figure 2-6(A) the control port is open, and spring pressure holds the supply port open and exhaust port closed. This is a reverse-acting controller. As the control port is partially closed, pilot pressure increases, forcing the pilot diaphragm down and closing the supply valve (Figure 2-6(B)). A further increase in pilot pressure opens the exhaust valve, causing a decrease in pressure in the control chamber. This decreases the force opposing the pilot pressure (positive feedback) and allows the exhaust valve to open completely and remain open until pilot pressure is reduced (Figure 2-6(C)).
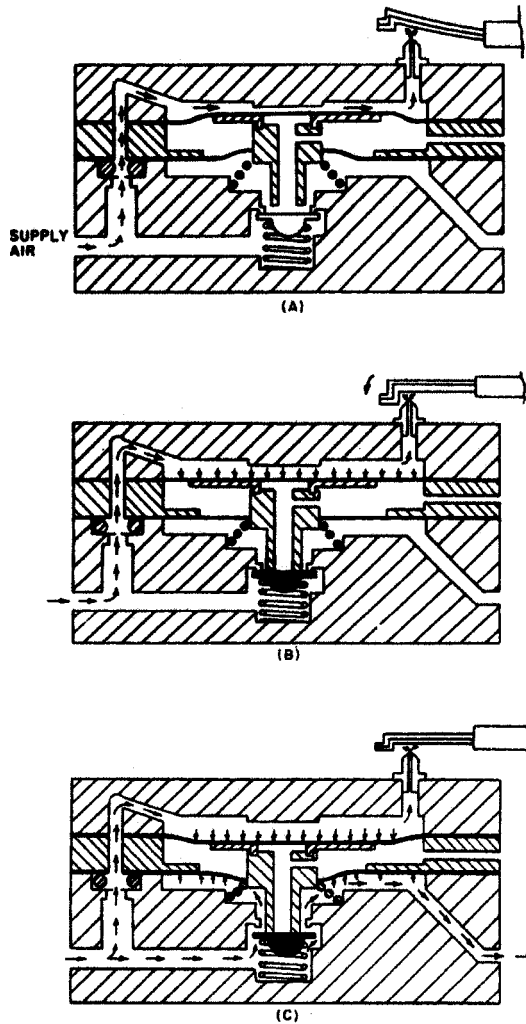
Figure 2-4 Operation of proportional relay. (*Courtesy Johnson Service Company.*) (A) When the control port is open, the exhaust valve between the control and exhaust chambers is open. Thus, air in the control chamber is at zero gage or atmospheric pressure. The supply valve is held closed by a spring and supply pressure. (B) When the sensing element moves closer to the control port, pressure begins to increase in the pilot chamber. At 3 psig, pilot pressure overcomes the force of the opposing spring and closes the exhaust seat. (C) As pilot pressure continues to increase, it forces the pilot diaphragm down and opens the supply port. This allows supply air to flow to the control line.

(D)
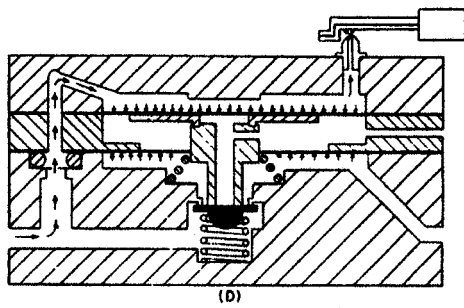
Figure 2-4 (Continued) (D) Pressure now increases in the control chamber, and acts against the control diaphragm to oppose pilot pressure. When the total of forces in each direction is equal, the supply valve is closed and the controller is balanced.
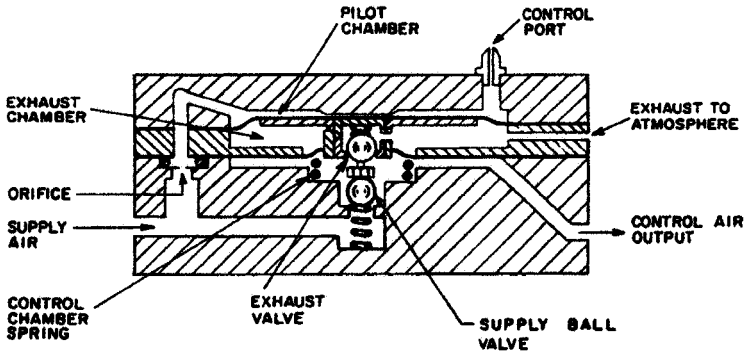
Figure 2-5 Two-position relay controller, pilot-bleed type. (*Courtesy Johnson Service Company.*)
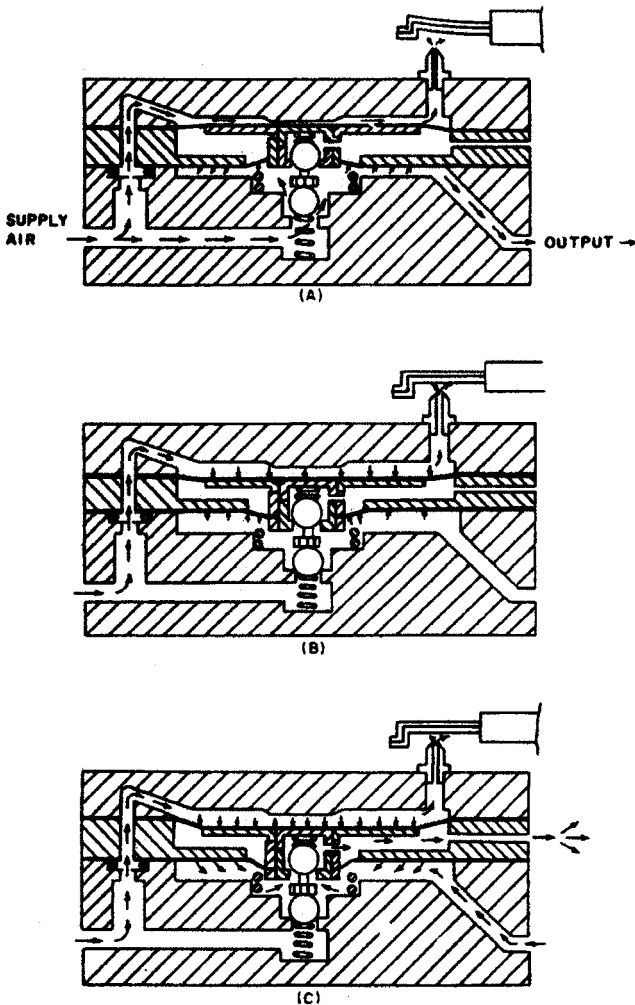
Figure 2-6 Operation of a two-position relay. (*Courtesy Johnson Service Company.*) (A) When the control port is open, supply air flows through the supply chamber and out the control chamber and the controlled devices. Spring and air pressure combine to hold the supply ball valve open and the exhaust valve closed. (B) As the control port is closed by the element, pressure in the pilot chamber increases. When pilot pressure is greater than the opposing forces, the pilot diaphragm moves, seating the supply ball valve. (C) Further movement of the pilot diaphragm opens the exhaust ball valve, so that control air is exhausted. This action reduces the forces opposing the pilot pressure (positive feedback) and causes the exhaust valve to open fully and remain open until pilot pressure is reduced (by opening the control port). The sequence is then reversed.

2.2.4 Sensor-Controller Systems

The principles represented by the forces and linkages shown above are found in sensor-controller systems used extensively in HVAC control practice. Single or dual sensor inputs may be used. A single-input sensor-controller is shown in Figures 2-7(A), 2-7(B), and 2-7(C). The system operates as follows:

In the balanced condition, Figure 2-7(A), the following conditions exist:

- The input diaphragm H force acting on the main lever I (which pivots about J) is balanced by the force of the set point adjustment T and the feedback force from diaphragm N, which acts through the proportional-band lever L and adjustment K.

- Exhaust nozzle F and main nozzle Q are both closed. In this condition, the pressure in the controller branch-line is at a value proportional to the requirements at the sensor.

On an increase in temperature, Figure 2-7(B), the brass tube B expands, tending to close flapper C against the nozzle V. This results in an increased pressure in the sensor-line G and a force on the input diaphragm H, which rotates the main lever I clockwise. The proportional band adjustment K rotates the proportional band lever I counterclockwise about fixed pivot M. Feedback diaphragm N is forced up, and flapper O rotates about nozzle P opening nozzle Q. Main air enters the feedback chamber W, increasing the branch-line pressure. As the pressure in the feedback chamber increases, the feedback diaphragm N is forced down until both nozzles P and Q are closed again, and the system is rebalanced at an increased branch-line pressure.
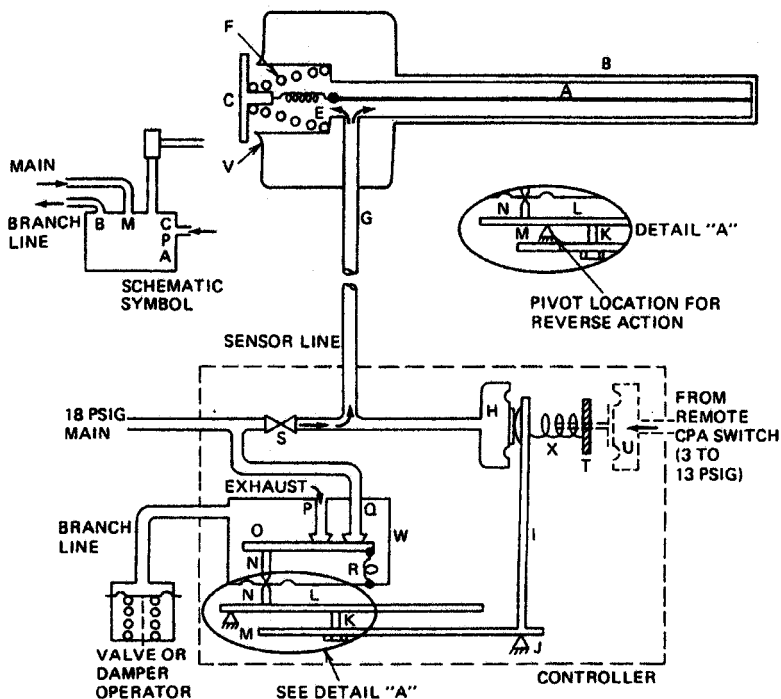
Figure 2-7 A. Diagram of a single-input, sensor-controlling system in a balanced condition.

On a decrease in temperature, Figure 2-7(C), the brass tube B contracts, tending to open the flapper C. This results in a decrease in pressure in the sensor line G and a decreased force on input diaphragm H. The net force on feedback diaphragm N now unbalances the lever system. Flapper O rotates counterclockwise following feedback diaphragm N and opening exhaust port P. Feedback diaphragm N, in turn, forces the proportional band lever L to rotate clockwise, and the main lever I and proportional band adjustment rotate counterclockwise. Air is exhausted from the controller branch line and the feedback chamber W. As the pressure in the feedback chamber decreases, the feedback diaphragm N moves up until both nozzles P and Q are closed again, and the system is rebalanced at a decreased branch-line pressure.
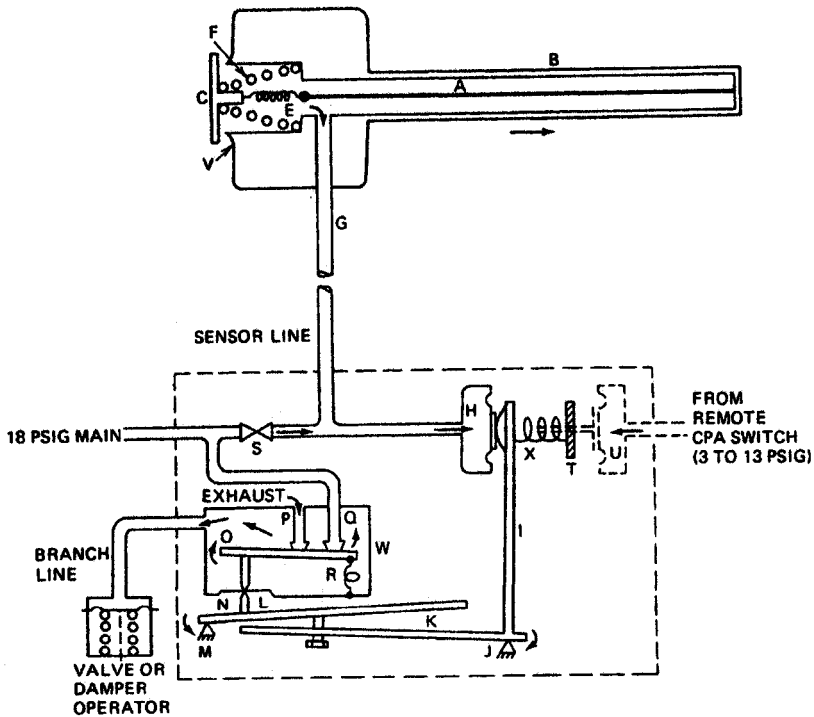
Figure 2-7 B. Diagram of a single-input, sensor-controller system increasing the branch-line pressure. (Courtesy Honeywell, Inc.).

The system is shown direct-acting, but can be changed to reverse-acting as shown in detail A of Figure 2-7(A).

The CPA is a remote control point adjustment, accomplished by varying the pressure at the CPA port using either a manual switch or a transducer controlled by a supervisory system.

For the dual-input controller shown in Figure 2-8, an additional arm is added to the main lever I to provide compensation or reset in the system. Also, an additional input chamber A is added to allow a second force to be applied to the lever system. This force acts in the same direction as the input 1 chamber H force, regarding the main lever I.
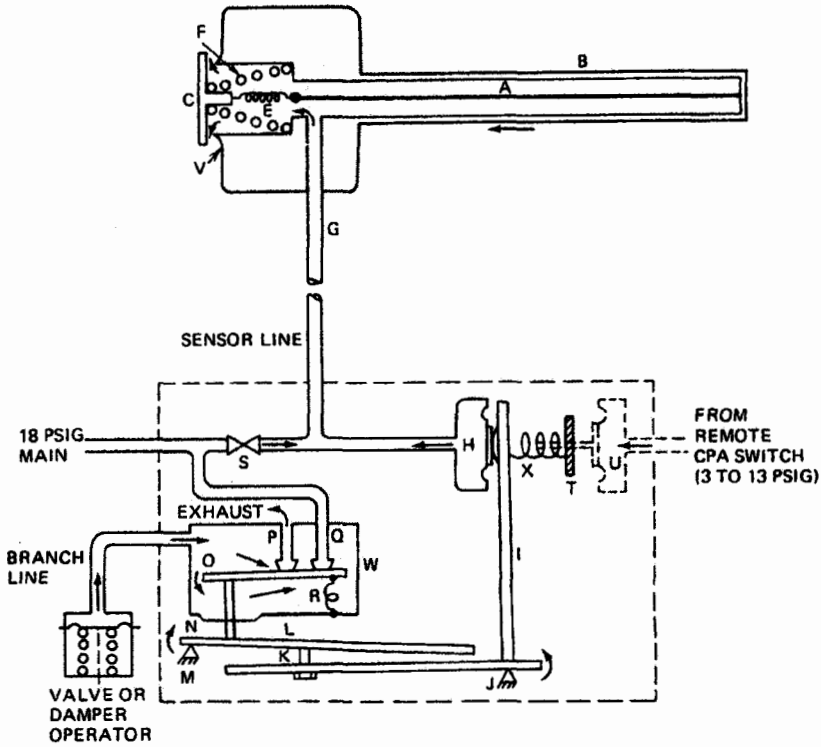
Figure 2-7 C. Diagram of a single-input, sensor-controller system decreasing the branch-line pressure. (Courtesy Honeywell, Inc.).

In a typical application, compensation may be provided with this dual-input controller by using two remotely located sensors. To reset hot water, for example, the input 1 sensor is in the supply water discharge and the input 2 sensor is in the outdoor air. A drop in outdoor-air temperature reduces the input diaphragm A force. This has a similar effect on the main lever to that of increasing the set point adjustment spring X force. In other words, the drop in outdoor-air temperature raises the control point of the system. By setting the authority adjustment B, the relative effect of the input diaphragm A force may be varied, compared to the effect of the input diaphragm H force.
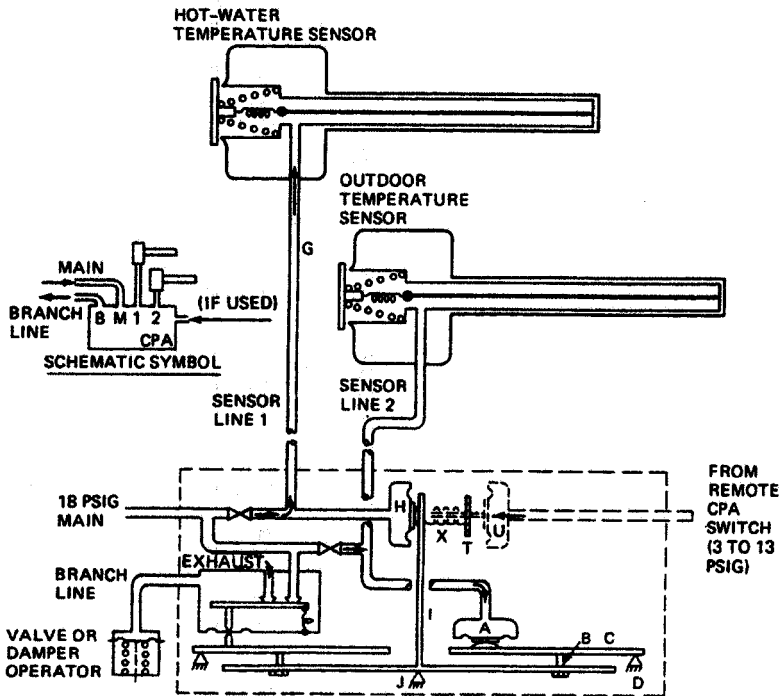
Figure 2-8. Diagram of a dual-input, sensor-controller, hot-water reset system with remote control point adjustment. (Courtesy Honeywell, Inc.).

An increase in the compensating medium temperature causes an increase in the input diaphragm A force. The resultant force acting on the main lever I through the authority lever C causes the main lever I to rotate clockwise about the pivot point J, effectively lowering the set point.

These examples show temperature sensors, but any kind of variable may be sensed -humidity, pressure, flow, and so on - with no change in the controller. The control differential and the authority of primary and reset sensors may be adjusted over a fairly wide range at the controller. The dual-input controller of Figure 2-8 can also be used with a single sensor, with or without the CPA. If the single sensor is connected to both sensor input ports the effect is to reset the set point when the normal proportional offset is present. The result is less offset from the

original set point, similar to the integral function. This is called "proportional with reset."

Some of the most modern pneumatic controllers use stacked diaphragms to implement force balances similar to those described above. Figure 2-9 shows a proportional plus integral sensor and controller system. We will illustrate how the system works by an example.
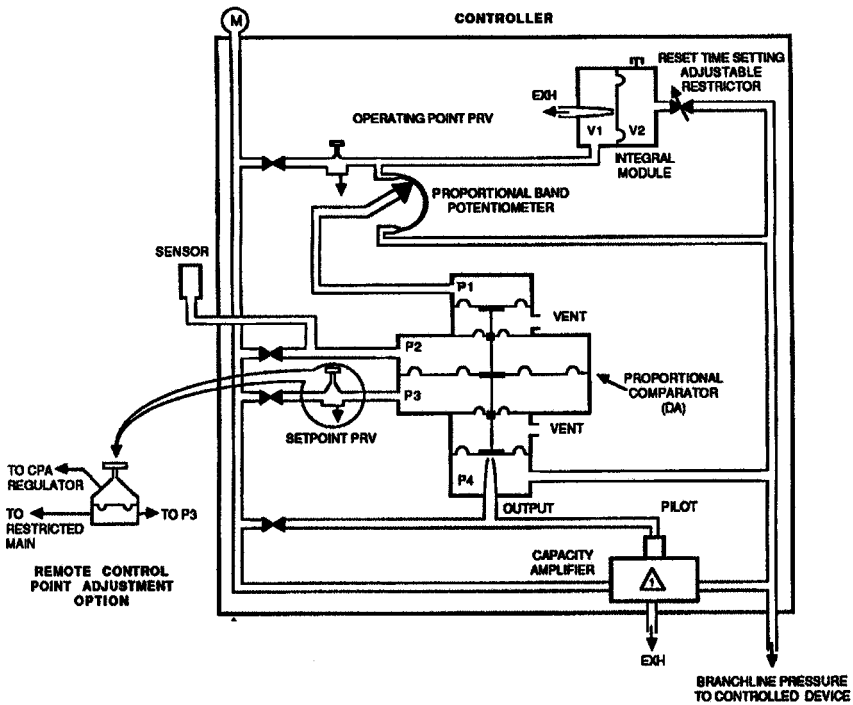
Figure 2-9. Single-input PI sensor-controller system (Courtesy of Honeywell, Inc.)

Suppose that the pressures in each chamber, P1, P2, P3, and P4 are initially equal at 9 psig. In this condition the air coming in through the nozzle into chamber 4 escapes through the capacity amplifier exhaust port or to the branch-line device.

Now suppose that the sensor pressure P2 suddenly increases to 10 psig. This will provide an additional downward acting force on the diaphragm assembly, closing the nozzle. The output pressure will build up causing the pressure P4 to build up. Assuming that pressure P1 changes slowly and that the diaphragm separating chambers 2 and 3 is twice the area of the other diaphragms, the pressure P4 would increase to 11 psig before the system reaches a temporary balance. The force balance is:

$$P1 + 2\,P2 = P4 + 2\,P3 \quad \text{or} \quad 9 + 2 \times 10 = 11 + 2 \times 9$$

However, part of the output pressure is fed back to chamber 1. If we assume that the proportional band potentiometer blends one-third of the output pressure to two-thirds of the integral module pressure to achieve P1 and that the integral module pressure changes slowly, then the force balance is:

$$(1/3)\,P4 + (2/3) \times 9 + 2\,P2 = P4 + 2\,P3 \quad \text{or} \quad P4 = 12 \text{ psig}$$

Now if the sensor chamber pressure remains at 10 psig, the output pressure P4 will cause the diaphragm separating V1 from V2 in the integral module to close the exhaust port in the integral module, resulting in a buildup of pressure from the integral module to P1. This will increase the pressure in P1 and result in a further increase in output pressure. However, an increase in output pressure should be having the effect of driving the sensed variable (the controlled variable) toward the set point. Let us assume that as the output pressure goes toward 13 psig, the sensor pressure finally returns to 9 psig. This new equilibrium condition has the pressures in P1 and P4 at 13 psig while the pressures in chambers 2 and 3 are equal at 9 psig. Notice that it is only when P2 and P3 are equal that the output pressure stops changing. Notice further how the integral action can lead to a new steady state output pressure while forcing the controlled variable to be equal to the set point.

2.2.5 Sensing Elements

Controllers can be operated in response to various stimuli, such as temperature, pressure or humidity. A brief discussion of the more common sensing elements follows.

2.2.5.1 Temperature-Sensing Elements

These elements include bimetal elements, vapor-filled bellows, and liquid-, gas- or refrigerant-filled bulb and capillary sensors.

The bimetal element, which is the simplest and most often encountered sensor, has two thin strips of dissimilar metals fused together. Because the two metals expand at different rates, the element bends as the temperature changes. The resulting motion typically can be used to vary the pressure at a pneumatic control port or to open and close an electrical contact. The device is most often made of brass, with a high coefficient of expansion, and invar metal, which has a very low expansion coefficient. Figure 2-10 shows the movement of a bimetal element as it is heated. The bimetal also may be shaped in a spiral coil to provide a rotary motion output. The bimetal is often used in wall-mounted thermostats where it will sense ambient air temperature.
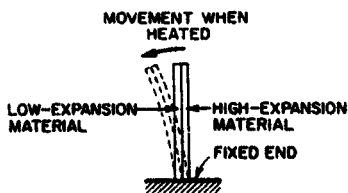
Figure 2-10. Bimetal sensor.

Another ambient temperature sensor is the vapor-filled bellows, usually made of brass and filled with a thermally sensitive vapor, which will not condense at the temperatures encountered. Temperature changes will cause the bellows to expand and contract. An adjustable spring is used

to control set point and limit expansion. The resulting movement can be used directly or through a linkage.

Bulb and capillary elements are used where temperatures must be measured in ducts, pipes, tanks, or similar locations remote from the controller. There are three essential parts of this device: bulb, capillary, and diaphragm operating head. The fill may be a liquid, gas, or refrigerant, depending on the temperature range desired. Expansion of the fluid in the heated bulb exerts a pressure that is transmitted by the capillary to the diaphragm and there translated into movement (Figure 2-11).
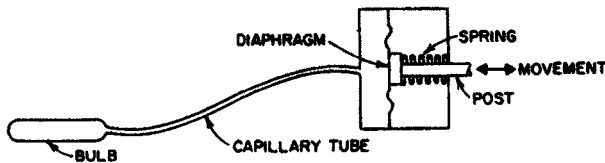
Figure 2-11. Bulb and capillary sensor.

The sensing bulb may be only a few inches long, as used in a pipe or a tank, or it may be as long as 20 feet when used to sense average air temperature in a duct of large cross section.

Special long bulbs are used for freeze protection. Refrigerant is used in this type of bulb, as the refrigerant will condense at freezing temperatures, causing a sharp decrease in pressure if any part of the bulb is exposed to low temperature.

Temperature-compensated capillary tubes are used to avoid side effects from the ambient temperature around the capillary. Capillaries may be as long as 25 or 30 feet.

2.2.5.2 Pressure Sensing Elements

These elements include diaphragms, bellows, and bourdon tubes.

The diaphragm is a flexible plate, sealed in a container so that fluid cannot leak past it. A force applied to one side will cause it to move or flex. A spring usually operates to control the movement and return the diaphragm to its initial position when the force is removed. Some diaphragm materials will spring back to the original shape without help. A variety of materials are used to cope with the various temperatures, pressures, and fluids encountered.

A bellows is a diaphragm that is joined to the container by a series of convolutions (folds) so that a greater degree of movement may be obtained (Figure 2-12). The bellows may be completely sealed, as in a temperature-sensitive unit, or it may have a connection for sensing pressure, either internally or externally. The bellows acts like a spring, returning to its original shape when the external force is removed. Frequently a separate spring is added for adjustment and to increase reaction speed.
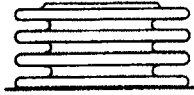
Figure 2-12. Bellows sensor.

The bourdon tube (Figure 2-13), widely used in pressure gages and other pressure instruments, has a flattened tube bent into circular or spiral form. One end is connected to the pressure source, and the other end is free to move. As pressure is increased, the tube tends to straighten out, and this movement may be used, through an appropriate linkage, to position an indicator or actuate a controller.
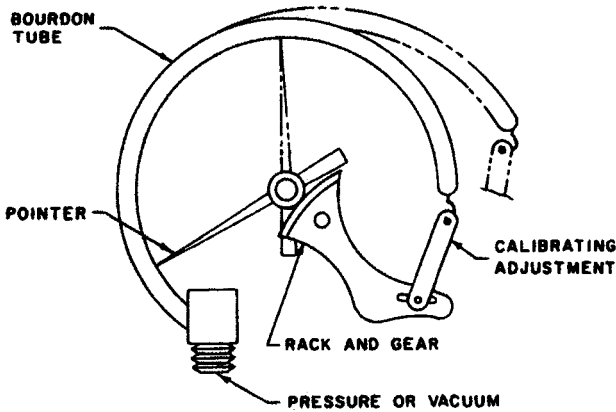
Figure 2-13. Pressure sensor, bourdon-tube type. (Courtesy Johnson Service Company.)

2.2.5.3 Humidity-Sensing Elements

Used with pneumatic controls, these elements are made of hygroscopic materials, which change size in response to changes in humidity. An element similar to a bimetal is made of two strips of unlike woods glued together. The different rates of hygroscopic expansion will cause the strip to bend as humidity changes. Yew and cedar woods are frequently used for this purpose.

Elements made of animal membrane, special fabrics, or human hair will increase or shorten their length as humidity changes, with the resulting movement mechanically amplified. Current practice is to use nylon or similar synthetic hygroscopic fabrics. But all of these have been largely superceded by electronic devices, see Chapter 4.

2.2.6 Pneumatic Actuators

Eventually the output of a pneumatic controller moves a pneumatic actuator that positions a valve or damper. A pneumatic actuator is simply a piston and spring in a cylinder (Figure 2-14). When control air

enters the cylinder, it drives the piston to compress the spring until the spring pressure and the load on the connecting rod balance the air pressure. The stroke may be limited by adjustable stops. The connecting rod may drive a valve stem directly, or operate a damper by means of a linkage. Different spring ranges are available for sequenced control; for example, full 3 to 13 psi range is most often used, but 3 to 8 psi, 8 to 13 psi and other ranges are available.
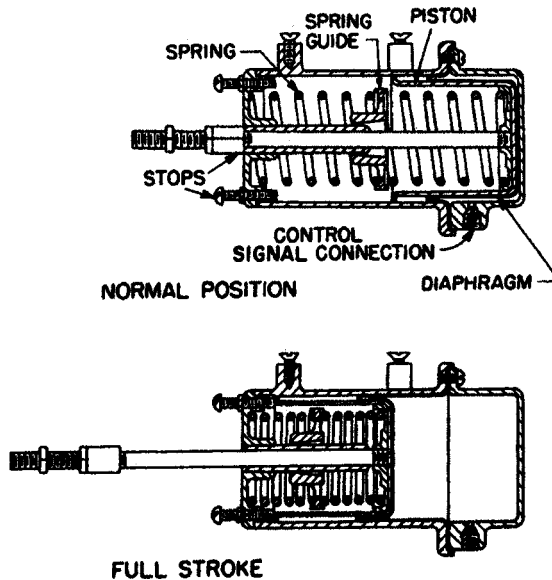
Figure 2-14. Pneumatic operator. (Courtesy Johnson Service Company.)

2.2.6.1 Positive Positioners

A standard pneumatic actuator may not respond to small changes in control pressure due to friction in the actuator or changing load conditions such as wind acting on a damper blade. Where accurate positioning of a modulating device in response to load is required, positive positioners are used.

A positive positioner (or positive-positioning relay) is designed to provide up to full main control air pressure to the actuator for any change in position. This is done by means of the arrangement shown schematically in Figure 2-15. An increase in branch pressure from the controller (A) moves the relay lever (B), opening the supply valve (C). This allows main air to flow to the relay chamber and the actuator cylinder, moving the piston (not shown). The piston movement is transmitted through a linkage and spring (D) to the other end of the lever (B); and when the force due to movement balances out the control force, the supply valve closes, leaving the actuator in the new position. A decrease in control pressure will allow the exhaust valve (E) to open until a new balance is obtained. Thus, full main air pressure is available, if needed, even though the control pressure may have changed only a fraction of a psi. The movement feedback linkage is sometimes mounted internally. Positioners may be connected for direct or reverse action. For large valves or dampers, main air to the actuator may be at a higher pressure than pilot air.
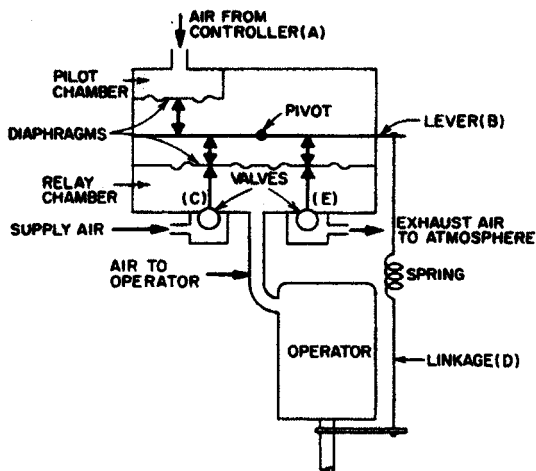
Figure 2-15. Positive positioner.

Some positioners have adjustments for start point and spring range, so that they may be used in sequencing or other special applications. Start point is the pressure at which the operator starts to move. Spring range is the pressure range required for full travel of the operator.

2.2.7 Relays

In pneumatic control jargon, a relay is a device that takes a signal from a controller, changes it in some way, and relays it to another controller or actuator.

Many different types of pneumatic relays are manufactured. Mostly they use some variation of the non-bleed controller shown in Figure 2-2. A pressure signal from another controller or relay replaces the force due to temperature or pressure change. Thus there may be a reversing relay (Figure 2-16) in which a direct-acting control input pressure may be changed to a reverse-acting output. The output of this and other relays also may be amplified or reduced with respect to input, so that one input to several relays may produce a sequence of varied outputs.
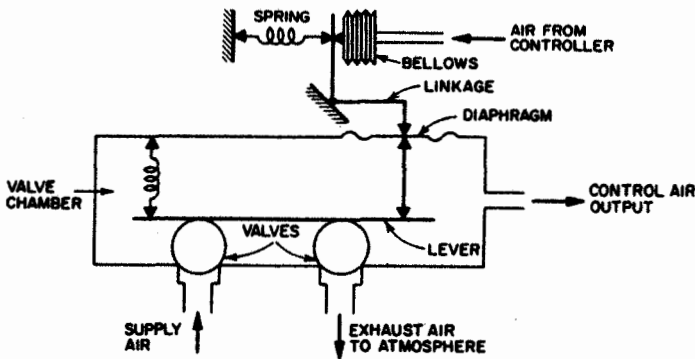
Figure 2-16. Reversing relay.

To illustrate sequencing, envision a controller with a 6 to 9 psi output over the desired control range. It is desired to operate three valves so that one goes from fully open to fully closed with a 3 to 8 psi signal, a second goes from closed to open with a 5 to 10 psi signal, and a third has a control range of 8 to 1 3 psi from closed to open. These are to operate in sequence over the 6 to 9 psi control range. The first valve

would use a relay that produced a 3 to 8 psi output from a 6 to 7 psi control input, the second relay would provide a 5 to 9 psi output over a 7 to 8 psi input and the third would provide an 8 to 13 psi output over the remaining 8 to 9 psi input change.

Another type of relay will produce an output proportional to the difference between two inputs (Figure 2-17), and yet another produces an output equal to the higher (or lower) of two pressures (Figure 2-18). Two-position relays use the principles illustrated in Figures 2-5 and 2-6.
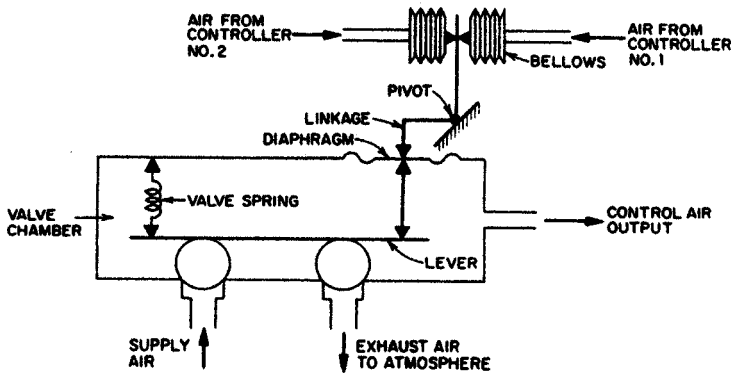
Figure 2-17. Relay: output is proportional to the difference between two signals.

The so-called discriminator relay will accept many input signals-from six or seven to twenty, depending on the manufacturer-and select and pass on the highest or lowest of the inputs. Some models will output both the highest and the lowest. These relays are widely used in multiple-zone HVAC systems for energy conservation. (See Chapter 7.)

An averaging relay will output the average of two to four input signals.

A switching relay is used to divert control signals in response to a secondary variable, typically outside air temperature. It is essentially a two-way valve. (See Figure 2-19.) With the switching signal above the

set point, the C port is connected to the normally closed (NC) port. With the switching signal below the set point the C port is connected to the normally open (NO) port.
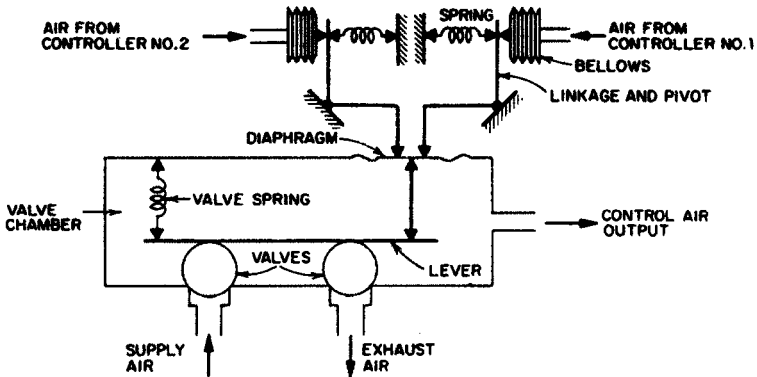
Figure 2-18. Relay: output proportional to higher of two pressures.
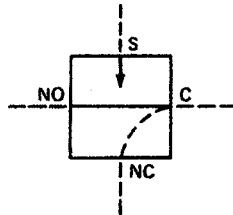
Figure 2-19. Switching relay.

2.2.8 Master-Submaster Thermostats

Although the master-submaster arrangement was used extensively in control systems, it is seldom used in current practice because a single controller with two sensors serves the same purpose. It has two thermostats, one of which (the master) senses some uncontrolled variable, such as outdoor temperature, and sends its output to the second (the submaster). This has the effect of resetting the set point of the submaster, which is sensing the controlled variable. Adjustments

are provided for basic settings of both thermostats and for setting up the reset schedule, that is, the relationship between submaster set point and master output signal. Figure 2-20 shows the system schematically. Both instruments are of the non-bleed type. The master also may be a bleed-type controller.

Figure 2-20. Master-submaster thermostat.

This principle may be extended to use a single non-bleed controller with two remote bleed-type sensors as in Figure 2-21. Here one sensor acts as a master to reset the set point of the other sensor. This arrangement is used extensively in current practice, both for master-submaster control and, with one sensor, for single-point sensing and control without reset. It has the advantage that the output of the sensor is independent of the controller action, and can be used to transmit information to other devices, such as supervisory panels or pneumatic thermometers.

Figure 2-21 Controller with two remote sensors.

2.2.9 Dual-Temperature Thermostats

This classification includes day-night and summer-winter thermostats. Two related temperature settings may be made and the transfer from one to the other is made by changing the supply air pressure. The settings cannot overlap, and there is a built-in differential,-adjustable or fixed, depending on the instrument.

A typical unit with two bimetal sensors is shown in Figure 2-22. The piloting relay is similar to those discussed previously. Two control ports are provided, and the bimetals are adjusted to close one port sooner than the other. The port in use is determined by a diaphragm-operated transfer mechanism that is actuated by changing the main air supply pressure, typically from 15 to 20 psi.

If both bimetal elements are direct-acting, that is, they bend toward the control port on increase of temperature, then the thermostat functions as a day-night instrument. When it is used for heating, a lower temperature will be provided at night than during the day. If one of the

bimetals is direct-acting and the other is reverse-acting, then the thermostat provides heating-cooling control, for year-round use.
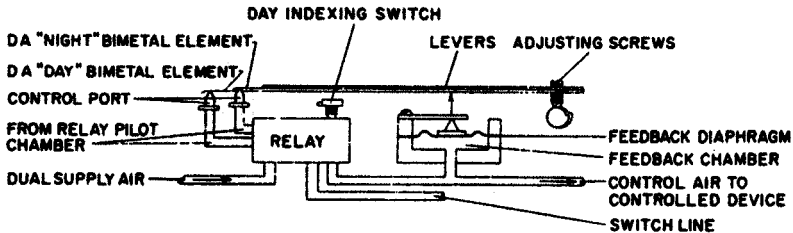
Figure 2-22. Dual-temperature thermostat. (Courtesy Johnson Service Company).

If both bimetal elements are direct-acting, that is, they bend toward the control port on increase of temperature, then the thermostat functions as a day-night instrument. When it is used for heating, a lower temperature will be provided at night than during the day. If one of the bimetals is direct-acting and the other is reverse-acting, then the thermostat provides heating-cooling control, for year-round use.

Changeover from one supply pressure to the other may be done in several ways: manually, by means of time clock or programmer, by means of an outdoor thermostat, or by means of a sensor that detects supply water temperature in a two-pipe system. (Obviously, it does little good to call for heating when chilled water is being supplied!)

2.2.10 Dead-Band Thermostats

The dead-band or zero-energy band thermostat was developed to conserve energy. Its design is based on the assumption that the comfort range of temperature is fairly wide, and no heating or cooling is required over that range. The dead-band thermostat therefore has a wide differential (dead-band) over which output remains constant as the temperature varies, with output changing only in response to temperature outside the differential range. (See Figure 2-23.) The set point is usually the upper limit of the dead-band range. The set point and the dead-band are field-adjustable.
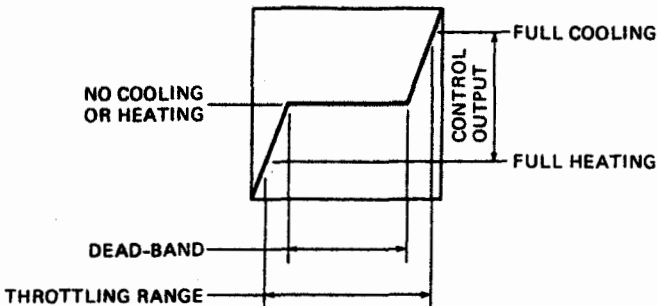
Figure 2-23. Dead-band thermostat performance.

2.2.11 Transducers

Broadly defined, a transducer is a device for transforming one form of energy to another. Because a pneumatic control system frequently must interface with electrical or electronic devices, transducers are necessary. They may be two-position or modulating.

The PE (Pneumatic-Electric) relay is simply a pressure switch that is adjusted to open or close an electrical contact at some value of the control pressure. Sequenced PE relays are frequently used for capacity control of refrigeration systems, for two-speed fan control, and similar functions.

The EP (Electric-Pneumatic) relay is a solenoid air valve, usually three-way to supply or exhaust air to or from the pneumatic control circuit. A three-way valve also may be used to change supply pressure (by changing from one supply source to another).

Modulating transducers change a modulating air signal to a variable voltage output, in either the electric or the electronic range; or a variable electric or electronic signal may produce a varying air pressure output. These deviceswill be discussed Chapter 3.

2.2.12 Manually Operated Pneumatic Switches

Manual switches are used extensively in pneumatic control systems. A simple two-position switch, shown in Figure 2-24, may be used to switch from one signal (A) to another (B). More commonly, control air is provided at port A, and port B is open to vent the line from AB when desired.

A more complex two-position switch is shown in Figure 2-25. In one position ports 1 and 2 are connected, as are ports 3 and 4. In the other position port 1 is connected to port 4 and port 2 to port 3.

Figure 2-26 shows a gradual switch that is used to provide a fixed output pressure (using control supply air as input). It is an adjustable pressure-reducing valve, whose primary function is to provide a minimum signal for a damper, valve, or speed controller. It is also used for set point adjustment in sensor-controller systems.
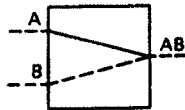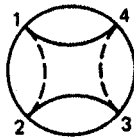
Figure 2-24. Manual switch; 2-position.

Figure 2-25. Manual switch; 2-position, 4 ports.
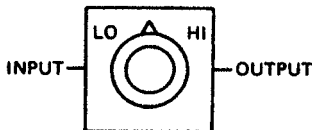
Figure 2-26. Hand "gradual" switch.

2.3   CONTROL CABINETS

With any control system that includes more than three or four devices it is desirable to provide a control cabinet in which all the controllers, relays, switches and indicating devices for the system will be mounted Control manufacturers supply standard cabinet sizes, up to about 36 by 48 inches and the cabinets usually are customized for the system. Indicating devices and switches are mounted on the door so that the cabinet may be closed and locked during normal operation.

Indicating devices may show the status, temperature, humidity, and control pressure at various points, as well as HVAC system static pressures, filter pressure drops, and even air flow rates. Pilot lights may be added to show the motor and safety control status.

Although a control cabinet adds some cost, it is extremely useful for monitoring system operation and for trouble shooting. Without some kind of status information it is very difficult to figure out what actually is happening in the system.

2.4   AIR SUPPLY

The air supply for a pneumatic control system must be carefully designed. It is of the utmost importance that the air be clean and dry, free from oil, dirt, and moisture. Thus it is essential to use air dryers, oil separators, and high-efficiency filters. Even small amounts of dirt, oil, or water can plug the very small air passages in modern commercial pneumatic devices, rendering them useless. See Figure 2-27.

Air consumption can be estimated from use factors for the components provided by the control manufacturer. Good practice requires that the compressor have a capacity at least twice the estimated consumption. In comparing compressor ratings it should be noted that displacement and capacity are different. Capacity will be from 60% to 80% of displacement.
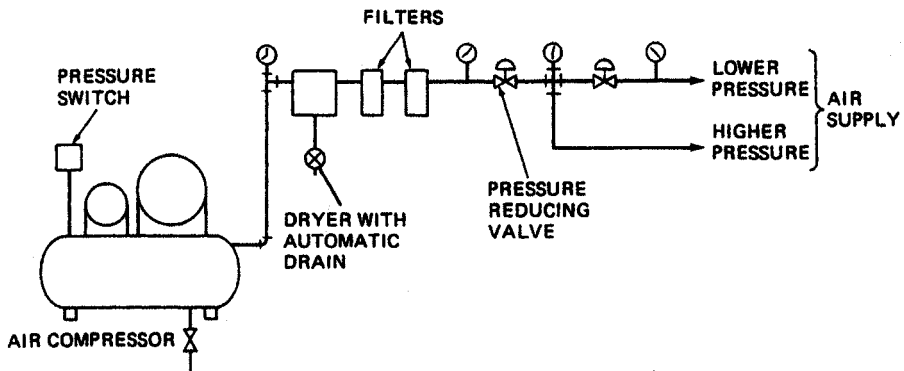
Figure 2-27. Compressed air supply.

If the system is very large, it is preferable to have two compressors to improve reliability. Generally air is compressed to 60 to 80 psig, and passes through dryers, filters, and a pressure-reducing valve. It is very important that the air be dry and clean, to minimize maintenance on the components.

2.4.1 Air Compressors

Compressors usually are reciprocating-type, single-stage, air-cooled. Typically they range in size from 1 hp to 10 hp, and the smaller units are mounted directly on the receiver tank. Motors are started and stopped by a pressure switch on the receiver, set for 60 to 80 psig with about a 10 psi differential. When two compressors are used, the receivers are crossconnected and a single pressure switch with an alternator controls both motors. The alternator is a sequencing device that provides for running a different motor on each start-stop cycle, thus equalizing motor running time.

2.4.2 Dryers and Filters

Two methods of drying are used. Refrigerated air dryers cool the air sufficiently to condense out the excess moisture that is then removed automatically through a trap, which is similar to a steam trap.

Chemical dryers are also used, with silica gel as the agent. For small systems an in-line unit may be used, and the chemical must be replaced at regular intervals. For large systems a double unit may be used, with one section being regenerated while the other is in use.

Oil may be removed in a coalescing filter, especially designed for oil mist entrapment and removal. The remaining oil and dirt may be removed in a three-micron high efficiency filter.

2.4.3 Pressure-Reducing Valves

The control manufacturer can and does supply the pressure reducing valves as part of the pneumatic control system. If a dual-pressure system is used, then two reducing valves will be used, in series, as in Figure 2-27.

2.4.4 Air Piping and Accessories

Air distribution piping systems often use soft copper tubing with soldered fittings. Compression fittings may be used at equipment connections. Branch piping is commonly 1/4 inch outside diameter.

Improvements in nylon-reinforced plastic tubing have led to acceptance of this material in many if not most recent pneumatic control projects. Specifications should call for a pressure test of the piping system at 30 psi.

Air gages are a necessity for adjusting, calibrating, and maintaining the control elements. Gages should be provided at main and branch

connections to each controller and relay and at the branch connection to each controlled device. In this manner the status of the component is readily determined.

3

Electric and Electronic Control Devices

3.1 ELECTRIC CONTROL DEVICES

Electrical controls are available in a variety of configurations, for every conceivable purpose in HVAC applications. All use one of four basic parts: the switch, the electromagnetic coil or solenoid, the two-position motor, and the modulating motor.

Although basic principles of electrical circuits are not discussed in this book, the manufacturers' handbooks listed in the bibliography include some very good presentations.

Any electrical circuit includes three elements: a power source, a switch, and a load (Figure 3-1). The switch serves to turn the power on and off. In an HVAC control system the load will be an actuator or a relay, and the switch will be the sensor or the controller. The power source is usually the building electrical power, which may be used at a normal 120 V or transformed to some lower voltage, typically 24 V. Some electrical devices use direct current (DC). This may be supplied by a battery or from an alternating current (AC) source by means of a transformer and a rectifier.
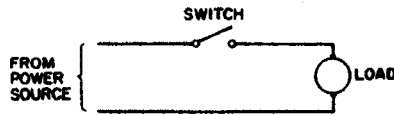
Figure 3-1. Electrical circuit.

3.1.1 Two-Position Controls

3.1.1.1 Sensors

Many of the sensors described in Section 2.2.5 "Sensing Elements," may be used for two-position (on-off) electric control.

The bimetal is very commonly used in electric thermostats because it can serve to conduct electricity. Figure 3-2 illustrates a simple single-pole, single-throw (SPST) bimetal thermostat. When it is used for heating, a decrease in room temperature will cause the bimetal to bend toward the contact. When the contact is almost closed, a small permanent magnet affects the bimetal enough to cause a quick final closure and to lock it in place. This magnet also causes a lag in the release, with resultant quick opening of the contact. This minimizes arcing and burning of the contacts, and eliminates chattering.

Figure 3-2. Bimetal sensor, electric.

The bimetal may also be arranged in a spiral, fixed at one end and fastened to a mercury switch at the other (Figure 3-3). The mercury switch is simply a glass tube partially filled with mercury and with wiring connectors at one or both ends. It is loosely pivoted in the

center so that when it turns past center the weight of the mercury running to the low end causes it to pivot farther. The mercury bubble acts as a conductor to connect the electrodes. This arrangement is often found in residential thermostats.

Figure 3-3. Mercury switch.
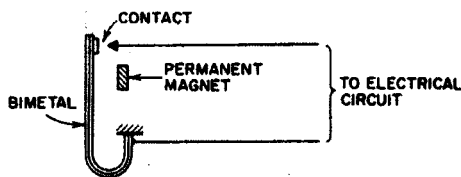
The mercury tube also can be used with a bourdon tube pressure sensor.

The diaphragm movement of bulb and capillary sensors can be used to trip an electric switch, and bellows sensors can be used in the same manner. The tripping action can be direct or through a linkage. A bistable spring or overcenter mechanism is required to provide the snap action.

Humidity sensors also can be used to trip switches, through either bending action or expansion.

3.1.1.2 Safety Controls

Safety controls are used in HVAC systems for the detection of abnormally high or low temperatures and for smoke detection.

High temperature sensors or smoke detectors are required in most systems by National Fire Protection Association and local codes. Low temperature sensors are used to prevent freeze-up.

A high temperature sensor will usually have a bimetal or rod-and-tube element designed for insertion in the supply or the return air duct. Factory temperature settings of 125°F to 135°F are provided. If the air temperature exceeds the control setting, a switch will open and remain

open until the device is manually reset. The control is commonly used to stop the air handling unit supply fan.

Smoke detectors for duct installation must be specifically designed for that use. The detector continuously samples the air stream in the duct and compares the sample with a standard. If products of combustion are detected, a control contact is opened. Additional contacts may be provided for alarm service and reporting. The smoke detector may be used to stop the supply fan, but often, in modern systems, is used to position dampers for smoke control and evacuation while the fan continues to run.

A low temperature duct sensor should have a long capillary so designed that a freezing temperature at any point will cause the sensor to open the relay contact. This prevents freezing due to stratified air streams. These devices may be automatically or manually reset.

3.1.1.3 Electromagnetic Devices

Sometimes called electromechanical devices, this class includes relays, solenoid valves, and motor starters.

These elements use the principle of electromagnetism. When an electric current flows through a wire, a magnetic field is set up around the wire. If the wire is formed into a coil then the magnetic field may become very strong, and a soft iron plunger placed in proximity to the end of the coil may be drawn up inside it. This is the solenoid that can then be used to operate a valve or a set of contacts.

Solenoid valves are made in many sizes and arrangements, for control of water, steam, refrigerants, and gases. Figure 3-4 shows a typical two-way valve. This valve is held in the normally closed position by fluid pressure.

When the coil is energized, the plunger is lifted and opens the valve. Some models are arranged with an internal pilot: a small port that is opened by the solenoid allowing fluid pressure to open the valve.

Three-way arrangements are common, and four or more ports are not unusual. The maximum size for a solenoid valve is about a 4 inch pipe size. Large sizes lead to problems of pressure and water hammer due to quick opening and closing.
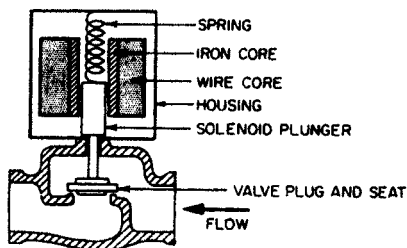

Figure 3-4. Solenoid valve.

Control relays are designed to carry low-level control voltages and currents, up to about 15 A and 480 V. The contact rating of a control relay will vary with the voltage and with the type of load. The rating will be higher for a resistive load than for an inductive load. A relay can make a circuit with a much higher current than it can break without arcing and burning the contacts. The breaking capacity of the relay should be the criterion.

One typical control relay configuration is the coil and solenoid shown in Figure 3-5. The figure shows double-pole, double-throw (DFDT) contacts, but many arrangements from single-pole, single-throw to as many as eight poles are available. The armature is spring-loaded so that it will return to the normal position when the power to the coil is turned off. Coils are available for most standard voltages, that is, 24, 48, 120, 208/240, and 440/480.

The solenoid-type relay is also available in latching arrangements, so that it may be driven to one position by a short-time energization of the coil and will stay in that position until returned by energizing a second coil. One latching system uses weak permanent magnets at each electromagnetic coil. These permanent magnets are not capable of displacing the solenoid, but will hold it in position (Figure 3-6).

Another method uses a mechanical latch or detent, which is tripped when the electromagnetic coil is energized. The advantage of the latching relay is that no power is required to hold it in position. The disadvantage is that it does not "fail safe" when power is removed.
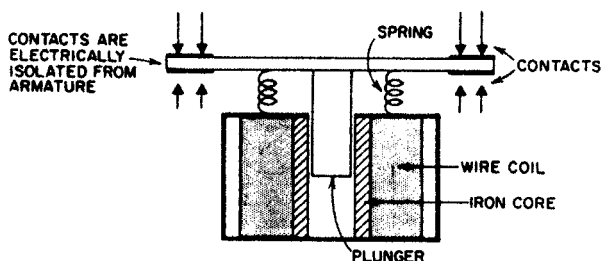
Figure 3-5. Relay coil and solenoid.
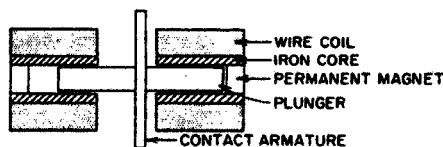
Figure 3-6. Permanent-magnet latching type relay

Electromagnetic coils are also used in clapper-type relays (Figure 3-7). The coil is mounted near a soft iron bar that is part of a pivoted, spring-loaded contact armature. When the coil is energized, the arm is pulled over to close the contact. Several contact circuits may be mounted on a single armature. Double-throw contacts are also supplied. This type of relay usually, but not always, has a lower current and voltage rating than the solenoid type. The units are available as reed relays, and some miniature versions are made for electronics work.

Contactors are similar in all respects to solenoid-type control relays, but are made with much greater current-carrying capacity. They are usually used for electric heaters, or similar devices with high power requirements. A special kind of contactor uses mercury switch

contacts, to allow for the frequent cycling required in electric heating applications.
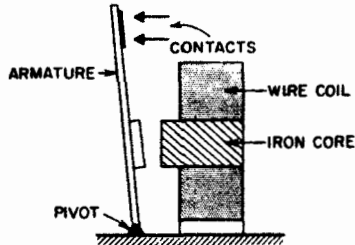
Figure 3-7. Clapper-type relay

Motor starters also use the solenoid coil actuator, and are similar to relays but with the addition of overload protection devices. These devices sense the heating effect of the current being used by the motor and break the control circuit to the coil if the current exceeds the starter rating. A typical across-the-line starter schematic is shown in Figure 3-8. This subject is discussed further in Chapter 8.
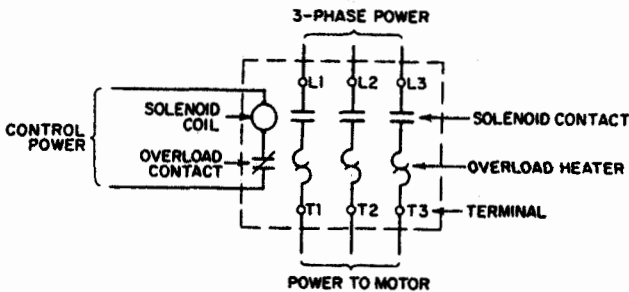
Figure 3-8. Motor starter.

Time-delay relays, as the name implies, provide a delay between the time when the coil is energized (on-delay) or deenergized (off-delay) and the time when the contacts open and/or close. This delay may range from a small fraction of a second to several hours. Three general classes of time delay relays are available: solid-state, pneumatic, and clock-driven.

Solid-state timers use electronic circuits to provide highly accurate delay periods. Timing ranges vary from 0.05 seconds to 15 minutes or more. Pneumatic timers use the familiar solenoid coil principles, but the movement of the solenoid is delayed by the diaphragm cover of a pneumatic chamber with a very small, adjustable leak port (Figure 3-9). In the on-delay sequence, when the coil is energized, the solenoid pushes against the diaphragm, forcing the air out of the chamber. The leak port governs the rate of escape, and therefore the time delay. When the coil is deenergized, a check valve opens to allow the chamber to refill rapidly. Delays from about 0.1 seconds to 60 minutes are available. Delay on deenergizing is also available.
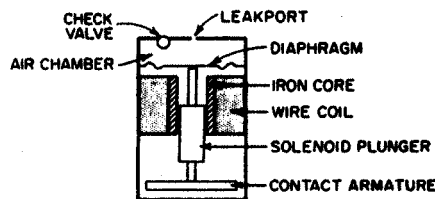
Figure 3-9. Time-delay relay, pneumatic type.

Clock timers use a synchronous clock motor that starts timing when the power is turned on. At the end of the timing period the control circuit contacts are opened or closed. When the power is turned off, the device immediately resets to the initial position. Clock timers are available with ranges from a fraction of a second to about 60 hours. Many special sequences and capabilities are used in process control.

Most of these time-delay relays may be provided with auxiliary contacts that open and/or close without delay, as in an ordinary relay.

Sequence timers include a synchronous motor that drives a cam shaft through a chain of reducing gears. Adjustable cams operate switches in any desired sequence. The timer may go through a single cycle, then stop; or it may run continuously, repeating the cycle over and over. Units with as many as 16 switches are available. Modulating or floating-type motor control can also be used if needed. Then the sequence can be stopped at any point with the HVAC system stabilized

at that condition. A typical use for this type of control is the sequencing of cooling tower fans.

3.1.2 Two-Position Motors

Two-position motors are used for operating dampers or for valves that need to open and close more slowly than a solenoid coil will allow. Motors may be unidirectional, spring-return or unidirectional, three-wire.

A spring-return motor is shown in Figure 3-10. When the controller closes its SPST switch, the motor winding is energized from A to B. This starts the motor, and it runs, driving a crankshaft and linkage (through a reduction gear) to open or close a valve or damper. A cam is mounted on the shaft, and at the proper position (usually 180° of rotation but sometimes less) the cam throws the limit switch from B to C. This added coil resistance reduces the current to a holding level, which will hold the motor in this position but will not cause damage. When the controller switch opens, the spring returns the motor to its original position.
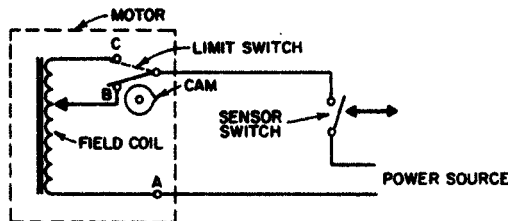
Figure 3-10. Two-position spring-return motor.

Figure 3-11 shows a three-wire motor. For discussion purposes assume that this motor is operating a heating valve and that the valve is closed in the position shown. A double-throw controller is required, and assume that this is a thermostat that closes contact B on temperature rise and contact A on temperature fall. The controller is shown in the satisfied position. The limit switches (marked SW1, SW2) are operated

by a cam of the motor crankshaft. On a fall in room temperature the thermostat closes contact A, establishing a circuit through SW2 to the motor field coil. The motor runs, and almost immediately SW1 closes, establishing a maintaining circuit to the coil. Now the motor will run 180° regardless of what happens at the controller. When a 180° stroke has been completed, SW2 is opened by the cam and breaks the circuit to the coil, stopping the motor. The valve is fully open. On a rise in temperature the controller breaks contact A and makes contact B, establishing a circuit through SW1 to the coil and restarting the motor. As the valve starts to close SW2 closes and makes the maintaining circuit. When the valve is fully closed, the cam opens SW1, stopping the motor.

Figure 3-11. Two-position motor, three-wire arrangement.

In case of power failure the three-wire motor will stop and stay wherever it may be. The spring-return motor will return to normal position.

These motors may be provided with cam-operated auxiliary contacts that open and/or close at any desired point in the rotation.

3.1.3 Modulating Motors

Modulating motors are used for proportional and floating controls. They must, therefore, be reversible and capable of stopping and holding at any point in the cycle. The motors used in these devices will be either reversible two-phase induction motors or reversible shaded-pole motors.

3.1.3.1 Reversible Induction Motors

Schematically, the two-phase induction motor (Figure 3-12) has two field windings directly connected at one end (C) and connected at the other by a capacitor. Power may be supplied at A or B on either side of the capacitor, with the other power connection at C. If, for example, alternating current power is connected across A and C, then coil 1 is directly powered and coil 2 is indirectly powered through the capacitor. The effect of the capacitor is to introduce a phase shift between coils 1 and 2; thus a rotary motion is imparted to the motor armature. If the power is applied across B and C, the phase is reversed, and the motor runs in a reverse direction.
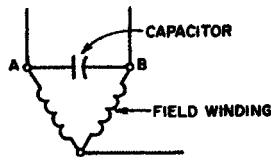

Figure 3-12. Reversible induction motor.

3.1.3.2 Shaded-Pole Motors

The shaded-pole motor is constructed with a main field coil that is directly energized. However, by itself this will not start the motor. To provide a biasing effect shading coils are added as shown in Figure 3-13. These are powered by transformation effect from the main field coil, and when both ends of the shading coil windings are shorted out, a phase lag is caused in part of the field. This produces a rotating field that starts the motor and improves its efficiency while running. This arrangement is unidirectional. To provide a reversing motor two additional shading coils are added (Figure 3-14) and wired as shown in Figure 3-15. Grounding one pair of shading coils causes the motor to run in one direction, grounding the other pair causes the motor to reverse rotation.
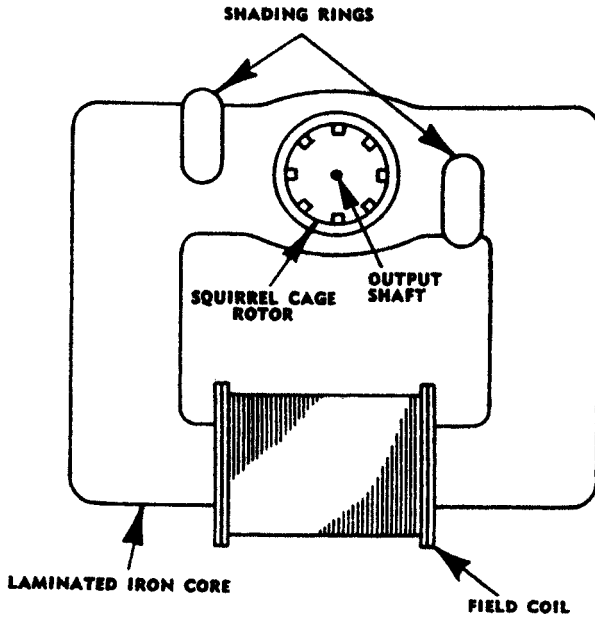
SHADING RINGS

SQUIRREL CAGE
ROTOR

OUTPUT
SHAFT

LAMINATED IRON CORE

FIELD COIL

Figure 3-13. Shaded-pole motor

SHADING COILS

LAMINATED
IRON CORE

SQUIRREL
CAGE
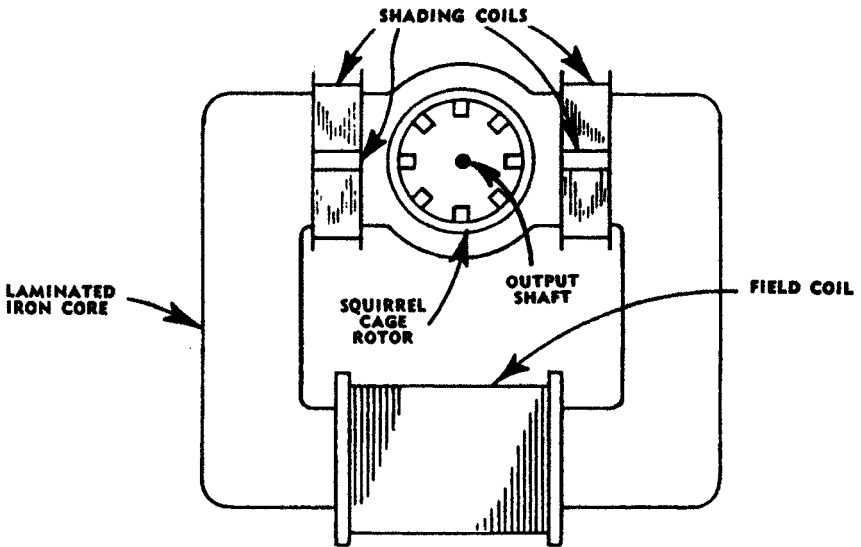ROTOR

OUTPUT
SHAFT

FIELD COIL
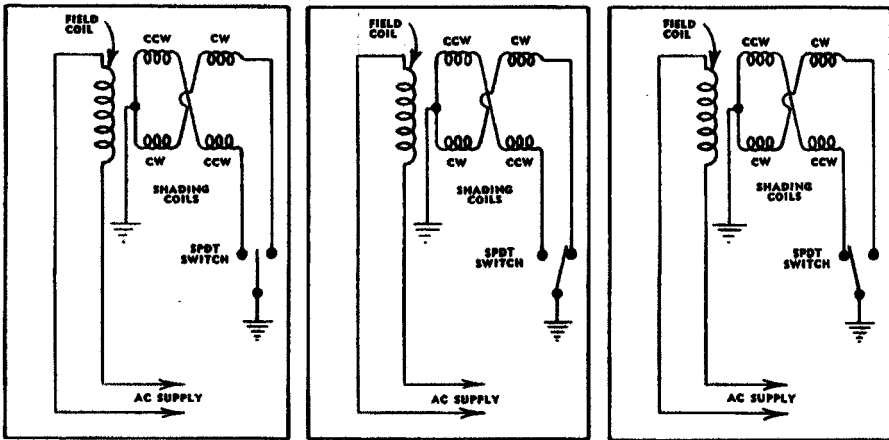
Figure 3-14. Reversible shaded-pole motor.

Figure 3-15. Shaded-pole motor wiring diagrams.

3.1.3.3 Modulating Motor Control

Two general types of control configurations are used with modulating motors.

Floating control includes a three-wire controller with a center dead spot and no feedback. The motor is geared down to provide a slow change in the controlled device, so that system response will usually cause the sensor to return the switch to the center-off position before the motor has traveled the full stroke. Figure 3-16 shows a shaded-pole motor with this type of control. The limit switches are cam-operated at the end of the stroke (as described under two-position motors). The sensor controller is commonly called a floating controller. This same controller may be connected in a similar manner to an induction motor.

Fully modulating control requires negative feedback from the motor, as in Figure 3-17. Figure 3-18, which shows only the potentiometers and relay coils, will aid one in understanding the following description. In the balanced condition shown, the wiper arm of the controller potentiometer is centered on its resistance winding, as is that of the feedback potentiometer. Under these conditions the currents through the two coils of the balancing relay are equal, and the relay arm is

centered between contacts (Figure 3-18(A)). If the controller responds to temperature (or pressure) change and moves its potentiometer wiper arm, the circuits are unbalanced, and the current differences in the two relay coils cause the relay arm to swing to one contact (Figure 3-18(B)). This starts the motor and causes it to actuate a valve or a damper to offset the variation from set point. The motor operation also moves the feedback potentiometer wiper, which offsets the effect of the controller. When balance is restored (with the motor in a new position), the relay coil currents are again equal, the relay arm is centered, and the motor stops (Figure 3-18(C)). The limit switches are needed to stop the motor at the end of the stroke if the sensed conditions are beyond the throttling range of the controller.
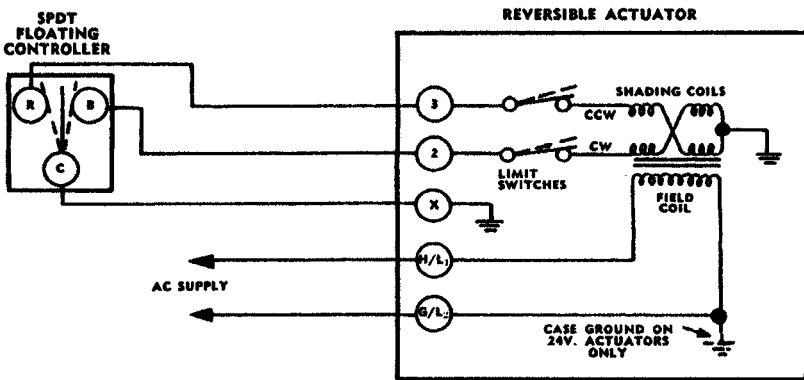

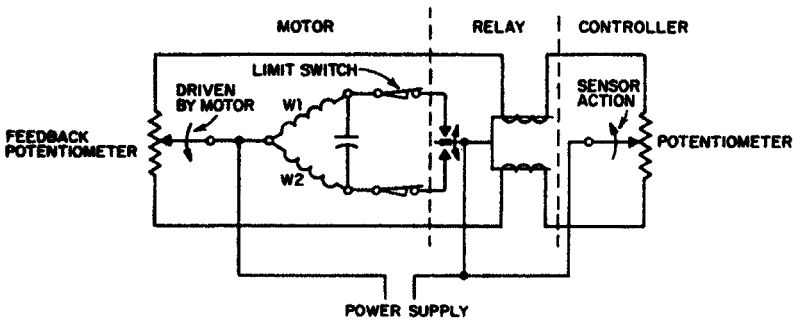Figure 3-16. Modulating motor control.
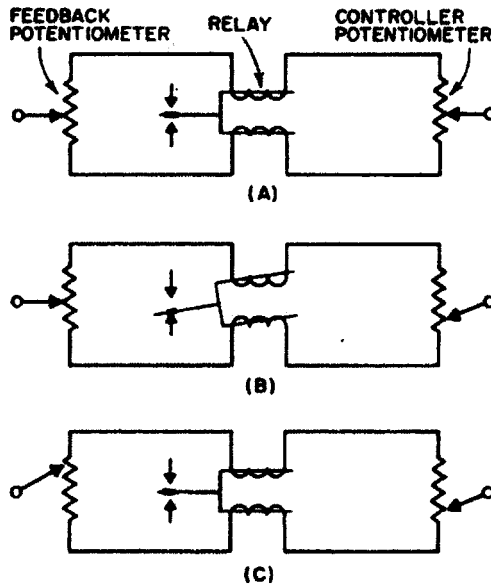

Figure 3-17. Modulating induction motor.

Figure 3-18. (A) Balanced position. (B) Controller moved, relay unbalanced. (C)Feedback moved, relay balanced.

The modulating motor may be equipped with cam-operated auxiliary switches that open or close at any point in the stroke. It also may drive an auxiliary potentiometer that provides input to make another modulating motor follow the action of the first motor driving another valve or damper.

3.2 ELECTRONIC CONTROL DEVICES

Electronic controls are distinguished from electrical controls by low voltages and solid-state circuitry. The power supply voltage is typically 24 V AC or DC, but signal level voltage ranges are commonly 0 to 5 or 0 to 10 V. Current is also used as a signal, the usual standards being 4 to 20 ma (milliamperes) or 10 to 50 ma ranges. Simplified discussions of operating principles are included here without details of construction and theory.

The increasing sophistication and the decreasing cost of electronic devices, and their easy interface to computer-based controls, are leading to their more frequent use in preference to pneumatic devices.

3.2.1 Bridge Circuits

The original and most commonly used bridge circuit is the Wheatstone bridge (Figure 3-19). The bridge is formed by four resistances connected as shown. Power is connected to two corners of the bridge, and output to the two opposite corners. One or more of the resistances may be variable. (R4 is shown here as variable by the arrow across it.) When all resistances are equal the output is zero.
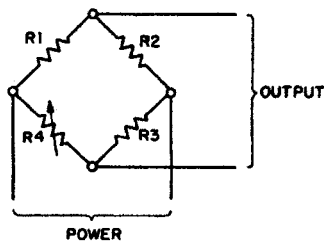

Figure 3-19 Wheatstone bridge.

If one or more of the resistances is changed, the bridge becomes unbalanced, and an output signal results that is approximately proportional to the resistance change.

In the ordinary electronic sensor, the variable resistor is the sensing element and is often mounted remote from the rest of the bridge and the amplifier (Figure 3-20). When remote mounting is required, some method may be needed to compensate for the resistance of the connecting wire.

This very simple arrangement does not allow for adjustment of the set point or calibration. For these functions it is necessary to add two more resistances (Figure 3-21). The set point adjustment is in series (or parallel) with the sensor resistor, and the calibration adjustment is a potentiometer with an adjustable wiper arm.
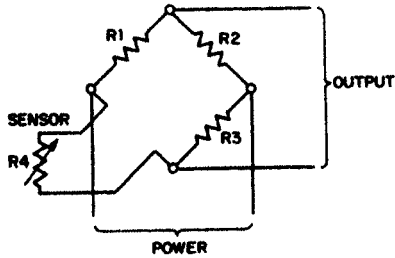
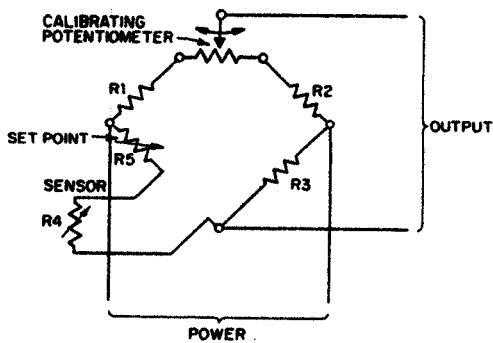Figure 3-20 Wheatstone bridge with remote sensor.

Figure 3-21. Bridge with calibration and set point.

To provide negative feedback for modulating controls a throttling range bridge must be added to the circuit. This is wired in series with the main bridge (Figure 3-22). It includes a variable potentiometer with a wiper arm that is driven by the controlled device motor. When the main bridge is unbalanced and causes the motor to run, this potentiometer adjusts the throttling range bridge to offset the effect of the sensor and rebalance the system. In a simpler arrangement, the motor-driven potentiometer may be placed in series with the sensor, but then the throttling range is not adjustable.
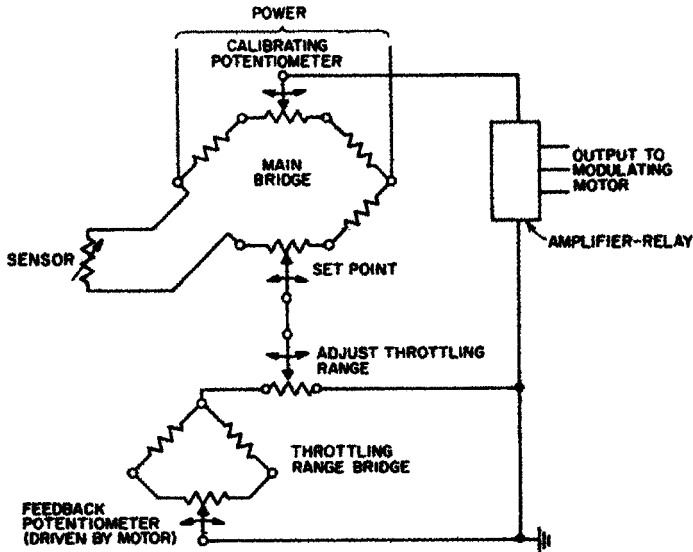
Figure 3-22. Throttling range bridge.

Sensing and relating of two control points, such as room temperature and discharge temperature, may be done with a single bridge circuit (Figure 3-23). Here the two sensors are shown on opposite sides of the bridge, but the room sensor has the same basic resistance as the other legs of the bridge. The discharge sensor usually has only 10% to 20% of the basic resistance, with the balance being furnished by fixed resistors. This gives the discharge sensor less authority, the amount being expressed as the ratio of discharge sensor resistance to room sensor resistance. Typical values are 1000 ohms for the room sensor and 100 ohms for the discharge sensor. Then the authority of the discharge sensor is 100:1000 or 10%, which means that a 10-degree rise in discharge temperature is necessary to rebalance the bridge after a one-degree fall in room temperature.

Thus, when the room temperature decreases, the resistance of the room sensor R1 is decreased. This unbalances the bridge and causes an output to the amplifier. This may cause a hot water valve to open, which increases the discharge air temperature. This is sensed by the discharge sensor R2 and results in an increased resistance at the sensor until the bridge is rebalanced.
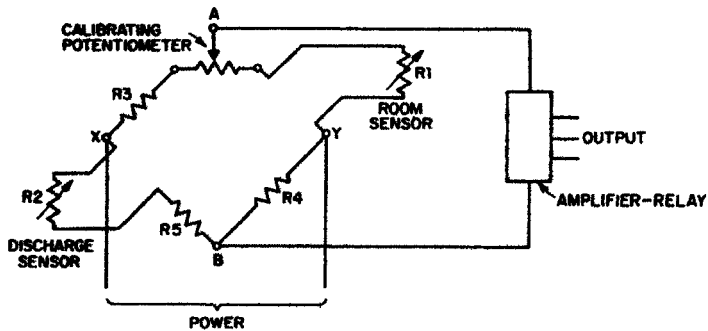
Figure 3-23. Bridge with two sensors.

Figure 3-24 will serve to clarify the situation. Figure 3-24(A) shows the bridge in the balanced condition, with a 12 V drop being shared equally by each of the two resistors in each side. Then there is no difference in potential across A-B; therefore there is no output. When the resistance of R1 decreases, the voltage drop across it will also decrease, say to 5 V. But to maintain the total 12 V drop, the drop across R3 must increase to 7 V (Figure 3-24(B)). Now there is a difference in potential from A to B, which the amplifier will sense and use to operate a valve motor. As the discharge temperature increases, resistance R2 will increase until the voltage drop across R2 + R5 equals 7 V (leaving 5 V across R4), and the bridge is rebalanced (Figure 3-24(C)). Output ceases and the valve motor is stopped at this position.

The system as shown is set up for heating. For cooling it is necessary to interchange the location of R4 with that of R2-R5.
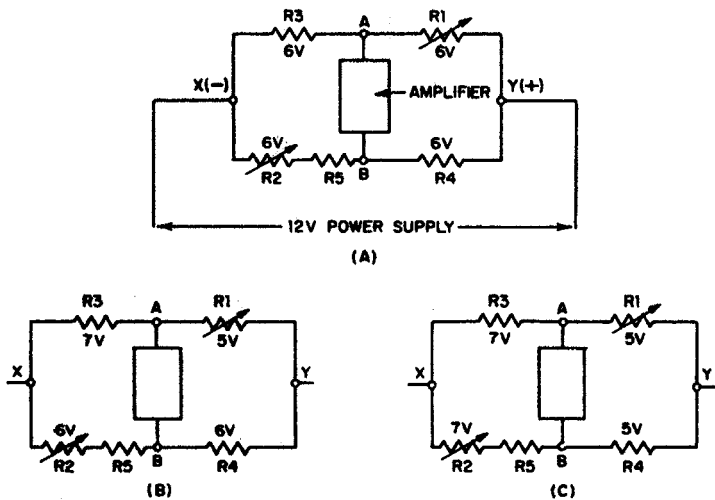
Figure 3-24. (A) Bridge balanced. (B) Room temperature decrease, bridge unballanced. (C) Discharge temperature increase, bridge rebalanced.

3.2.2 Electronic Sensors

3.2.2.1 Temperature Sensors

Electronic sensors for temperature sensing in modern HVAC systems are usually resistance temperature detectors (RTD), in which resistance varies with temperature. The least expensive and most common RTD is the thermistor, which uses a solid-state device. A thermistor has a high reference resistance, typically 1000 ohms at 0 °C and a high change in resistance per degree change in temperature. Low-cost thermistors tend to drift (get out of calibration), especially when subjected to thermal cycling. They require frequent calibration. Special caution is needed to be sure a replacement thermistor has the same operating characteristics as the thermistor being replaced. Despite these maintenance disadvantages, thermistors are popular because they are low-cost and can be used with lower-quality electronics.

Platinum RTDs, using pure-platinum wound-wire resistors, will retain their calibration indefinitely. Typically, the platinum RTD has a reference resistance of 100 ohms. This low resistance makes the

resistance of the leads significant, and three- or four-wire leads are used to compensate. Higher-quality electronic circuits are needed because the resistance of platinum RTDs changes only slightly with temperature.

The thin-film platinum RTD is a temperature-sensitive resistor made by deposition of a thin platinum film on a substrate. The device uses very little platinum but has a high reference resistance of 1000 ohms or more.

Thermocouples are seldom used in electronic control for HVAC. However, there is one thermocouple application that is common in small residential gas-fired heating systems. A special form of thermocouple called a thermopile (several thermocouples wired in series) is inserted in the gas burner pilot flame. The heat of the flame generates enough electric current to power a special gas valve, through a bimetal thermostat. Because the small available current will not operate a conventional solenoid valve, the valve used is a balanced diaphragm type. A small solenoid pilot valve opens to allow gas pressure from the supply line to move the diaphragm and open the valve.

3.2.2.2 Velocity Sensor

A hot wire anemometer is an electronic velocity sensor made from two temperature sensitive wires. One simply senses the air temperature whereas the other wire is heated by passing a current through it so that its temperature is a fixed amount higher than the air temperature. Because the heat transfer coefficient from the heated probe to the air is a function of air velocity over the probe, the amount of current required to keep the probe at an elevated temperature is proportional to the air velocity.

Hot wire anemometers can measure very low velocities (a few feet per minute) and can be fast in responding if the wires are thin.

3.2.2.3 Humidity Sensors

There are several types of electronic sensors for humidity. Synthetic fabrics that change dimensions with humidity changes are still used, but they have poor accuracy and need frequent calibration. A good device for measuring the dew point uses a tape impregnated with lithium chloride and wound with two wires that are connected to a power supply. As the lithium chloride absorbs moisture from the atmosphere, it creates an electrical circuit that heats the system until it is in balance with the ambient moisture. The resulting temperature is measured. The device is quite accurate but requires frequent maintenance to retain its accuracy.

Solid-state humidity sensors use thin or thick polymer film elements so that resistance or capacitance varies with relative humidity.

One of the most accurate dew-point sensors is the chilled-mirror type polished mirror is provided with a small thermoelectric cooling system and a light beam is reflected from the mirror to a photo cell (Figure 3-25). When the mirror is cooled to the ambient dew-point temperature, moisture condenses on it, changing the mirror from a specular to a diffuse relector. The resulting change in light reflectivity serves as feedback to a circuit that controls the temperature of the mirror so that it is at the dew point. The mirror temperature (the dew-point temperature) is measured by a platinum RTD. The only maintenance required is periodic cleaning of the mirror. By simultaneously measuring the dry bulb temperature, one can compute the relative humidity. Instruments are available that make the needed calculation and provide relative humidity as an output.

3.2.2.4 Pressure Sensors

Electronic pressure sensors can use the same sensing elements as pneumatic and electric devices while providing an electronic-level signal. Peculiar to electronic circuitry is the strain gage where a small solid-state device is connected to the diaphragm of a pressure sensor. When it is distorted by pressure changes, the resistivity of the device

varies. A small distortion produces a significant change so that very small pressure changes can be measured, as low as a few hundredths of an inch of a water column.
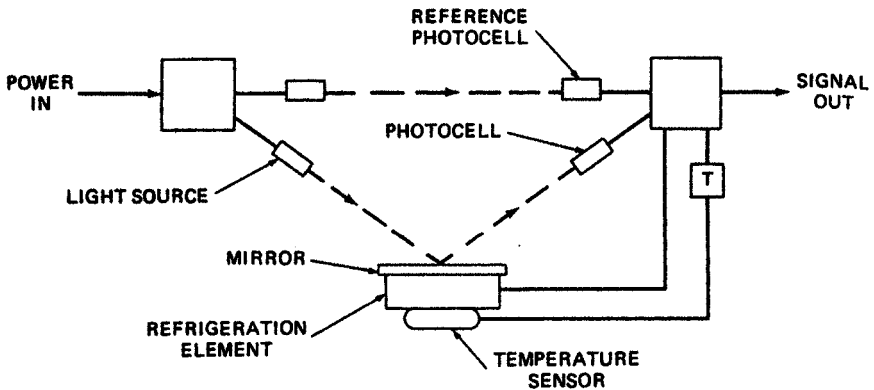

Figure 3-25. Principle of chilled dewpoint mirror sensor.

Figure 3-26 shows a pressure sensing device that uses two hot wire anemometers to measure flow through a small tube connecting two adjacent rooms or spaces. The pressure difference between the spaces is proportional to the square of the velocity through the tube. By measuring this velocity the pressure difference can be determined. Since hot wire anemometers can measure very low velocities, this is a very sensitive pressure-measuring instrument.

By using two anemometers the direction of flow also can be determined because one anemometer is in the wake of the other. This allows this pressure instrument to determine which space is at a higher pressure.

3.2.3 Amplifiers and Transducers

The output of an electronic sensor usually is so low that an amplifier is required. Amplifiers use bridge circuits and other techniques to condition the signal-including linearization if needed-and raise it to a

level adequate for transmission and use by controllers. (See Figure 3-27.)

An electronic-to-pneumatic (E/P) transducer produces a pneumatic output signal proportional to input voltage or current. One type has a restricted air supply fed to a nozzle covered by a plunger that is surrounded by an electromagnetic coil, as shown in Figure 3-28. The force that the plunger exerts on the nozzle is proportional to the current or the voltage supplied to the coil. In equilibrium, this force is just balanced by the pneumatic output pressure times the area of the nozzle opening. Thus the output pressure will vary with the electronic input signal.
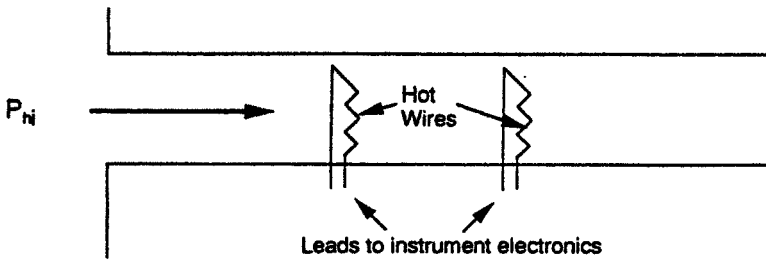
Figure 3-26. Hot wire anemometers used to measure pressure.
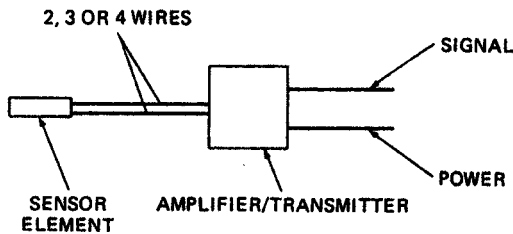

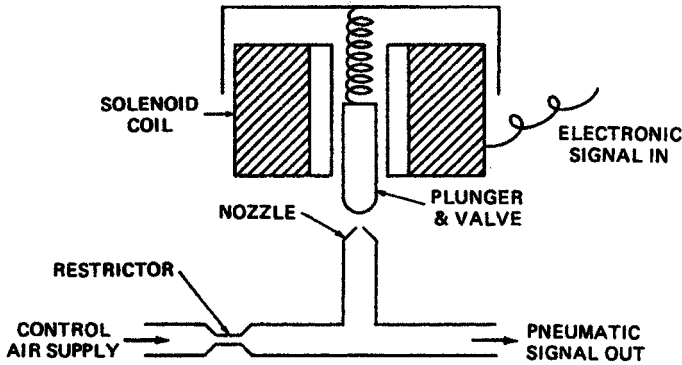Figure 3-27. Sensor-transmitter.

Figure 3-28. E/P (or I/P) transducer, variable bleed.

Another type of E/P transducer uses a stepper-motor to turn the stem on a small pressure regulator.

A pneumatic-to-electronic transducer (P/I) uses a potentiometer with the wiper arm driven by a bellows that senses the output of the pneumatic controller. Then the potentiometer resistance varies with branch-line pressure. Solid state P/I transducers also are available, using the principle that current flow in solid-state devices will vary as physical pressure is applied to the device.

3.2.4 Electronic Controllers

The basic element in an electronic controller is a device called an operational amplifier (OP Amp). The OP Amp is a solid-state amplifier that can provide a large gain while handling signals that vary with time and over a wide frequency range. The gain of the OP Amp is the negative of the ratio of voltage out to voltage in; thus:

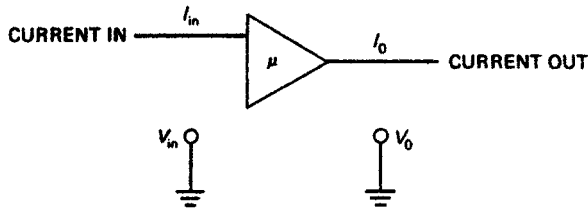$$\mu = -(V_o/V_i)$$

(See Figure 3-29.)

Figure 3-29. Ideal operational amplifier (OP amp).

Because the gain of an OP Amp is very high (essentially infinity in the idealization of an OP Amp), any small input current or corresponding input voltage will produce a very large negative output voltage.

To make the OP Amp useful, resistors and capacitors are used as parts of the input and feedback circuits.

Figure 3-30 shows a proportional amplifier. To understand how this device works, recall that any input current to the OP Amp will create a large negative output voltage. Acting through the feedback resistor, this negative voltage will cause the input voltage to drop to a value very near zero. For this to occur, almost all the current must flow around the OP Amp though the feedback resistor. We can now see how the proportional amplifier functions. For example, suppose that both the input resistor and the feedback resistor have the same value and V_i is 5 V. If all the current is to flow around the OP Amp, and the input voltage to the OP Amp is to be near zero, the output of the OP Amp must be -5 V. Similarly, if V_i is 10 V, the output of the circuit will be -10 V.

Now suppose that the feedback resistor has twice the resistance of the input resistor. We see that in this case if V_i is 5 V, the output of the circuit will be -10 V. This shows how the gain of the proportional amplifier is determined by the ratio of the feedback to input resistors.

The symbol for a proportional amplifier used in control diagrams is also shown in Figure 3-30.
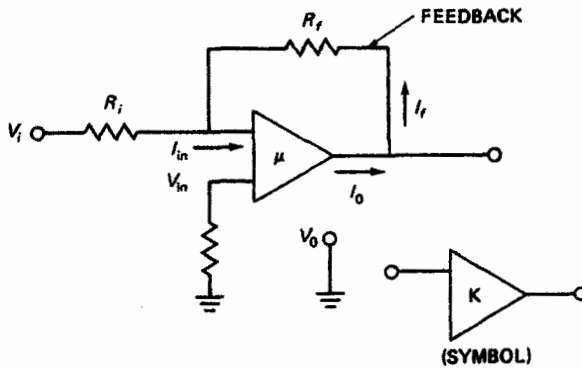
Figure 3-30. Basic proportional OP amp.

To build circuits that integrate or differentiate an input signal, a combination of capacitance and resistance must be used. Figure 3-31 shows the integral mode arrangement with an input resistor and a feedback capacitor. The gain becomes a function of the charging time of the capacitor. Figure 3-32 shows a derivative mode arrangement with an input capacitor and a feedback resistor.
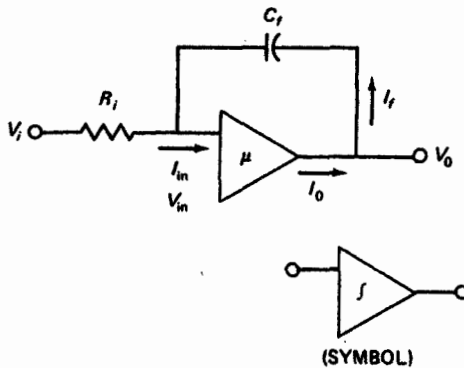
Figure 3-31. Integrating OP amp.

To combine the various modes, other OP Amps are used as summers. The simple "summer" circuitry is shown in Figure 3-32. Subtraction is shown in Figure 3-33.
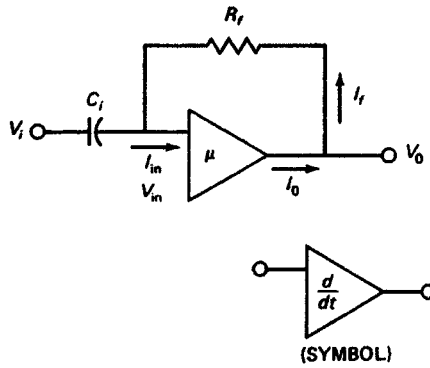
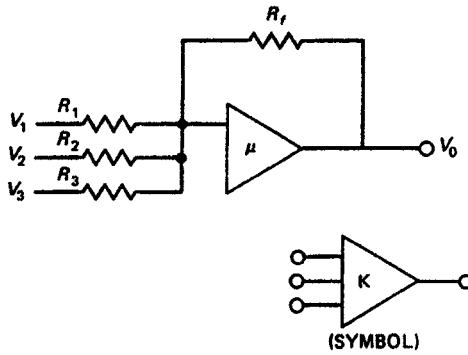Figure 3-32. Derivative OP amp.
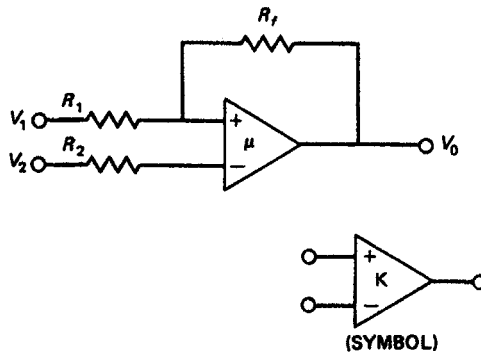

Figure 3-33. Summing OP amp.

Figure 3-34. Subtraction OP amp.

To form a complete electronic controller, two input voltages are added algebraically (with due regard for negative and positive signs) to compare the set point and the measured value of the variable (the measured value might be the output from a Wheatstone bridge circuit, for example). The output from this summer is fed to up to three other OP Amp circuits where proportional, integral, and derivative control signals are produced. These are fed to a summer to produce the final output as shown in Figure 3.35.
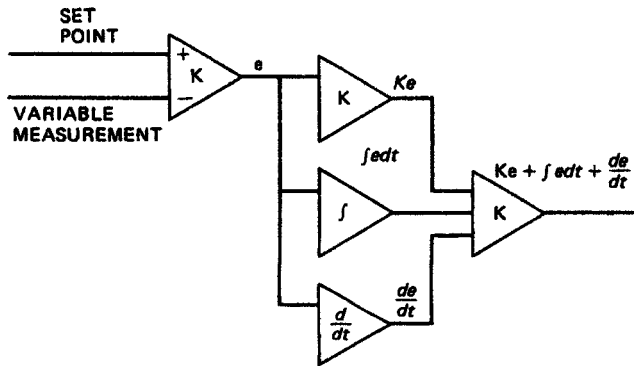

Figure 3-35. Ideal OP amp PID controller.

All the circuits illustrated are in simplified form, showing only the essential elements. In practice, additional circuitry is required to power the circuits, provide constant voltage for bridge circuits, provide sufficient output power, filter out high frequency noise, signals and provide other conveniences. Adjustable resistors and capacitors are needed to allow operators to adjust set points and proportional, integral, and derivative gains.

4

Fluidic Control Devices

4.1 INTRODUCTION

The first use of fluidic principles in control applications occurred about 1960. Their use in HVAC controls is somewhat limited. So-called self-powered control systems using duct air supply pressures are essentially fluidic in nature. Although they use lower pressures than pneumatic devices, they consume more air.

Fluidic devices use the dynamic properties of fluids, in contrast to pneumatic and hydraulic devices that depend primarily on static properties. Almost any gas or liquid can be used, but compressed air is the preferred medium. Logically, fluidic devices resemble electronic devices, and the same logic terms frequently are applied to both. The principles most commonly used in fluidic devices are wall attachment, turbulence amplification, and vortex amplification.

4.2 WALL ATTACHMENT DEVICES

A wall attachment device functions because of the Coanda effect: the property of a jet to attach itself to the surface of an adjacent plate or wall. A jet of air flowing from a nozzle entrains air with it (Figure 4-1). If the jet is confined between two parallel surfaces (assumed to be in the plane of the page), then the entrainment takes place only in that plane. If a plate or a wall surface is added at one side of the jet

(Figure 4-2), air entrainment on that side is reduced. This reduces the air pressure near the wall, and the jet is "bent" until it is attached to the wall. If a similar and symmetrical wall is provided on the other side of the jet, then it may become attached to either side with only a slight nudge. Although in theory the shape of the jet is not important, in practice a rectangular jet is found to work best.
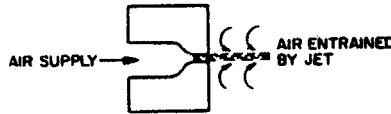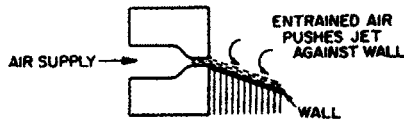

Figure 4-1. Elementary air jet.

Figure 4-2. Wall attachment principle.

This effect can be used to create a logic relay-amplifier with a memory (Figure 4-3). If air is supplied at control port Cl, the jet is deflected away from the port and attaches itself to the opposite wall, providing an output signal at 02. If the control signal is removed, the jet will continue to supply 02. If now a signal is applied at C2, the jet will switch to the opposite wall, and the output signal will appear at 01. Amplifications of about 3 or 4 to 1 (output to control signal) are possible. The vents are provided to allow continuing airflow if the output is blocked because interruption of airflow would cause the device to operate improperly. This device is called a bistable amplifier with memory.

A slightly different device is the OR-NOR amplifier of Figure 4-4. With no signal at C1 or C2 the output is at 02 (NOR). With a signal at C1 or C2 or both, the output is at 01 (OR). When the signal or signals are removed, the output returns to 02.

Figure 4-5 is similar to the bistable amplifier of Figure 4-3 but with the walls cut away just beyond the control ports. With no walls and no

control signal, the supply jet goes straight out and produces two equal outputs at O1 and O2. If a control signal is applied at C1 or C2 the jet is deflected in proportion to the strength of the signal, and the outputs at O1 and O2 become unequal. This is a "proportional beam deflection amplifier."
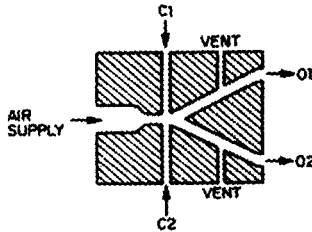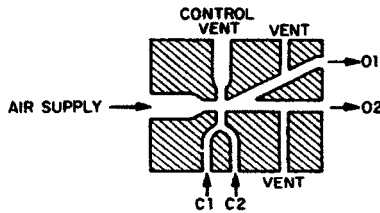

Figure 4-3. Bistable amplifier.

Figure 4-4. OR-NOR monostable amplifier.

All the above devices operate best at low pressures: 1 to 5 psig supply and 0.1 psig or less control. Output pressures are usually less than 1 psig.
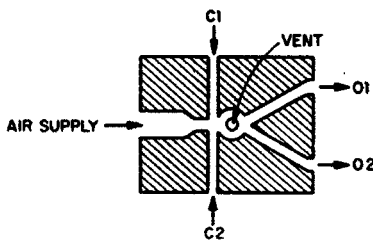

Figure 4-5. Proportional jet amplifier.

4.3 TURBULENCE AMPLIFIERS

A jet issuing freely from a nozzle can be adjusted to a fairly long laminar flow pattern (1 inch or more). A receiving nozzle in the path of

this pattern will then receive some flow and pressure (Figure 4-6). A small control jet directed at the laminar jet will cause it to break up, thus decreasing the pressure felt at the receiver. This change in output is proportional to the control pressure change, with considerable amplification (as much as 10:1). Also, more than one control jet can be used. The turbulence amplifier is a very low pressure device, typically using supply pressures of 10 inches of water and control pressures of 0.4 inches of water.
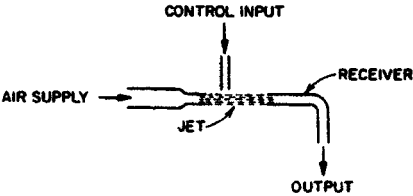

Figure 4-6. Turbulence amplifier.

4.4 VORTEX AMPLIFIERS

A vortex amplifier has a cylindrical body with supply air introduced at the side and output at one end (Figure 4-7). With no control flow, the air flows directly to the outlet, and flow is not restricted. When a control signal is applied at a tangent to the cylinder wall and at right angles to the supply flow, a swirling action is created, forming a vortex with high resistance and, therefore, lower flow. The control pressure must be higher than the supply flow, and supply output can be reduced, in a proportional manner, to as little as 10% of supply input. The device is most often used as a valve or variable restrictor.
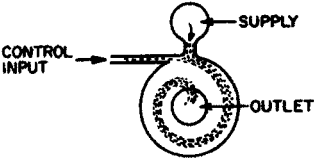

Figure 4-7. Vortex amplifier.

4.5 RADIAL JET AMPLIFIER

A variation of the turbulence amplifier is the radial jet amplifier (Figure 4-8). Two jets are arranged to oppose one another. At the point where they meet a radial pressure area is created, and when this is confined in a chamber an output signal can be produced. The diagram shows a reference jet (B) that can be adjusted to a desired pressure (set point), and a signal input jet (A) from the sensor. When A is less than B, the radial jet forms in the left-hand chamber and is vented to atmosphere. When A is greater than B, the pressure is developed in the right-hand chamber, and an output signal is generated that can power an actuator or reset a relay.
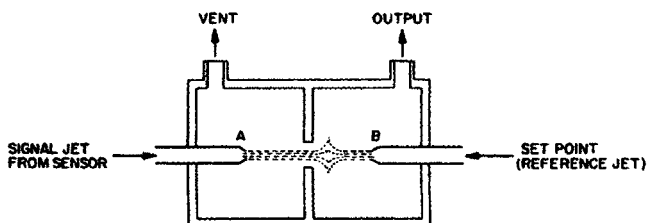

Figure 4-8. Radial jet amplifier.

4.6 FLUIDIC TRANSDUCERS

You will have noticed that fluidic control pressures are very small. To operate controlled devices such as valves or motors, it is necessary to provide transducers. Fluidic-to-pneumatic transducers are most common, as both use air, though at greatly different pressures. All fluidic transducers operate on one of two basic principles: direct force or assisted direct force.

In Figure 4-9 the fluidic signal operates against a diaphragm to deflect it. This direct force action could close an electrical switch or move a pilot valve in a pneumatic relay. Even with a large diaphragm the available force is small.
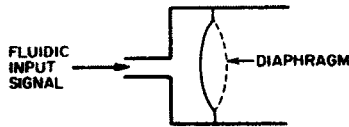
FLUIDIC
INPUT ────► ◄── DIAPHRAGM
SIGNAL

Figure 4-9. Direct force transducer

In Figure 4-10 a high-pressure air supply is provided to amplify the fluidic signal. With a low fluidic signal the high-pressure air is vented. When the fluidic signal is increased, diaphragm 1 moves to restrict or close the vent nozzle, increasing the pressure on diaphragm 2. This higher pressure is adequate for positioning valves or dampers directly. The action can be proportional or two-position, depending on the construction of the diaphragm.

HIGH-PRESSURE
AIR SUPPLY

FLUIDIC
INPUT ────► ◄── OUTPUT
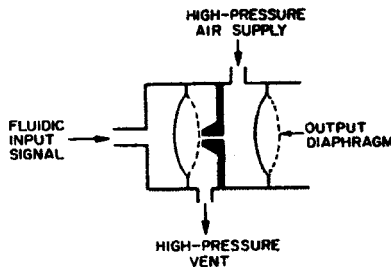SIGNAL   DIAPHRAGM

HIGH-PRESSURE
VENT

Figure 4-10. Assisted force transducer.

Figure 4-11 shows a three-way two-position pneumatic valve, driven both ways by two assisted fluidic signals operating on a piston. This also could be constructed to use a single signal with spring return. Such a valve has a very fast response time.

Fluidic-electronic transducers are also available. These devices generally use strain gages or pressure-sensitive transistors to modify the low-power electronic signal. They are not generally used in HVAC systems at this time.

4.7 MANUAL SWITCHES

A simple fluidic pushbutton switch operates on the bleed principle (Figure 4-12). The air supply is vented through an orifice and chamber under the button. When the button is depressed, the vent is closed, and the output signal increases.
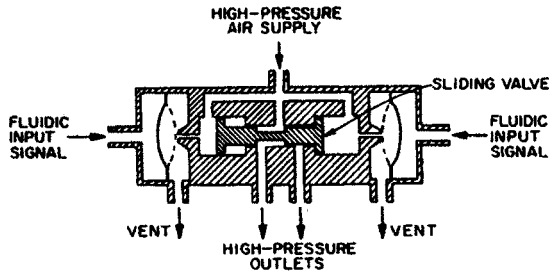
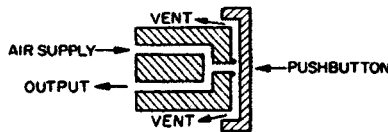Figure 4-11. Three-way valve (with assisted force operation).

Figure 4-12. Fluidic pushbutton.

Figure 4-13 shows a selector switch that provides a choice of two outputs. This can be built in a rotary or a slide arrangement, with a choice of many outlets or combinations. Any of the manual or automatic air valves commonly used for pneumatic systems also can be used with fluidic controls.
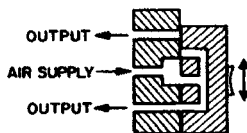
Figure 4-13. Selector switch.

5

Flow Control Devices

5.1 DAMPERS

Dampers are used for the control of airflow to maintain temperatures and/or pressures. Some special types of dampers are used in HVAC equipment, such as mixing boxes and induction units, but the control dampers in air ducts and plenums are almost invariably the multi-leaf type. Two arrangements are available: parallel-blade and opposed-blade. As Figure 5-1 shows, in parallel-blade operation all the blades move in the same (or a parallel) way. In opposed-blade operation adjacent blades move in opposite directions.
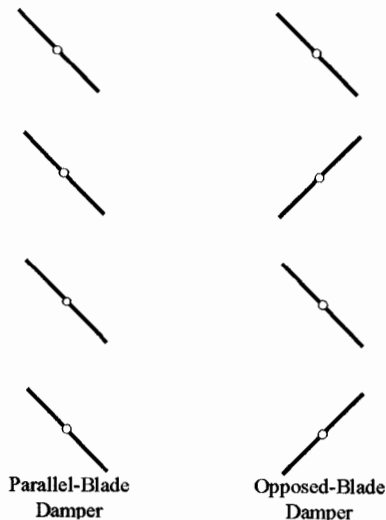

Parallel-Blade
Damper

Opposed-Blade
Damper

Figure 5-1 Airflow Dampers

5.1.1 Pressure Drop

The pressure drop through dampers can be characterized by the dimensionless loss coefficient, C_d, which is the ratio of the total pressure loss through the damper to the local velocity pressure:

$$C_d = \frac{\Delta p_t}{p_v} = \frac{\Delta p_t}{\rho (V/1097)^2} \qquad (5\text{-}1)$$

where

$\quad C_d$ = loss coefficient, dimensionless

$\quad \Delta p_t$ = total pressure loss, inches of water

$\quad V$ = velocity, feet per minute

$\quad p_v$ = velocity pressure, inches of water

$\quad \rho$ = density of air, pounds per cubic foot

The loss coefficient is a function of the blade angle and the damper type. Data for opposed and parallel blade dampers can be found in older editions of the ASHRAE Handbook of Fundamentals (ASHRAE, 1989). Part of the data is shown in Figure 5-2. Over most of its operating range, the loss coefficient is an exponential function of blade angle for both parallel and opposed blade dampers (Legg, 1981). That is:

$$C_d = k_1 e^{k_2 \theta} \qquad (5\text{-}2)$$

where k_1 and k_2 are constants, and θ is the blade angle in degrees.

Notice that the loss coefficient increases more slowly as the damper closes for parallel-blade dampers than for opposed blade-dampers - parallel blades act more like turning vanes whereas opposed blades act more like orifices. This characteristic influences the way in which a damper modulates air flow in a damper/duct system. It can be shown that for a constant pressure difference, the flow through a duct and damper system varies according to the following:
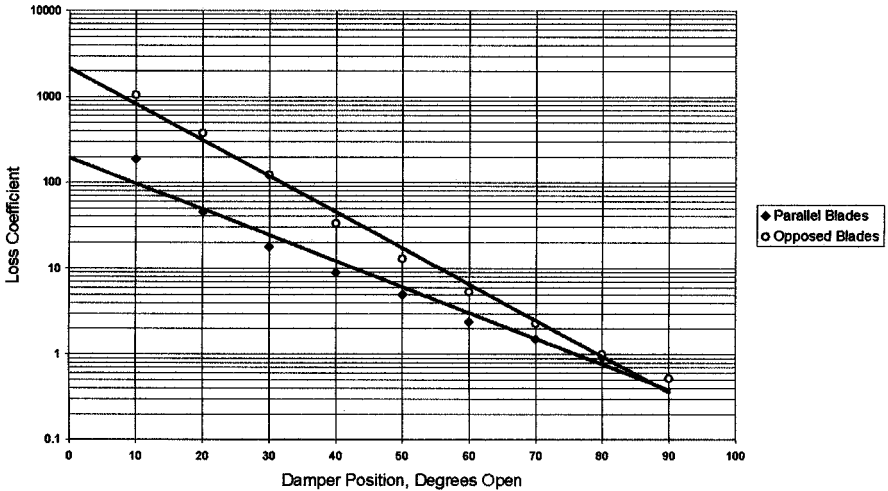
Figure 5-2. Damper flow characteristics.

$$\text{Fraction of full flow} = \sqrt{\frac{C_{do}}{f \cdot C_d + (1 - f) \cdot C_{do}}} \qquad (5 - 3)$$

where:

C_{do} = loss coefficient for the wide open damper

C_d = loss coefficient, a function of blade angle

f = open damper resistance as a fraction of

the total system resistance

Figures 5-3 and 5-4 show fraction of full flow versus blade angle for parallel and opposed blade dampers for various values of ∂ (∂ is expressed in percent of total system resistance). Notice that for a typical parallel blade damper to achieve nearly linear modulating control it is necessary that the pressure drop through the damper in the full open position be from 20% to 50% of the total system pressure. To get almost equivalent linearity with a typical opposed-blade damper, the wide open pressure drop through the damper need be only about 5% to 10% of the system pressure drop.
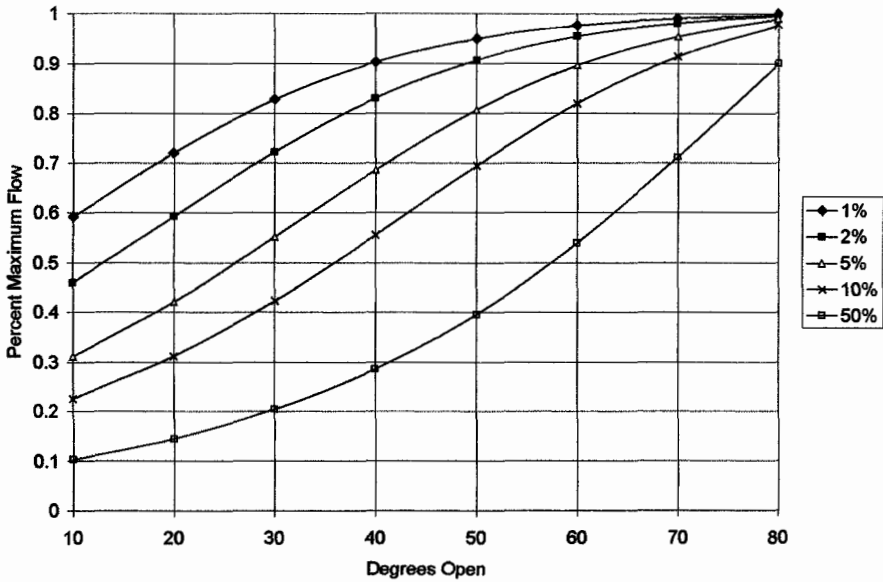
Figure 5-3. Parallel-blade damper characteristics.

As most of the energy loss creates noise, the opposed-blade damper is preferable for modulating service from both noise and energy loss standpoints.

In most applications, the velocity through the wide open damper must be fairly high in order for the damper to be a significant portion of the system pressure loss. For example, if the total loss at full flow is one inch of water gage and the wide open loss coefficient is about 0.4, then the velocity through the damper must be about 2000 feet per minute (fpm) if the damper pressure drop is to be 10% of the total. By comparison, many building cooling systems have coil face velocities of approximately 500 fpm. This means that dampers in outdoor or return ducts probably should be less than one-fourth of the area of the coil.
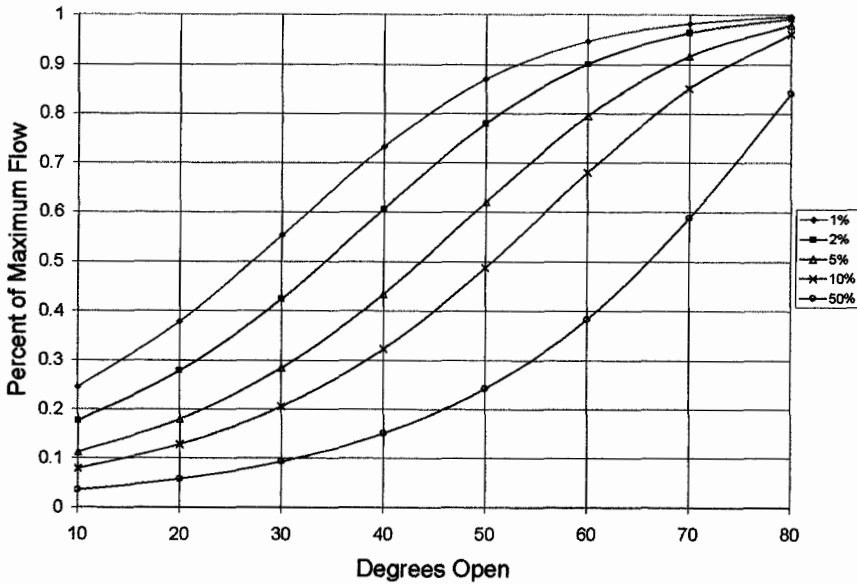
Figure 5-4 Opposed blade damper characteristics.

5.1.2 Leakage

Dampers are inherently leaky, the amount of leakage at close-off being a function of the damper design. Minimizing leakage increases cost; so system requirements should be the basis for selecting a damper. Most manufacturers publish leakage ratings. A simple inexpensive damper may have a leakage of 50 cfm per square foot of face area at 1.5 inches of water pressure, whereas a very carefully designed and constructed damper may leak as little as 10 cfm per square foot at 4 inches of water pressure. Damper leakage can become a serious problem in some applications. For example, outside air dampers must have minimal leakage to prevent equipment damage due to freezing air.

5.1.3 Operators

Damper motors may be pneumatic, electric or hydraulic. They must
have adequate power to overcome bearing friction and air resistance,
and for tight-fitting low-leakage dampers, to overcome the binding
friction of the fully closed damper.

5.1.4 Face and Bypass Dampers

Face and bypass dampers frequently are used with preheat coils, and
less often with direct-expansion cooling coils. Figure 5-5 shows a
typical preheat installation. The preheat coil is sized with a face velocity
based on the manufacturer's recommendations, and with all the air
flowing through the coil. The open face damper will have this face
velocity. The face damper and coil pressure drops then can be
determined from manufacturer's information (see also Brown, 1960).
The bypass damper should be sized so that its wide-open pressure drop
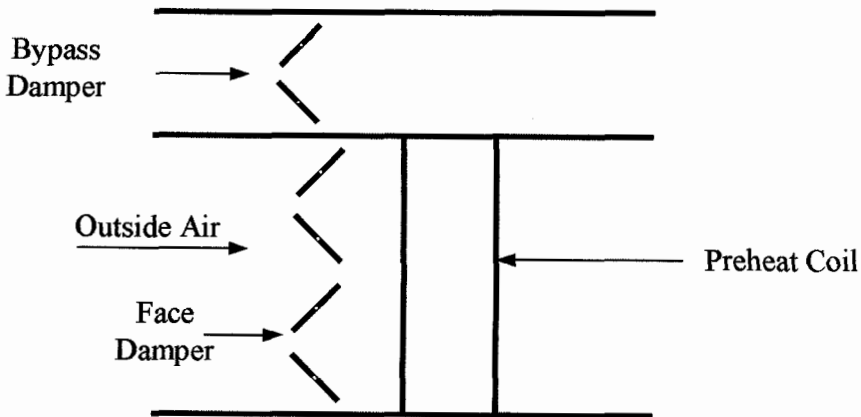is equal to the sum of the face damper and coil pressure drops.

Figure 5-5. Face and bypass dampers.

5.2 STEAM AND WATER FLOW CONTROL VALVES

The proper selection of valves for the control of steam and water flow requires an understanding of both the valve characteristics and the system in which the valve will be used. This section discusses valve types and sizing methods.

The size of a heat exchanger or a coil and the fluid flow rate through it must be based on some maximum design load. But, in practice, the equipment usually operates at part load; so the valve must control satisfactorily over the whole range of load conditions.

5.2.1 The Flow Coefficient

Most manufacturers publish valve capacity tables based on the flow coefficient, C_v. In inch-pound units, this is the flow rate in gallons of 60°F water that will pass through the valve in one minute at a one-pound pressure drop. The flow rate at pressure drops other than one pound is found by the formula:

$$GPM = C_v \sqrt{\Delta P} \tag{5-4}$$

where ΔP is the pressure drop across the valve.

Valve capacity tables usually show C_v and then flow rate at various pressure drops.

The rated C_v is established with the valve fully open. As the valve partially closes to some intermediate position the C_v will decrease. The rate at which it decreases determines the shape of the curve in Figure 5-6. Some industrial valve manufacturers publish C_v data for partially open valves.

The change in pressure drop and flow in relation to stroke, lift or travel of the valve stem is a function of the valve plug design. Different types of valve plugs are required to fit different control methods and fluids.
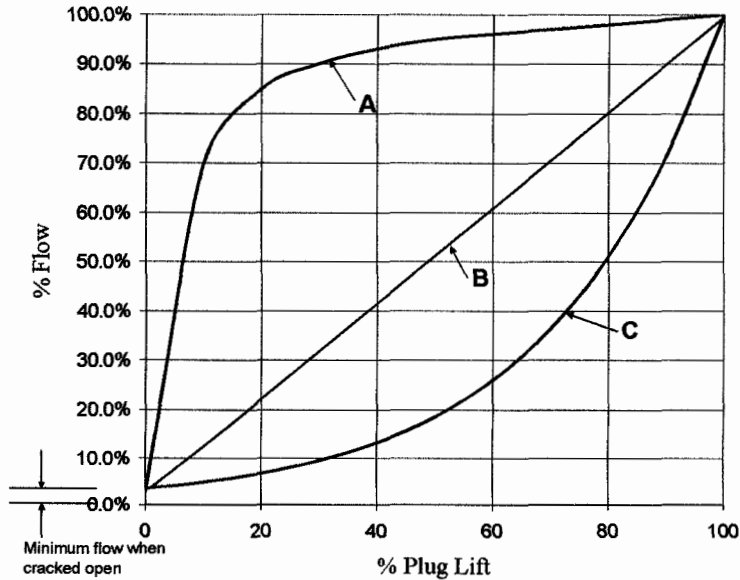
Figure 5-6. Valve characteristics.

5.2.2 Two-Position Valves

These valves should be of the flat-seat or quick-opening type shown in Figure 5-7. The accompanying graph of percent flow versus percent lift (Figure 5-6, curve A) shows that nearly full flow occurs at about 20% lift. Two-position valves should be selected for a pressure drop of 0% to 20% of the piping system pressure differential, leaving the other 80% to 90% for the heat exchanger and its piping connections.

Figure 5-7. Quick opening valve.

5.2.3 Modulating Valves

The position of a modulating valve is usually what is adjusted by the control system to control the controlled variable. If the valve is designed with a linear characteristic, as in Figure 5-8 and curve B in Figure 5-6, then the flow rate will vary directly with lift (or nearly so) *for a constant pressure drop across the valve.* By varying the angle of this V-port arrangement, curves above and below the linear relation can be obtained.
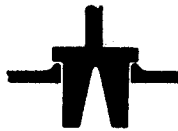

Figure 5-8. Linear or V-port valve.

There is a minimum flow that is obtained immediately upon cracking the valve open. This is due to the clearances required to prevent sticking of the valve. This minimum flow is generally about 3% to 5% of maximum flow and is the turndown ratio of the valve. At 5% the turndown ratio would be 100:5 or 20:1 This is a typical ratio for "commercial"-quality valves. "Industrial" valves with turndown ratios of 50 or 100 to 1 or even higher are available. Higher ratios are needed only for very close control and are not justified in most HVAC applications.

A linear characteristic valve is excellent for proportional control of steam flow because the heat output of a steam heat exchanger is directly proportional to the steam flow rate. This is so because the steam is always at the same temperature, and the latent heat of condensation varies only slightly with pressure change.

Hot water coils, however, create a different requirement, as reduction in flow will be accompanied by an increase in the temperature change of the water. The net result may be only a small reduction in heat

exchange for a large reduction in flow. Figure 5-9 shows fractional capacity versus water flow for a typical hot water coil with 160°F entering water and 60°F entering air and a design water temperature drop of 35°F at full load. The air volume flow rate is assumed to be constant.   This shows that a 50% reduction in water flow will cause only about a 20% reduction in heat output, with an increase in the water temperature drop to nearly 60°F. To cut the capacity by 50% the water flow must be reduced by 80%.   The data were generated by using a manufacturer's heating coil selection program.

The gain of the coil is the ratio of the change in air temperature rise to the change in water flow rate.   This is the derivative or slope of the curve of Figure 5-9. For this coil and the specified entering water and air temperature and for a fixed air flow rate, the gain at 5% water flow is more than ten times the gain at 95% water flow.
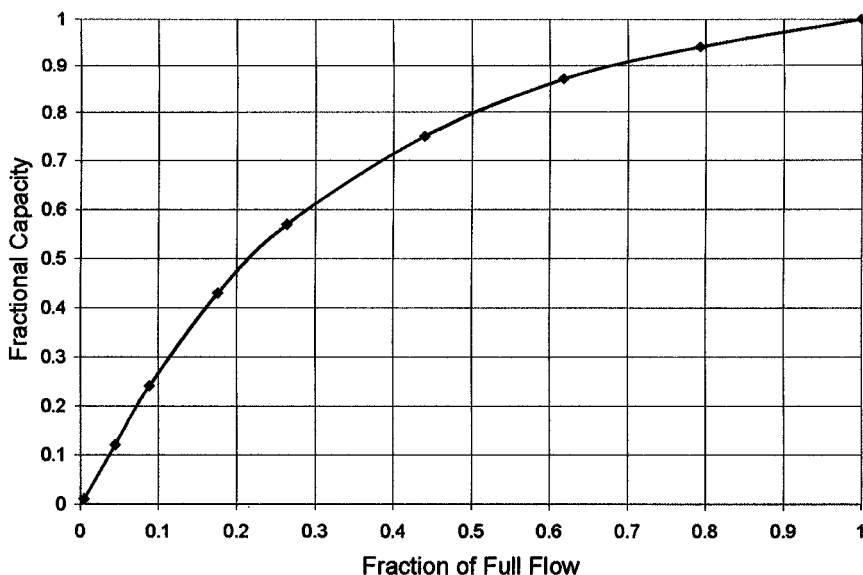

Figure 5-9. Flow vs. heating capacity for a heating coil.

To get a better relationship between valve position and heat output for this case, an equal percentage valve is used (curve C of Figure 5-6). The equal percentage valve has a plug shaped so that a percentage

change in valve position produces a corresponding percentage change in flow (see Figure 5-10). For example, for a fixed pressure drop across the valve, a valve might produce a 40% change in flow for a 10% change in position.

Figure 5-10. Equal percentage valve.

It can be shown that a valve will have an equal percentage characteristic if the valve coefficient varies with valve position according to the following:

$$\frac{C_v}{C_{v\,max}} = e^{k(1-X)}$$

(5-5)

where :

C_{vmax} = the wide open valve coefficien t

X = the valve position, 1 is open, 0 is closed

k = a constant

For example, for a valve with a 40% change in flow for a 10% change in position, k is equal to -3.3648.

It turns out that the above valve works well with the coil shown above to produce a nearly linear relationship between heat output and valve position. Figure 5-11 shows percent of capacity for the coil and percent of flow for the valve versus valve position. This figure shows how valve characteristics can be tailored to the application but we have assumed that the most of the pressure drop is due to the valve. In the next section we will discuss the impact of authority, the ratio of valve pressure drop to total system pressure drop.
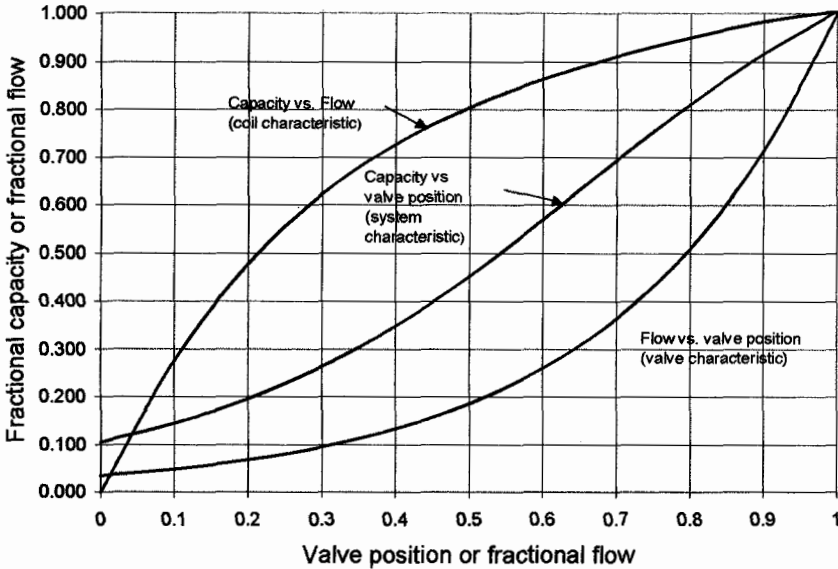
Figure 5-11. Matching heating coil and valve characteristics.

5.2.4 Valve Pressure Drop

To figure out the required pressure drop for a modulating water valve at full design load it is necessary to look at the entire system. Three conditions are possible:

1. Throttling or closing off any individual control valve has little or no effect on the pressure differential from supply main to return main.

2. There is only one control valve, so that throttling or closing off this valve changes the flow correspondingly in the entire piping system.

3. Other conditions between these two extremes.

In the first case the heating coil or heat exchanger and related piping between supply and return mains can be considered as a subsystem

(Figure 5-12(A)). First consider the subsystem without a control valve. Now if somehow the pressure differential between supply to return mains is varied, the flow rate in the subsystem will vary in proportion to the square root of the pressure drop, as shown in Figure 5-12(B). But a constant differential pressure between supply and return mains has been assumed, with a control valve as shown in Figure 5-13(A). Depending on the valve characteristic, most of the pressure drop must be taken by the valve to provide adequate control at all conditions.
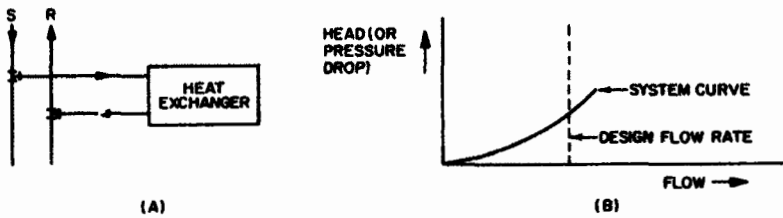
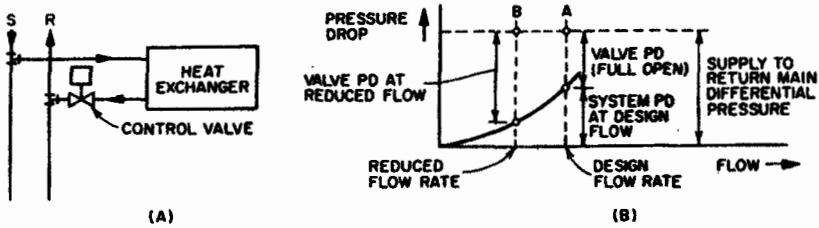Figure 5-12. A heat exchanger subsystem.

Figure 5-13. Subsystem with control valve

It can be shown that the flow rate through a valve and coil system where the supply to return main pressure drop is constant can be found from the following equation:

$$\frac{Q}{Q_{max}} = \sqrt{\frac{1 + \left(\dfrac{\Delta P_{valve}}{\Delta P_{total} - \Delta P_{valve}}\right)}{1 + \left(\dfrac{\Delta P_{valve}}{\Delta P_{total} - \Delta P_{valve}}\right)\left(\dfrac{C_{v\,max}}{C_v}\right)^2}} \qquad (5\text{-}6)$$

where:

$$\frac{Q}{Q_{max}} = \text{fraction of full flow through the system}$$

$$\left(\frac{\Delta P_{valve}}{\Delta P_{total} - \Delta P_{valve}}\right) = \text{ratio of the pressure drop through the valve to the}$$

$$\text{pressure drop through the rest of the system}$$

$$\left(\frac{C_{v\,max}}{C_v}\right) = \text{ratio of the wide open valve coefficient to the partially}$$

$$\text{open valve coefficient}$$

Figure 5-14 shows how valve authority influences flow for a linear valve. Such a valve might be used on a steam-to-hot-water converter where capacity varies almost linearly with steam flow. Notice that for the total system to behave more or less linearly, the pressure drop through the wide open valve should be about half the total system pressure drop. If the valve represents only 10% of the pressure drop, the overall system response is more quick opening than linear.
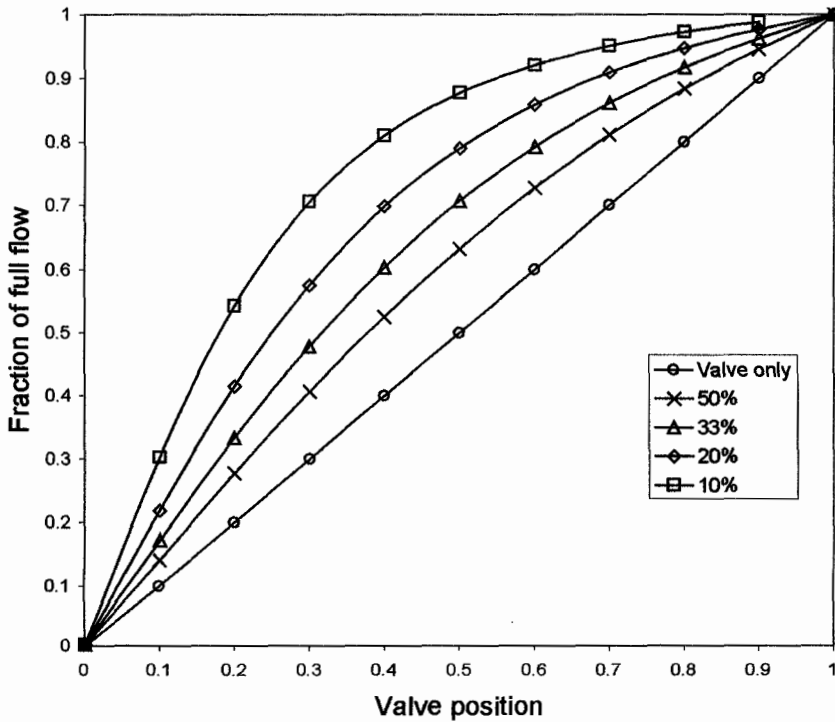
Figure 5-14. Flow vs. valve position for different valve authority, linear valve.

Figure 5-15 shows a similar plot for an equal percentage valve like the one described above, for use with a hot water heating coil. As with the linear valve, equal percentage behavior cannot be maintained unless the valve represents a significant proportion of the total pressure drop.

Figure 5-16 shows fractional capacity versus valve position for a heating coil and valve system. It is similar to Figure 5-11 but includes the effect of valve authority. This figure illustrates the importance of valve authority on system performance. Oversizing a control valve will lead to poor, even oscilatory control.
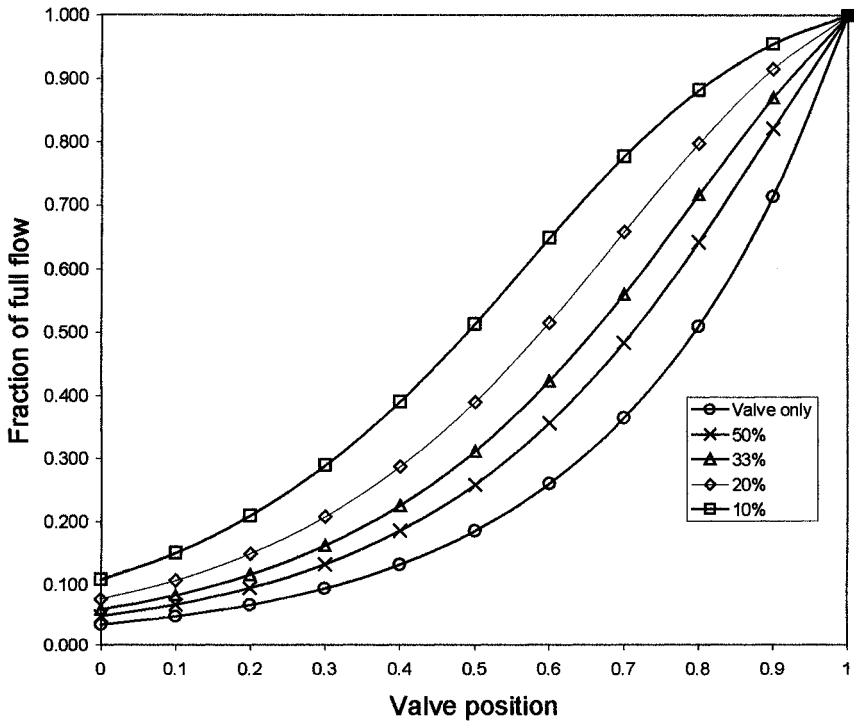
Figure 5-15. Flow vs. valve position for different valve authority, equal percentage valve.
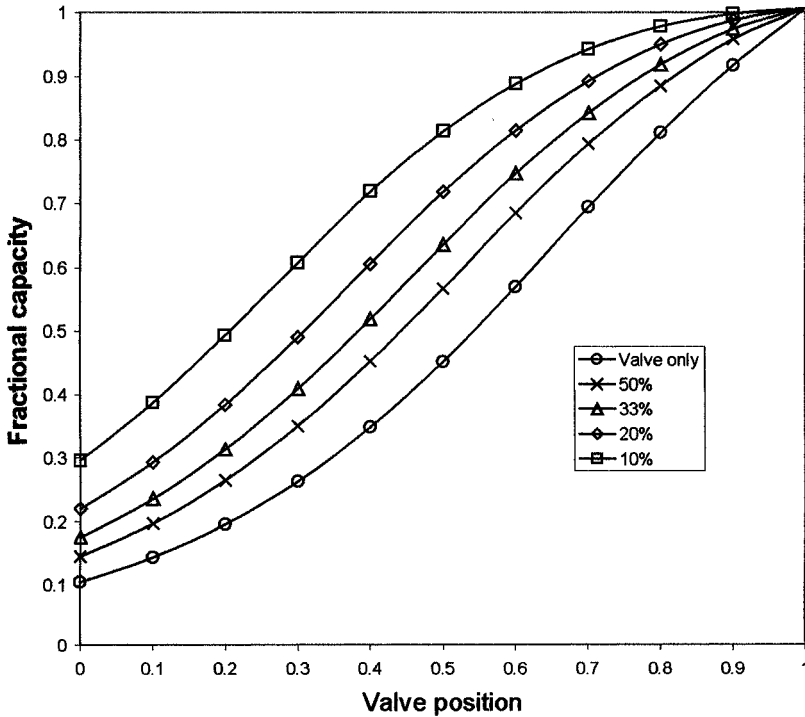
Figure 5-16. Heating capacity vs valve position for different valve authority.

In the case with only one control valve our system would look schematically like Figure 5-17. For this system, flow through the coil and valve is again related to the square root of the pressure drop as in Figure 5-12(B). A typical pump performance curve of head versus flow will look like Figure 5-18. With the control valve full open these two curves will establish some operating flow rate A, as in Figure 5-19. As the control valve modulates toward the closed position, flow decreases to B. However, the pump head must increase as flow decreases. This means simply that the control valve pressure drop has had to increase more for the same flow reduction than in the first case we considered. That is, the valve will need a different plug characteristic to obtain the same flow reduction as in the first case.
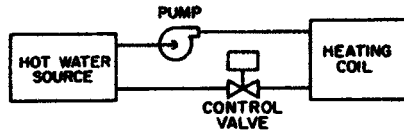
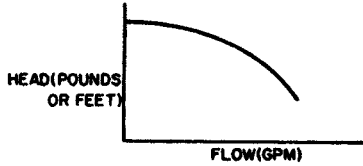Figure 5-17. A simple system with one control valve.

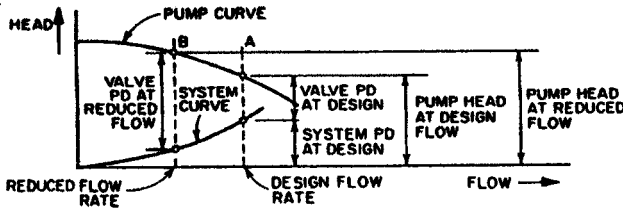Figure 5-18. A typical pump performance curve of head versus flow.

Figure 5-19. Simple system at design and reduced flow rates.

Cases that fall between these two extremes result from small systems with only a few control valves or from systems in which a few control valves handle most of the water flow. Then any one of these control valves will have a noticeable but not completely governing effect on the pump pressure and flow.

These last two cases can be avoided by using three-way valves or pressure-controlled bypass valves, which will maintain the total flow rate (and therefore the supply to return main differential pressure) essentially constant. The use of three-way valves should be minimized in the interest of energy conservation, as noted in discussions in Chapters 6, 7, and 12.

5.2.5 Shutoff Head and Static Head

Any flow control valve will have a flow direction symbol on the outside of the body. For a modulating valve the direction of flow should always be such that the flow and pressure tend to hold the valve open (Figure 5-20). If the flow and the pressure are arranged to help close the valve, the valve tends to slam shut and oscillate. This produces chatter, which is usually a result of installing the valve backward.
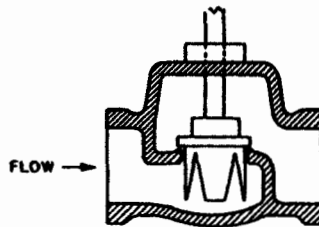

Figure 5-20. V-port valve.

When the valve is properly installed, the closing force must overcome both velocity and static pressure. When the sum of these two pressures reaches a maximum, it is known as the close-off pressure and is one criterion for valve selection.

The static pressure rating of a valve is the highest pressure the valve body will stand without damage or leaking.

5.2.6 Three-Way Valves

Three-way valves are generally used to provide roughly constant flow rate through the piping system while varying the flow rate through the heat exchanger. They also may be used to provide constant flow through the heat exchanger while the primary flow rate varies.

The heat exchanger in this context may include an air-to-fluid heat exchange coil, a chiller, a heater, a condenser, a cooling tower or a fluid-to-fluid heat exchanger.

Three-way valves are made in two types: mixing and diverting. The internal designs are different (see Figures 5-21 and 5-22), the difference being necessary so that the valve will seat against flow, as discussed in the section on shutoff head above. The mixing valve uses a typical linear V-port plug with an added taper on top to seat in the second inlet port. The diverting valve uses two V-port plugs that seat in opposite directions and against the common inlet flow. Using either design for the wrong service would tend to cause chatter.
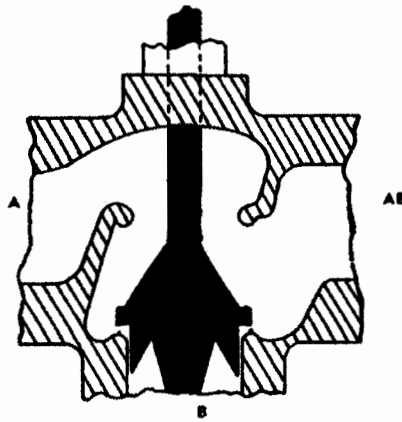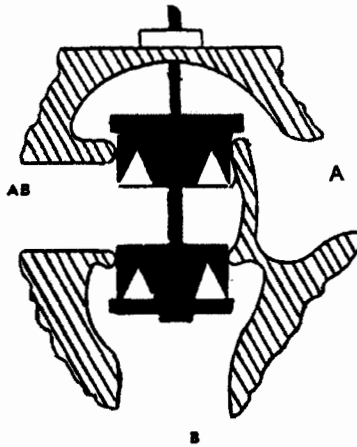
Figure 5-21. Three-way mixing valve.

Figure 5-22. Three-way diverting valve.

Although the valve connections will be marked A, B, and AB, for logic purposes they are usually designated normally closed, normally open and common. Either the A or the B connection may become normally open, depending on the arrangement of the motor operator. Typically, however, the B connection is normally open.

5.2.7 Valve Operators

Valve operators may be pneumatic, electric or electronic motors. Such motors have been described in Chapters 2 and 3. Valves also may be self-contained; that is, the operator-motor will derive its power from the expansive force of a thermally sensitive fluid in a bulb and capillary tube arrangement (Figure 5-23). Another operator, which is technically electronic, is actually hydraulic, and contains an oil reservoir and a tiny pump that pumps oil from one side of a diaphragm to the other.

5.2.8 Criteria for Valve Selection

Flow control valves should be carefully selected to match the characteristics of the system that they are to control. Necessary criteria include flow rate, pressure drop, close-off pressure, static pressure, and type of action. The type of operator used should be compatible with the rest of the control system. Close-off pressure is most often overlooked, especially with large valves (3 inch and larger). It often is necessary to use oversize operators or higher operating air pressures to provide adequate close-off pressures.
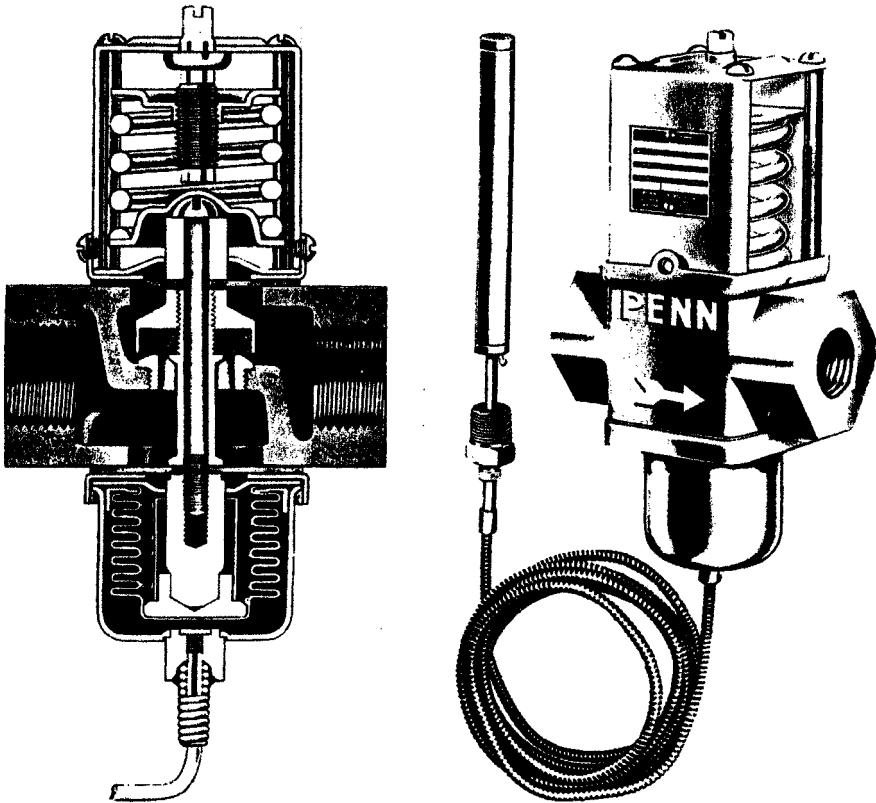
Figure 5-23. Self-contained valve. (Courtesy Penn Controls, Inc.)

6

Elementary Control Systems

6.1 INTRODUCTION

The preceding chapters discussed the various control elements or units, and how they function. The next four chapters will consider the application of these units to form combinations or "control systems" for the control of HVAC. This chapter will consider only small segments of the larger complete systems. Each of these is a complete control system, and large systems are made up of combinations of these subsystems.

The following discussions will deal with function only, and, except in a few cases, assume that the function can be accomplished by electric, pneumatic, electronic, or digital control hardware.

6.2 OUTSIDE AIR CONTROLS

Before deciding how to control the amount of outside air it is necessary to figure out how much is required by the HVAC system and why. For example, certain areas such as laboratories and special manufacturing processes may require 100% exhaust and make up. Clean rooms require that a positive internal pressure be maintained to prevent infiltration from surrounding areas, whereas spaces such as chemical

labs and plating shops require a negative pressure to prevent exfiltration.

When there are no special requirements, the minimum amount of outside air required is that needed to meet the code requirements for ventilation rates. See ASHRAE Standard 62-2001, Ventilation for Acceptable Indoor Air Quality. Typically 15 cfm per person is required, though more may be necessary in certain applications and where smoking is allowed. Where the outdoor air quality is suitable (not too humid or dirty), "economy cycle" control often is used, as described below.

Once the criteria have been determined, one of the following methods of control can be selected.

6.2.1 Minimum Outside Air

The simplest method of outdoor air control is to open a "minimum outside air" damper whenever the supply fan is running (Figure 6-1). This provides the air required for ventilation or exhaust makeup but does not allow more than the minimum outside air to be introduced, even when use of cool outside air could reduce the need for mechanical refrigeration
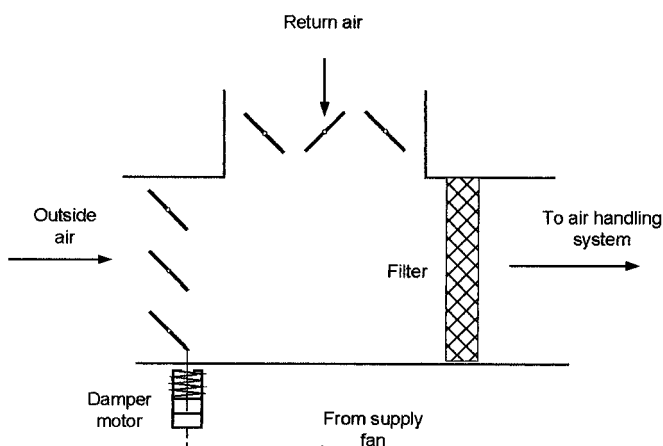

Figure 6-1. Outside air; two-position.

6.2.2 Economy Cycle Outside Air

When fixed amounts of outside air are used, there may be times when it is necessary to operate the cooling coil even when outdoor air temperatures are cool. This need gave rise to the so-called economy cycle (Figure 6-2), with outside, return, and relief dampers controlled by air temperature. When the outside air is at the design winter temperature, the outside air damper and relief dampers are usually in the minimum open positions (as determined by ventilation and exhaust requirements), and the return air damper is wide open. As outside air temperature increases, the mixed air thermostat (T1) gradually opens the outside air damper to maintain a constant mixed air temperature. Return and relief dampers modulate correspondingly. At some outside temperature, usually between 50°F and 60°F, 100% outside air will be provided and used for cooling. As the outside air temperature continues to increase, at 70°F to 75°F an outdoor air thermostat (T2) is used to cut the system back to minimum outside air, thus decreasing the cooling load. An interlock from the supply fan is provided in most outside air control systems so that the outside air damper will close when the fan is off. In the schematic, a solenoid relay interupts the air supply to the pneumatic damper actuators.

Figures 6-3 and 6-4 show two other methods of economy cycle control. Figure 6-3 shows two thermostat/controllers, one direct-acting and one reverse-acting, controlling the dampers. T1 modulates the outside and return air dampers to maintain the desired mixed air temperature until an upper bound is reached and T2 begins to reduce air pressure to close the outside air and relief dampers and open the return air damper. If a minimum amount of outside air is required, the damper linkage is set up to provide this amount even when the damper actuator is in its normally closed position. This scheme only works where there is no danger of freezing.

Figure 6-2. Outside air; economy cycle, adjustable minimum.

Figure 6-3. Outside air; economy cycle, no minimum.

 Figure 6-4 is shows another arrangement, with a fixed minimum outside air damper that opens whenever the fan runs and is not affected by the temperature controls. The rest of the system operates as described above.

The above discussion supposes a fixed mixed air temperature set point. It can be shown that this may not be the best approach for energy conservation because it can increase heating requirements as compared to a fixed minimum outside air. To solve this problem it is necessary to reset the controller set point as a function of the building heating and cooling load. Some examples are shown in Chapter 7.

Figure 6-4. Outside air; economy cycle, fixed minimum.

6.2.3 Enthalpy Control

In theory, outside air "economy cycle" control based on dry bulb temperatures is not always the most economical approach. In very humid climates the total heat (or enthalpy) of the outside air may be greater than that of the return air even though the dry bulb temperature

is lower. For example, in Figure 6-5, the psychrometric chart shows outside air nearly saturated at 73°F dry bulb (DB) whereas return air at 80°F DB though much drier, has a lower enthalpy. Since the cooling coil usually must remove the total heat from the air to maintain the desired condition, it is more economical in this case to hold outside air to a minimum.
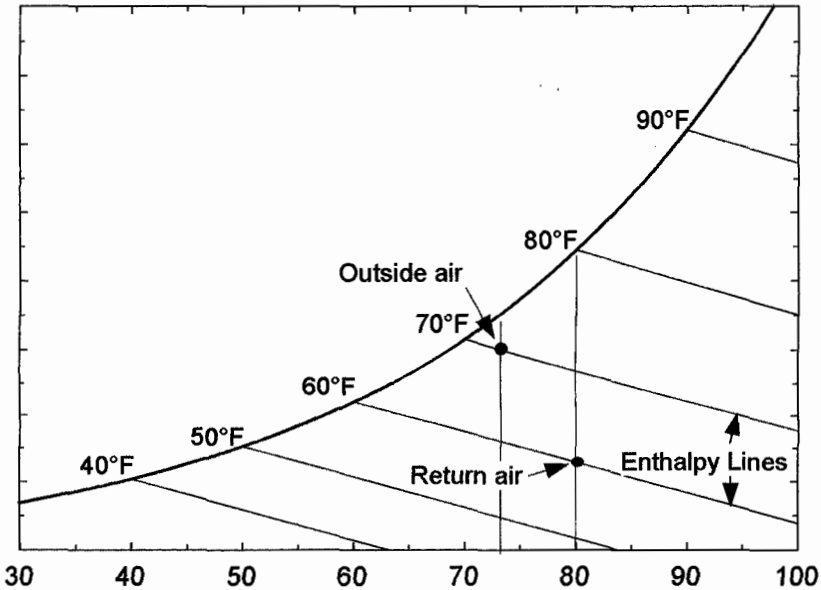

Figure 6-5. Psychrometric chart

To measure enthalpy it is necessary to sense dry bulb temperature and either wet bulb temperature, relative humidity, or dew point. Several manufacturers now have instruments that simultaneously sense dry bulb and dew point, figure out enthalpy for return and outside air, and provide an output to control the dampers. (See Figure 6-6.)

Although enthalpy control has some potential benefits, the energy savings when compared to a temperature-based economy cycle are small (Spitler et. al., 1989). An enthalpy economy cycle is also difficult to implement. The accuracy of commercial humidity sensors is difficult to maintain without frequent calibration and the accurate calculation of enthalpy is usually limited to modern digital control hardware. It often

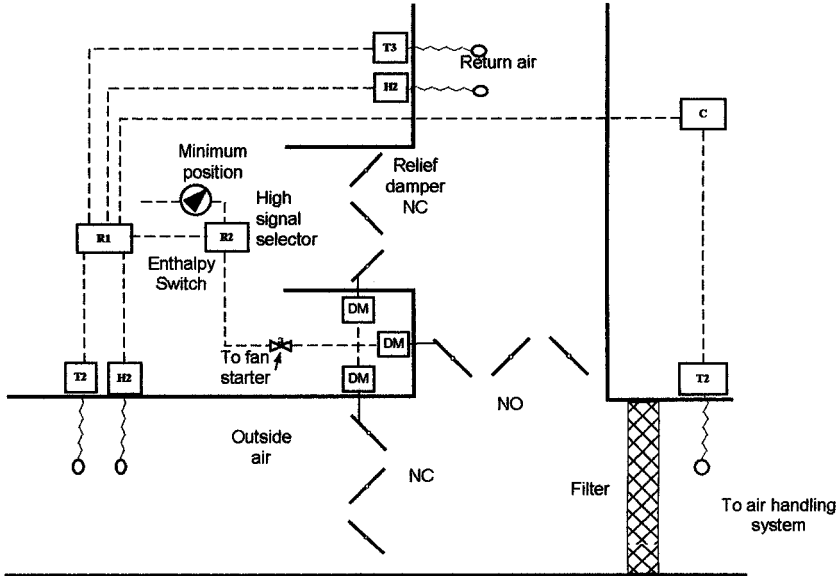is difficult to justify the additional complexity and cost of enthalpy control.

Figure 6-6. Outside air; enthalpy economy cycle.

6.2.4 Static Pressure Control

For those spaces requiring a constant positive or negative pressure with respect to their surroundings, the outside, return, and relief air dampers will be controlled by static pressure controllers. This will usually be part of a larger system as described later in this chapter. In its simplest form (Figure 6-7), the static pressure controller senses the difference in pressure between the controlled space and a reference location (either next to the controlled space or outdoors) and adjusts the dampers to maintain that pressure differential. The amount of outside air provided must be sufficient to make up any exhaust and to pressurize the space. Proportional plus integral controls are required because low proportional gain is needed to prevent instability due to pressure surges that occur when doors are opened. Other pressure balance control systems are described in Chapter 9.
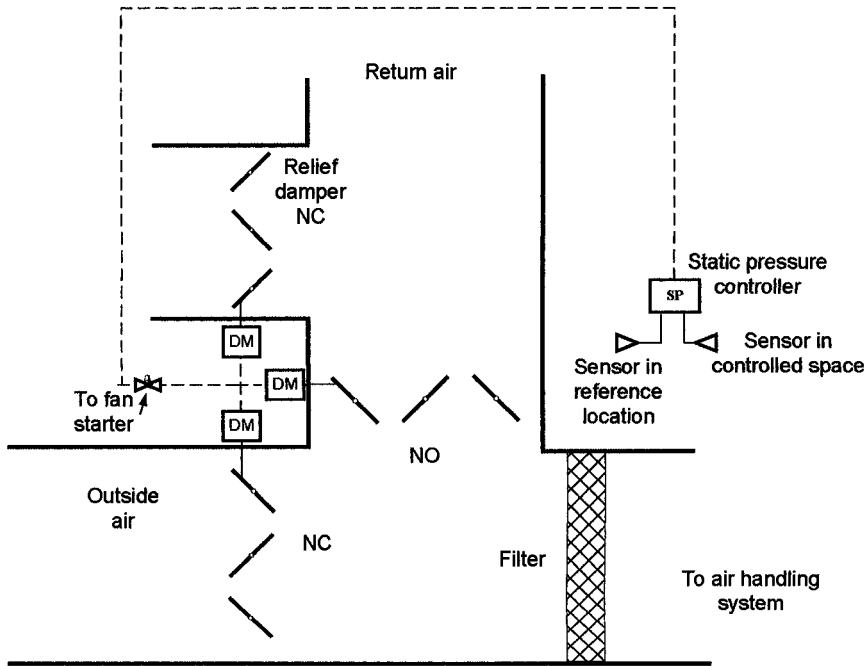
Figure 6-7. Outside air; static pressure control.

6.2.5 Outside Air and Variable Air Volume Systems

With the prumulgation of ANSI/ASHRAE Standard 62-2001, Ventilation for Acceptable Indoor Air Quality, renewed interest has been placed on providing minimum required ventilation air when using variable air volume (VAV) systems. The scheme shown in Figure 6-4 will provide minimum outside air for constant volume systems. It fixes a minimum position for the outside air damper that is adjusted during comissioning to provide the required air flow. However, when used in a VAV system, it provides a minimum *fraction* of flow not a miminum absolute amount. Several approaches to insure that the minimum amount of outside air is maintained under varying total air volume flow rates have been proposed.

One scheme is show in Figure 6-8. The scheme is similar to that shown in Figure 6-4 except that the return damper is controlled to maintain a constant negative pressure in the mixing box. With the minimum outside air damper open, this insures that the minimum fresh air amount will be achieved.
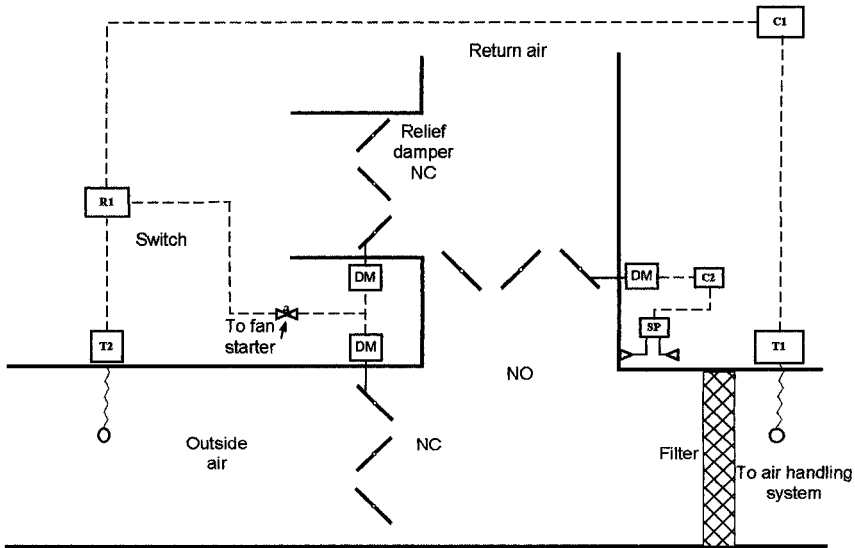

Figure 6-8. Return damper controls mixing box pressure.

Another approach, shown in Figure 6-9, is to measure the outside air flow and use the output of the flow measuring station in place of a miniimum position setting.

6.3 AIR STRATIFICATION

Stratification of return air and outside air streams in "mixing" plenums can be a serious problem. In a worst case, such as in Figure 6-10, the two air streams will not mix and will remain separate for a long distance (through filters, coils, and even centrifugal fans, for example). If the outside air temperature is below freezing, the separate air stream can cause localized freezing in heating and cooling coils or trip low

temperature safety controls. Even if damage or shutdown does not occur, poor mixing usually creates control problems because the "mixed" air temperature cannot be sensed, even with so-called averaging sensors.
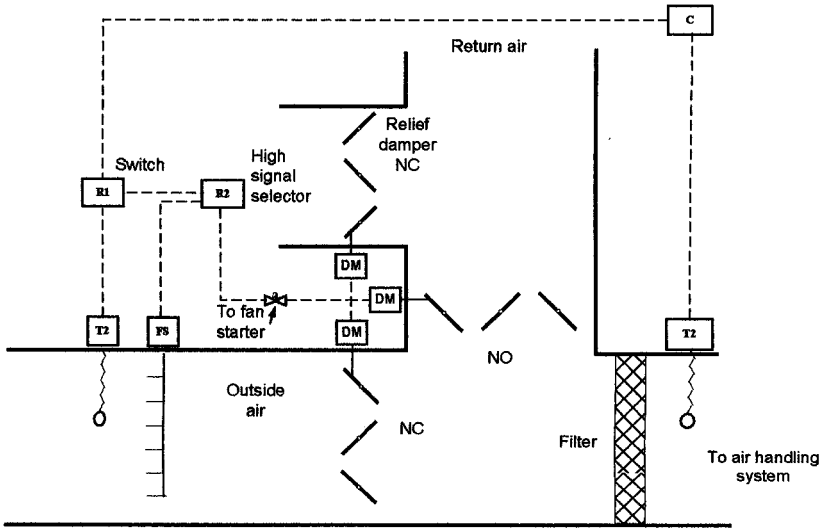
Figure 6-9. Outside air is measured to maintain minimum amount.
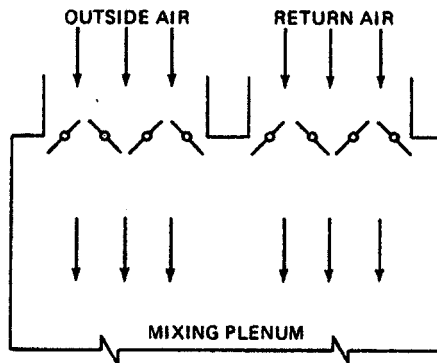
Figure 6-10. Air streams side by side; no mixing.

The mixing plenum and its dampers should be designed to promote good mixing of the air streams. One of the simpler methods is shown in Figure 6-11. Parallel-blade dampers are so arranged that the air streams meet head-on. We showed earlier, however, that parallel-blade dampers usually are not appropriate for modulating control.
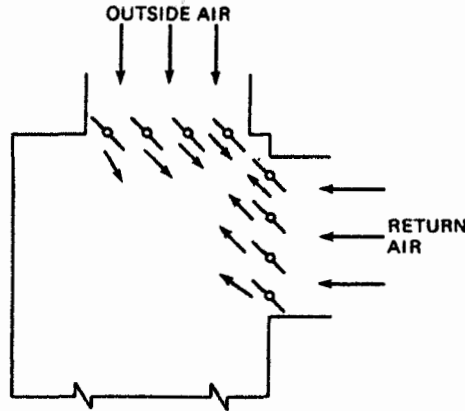
Figure 6-11. Air streams at 90° angle, good mixing.

If the air streams enter opposite sides of the mixing plenum, as in Figure 6-12, then good mixing may occur. Good mixing is more likely if the comparatively high damper velocities recommended in Chapter 5 are used.
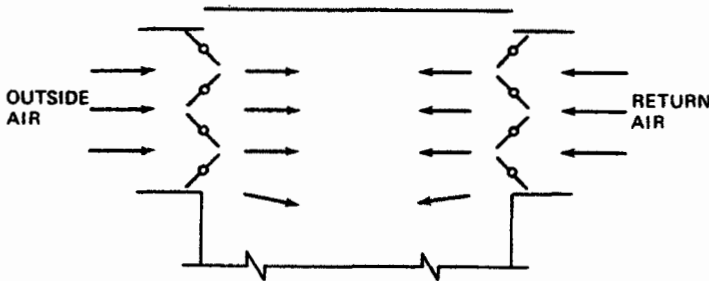
Figure 6-12. Opposed air streams; good mixing.

Unfortunately, in many existing systems damper velocities can be low and outdoor and return air stream will not mix, especially in cold weather, without modifying the mixing box. One approach is to block off part of the dampers to increase velocities.

Static mixers also can be used. They impart a whirling, mixing motion to the air, but complete mixing does not occur for several diameters past the mixer. If an obstruction such as a filter or heating or cooling coil is encountered, mixing stops. Some system pressure drop is added.

Another approach is to add baffles to promote mixing, as shown in Figure 6-13. Experience has shown that this method is the most effective in producing a fully mixed air stream at the cost of some additional system pressure drop.
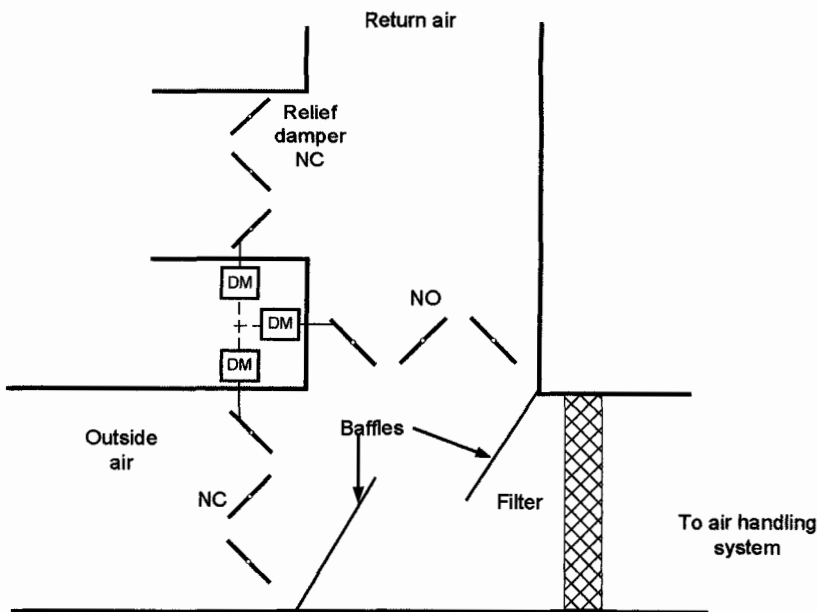
Figure 6-13. Baffles to improve mixing.

6.4 HEATING

Heating in HVAC systems usually is provided by steam or hot water coils with remote boilers. Electric heating coils, heat pumps, and direct gas-fired duct heaters also are used, and are discussed in other sections of this book.

Heating may be done to preheat outside air or heat mixed air, to heat part of the air stream, or to reheat for humidity control or individual zone temperature control.

6.4.1 Preheat

Preheating is used when large percentages of outside air could cause freezing of downstream heating and cooling coils. The main problem in preheating is freeze-up of the preheat coil itself. Several methods are used to prevent this.

Figure 6-14 shows the simplest approach. This is a two-position valve in the steam or hot water supply with an outdoor thermostat that opens the valve whenever the outdoor temperature is below 35°F or 40°F. (This, incidentally, is an open-loop control.) The filter is downstream of the coil to prevent snow loading in severe winter storm weather. Because no control of leaving air temperature is provided, the preheat coil must be carefully selected to prevent overheating at, say, 30°F outside, while still providing adequate capacity at perhaps -10°F or -20°F outside design conditions. This is a difficult, if not impossible, compromise.

Face and bypass dampers are added at the coil and controlled by means of a downstream thermostat (T2, Figure 6-15) to provide a controlled mixture temperature. The difficulty here is stratification of the two air streams. In some cases where a downstream cooling coil has been frozen by a bypass air stream while the preheat coil was in full operation. The preheat coil should always be located in the bottom of the duct, and, even so, it is desirable to provide mixing baffles. Given

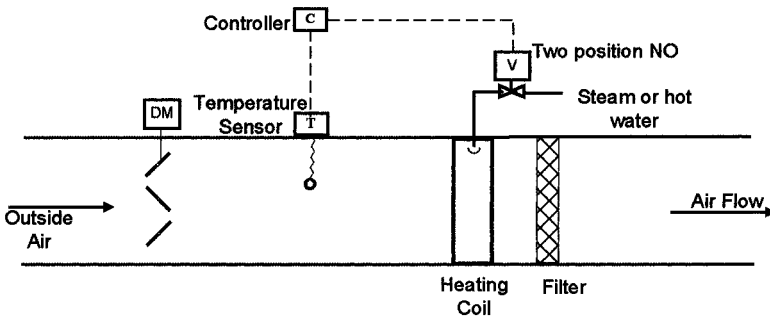sufficient distance in which to provide adequate mixing, this system works well.

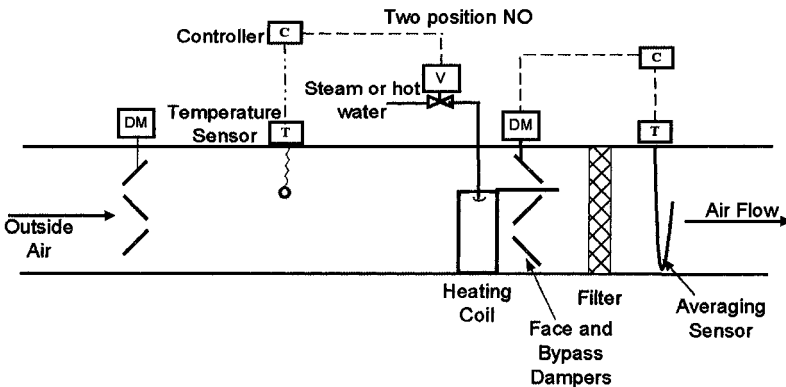Figure 6-14. Preheat: outside air thermostat.

Figure 6-15. Preheat face and bypass dampers.

Often enough distance is not available. The best solution here is to use hot water with a recirculating pump (Figure 6-16). Now there can always be full flow through the coil with the temperature of the water varied to suit requirements. No air is bypassed; so there are no mixing problems. Very accurate control of the air temperature is possible. Notice the opposed flow arrangement with the hot water supply entering the side of the coil where the air leaves.

An alternative pumping arrangement is shown in Figure 6-17. This allows the use of a straight-through valve. The pump head and

horsepower may be somewhat less than in the three-way valve arrangement. (See Haines, 1971 for a full discussion of comparative performances of these two arrangements.) This is particularly true in large systems with multiple chillers and for boilers and circulating pumps. Three-way valves would require that full pumping capacity always be available, even at light loads. With straight-through valves the flow is reduced at part loads and some of the pumps and chillers can be shut off, with a savings in energy use. See Chapter 12 for a full discussion of this topic.
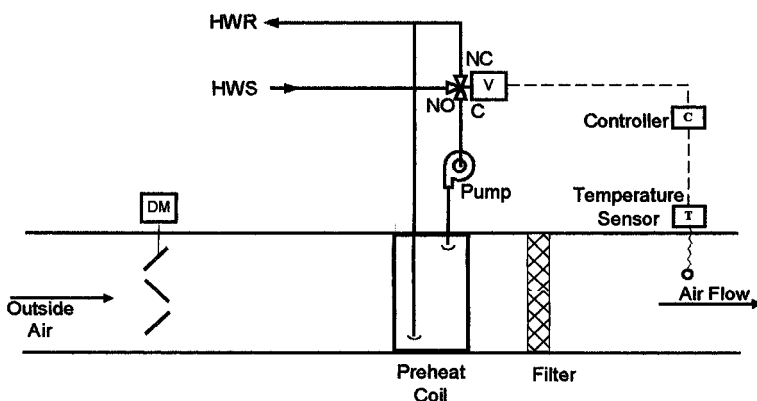

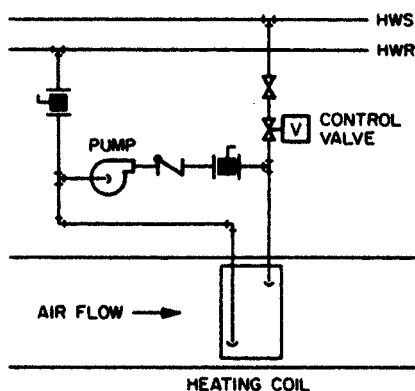Figure 6-16. Preheat; secondary pump and three-way valve.

Figure 6-17. Preheat coil with circulating pump.

In dealing with freezing air, certain precautions are necessary. For hot water, it has been shown experimentally that water velocities of $2\frac{1}{2}$ to 3 feet/second in the coil tubes are sufficient to prevent freezing at outdoor temperatures down to about -30°F provided some hot water is being added. However, a pump failure could lead to a frozen coil! Use of a glycol solution might be a safer alternative. If it is necessary to contend with temperatures of -40° F or below, the use of direct-fired systems, gas, oil or electric, is recommended.

Steam coils in freezing air should be the double-tube distributing type with a good slope or vertical arrangement to drain condensate, as well as adequate trap capacity and vacuum breakers. Even then, problems may occur if the steam flow is modulated. Traps and drain lines must be insulated if exposed to freezing air.

6.4.2 "Normal" Heating

"Normal" heating refers to the coil in a single-zone, multizone, or dual-duct air system that handles all or most of the system air at entering temperatures of 45°F to 50°F or higher. For a single-zone unit (Figure 6-18) the supply valve is controlled by a room thermostat (T1), frequently with a high-limit discharge thermostat (T2) added.
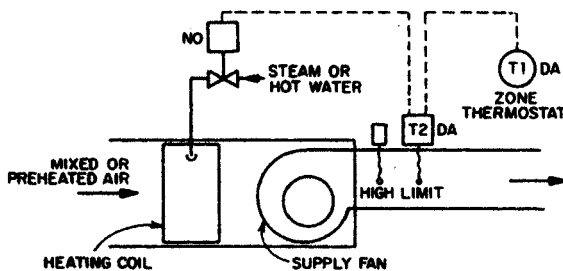

Figure 6-18. Heating, single-zone.

Alternatively, the supply valve may be controlled to provide a variable discharge air temperature with reset from the zone temperature (Figure

6-19). Either of these systems can be used for cooling, heating, or a combination of the two, with heating and cooling coils in series (see Chapter 7).
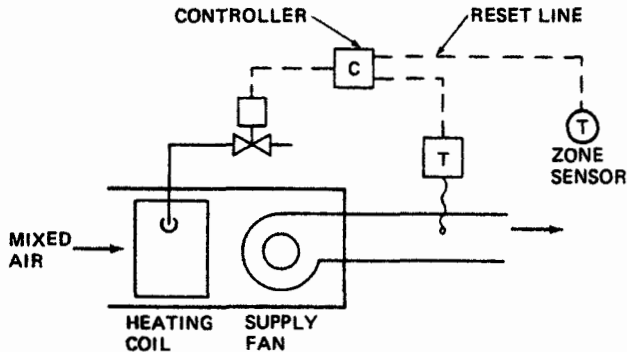

Figure 6-19. Heating, single-zone, discharge control.

In dual-duct or multizone systems the supply valve is controlled by a hot plenum thermostat (Figure 6-20). To improve overall controllability, it is desirable to add outdoor reset, decreasing the hot plenum temperature as the outdoor temperature increases. Discriminator control is also used for reset, as described in Chapter 7.
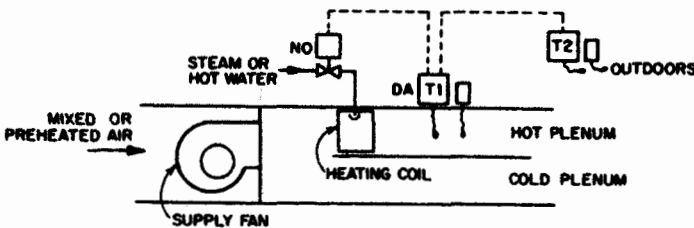

Figure 6-20. Heating, multizone or dual-duct.

If a recirculating pump system is used as in Figures 6-16 or 6-17, one coil sometimes may serve for both preheat and normal heating in a single-zone air handler. The choice will depend on the degree of control required and the economics.

Reheat is used for humidity control or individual zone temperature control. In either case, control of steam or hot water supply valves is

usually by room thermostat, sometimes in series with a supply duct high-limit thermostat.

6.5 COOLING COILS

Cooling coils generally are confined to the air handling unit although occasionally recooling coils are required, as, for example, with chemical dehumidifiers. There are two types: direct-expansion (DX) coils and those using chilled water or brine.

6.5.1 Direct-Expansion Coils

DX coils must, by their nature, use two-position control with its inherently wide operating differential. Nonetheless, this system is often used, particularly in small units and where close control is not required. Figure 6-21 shows a typical DX coil control. The room thermostat opens the solenoid valve, allowing refrigerant liquid to flow through the expansion valve to the coil. The expansion valve modulates according to its setting to try to maintain a minimum refrigerant suction temperature. A low-limit discharge thermostat, T2, keeps the supply air temperature from becoming too cold.
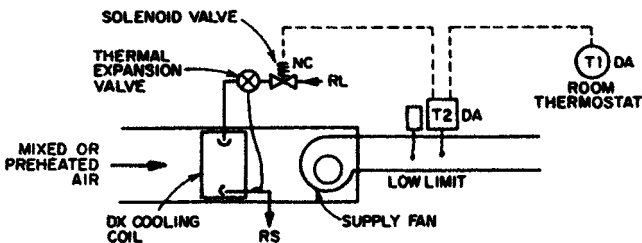

Figure 6-21. Direct-expansion cooling; two-position control.

Controllability can be improved by providing face and bypass dampers (Figure 6-22), but this may lead to complications such as lack of humidity control and coil icing at high bypass rates. Maximum bypass

rates must be established, and the system may not provide adequate control at very light loads.

Figure 6-22. Direct-expansion cooling; face and bypass control.

A different approach adds a variable back-pressure valve in the refrigerant suction line, controlled by the room thermostat (Figure 6-23). As the room temperature decreases, the valve is throttled, increasing the suction temperature at the coil and decreasing the coil capacity. A reversing relay allows the back-pressure valve to be normally open, a necessary condition when the solenoid valve is first opened.

Figure 6-23. Direct-expansion cooling; suction pressure control.

This scheme can lead to problems in the refrigerant circuit and should be used only by an expert in the refrigerant piping design.

Hot gas bypass also may be used for capacity control, as Figure 6-24 shows. A constant pressure expansion valve is used to maintain the evaporator pressure (and temperature) at a constant level, regardless of

load. There are limitations on the percentage of total refrigeration flow that may be bypassed, and on pressure drops in the piping system. Consult a good manual on refrigeration practice.
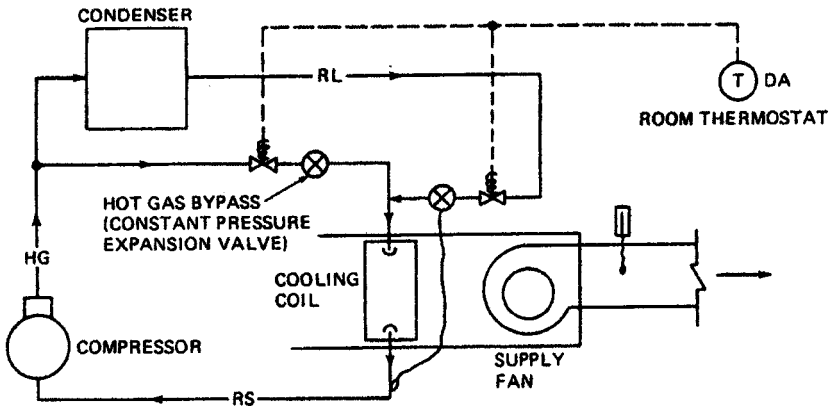

Figure 6-24. Hot gas bypass capacity control.

Two-stage direct expansion will often provide adequate capacity control. The stages should be made by rows of coil rather than by sectioning the coil. Otherwise the active section may ice up, forcing most of the air flow through the inactive section and reducing the coil capacity.

In a multirow coil the first stage should be the first row in the direction of air flow and the second stage the rest of the rows, since the first row of a three or four row coil does at least half the cooling. A two-stage thermostat is used (Figure 6-25).

6.5.2 Chilled Water Coils

Chilled water or brine coils are controlled in much the same way as heating coils, with a three-way or straight-through valve, modulating or two position. Generally, cooling coil control valves should fail in the closed position because this allows the use of direct-acting controllers.

The three-way valve arrangement would then appear as in Figure 6-26 or, if a recirculating pump is used, as in Figure 6-27 or Figure 6-17.
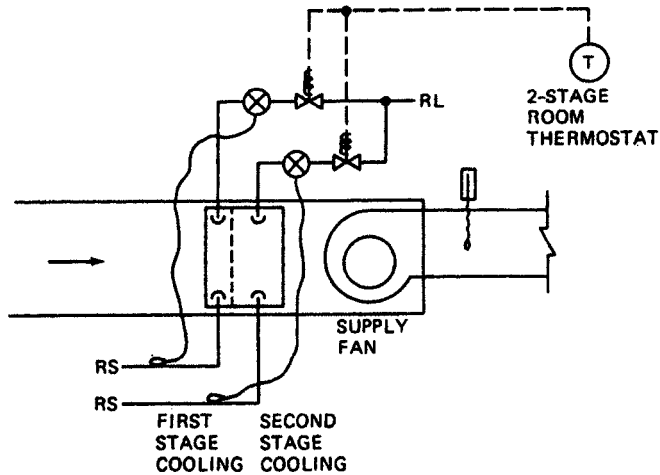
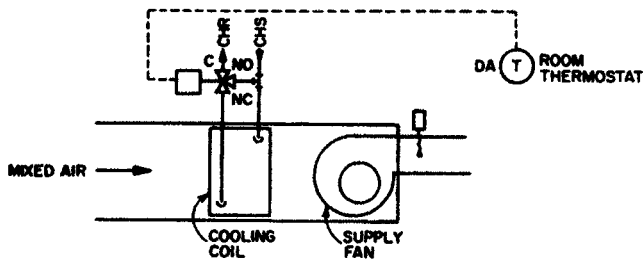Figure 6-25. Direct expansion cooling; two-stage control.

Figure 6-26. Cooling, chilled water, three-way valve.

The recirculating pump arrangement is very useful in two cases: (1) for extremely accurate temperature control and (2) to avoid freezing in those situations where system geometry may make it impossible to avoid stratification of mixed or partially preheated air.
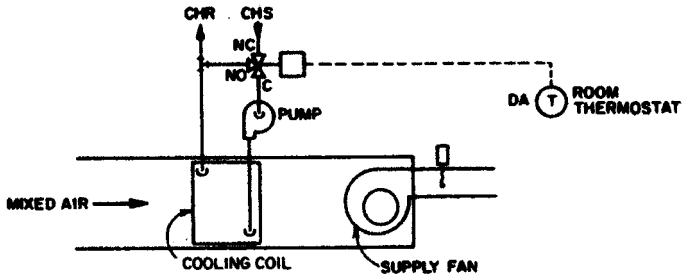
Figure 6-27. Cooling; chilled water, three-way valve and circulating pump.

6.5.3 Parallel and Counter Flow

Notice in both the preceding figures that water flow is shown counter to airflow. As with hot water heating coils, this is very important in maintaining heat exchanger effectiveness. Consider the cooling coil with air flowing through it, decreasing in temperature from 80°F to 55°F and with water flowing parallel to the air and increasing in temperature from 42°F to 52°F. On a graph of temperature versus distance this process would appear as in Figure 6-28(A). For counter flow, Figure 6-28(B) applies.
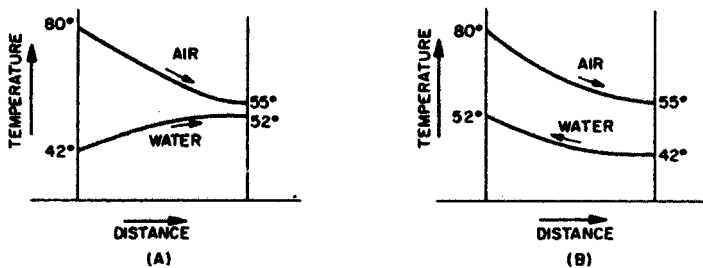
Figure 6-28. Chilled water flow in a cooling coil. (A) Parallel flow. (B) Counter flow.

From heat transfer theory we learn that the heat transfer from air to water in a coil is a function of the resistance through the tube wall and air and water films, the total finned surface area, and the log mean

temperature difference (LMTD). Given fixed rates of air and water flow, LMTD can be calculated by using the equation below:

$$LMTD = \frac{GTD - LTD}{\ln\left(\dfrac{GTD}{LTD}\right)}$$
(6-1)

where :

GTD = greatest temperature difference between the water and the air

LTD = least temperature difference between the water and the air

If we calculate the LMTD for each flow arrangement, we get:

1.   Parallel flow:

        GTD = 80 - 42 = 38
        LTD = 55 - 52 = 3
        LMTD = 13.8

2.   Counter flow:
        GTD = 80 - 52 = 28
        LTD = 55 - 42 = 13
        LMTD = 19.5

Because the heat transfer is proportional to area and LMTD for a dry coil, the counter flow arrangement requires a coil that is only 71% (13.8/19.5) as large as would be required with parallel flow.

The LMTD equation also holds when used with a condensing or an evaporating fluid. But now counter flow does not apply because condensing steam or evaporating refrigerant provides essentially a constant temperature in the coil.

6.6 HUMIDITY CONTROL

It may be necessary to raise or to lower the humidity of the supply air to maintain selected humidity conditions in the air-conditioned space.

6.6.1 Air Washer

Consider first the air washer (Figure 6-29). Often used for its sensible cooling capability, it is also known as a direct evaporative cooler. Whether an inexpensive wetted-pad residential-type unit or a large industrial unit with an elaborate system of sprays and eliminators, any air washer operates on the adiabatic cooling principle. That is, the cooling is done by using the sensible heat of the air to evaporate water. Thus, the air passing through the washer changes conditions along a constant web bulb line, with the final state being dependent on the initial state and the saturation efficiency of the washer (generally 70% to 90%). There is no control of humidity. This is shown in the psychrometric chart of Figure 6-30.
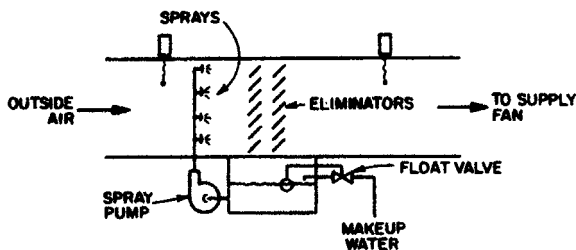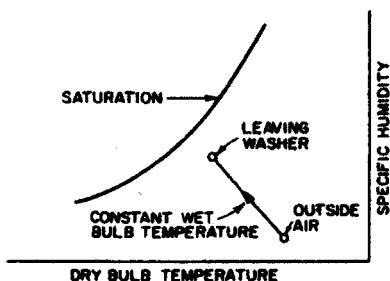

Figure 6-29. Evaporative cooling (air washer).

Figure 6-30. Psychrometric chart for evaporative cooling.

6.6.2 Two-Stage Evaporative Cooling

Two-stage evaporative cooling may be used as an alternative to mechanical refrigeration, when outdoor conditions allow it. This system provides lower dry bulb temperatures and relative humidities than can be obtained with ordinary evaporative cooling. Figure 6-31 shows the arrangement and control of a two-stage evaporative cooling system. Figure 6-32 is the psychrometric chart of the cycle. The dashed lines on the chart show a single-stage cycle as in Figure 6-30.
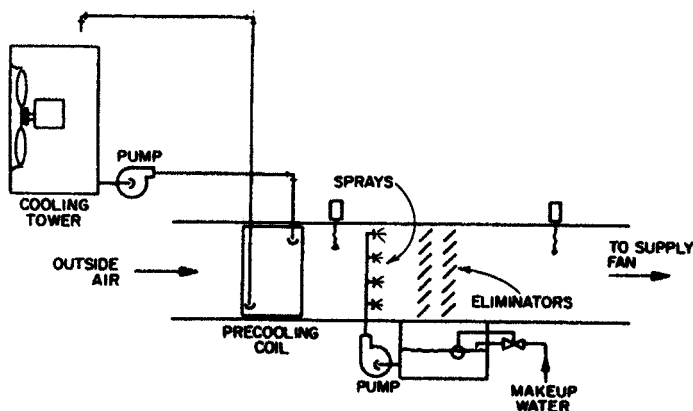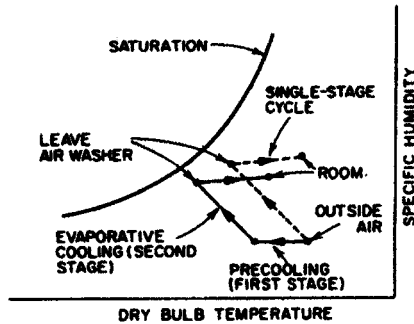

Figure 6-31. Two-stage evaporative cooling

Figure 6-32. Psychrometric chart for Figure 6-31.

The cooling tower and the precooling coil are the first stage. This sensible cooling reduces both the wet and the dry bulb temperatures of the air, so that in the second stage a lower dry bulb temperature may be obtained. The room thermostat will be a two-position type, with two-stage control optional. The controllability of the system depends on the condition of the outside air.

6.6.3 Air Washer with Preheat

About the only control that can be applied to the ordinary air washer is to turn the spray water (or pump) on or off. If a minimum humidity is required, it is sometimes necessary to preheat the air to the desired wet bulb temperature. Such a control system is shown in Figure 6-33. The room humidistat senses low humidity and turns on the spray pump and then opens the preheat coil supply valve. As room humidity increases, the preheat valve is closed first, then if the increase continues, the sprays are shut down. Under high outdoor humidity conditions the cooling capacity is limited. Final room temperature control is provided by reheat coils. The psychrometric chart in Figure 6-34 shows this cycle.
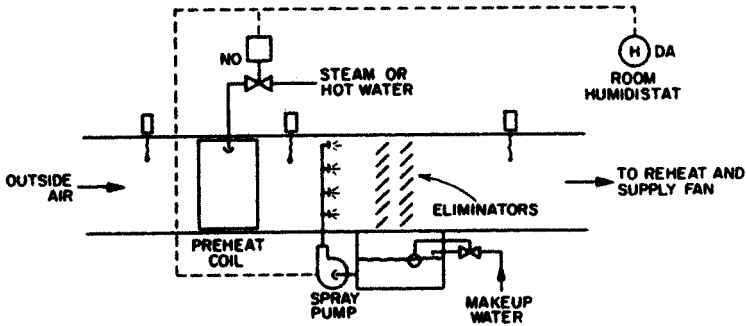
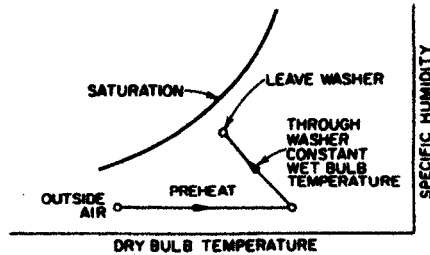Figure 6-33. Minimum humidity; air washer with preheat.

Figure 6-34. Psychrometric chart for Figure 6-33.

6.6.4 Air Washer with Preheat and Refrigeration

An air washer with preheat will not control the upper limits of humidity. To accomplish this two choices are available, both requiring refrigeration:

1. Heat and/or cool the spray water with a shell and tube heat exchangers (Figure 6-35). By allowing the humidistat to control the heat exchanger supply valves it is possible to obtain very accurate control of the humidity as well as the air temperature leaving the washer. Figure 6-36 is the psychrometric chart diagram of this process. Reheat is required because the temperature leaving the washer is constant and is not a function of the building load.
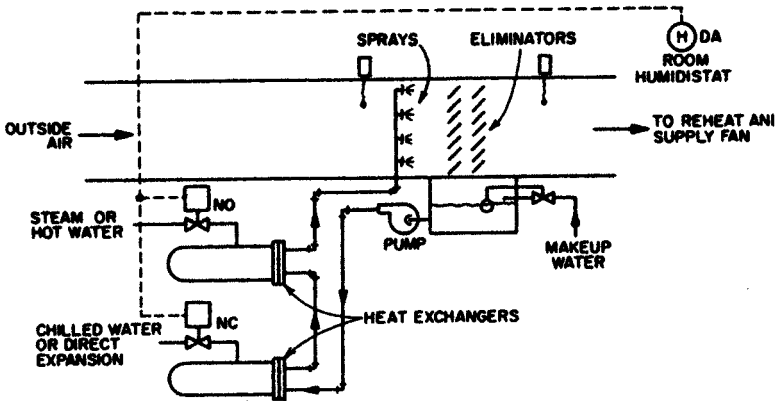
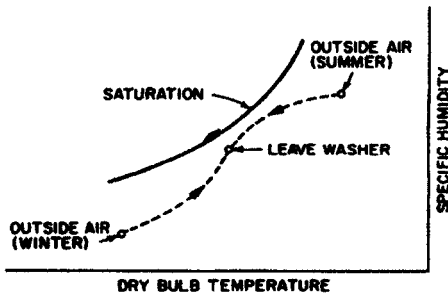Figure 6-35. Humidity control by heating or cooling spray water.

Figure 6-36. Psychrometric chart for Figure 6-35.

2. Add a cooling coil in the air washer, creating a sprayed-coil dehumidifier (Figure 6-37). This designation is not incorrect, because at high humidity conditions dehumidification does take place, but the device also acts as a humidifier if necessary. Because the addition of the coil increases the saturation efficiency to between 95% and 98%, it is possible to do away with the humidistat and use a simpler, less expensive dry bulb thermostat (T1) in the air leaving the coil (the so-called dew point thermostat). This is set to maintain a fixed condition of dry bulb and relative humidity. When the mixed-air wet bulb temperature is above the required wet bulb temperature of the air leaving the coil, then refrigeration must be used. Reheat is necessary for final space temperature control. If all control settings are properly

selected, it is possible to maintain very accurate control of the space temperature and humidity. Figure 6-38 shows the cycle on a psychrometric chart, assuming cool, low-humidity outside air. The dew point thermostat senses a decrease in the coil leaving-air temperature and shuts down the chilled water flow to the coil and then opens the preheat coil valve. Evaporative cooling is used. As the coil leaving-air temperature increases, the preheat coil valve is gradually closed. A further increase in the controlled temperature above the thermostat setting will cause the chilled water valve to open, using refrigeration for cooling. Figure 6-39 is a psychrometric chart showing what happens with high-temperature outside air. As long as the entering-air wet bulb is above the dew point thermostat setting, chilled water will be required, and the coil leaving-air temperature will be maintained by varying the flow of chilled water through the coil. Direct expansion may be used instead of chilled water.
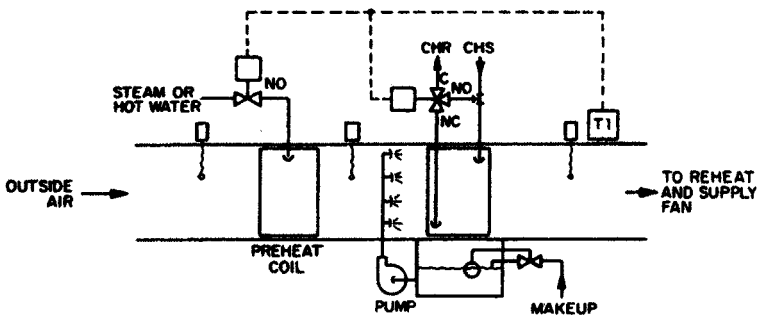

Figure 6-37. Humidity control, sprayed coil and preheat.

6.6.5 Air Washer with Mixed Air and Refrigeration

Mixed air can be used rather than 100% outside air if the control system is altered as shown in Figures 6-40 and 6-41. As long as the outside air wet bulb temperature is below the coil leaving-air set point of T1, the desired condition can be maintained by adjusting the mixing dampers to obtain a mixture that falls on the coil leaving-air wet bulb line, and using the evaporative cooling effect of the sprayed coil. When the outside air damper is fully open and the coil leaving-air temperature

increases above the set point of T1, then refrigeration must be used, either chilled water as shown or direct expansion. Again, reheat is required for space temperature control.
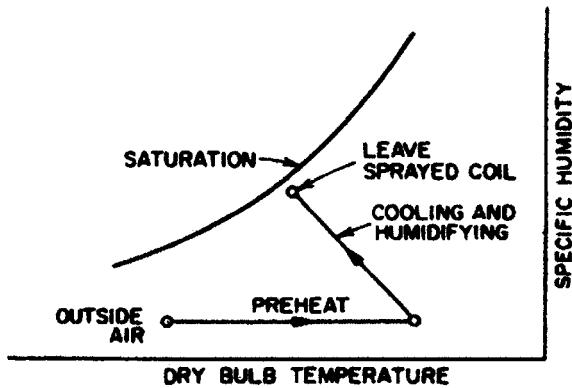

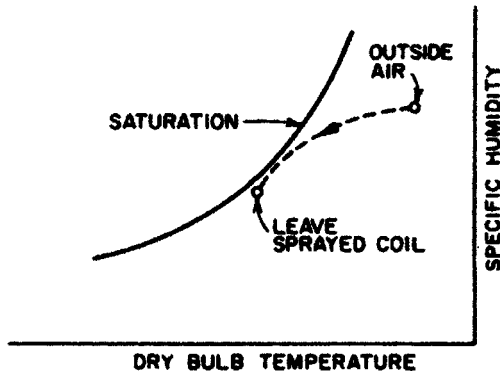Figure 6-38. Psychrometric chart for figure 6-37 (winter).

Figure 6-39. Psychrometic chart for Figure 6-39 (summer).

All of these systems using open sprays require chemical treatment of the spray water to minimize solids deposition. The systems with finned coils pose a severe maintenance problem due to deposition. The only satisfactory solution is to use demineralized water in the spray system.

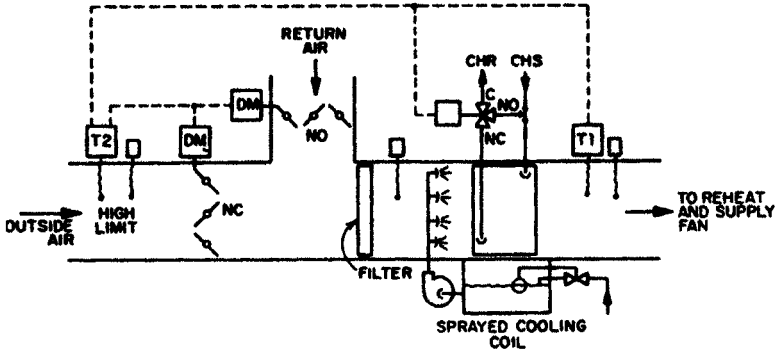Sprayed coils may not be allowed under some codes because of the potential for bacterial contamination.

Figure 6-40. Humidity control; sprayed cooling coil with mixed air.
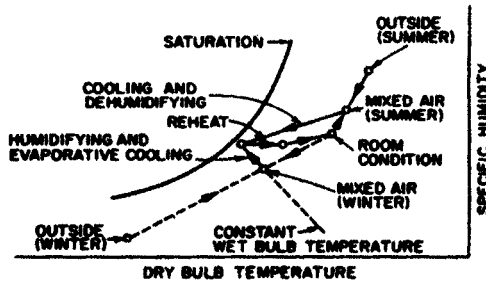

Figure 6-41. Psychrometric chart for Figure 6-38.

6.6.6 Steam Humidifiers

Steam humidifiers are often used because of their simplicity. A piping manifold with small orifices is provided in the air duct or plenum (Figure 6-42). The steam supply valve is controlled by a space or duct humidistat. Avoid the use of steam that has been treated with toxic chemicals. Almost any humidity ratio, up to saturation, can be obtained in the supply air stream. If a space humidistat is used a duct high-limit humidistat should be provided to avoid condensation in the duct.
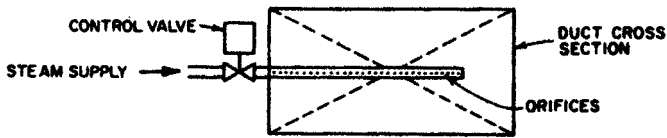
Figure 6-42. Steam grid humidifier.

6.6.7 Pan Humidifiers

Evaporative pan humidifiers vary in capability and controllability. The old-fashioned evaporator pan found in many residential warm-air heating systems is not very efficient, and lacks any kind of control. The addition of a heater, of immersion or radiant type, improves efficiency and allows a limited amount of control. Even with heaters, however, this type of humidifier will not usually provide space relative humidity of over 40%.

6.6.8 Atomizer Humidifiers

Slinger or atomizer humidifiers may be controlled on-off by a humidistat or manually. Again, space relative humidity will seldom exceed 40%. During "off" periods the lime deposits left from the water evaporation may be entrained in the air stream as dust. When evaporative or atomizer humidifiers are used, the water supply should be distilled or deionized. Mineral deposits from evaporation are a serious maintenance problem. "Rental" deionizers are very satisfactory for small installations. (See also Section 7.2.)

6.7 DEHUMIDIFIERS

As noted above, a sprayed cooling coil may serve as a dehumidifier if the entering air is sufficiently humid. However, to maintain very low humidities at normal air conditioning temperatures it is necessary to employ chemical dehumidifiers or low temperature refrigeration.

6.7.1 Chemical Dehumidifiers

Chemical dehumidifiers use a chemical adsorbent, usually with provision for continuous regeneration (drying) to avoid system shutdown. One form of dehumidifier (Figure 6-43) uses a wheel containing silica gel that revolves first through the conditioned air stream, absorbing moisture, and then through a regenerative air stream of heated outside air that dries the gel. In the process, heat is transferred to the conditioned air and recooling is necessary. The dehumidification efficiency is largely a function of the regenerative air stream temperature. Figure 6-44 shows the control system. The space humidistat controls the heating coil in the regenerative air stream. Final control of space temperature is accomplished by the room thermostat and the recooling coil. Very low humidities may be obtained in this way.
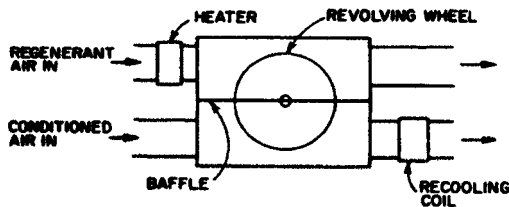

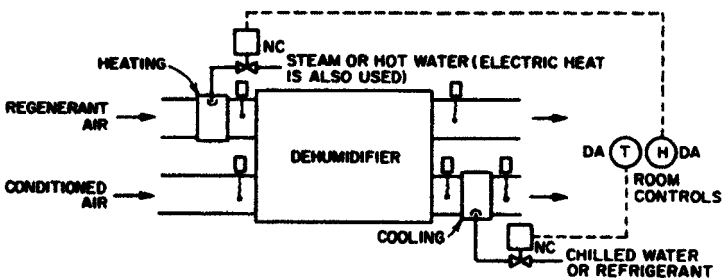Figure 6-43. Control of chemical dehumidifier.

Figure 6-44. Control of chemical dehumidifier.

Another type of system uses a water-absorbing liquid chemical solution that is sprayed over a cooling coil in the conditioned air stream, thus absorbing moisture from the air. A portion of the solution is continuously pumped to a regenerator where it is sprayed over heating

coils and gives up moisture to a scavenger air stream that carries the moisture outside. Control of the specific humidity of the leaving conditioned air is accomplished by controlling the temperature of the solution. Because this is a patented, proprietary system, the control system recommended by the manufacturer should be used.

6.7.2 Dehumidifying by Refrigeration

Low temperature cooling coils may also be used to reduce humidity to low values. Because coil surface temperatures may be below freezing, with ice formation resulting, special DX coils with wide fin spacing must be used, and provision must be made for defrosting by hot gas, electric heat or warm air. This approach tends to be inefficient at very low humidities and requires intermittent shutdown for defrosting, or parallel coils so that one may operate while the other is being defrosted. Reheat is necessary for control of space temperature. Because space humidity is largely a function of the coil temperature, fairly good control may be achieved through humidistat control of a variable back-pressure valve (Figure 6-45). The selective relay (R) allows the room thermostat to operate the cooling coil at minimum capacity when the humidistat is satisfied.
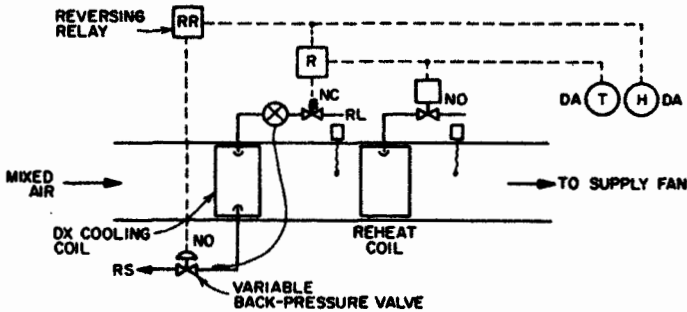

Figure 6-45. Dehumidifying with low temperature cooling unit.

6.8 STATIC PRESSURE CONTROL

Static pressure controls are used to provide a positive or a negative pressure in a space with respect to its surroundings. For example, a clean room will be positive to prevent infiltration of dust, whereas a chemical laboratory or plating shop will be negative to prevent exfiltration of fumes. Many nuclear processes require careful space segregation with fairly high pressure differentials. For most ordinary spaces it is impractical and unnecessary to design for pressure differentials greater than 0.1 inch water gage. Swinging doors are difficult to open and/or close even at this small pressure, and special sealing methods are necessary to maintain higher pressures. Air locks often are used.

6.8.1 Single Room

A pressure-controlled space may be served by its own air-conditioning unit. Outside air may have minimum or economy cycle control by temperature, or may be simply 100%. Relief air is usually provided by power exhaust although positive pressures in the space allow gravity relief. In either case, the static pressure controller controls the relief damper, and if 100% outside air is used, may also control the intake damper. Such a control system is shown in Figure 6-46. (See also Figure 6-7.) The reversing relay is needed so that both dampers can be normally closed.
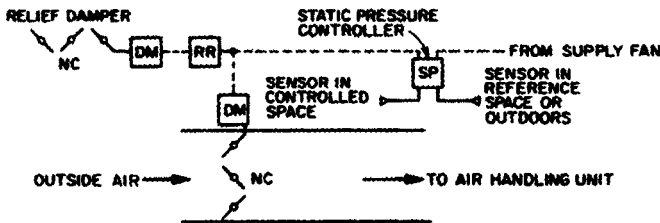

Figure 6-46. Outside air; static pressure control.

6.8.2 Multiple Rooms

If several rooms are to be served by a single air-conditioning unit, then variable-volume dampers on supply and relief to each room will be operated by the static pressure controller for that room. Individual exhaust fans or one large central exhaust may be used.

6.9 ELECTRIC HEAT

Electric heaters may be controlled on the same basic cycles as other heating devices-two-position, timed two-position, and proportional. Because of the use of electricity as the energy source, certain special considerations are necessary. Any electric heater must be provided with a high-limit control. Some codes require both automatic reset and manual reset high limits. Forced-air heaters should have airflow switches to prevent the heater from operating when the fan is off or when air flow is below the minimum rate required to prevent overheating of the electric element.

6.9.1 Two-Position Control

Two-position control of small-capacity heaters may be provided by a heavy-duty line-voltage thermostat. More common, however, is the piloted system (Figure 6-47) with the thermostat operating a contactor. For large heaters it is common to use a multistage thermostat or sequencing switches with several contactors, each controlling current flow to a section of the heating coil.

6.9.2 Proportional Control

True proportional control may be obtained by using a saturable core reactor or variable autotransformer (Figure 6-48). A very small change in the DC control current can cause a large change in the load current.

Because the controller must handle the full-load current it may become physically very large. Efficiencies are poor at part load.
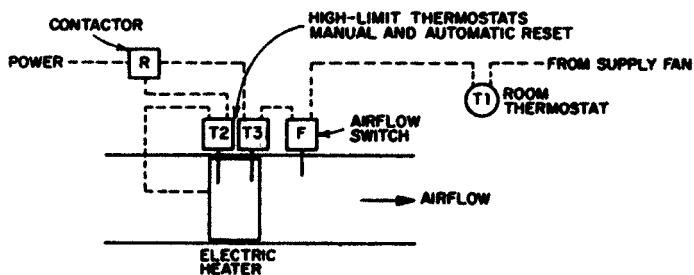

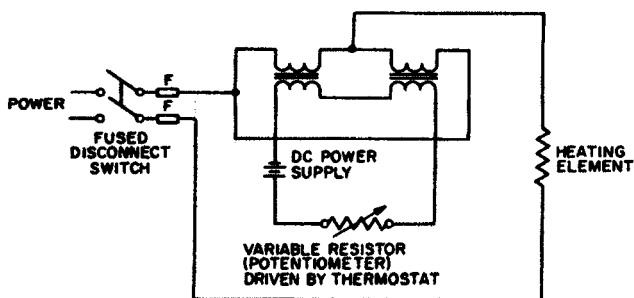Figure 6-47. Electric heater; piloted control, two-position.

Figure 6-48. Electric heater; saturable core reactor control.

6.9.3 Timed Two-Position Control

Timed two-position control can be achieved by using a timer and the same contactors used for ordinary two-position control (Figure 6-49). The timer may be mechanical or electronic and provides a fixed time base (usually adjustable) of from one-half minute to five minutes. The percentage of on-time varies according to the demand sensed by the room thermostat. The contactors will cycle on and off once in each time base period but the length of on-time will be greater if the room temperature is below the thermostat setting.
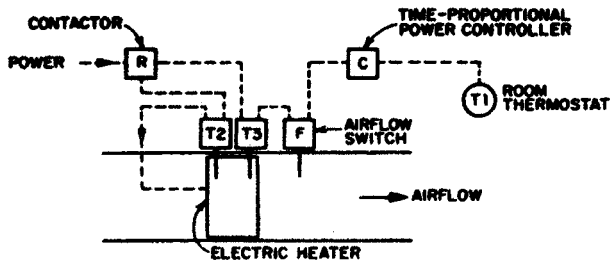
Figure 6-49. Electric heater, time-proportional control.

Rapid cycling may lead to maintenance problems, even with a mercury switch contactor. The preferred method is to use solid-state controllers; SCR's (silicon controlled rectifiers) or Triacs. These devices can provide extremely rapid cycling rates, so that control is, in effect, proportional (Figure 6-50). The solid-state controller has an electronic timer, which provides an extremely short time base. The load current is handled directly by the controller through semiconductor switching devices. The thermostat demand will vary the percentage of on-time. Power regulation may be accomplished by phase control or burst control. The SCR may be piloted by any type of proportioning thermostat, to vary the cycle time rate. The vernier control system (Figure 6-51) combines sequence control with solid-state control to obtain near-proportional control. It is especially economical if the heater is large. The coil is divided into several small sections with one controlled by the solid-state unit and the others by a sequencing step controller. As the space temperature decreases below the proportional thermostat setting, the solid-state unit modulates its section of the coil from off to 100% on. If heating demand continues to increase, the first section of the sequence controlled portion is turned on, and the power to the solid-state section may be cut back. This continues until all sections are on, and, of course, the opposite sequence occurs on a decrease in heating demand. The modulated section should have 25% to 50% greater capacity than the sequenced sections to prevent short-cycling at changeover points.
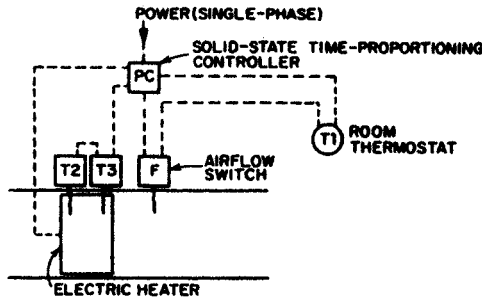
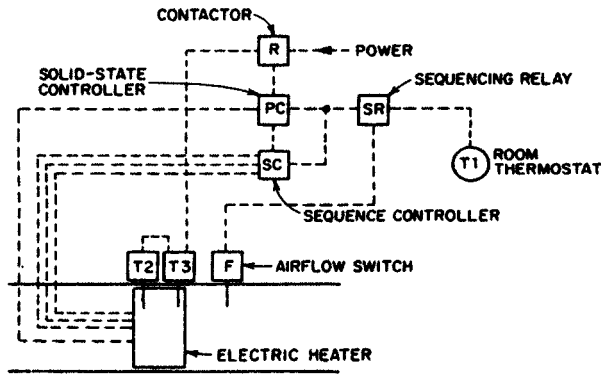Figure 6-50. Electric heater; solid-state controller

Figure 6-51. Electric heater; Vernier control system.

6.10 GAS-FIRED HEATERS

Direct gas-fired heaters are, generally, packaged equipment complete with controls. Usually the complete package, including controls, is approved by the American Gas Association (AGA). Any change in the system, however minor, will void this approval. There are still several options among the approved packages.

6.10.1 Two-Position Control

Two-position controls are most common (Figure 6-52). A variation of this is multistage two-position, with the gas burners sectionalized.
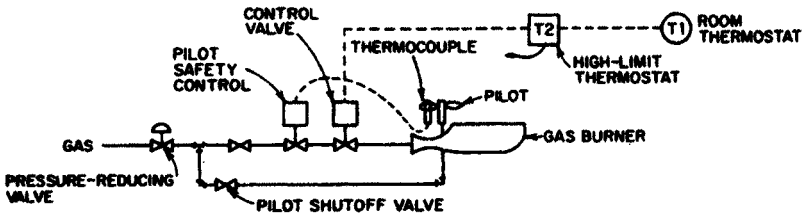

Figure 6-52. Gas heater; two-position control.

6.10.2 Proportional Control

Modulating gas valves are available. AGA approval for small systems is limited to self-contained types, with the sensing bulb mounted in the discharge air stream. Minimum capacity is about 30% of total capacity because the gas flame becomes unstable at lower rates (Figure 6-53). At this lower limit the valve becomes a two-position type.
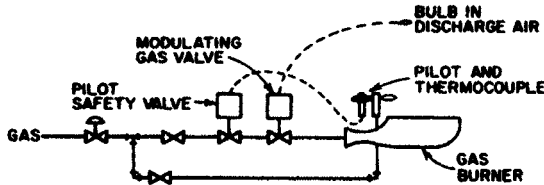

Figure 6-53. Gas heater; modulating control.

6.10.3 Safety Controls

Any gas control system must include such safety controls as high-temperature limits and pilot flame proving devices. Forced-draft heating boiler controls may also include timed pre-purge and post-purge cycles to prevent gas accumulation and explosion. These

specialized cycles usually are programmed by special-purpose electronic controls.

6.10.4 Forced Draft Burners

The above discussions deal with atmospheric-type gas burners. If forced draft or pressure burners are used, the control systems will be furnished by the manufacturer to meet the requirements of one of the national insuring agencies such as Factory Mutual (FM). Many elements described above will be present in such systems. More efficient control of the combustion process is possible, but discussion of these systems is beyond the scope of this book.

6.11 OIL-FIRED HEATERS

Oil-burning heaters, like gas-fired units, generally come complete with a control package with UL or FM approval. Two-position or modulating controls are available, depending on the type and the size of the equipment. This is a highly specialized area. Residential and small commercial control systems will be discussed in Chapter 7.

6.12 REFRIGERATION EQUIPMENT

Central refrigeration equipment may include compressors, condensers, and chillers. Compressors may be reciprocating, screw- or rotary type, scroll, or centrifugal. Often, the compressor is part of a package water chiller as described below. Reciprocating compressors, up to about 50 tons capacity, may be used directly with DX coils. Accurate control of DX systems above 25 or 30 tons is difficult, and systems larger than 50 tons are unusual. Larger reciprocating machines, positive displacement units, and centrifugal compressors will be part of a package chiller system, complete with factory-installed controls. Absorption machines

also come as part of a package chiller except for small residential units. Condensers may be air-cooled, water-cooled, or evaporative, and are sometimes part of the chiller package.

6.12.1 Reciprocating Compressors

Reciprocating compressors, except in very small sizes, have multistage capacity control. This is generally done by loading and unloading cylinders under control of a suction pressure or chilled water temperature controller, by raising the suction valve off its seat (Figure 6-54). Unloading devices may be mechanical or electrical and are always a part of the compressor package. The number of steps is determined by the manufacturer as a function of size, number of cylinders, and machine design.

Starting and stopping the compressor may be done directly by the room or chilled water thermostat, but is more often done on a pump-down cycle. On a rise in temperature of the controlled medium the thermostat opens a solenoid valve in the refrigerant liquid line to the chiller or cooling coil. Refrigerant flow raises the suction pressure. The low-pressure switch closes and starts the compressor. When the thermostat is satisfied and closes the solenoid valve, the lowering of the suction pressure as refrigerant is pumped out of the evaporator opens the low-pressure switch, stopping the compressor. Figure 6-55 shows a typical electric control system for a medium-sized reciprocating compressor with safety and operating controls.
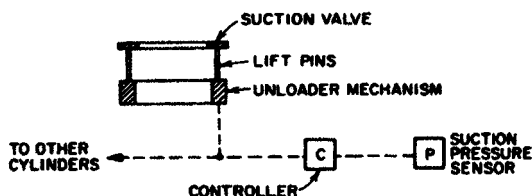

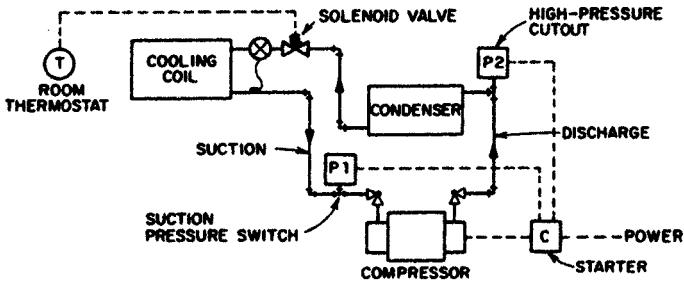Figure 6-54. Refrigeration compressor unloader.

Figure 6.55. Refrigeration compressor; "pump-down" control.

6.12.2 Centrifugal, Positive-Displacement, and Absorption Chillers

Because centrifugal and absorption machines always occur as part of a chiller package, the complete control system will be furnished by the manufacturer. These systems usually include fully modulating capacity controls with a low limit of 20% to 30% of maximum capacity, and provide elaborate safety and interlock controls to protect the equipment. A typical control system for a centrifugal package is shown in Figure 6-56. The chilled water and condensing water pumps are started manually (or the condensing water pump may be interlocked to start when the chilled water pump is started). When the chilled water thermostat calls for cooling, the compressor will start, provided that flow switches and safety switches are closed. The thermostat will then modulate the inlet vane capacity controller, with the capacity limiting controller acting as a maximum capacity limiting device. When the load falls below about 20% of machine capacity, the thermostat will stop the compressor. A time-delay relay, not shown, usually is provided to prevent restarting the compressor at less than 30-minute intervals. This is necessary to prevent damage to the motor.
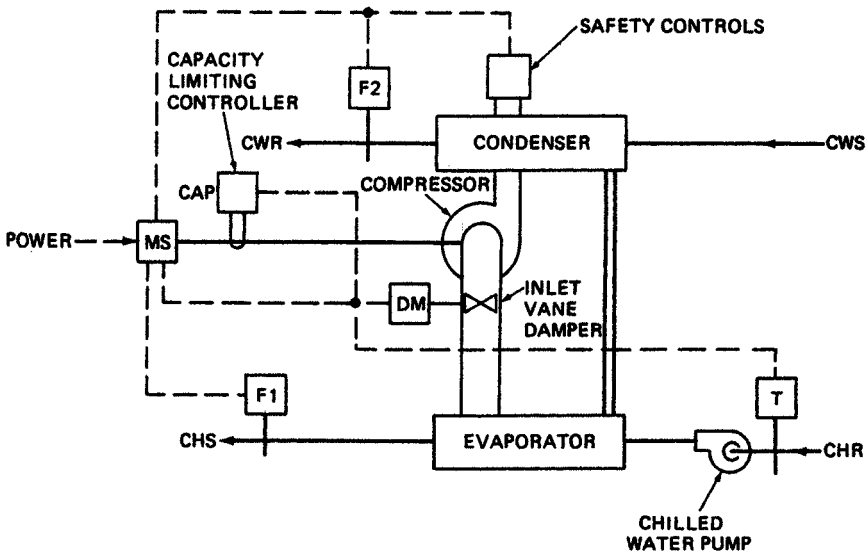
Figure 6-56. Centrifugal water chiller control.

6.12.3 Air-Cooled Condensers

Air-cooled condensers use ambient air forced across a condensing coil to cool and liquefy the compressed refrigerant gas. The condenser fan is controlled to start automatically whenever the compressor runs. Condensing pressures at high ambient temperatures will be higher than with water-cooled or evaporative condensers, because these condensers depend on wet bulb temperatures. But, at low ambient temperatures, this condensing pressure may fall so low as to cause operating problems in the refrigerant system. Any air-cooled system that must operate at low ambient temperatures must be provided with head-pressure control.

A simple head-pressure control system (Figure 6-57) uses modulating dampers to reduce airflow, finally stopping the fan when the damper is nearly closed. Alternatively, a variable-speed fan may be used.

A more elaborate and effective system uses the flooding principle, in which a throttling valve slows down the flow of liquid refrigerant, causing the condenser coil to be partially or wholly filled with liquid,

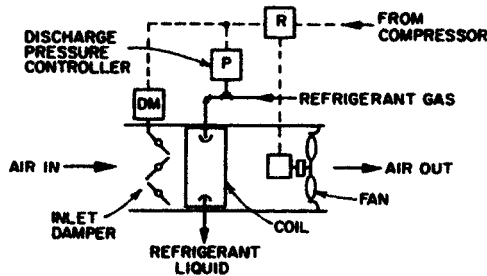thus reducing its capacity. There are several flooding control systems, mostly patented by manufacturers.

Figure 6-57. Air-cooled condenser.

6.12.4 Water-Cooled Condensers

Water-cooled condensers use water from a cooling tower or other source to cool and liquefy the refrigerant gas. Condensing pressure can be closely controlled by modulating water flow or controlling the supply water temperature by mixing condensing return water with supply water. If a cooling tower is used, the water temperature may be controlled by cycling the tower fan or bypassing some of the tower flow (Figure 6-58). With the temperature of the condensing water supply (CWS) above the thermostat set point, the flow valve V1 is open, bypass valve V2 is closed, and the fan is running. As the CWS temperature decreases below the set point, the fan is turned off. On a further decrease valve VI closes, and V2 opens. As the CWS temperature increases, the reverse cycle takes place.

A wide throttling range should be used because refrigeration system efficiency increases as the condensing temperature decreases. A typical control range is 70°F to 85°F. Too low a condensing temperature can cause surge problems with the refrigerant compressor.
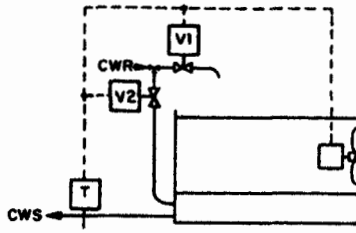
Figure 6-58. Cooling tower; bypass valve control.

Absorption chillers require close control of the condensing water temperature, with mixing most often used (Figure 6-59). The CWS thermostat modulates the mixing valve to maintain a constant supply water temperature. When the valve is in full bypass condition, the cooling tower fan is stopped. In order for the bypass valve to function properly it is necessary to have several feet of gravity head above the valve to the tower. If a good static head is not available, a diverting valve may be used (Figure 6-60).
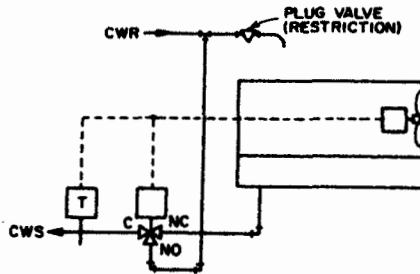
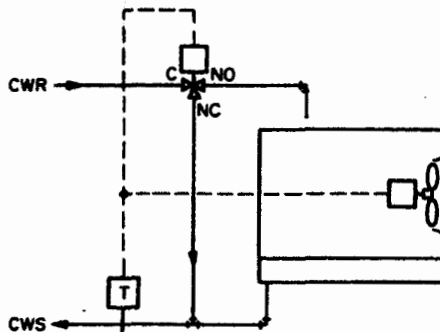Figure 6-59. Cooling tower; three-way valve control.

Figure 6-60. Cooling tower; three-way diverting valve.

6.12.5 Evaporative Condensers

Evaporative condensers take advantage of wet bulb temperatures by spraying water over the refrigerant condensing coil. Air is forced or drawn across the coil, and the resulting adiabatic saturation process provides efficient condensing. Control of the condensing temperature is achieved by varying the airflow (Figure 6-61). The head-pressure controller modulates the outside air damper to provide constant head pressure. When the damper is nearly closed, the fan is stopped. The condenser coil should be all "prime surface" (no fins) to minimize maintenance problems caused by mineral deposits.
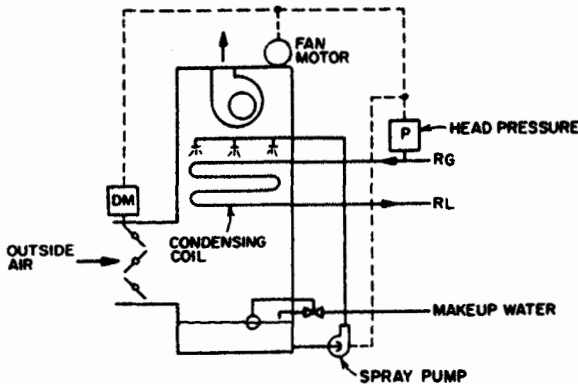

Figure 6-61. Evaporative condenser control.

6.12.6 Water Chillers

Chillers may not be part of a package. When they are installed as a separate piece of equipment, two arrangements are possible. Flooded systems use a surge tank with a low- or high-pressure float to control the refrigerant feed. Direct-expansion systems use a thermostatic expansion valve. In either case refrigerant flow will be started or stopped by a solenoid valve in the refrigerant liquid line, controlled by a chilled water thermostat. Interlocks to assure chilled water flow and prevent freezeup are required. Figure 6-62 shows a direct-expansion chiller with water flow and low water temperature interlocks. The

operating thermostat, T1, in the chilled water return will open the solenoid valve in the refrigerant liquid line, provided that the water is flowing and not near freezing.
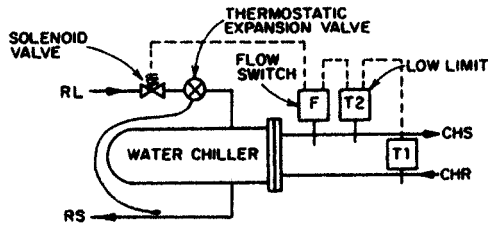

Figure 6-62. Water chiller control.

6.12.7 Cooling Towers

Cooling towers that will operate only during the cooling season are usually provided with fan control only. A condensing water supply thermostat will start or stop the fan as the temperature rises or fall. On larger towers two-speed fans may be used, with two- or three-stage thermostatic control.

Towers that are used all year need more extensive control systems, including bypass valves and heating to prevent freeze-up. Figure 6-63 shows a year-round system with two-speed fan control, bypass, and heating. On small or medium-sized systems (to about 200 tons) towers can sometimes be installed indoors so that modulating dampers to vary airflow will provide adequate control (Figure 6-64). When the damper is fully closed, the fan will be stopped. An outdoor tower with an indoor sump may also be used. In any case a low water flow rate through the tower in freezing weather may cause ice buildup in the tower fill, with resulting damage to the tower. Modulating water flow should no be used.
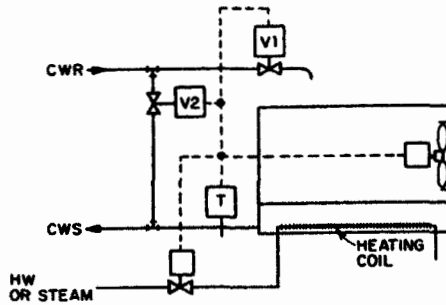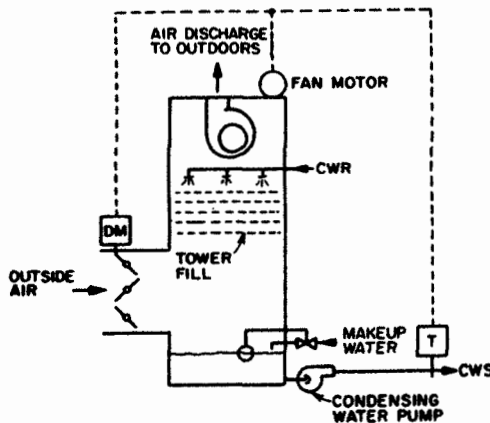
Figure 6-63. Cooling tower; water operation.

Figure 6-64. Cooling tower; indoor arrangement.

6.13 FIRE AND SMOKE CONTROL

Motorized fire and smoke dampers are used for fire separation and for control and evacuation of smoke. The basic design of these devices is controlled by the National Fire Protection Association and local codes. The motor operators are installed to hold the dampers open, so that they close on loss of control air or power. Modern smoke control technology provides for opening and closing smoke dampers during a fire so that the smoke generated will be evacuated from the building and not allowed to flow to areas within the building next to the fire.

For example, in Figure 6-65 if smoke is detected in zone 2, the following sequence would occur: Supply air damper 2-1 would close; supply air dampers 1-1 and 3-1 would remain open; exhaust dampers 1-2 and 3-2 would close; exhaust damper 2-2 would remain open.

The effect is to create a negative pressure in zone 2 and a positive pressure in zones 1 and 3, so that smoke will be removed from zone 2 to exhaust and will be replaced with clean air from zones 1 and 3. The controller is simply a logic device built up using relays, or, for many zones, it may be a programmable controller. A computer based supervisory control system can include smoke control among its many functions.

It must be emphasized that the typical HVAC system is not designed for smoke control and must not be used for this purpose. Only when the HVAC system is an engineered smoke control system can true smoke control be accomplished. (See "Design of Smoke Management Systems," ASHRAE 1992.)
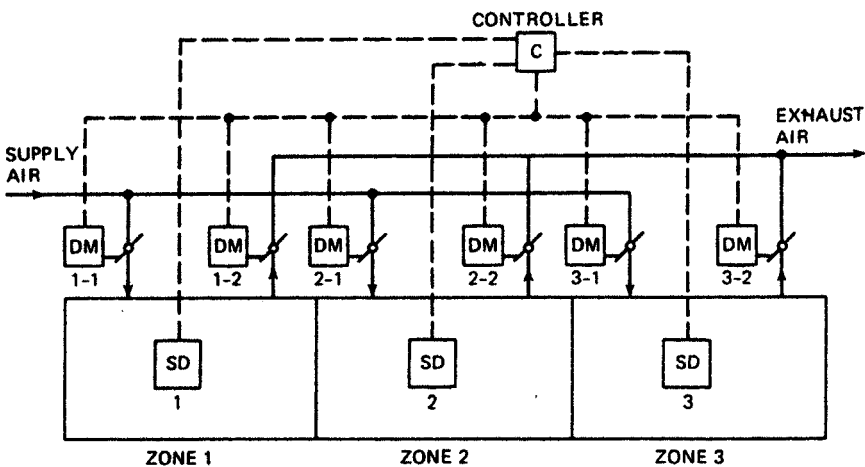

Figure 6-65. Smoke control system.

6.14 ELECTRICAL INTERLOCKS

As shown in several preceding examples, at some point it will be necessary for the temperature and pressure controls to interface with electric motor controls. Small motors may sometimes be operated directly by electric controllers. If the motors are large, or of a different voltage than that of the control circuits, or if pneumatic or electronic controls are used, then relays are required. This subject is discussed more fully in Chapter 8.

6.15 LOCATION OF SENSORS

The proper location for the sensor can best be determined by asking specifically what is to be controlled. If it is room temperature, then the location should be such that the sensor reads an average room temperature, with a minimum of exposure to supply air, drafts or radiant effects from windows or equipment. Putting the sensor near return air openings or in the return air duct is recommended.

In a large work area, you may want the best control at a particular work station; so mount the sensor there. Sensors have been mounted on movable frames with long cables and relocated as the critical work station changed. The main problem to avoid is side effects that may prevent the sensor from seeing conditions correctly. For temperature sensors these may be radiation (cold or hot), drafts, lack of adequate air circulation, or heat transfer through the mounting (as on an outside wall).

In systems with several rooms on a single zone, it is essential to select an "average" room for the sensor location. Conference rooms or rooms with large load variations should be on separate zones, but, if this is not possible, do not let them be the sensing point for the zone. Given a choice between large and small rooms, select the larger space. All rooms on a single zone should have comparable outside exposures with

a common orientation (north, south, east or west, but not a mixture of these).

6.16 SUMMARY

This chapter has discussed elementary control systems or, more correctly, subsystems. Larger systems are built up from combinations of these elements. Chapters 7 and 9 show how this is done. It is hoped you will be able to recognize these smaller elements in the larger systems.

7

Complete Control Systems

7.1 INTRODUCTION

In the previous chapter we discussed elementary control systems, which are the pieces and parts that can be fitted together to produce complete control systems for specific applications. This chapter will discuss many of these applications. The discussions begin with simple single-zone systems and proceed to more elaborate arrangements. Some highly complex systems will be dealt with in Chapter 9.

7.2 SINGLE-ZONE SYSTEMS

Single-zone systems are intended to control the temperature and sometimes the humidity in one space or a group of spaces that have similar heating and cooling load variations.

7.2.1 Single Air Handling Unit

Consider first a simple system consisting of a single air handling unit (AHU) with a room thermostat as the basic controller. For year-round air conditioning this AHU will have heating and cooling coils and sufficient outside air for ventilation. If the coils use hot and chilled

water, then the simple system might look as shown in Figure 7-1. When the supply fan is started, the minimum outside air damper opens and the temperature control circuit is energized. The room thermostat then opens and closes the hot and chilled water valves in sequence to provide heating or cooling as required. Notice that only one of these valves may be open at one time. In older buildings, a fire safety switch will stop the fan if a high (about 125°F) return air temperature is sensed. In modern buildings, a smoke detector is required by most building codes. The diagram shows the heating coil preceding the cooling coil. This is to minimize the chance of freezing a coil. Unless humidity control is required, the heating coil should be upstream of the cooling coil.
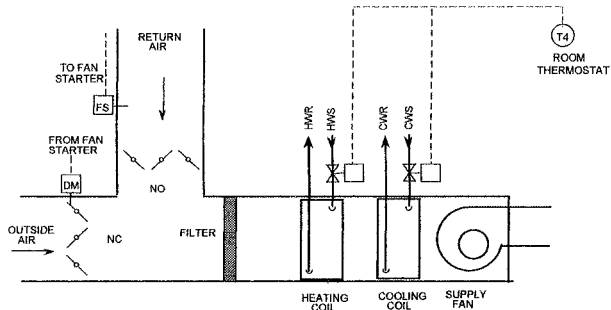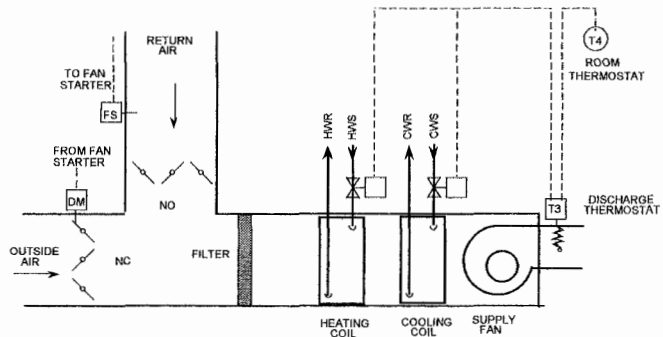
Figure 7-1. Single-zone AHU; minimum outside air.

The control system in Figure 7-1 works for close coupled situations where any change in the heating or the cooling of the air will quickly change the room temperature. This quick feedback will keep the system stable without a wide throttling range. In cases where spaces are large or duct runs are long, the room may respond sluggishly, and the system just described will have a rather wide operating differential. To avoid this the system is frequently modified by adding a thermostat in the air stream leaving the AHU (Figure 7-2). When the room thermostat senses the need for cooling, instead of controlling the valves directly, it causes a decrease in the set point of the discharge thermostat. And, when heating is required, the reverse is true. Because the discharge thermostat senses changes in the supply air temperature very quickly, the system differential is decreased.

7.2 Single-Zone AHU; minimum outside air, discharge thermostate.

As noted in Chapter 6, many systems use economy cycle control of outside air. Figure 7-3 show a single-zone system with an economy cycle. The outside, return and relief dampers modulate in response to the mixed-air controllers. The room thermostat can be used to reset the mixed air controller. This will provide greater energy conservation than can be obtained with a fixed set point. (See Section 6.2.2 and Haines, Apr. 1981)

The discharge thermostat controls the hot and chilled water valves in sequence. The room thermostat resets the set point of the discharge air temperature controller. Note that the diagram shows a single controllers with two inputs. This is the way many manufacturers now provide this control, but it also may be two separate units in a master-submaster arrangement.

7.2.3 Single-Zone Unit, Static Pressure Control of Outside Air

By the simple addition of a static pressure controller to the above cycle it is possible to maintain a positive or a negative pressure in the room (Figure 7-4). This controller operates the relief damper independently of the return and outside air dampers, thereby maintaining the set pressure.
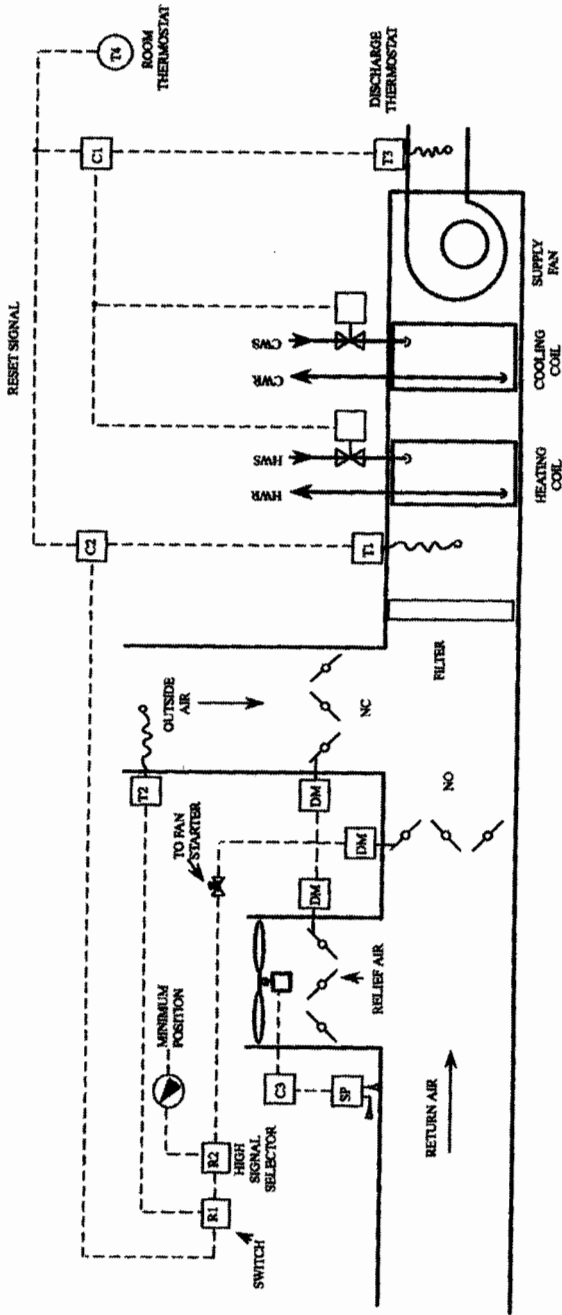
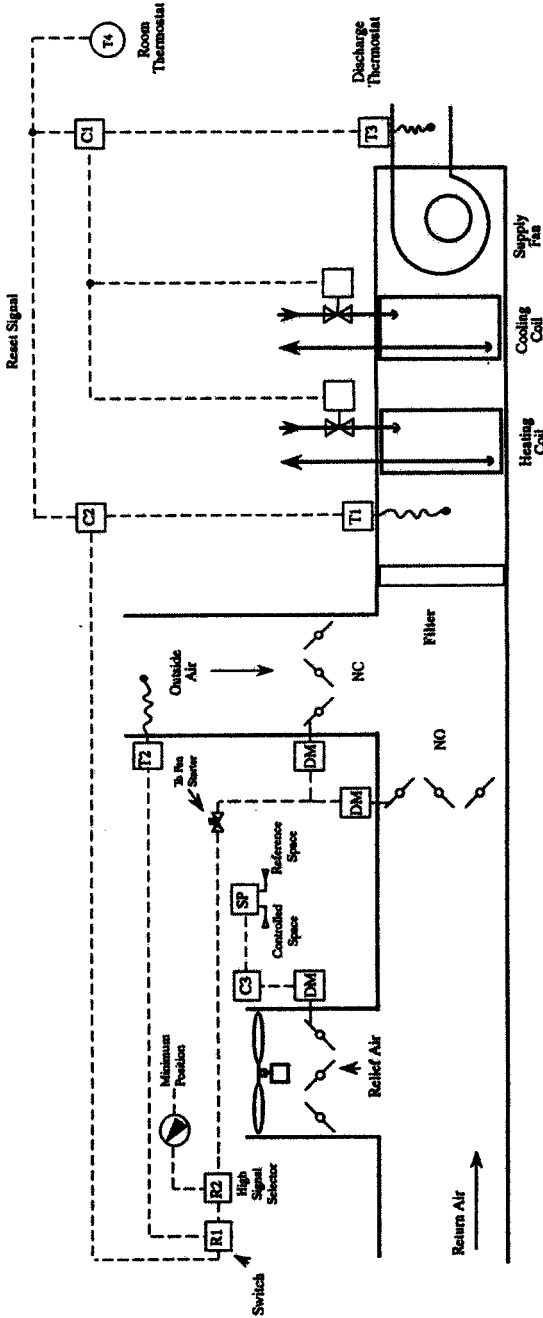Figure 7-3. Single-zone AHU; economy cycle outside air, discharge controller.

Figure 7-4. Single-xone AHU; economy cycle, discharge controller, static pressure.

Figure 7-5. Single-zone system with return fan.

7.2.4 Return-Relief Fans

Large air handling systems, with lengthy return air ducts, may have a significant pressure drop in the return air duct. In the past, designers have put in return air fans in the arrangement shown in Figure 7-5. The arrangement often leads to trouble, for this reason: The mixing box will normally be at a slight negative pressure in order to suck in outside air. Depending on the pressure drop in the outside air duct work (if any), the pressure upstream of the outside air damper will be near atmospheric pressure. However, the pressure upstream of the return dampers will have to be slightly positive if relief air is to escape the building. This can make damper selection and control difficult. In many cases poor commissioning leads to a significant positive pressure upstream of the return damper, meaning that little or no outdoor air is introduced unless the return dampers are closed and do not leak.

The relief fan configuration of Figure 7-3 is the preferred approach. In this design, the supply fan maintains a negative pressure in the mixing box that is low enough to bring the return air through the wide open return air damper. As the outdoor, return, and relief air dampers modulate, the exhaust fan is controlled to maintain the pressure upstream of the return dampers at a level less negative than that of the mixing box by the amount of the wide open pressure drop through the return air damper. This ensures that full flow is maintained through the return duct no matter what the damper position.

Return air quantity is equal to supply air quantity less any fixed exhaust and exfiltration requirements.

7.2.5 Single-Zone Humidity Control

Control of maximum humidity can be provided by placing the cooling coil ahead of the heating coil in the airflow sequence. Then a space humidistat is added to the control sequence (Figure 7-6). While room
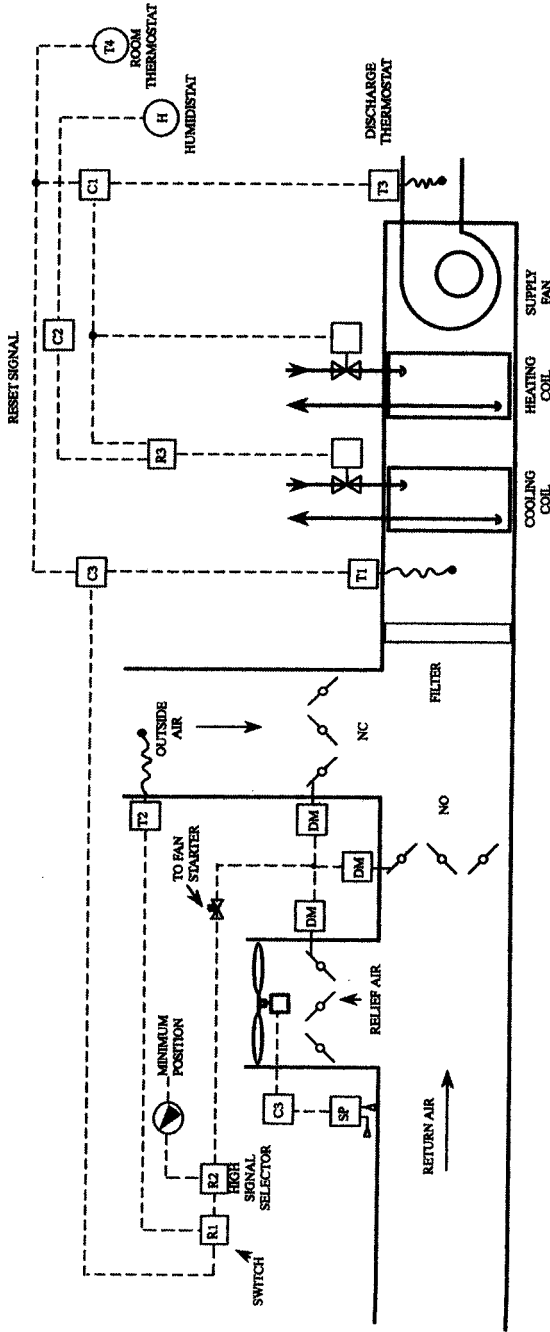
Figure 7-6. Single-zone AHU; maximum humidity control by reheat.

humidity is below the humidistat setting, the system operates as in Figure 7-3. If the humidity goes above the set point, the humidistat, through the selector relay, takes over control of the cooling coil, and calls for additional cooling, which provides dehumidification. If the room then becomes too cool, the thermostat will call for reheating. The psychrometric chart in Figure 7-7 shows the cycle graphically.
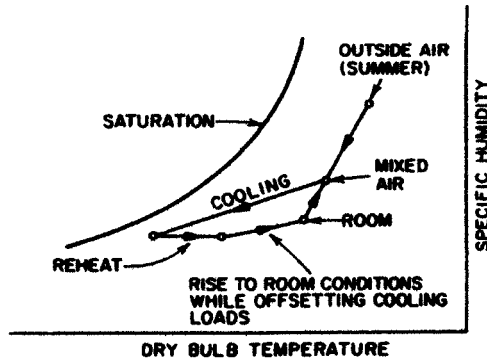

Figure 7-7. Psychrometric chart for Figure 7-6.

Year-round humidity control within fixed high and low limits can be achieved in several ways. A method used frequently in the past employed the sprayed-coil dehumidifier described in Chapter 6. This is no longer popular because of problems with solids deposition and bacterial contamination. An alternative is to use a humidifier to maintain the low limit, with high limit control as described above or by means of a dew-point temperature sensor and controller.

Figure 7-8 shows a single zone system using 100% outside air, with controlled humidity. The humidifier is in the air handling unit. It could be installed in the supply duct but must always be upstream of the duct temperature sensor because the humidifier adds some heat, as shown in Figure 7-9, the psychrometric chart for the system. This system will easily maintain 50% to 60% RH maximum in summer and 35% to 40%
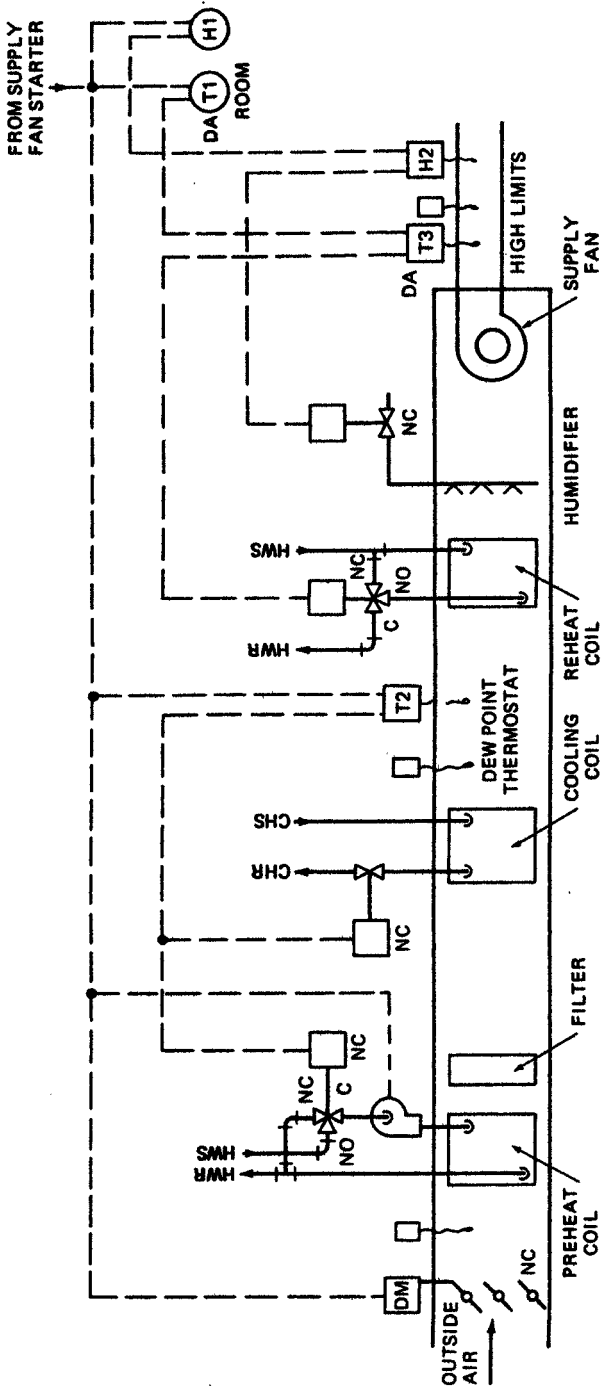
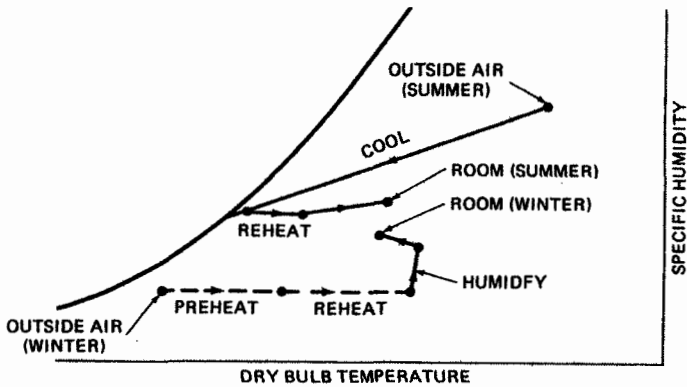Figure 7-8. Single-zone AHU; humidity control, 100% outside air.

Figure 7-9. Psychrometric chart for Figure 7-8.

minimum in winter. Lower maximum humidities may be obtained by decreasing the chilled water temperature or by using brine or direct-expansion cooling. Minimum humidities higher than about 45% are difficult to maintain and require the use of large steam grid humidifiers.

A similar control cycle can be applied if the system requires less than 100% outside air (Figure 7-10 and the psychrometric chart of Figure 7-11). The preheat coil is needed only if the minimum outside air is such a large percentage of outside air that the mixed-air temperature would be too cold. The low-limit mixed-air controller may be reset by the room humidistat to minimize the use of the humidifier. Otherwise, the outside air control is a standard economy cycle system.

Humidifiers require extra maintenance. Any evaporative system, such as a pan or sprayed coil, will leave deposits of solids behind from the evaporated water. A demineralizer on the makeup water will often pay for itself by saving the cost of cleaning up coils and pans. Steam humidifiers should not be used where the steam is treated. Many steam additives are toxic, and they cause odors and unsightly deposits. Where a central steam or high-temperature water supply is provided and a steam humidifier is to be used, a small heat exchanger can be used to generate steam from demineralized water. Whenever a duct humidifier is used, a duct humidistat should be provided and used as a high limit to avoid condensation in the duct.
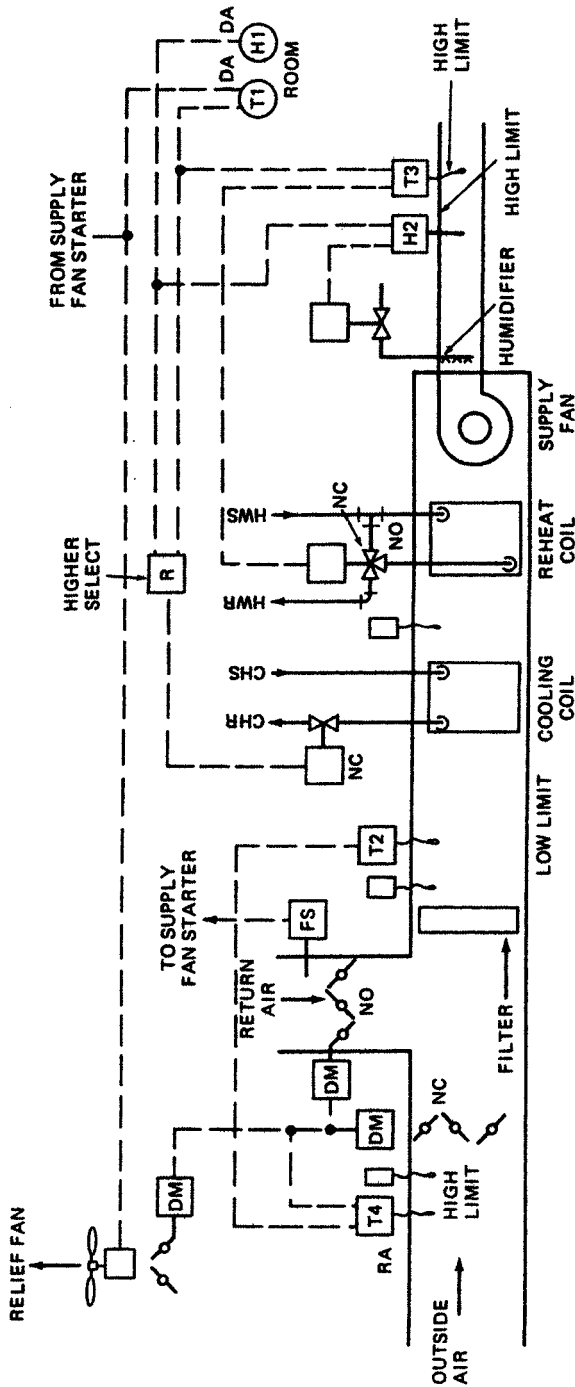
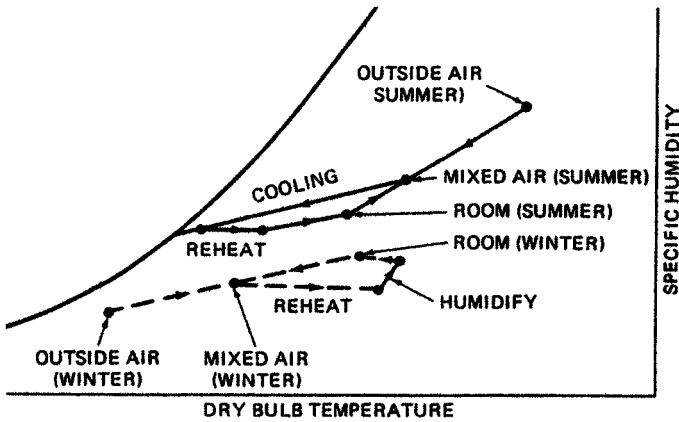Figure 7-10. Single-zone AHU; humidity control with mixed air.

Figure 7-11. Psychrometric chart for Figure 7-10.

7.3 MULTIZONE AIR HANDLING SYSTEMS

Figure 7-12 shows a schematic of a multizone system. Air is blown over a heating coil and a cooling coil arranged in parallel. Each room thermostat controls a pair of interconnected dampers which mix hot and cold air to provide air to the zone at the required temperature. The temperatures of the hot and cold air streams are controlled by independent control loops.

Multizone systems can be less costly than reheat and variable air volume systems because separate heating coils are not needed for each zone. However, a separate duct must run from the multizone unit to each zone. Consequently, the space required for the duct work and the zone dampers limits the number of zones that can be served by a single multizone unit.
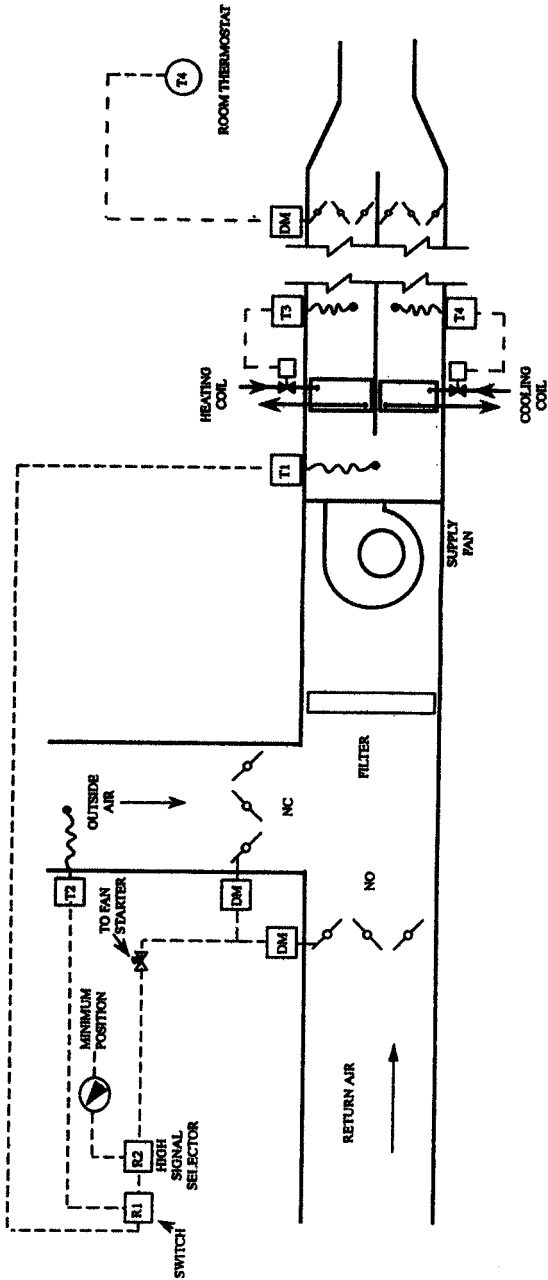
Figure 7-12. Multizone system.

7.3.1 Multizone Control

Using the simplest control scheme, the heating and cooling coil controllers have fixed set points, and the respective discharge air temperatures are more or less constant ,subject to the throttling ranges

In this configuration, the system is thermodynamically similar to a reheat system -- when zone loads are low, heated air is blended with cooled air to avoid overcooling.

The efficiency of a multizone unit is strongly influenced by the temperature of the air leaving the heating coil (the "hot deck" temperature). To illustrate this, suppose that a room thermostat controls the multizone mixing dampers according a linear profile between 72°F and 76°F. We can compute the fraction of cold air in the mixed stream going to the zone as:

$$F_c = ((T_r - T_{rset})/(\text{Room Throttling Range})) + 0.5 \qquad (7\text{-}1)$$

where:
T_r = the room temperature and
T_{rset} = the room temperature set point.

If there is no cooling or heating required in the room then the delivery air temperature and supply air temperature must be the same. The supply air temperature is simply the weighted average of the hot and cold deck temperatures:

$$T_s = T_r = (1\text{-}FC)(T_h) + (FC)(T_c) \qquad (7\text{-}2)$$

where:
T_s = temperature of air delivered to the zone,
FC = fraction of cold air,
T_h = delivery air temperature off the heating coil and
T_c = delivery air temperature off the cooling coil.

If we let T_C equal 55°F we can solve the above equations simultaneously to find FC as a function of T_h, the hot deck temperature. The lower curve of Figure 7-13 shows this result. Notice that even though there is no cooling load in the zone, the load on the cooling coil is almost 80% of the cooling capacity if T_h is high. It is only when the hot deck temperature is near the room temperature, that the load on the coil begins to track the room load. In this example, if the hot deck temperature is above 95°F there will be more total thermal energy transferred to the air when there is no cooling required in the zone than when full cooling is required.
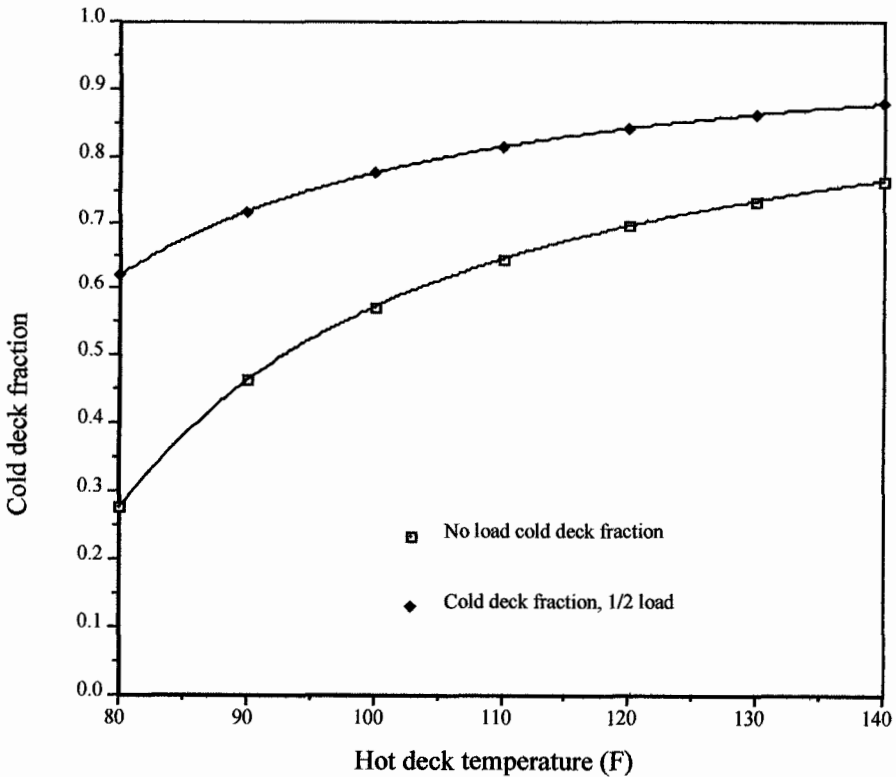
Figure 7-13. Cold deck fraction vs. hot deck temperature for a multizone system.

Like reheat systems, multizone systems can be terribly inefficient.

We can carry our example further to help expose the effect of high hot deck temperature on control stability. If the zone now has a cooling requirement of one half its capacity, we see from the upper curve of Figure 7-13 that the fraction of cold air in the total air to the zone is very high if the hot deck temperature is warm. More than 80% of the air comes from the cold deck if the hot deck temperature is above 110°F. If most of the zones on a system require some cooling, then the total flow over the heating coil could be less than 20% of the maximum design flow.

As the flow through a coil diminishes, the process gain increases (a small change in hot water flow causes a large change in the discharge air temperature). The time constants of the coil and the temperature sensor become longer. A simple proportional controller used to control a heating coil for a multizone system must have a very low gain if the control system is to be stable when only a little air is flowing over the heating coil. However, a low gain corresponds to a large throttling range, which leads to large steady state error under low load. If proportional only control is used, the heating coil discharge air temperature must rise well above the set point before the heating coil valve will close. This higher discharge temperature means that even less hot air is required, and the flow over the hot coil will be reduced even further, pushing the system toward instability while increasing energy consumption.

It is very difficult to set up a stable proportional only control loop for the heating coil of a multizone system.

An additional source of difficulty occurs at light-load conditions with modulating controls. Then, the temperature gradient across the face of the coil, from one end to the other, may be 5°F or 10°F or more. Poor control may result, especially for zones that feed off the plenum near the ends of the coils. This can be solved by using circulating pumps with water coils or by feeding both ends of long coils.

Similar difficulties can occur with the control of the cooling coil discharge air temperature. This can be avoided by simply using no control at all on the cooling coil. Cooling coils can be used without control because the temperature of the chilled water entering the coil can be controlled to be only about 10°F colder than the desired supply air temperature. If this is done carefully, then the droop that occurs under low load cannot be more than about 10°F even if no control is used for the discharge air temperature. Of course, if the chilled water temperature drops under low part-load conditions, then the uncontrolled discharge air temperature will drop correspondingly.

Unless the application ensures that there will be substantial air flow over the cooling coil whenever it is on, it is probably better to let the cooling coil run "wild" than to use a proportional only controller. If no control is used, the system will at least be stable.

A better approach is to use proportional plus integral control for both the hot and the cold discharge air temperature control loops. This will permit low proportional gain settings to be used without the steady state error inherent in proportional only control.

We now turn to the task of making multizone systems more efficient. The simplest way to make real headway is to turn the heating coil off in the summer and turn the cooling coil off in the winter. This stops the battling of hot and cold air streams, but it means that one zone cannot be heated while another is cooled. In some applications, however, simultaneous heating and cooling may not be required, and seasonal changeover from heating to cooling is simple and effective.

Although it is possible to have wintertime cooling loads in a building (interior rooms, for example), it is not possible to have summertime heating loads. Hence, the heating coil should always be shut off in the summer.

In addition to summer shutdown of the heating coil, reset of the hot deck supply air temperature controller is also important because:

- If both the heating coil and the cooling coil are on, and cooling or heating loads are low, a multizone system will consume disproportionate amounts of energy if the hot deck temperature is high.

- All zone dampers in a multizone system leak. Consequently some air from the hot deck is always mixed with air from the cold deck (at least 10% leakage is typical). If the hot deck temperature is high, energy waste due to damper leakage will be high and overheating of spaces may result, especially if the cooling coil is turned off during mild winter weather.

- The hot deck reset keeps a reasonable quantity of air flowing over the heating coil during periods of light load. This improves the stability of the heating coil control loop.

Two schemes are common for hot deck reset. In one, the discharge air temperature controller is reset based on the outdoor temperature (see Figure 7-14). A PI controller must be used here because the steady state error that would occur if proportional only control is used would eliminate much of the beneficial effects of the reset scheme.

In the other scheme, reset is based on zone temperature (see Figure 7-15). The lowest signal from the zone thermostats is used to reset the hot deck controller. Zone controlled reset will be effective only if the throttling range of the room thermostats is divided into two regions -- one used to modulate the zone dampers and the other to reset the hot deck controller. The zone controlled reset scheme may produce greater energy savings than the outdoor temperature reset scheme because room temperature is a better indicator of heating requirements than is outdoor temperature. However, if the heating and cooling coils are not allowed to run simultaneously, there will be almost no difference in savings between the two reset schemes. Zone reset is the more complicated and of these methods more likely to fail or malfunction.

In addition to the resetting of the heating coil controller, resetting the temperature of the hot water supplied to the heating coil based on
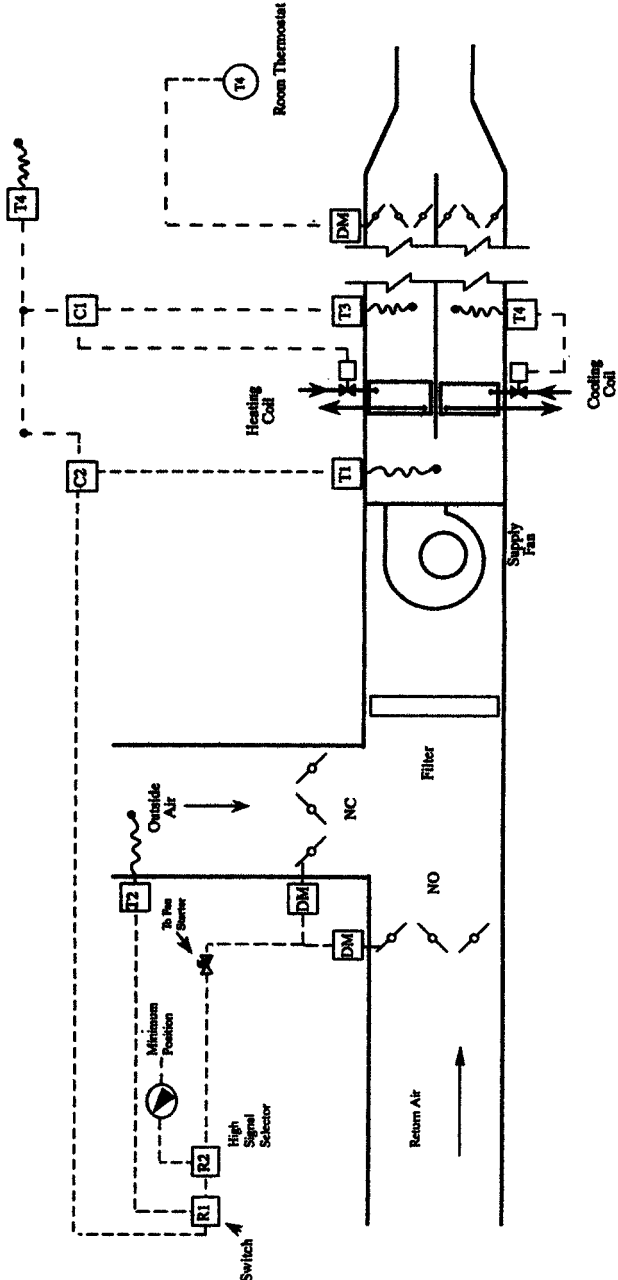
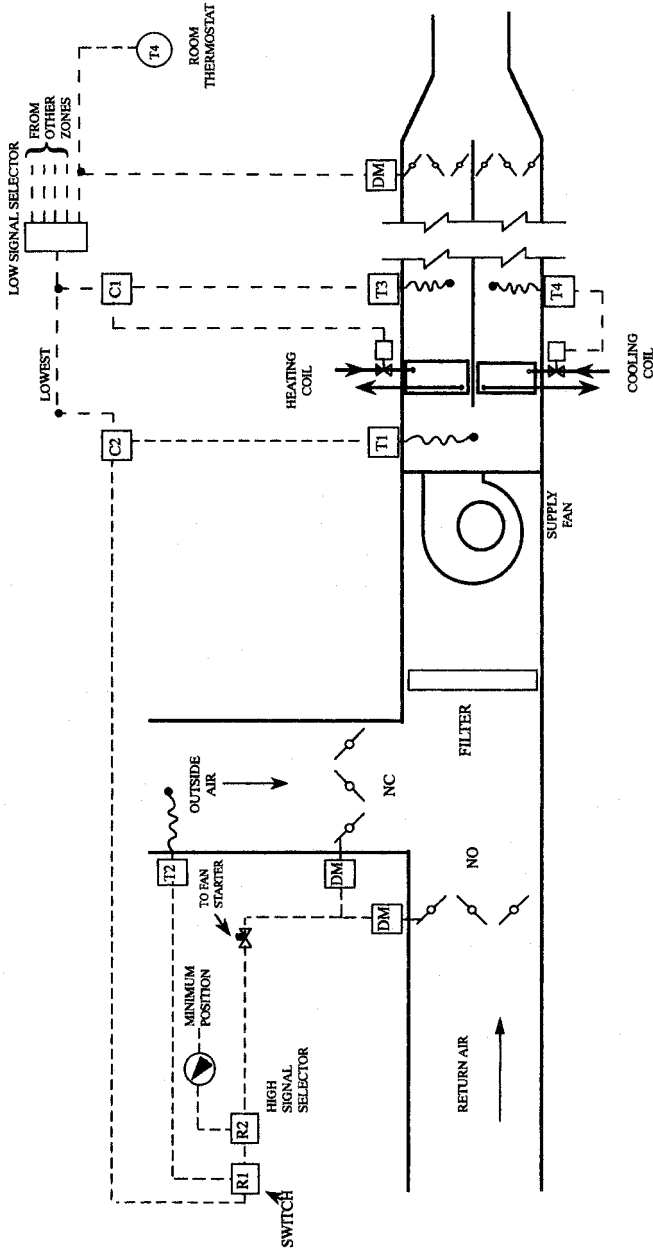Figure 7-14. Multizone system with hot plenum reset.

Figure 7-15. Multizone system with discriminator control of hot plenum.

outdoor temperature usually is required to help keep the heating coil control loop stable. If the hot water temperature is not reset but the heating coil controller is, then, during mild weather, the capacity of the heating coil will be very high compared to the load on the coil, and there will be a large difference between the temperature of the hot water supplied and   the desired discharge air temperature. The resulting increase in the gain of the process tends to drive the heating coil control loop toward instability.

Another frequently proposed conservation measure for multizone systems is to reset the cold deck temperature using a scheme similar to one of the above schemes for hot deck reset. However, resetting the cold deck temperature based on outdoor air temperature is almost always a poor strategy because the cooling load in many of the zones served by the system may be high even if the outdoor temperature is moderate. On a sunny but cool day the cold deck temperature will be too high to meet the zone loads.

Resetting the cooling coil controller based on the highest signal from the zone thermostats provides improved performance, at least   in theory. However, the system is complicated, so that it is difficult to commission and prone to failure. Again, the throttling range of each zone thermostat must be divided into three regions-one for cold deck reset, one for zone damper modulation, and one for hot deck reset.

So far we have described the hot and cold deck control loops only. We now consider the use of economy cycles with multizone systems.

The use of economy cycles on a multizone system poses a dilemma. Although the use of the appropriate economy cycle can save cooling energy, it will increase heating energy when applied to a multizone system because part of the mixed air stream (the mixture of outdoor and return air) passes over both the heating coil and the cooling coil. If the mixed air temperature controller used to implement an economy cycle has a set point low enough to minimize cooling energy requirements then the air entering the heating coil will usually be

colder than it would be if no economy cycle were used. Thus more heating will be required to bring the air flowing over the heating coil up to its discharge temperature. The heating energy consumption penalty in the winter will be large indeed if the mixed air temperature controller has a fixed set point.

To avoid this penalty, the economy cycle should be deactivated during cold weather. This can be done manually by the building operator or through the use of a low-limit two-position thermostat sensing outdoor air temperature.

Another way to avoid some or all of the heating energy consumption penalty associated with an economy cycle is to reset the mixed air controller and the cold deck controller on the basis of the highest signal from the zone thermostats. Although this control system has the potential of maximizing the efficiency of a multizone system, it is very complicated, difficult to commission, and prone to failure. As with all zone-temperature-based reset schemes, the failure of a single room thermostat or high signal selector or a single broken connection can lead to control system malfunction. Experience has shown that failure modes often are inefficient and can go undetected for indefinite periods unless the performance of the systems are routinely and skillfully monitored.

7.3.2 Multizone Control 100% Outside Air

Multizone units may be used in applications requiring 100% outside air. Here, preheat is required, usually to 50°F or 55°F, which provides a cold plenum temperature adequate for cooling interior zones.

7.3.3 Multizone with Humidity Control

Figure 7-16 shows a suitable arrangement of a multizone unit with humidity control. For maximum humidity control all the air must be cooled to a dew point corresponding to the desired relative humidity.

Figure 7-16. Multizone system; humidity control systems.

When humidity is to be added, it must be supplied by humidifiers in the individual zone ducts. The psychrometric chart would be similar to Figure 7-11.

7.3.4 Three-Plenum Multizone System

Many energy codes now prohibit the use of "reheat" in air conditioning; this effectively disqualifies two-plenum multizone.

The three-plenum multizone system eliminates the use of new energy for reheat by providing a bypass plenum in addition to the usual hot and cold plenums (Figure 7-17). The control sequence is as follows: At full heating load the hot damper is fully open, and the bypass and cold dampers are closed. As zone temperature rises the hot damper modulates toward closed and the bypass damper modulates toward open. With the zone thermostat satisfied, the bypass damper is fully open, and both hot and cold dampers are closed. If the zone temperature rises above the set point, the switching relay R2 causes the signal from the reversing relay to control the bypass damper so that it will close as the cold damper opens. Cold and hot plenum air streams are never mixed with one another.
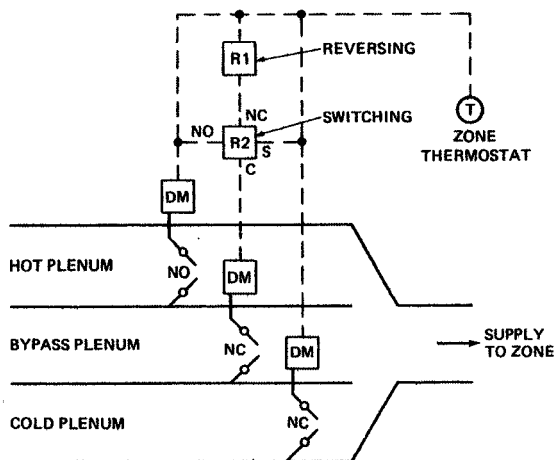
Figure 7-17. Three plenum multizone system.

7.4 DUAL-DUCT SYSTEMS

Dual-duct systems are thermodynamically similar to multizone systems. Mixing of hot and cold air streams simply takes place near the zones being served instead of at the air handler (see Figure 7-18). Both hot and cold air ducts are run to each zone (hence the name dual-duct). Using this approach, many more zones can be served by a dual-duct system than is possible with a multizone system. High pressure and high air velocities have characterized many dual duct systems - duct-work sizes were reduced at the expense of fan energy.

Room temperature control is accomplished by blending hot and cold air in a mixing box near each zone. Because these boxes may be some distance from the fan, variations in flow and duct resistance cause pressure variations at the inlets to the boxes.

7.4.1 Dual-Duct Static Pressure Control

Static pressure differentials may be minimized by static pressure control dampers in the main ducts, near the air handling unit. The sensor measures static pressure near the end of the duct and modulates the damper between full open and some minimum closure position to maintain the desired end-of-main pressure. This must be set high enough to feed the most remote mixing box. When both ducts have such controls the end-of-main differential should be small. It should be noted that many systems operate satisfactorily without static pressure controls, and better quality mixing boxes work very efficiently even with large pressure differentials.

7.4.2 Mixing Units

Mixing boxes for high-pressure systems should be selected with a high enough pressure drop for good control, without being noisy. Between one and two inches of water pressure drop is common. Most mechanical constant-volume controllers do not give satisfactory operation below about 0.75 inch of water pressure drop. Many older

systems used two-motor, constant-volume controllers and these will operate satisfactorily at pressure drops as low as 0.25 inch of water.
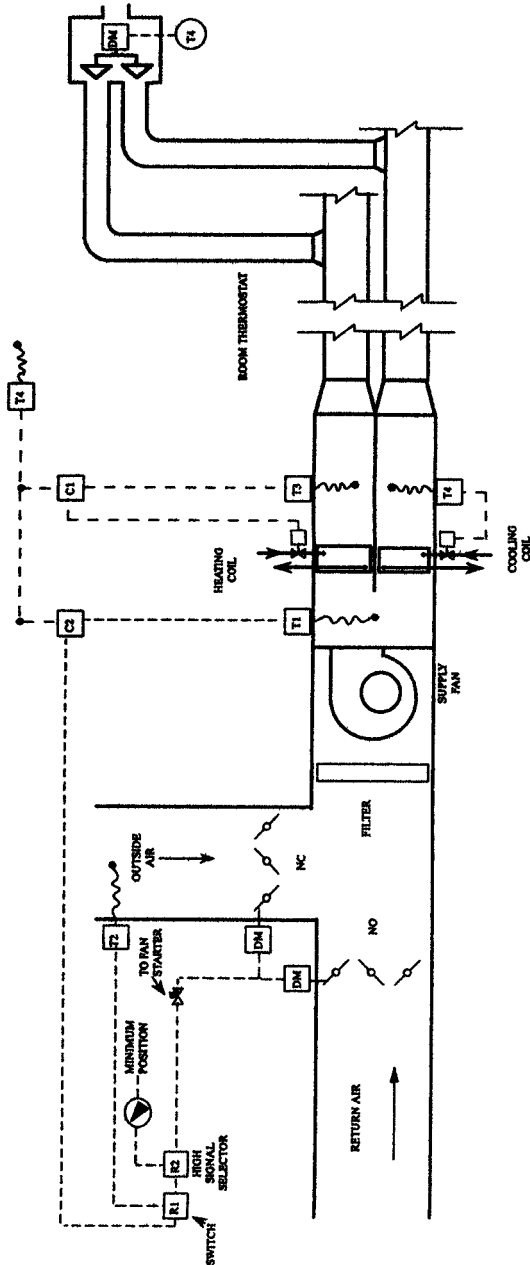

Figure 7-18. Dual-duct system.

Most systems today use mechanical constant-volume controllers. These take various forms, but are all basically spring-loaded dampers that open on a decrease in upstream pressure and modulate toward a minimum closure position as pressure increases.

Figure 7-19 shows a one-motor mechanical constant-volume mixing box control. The dampers are operated in parallel by the motor, so that as one closes, the other opens. Pressure changes that occur during the cycle, because of the pressure differential between the hot and cold ducts, are compensated for by the mechanical constant-volume controller. The room thermostat positions the damper motor to provide more hot or cold air as required.
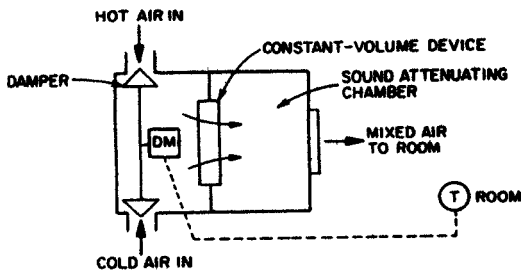
Figure 7-19. Mixing box; mechanical constant volume.

Figure 7-20 shows a two-motor constant-volume controlled mixing box. The room thermostat modulates the hot damper motor toward the open or the closed position as required to maintain room temperature. The constant-volume controller senses the change in pressure drop through the mixing box, caused by the hot damper change, and positions the cold damper to compensate and thus maintain a constant flow through the box. The relay allows the volume controller to override the room thermostat and partially open the hot air valve when the cold air valve is full open but total volume is too low. This happens occasionally on high cooling demand.
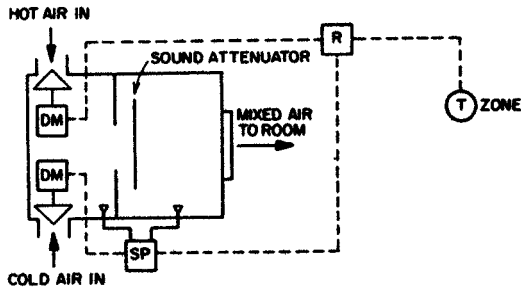
Figure 7-20. Mixing box; two-motor, constant volume.

This type of system is still used for zones larger than can be handled by a standard mechanical constant-volume mixing box.

Low-pressure mixing box systems are also used, though not widely, as they are usually limited in size because of economics and the space available for duct work. One-motor boxes without constant-volume control normally are used because available pressure drops are not sufficient for constant-volume control.

7.4.3 Dual-Duct Air Handling Unit Controls

All of the control schemes previously described for multizone systems can be applied to dual-duct systems, and each scheme has the same advantages and disadvantages. However, dual-duct systems that serve many zones are usually not suited to zone-based hot and cold deck reset schemes; there are simply too many room thermostat signals from which the reset signals must be selected.

7.4.4 Two-Fan Dual-Duct System

One variation on the conventional dual duct system uses separtate fans for the hot and cold air supplies (see Figure 7-21). This system has the advantage that an economy cycle can be implemented without imposing added heating loads. However, static pressure control must be provided for each fan.
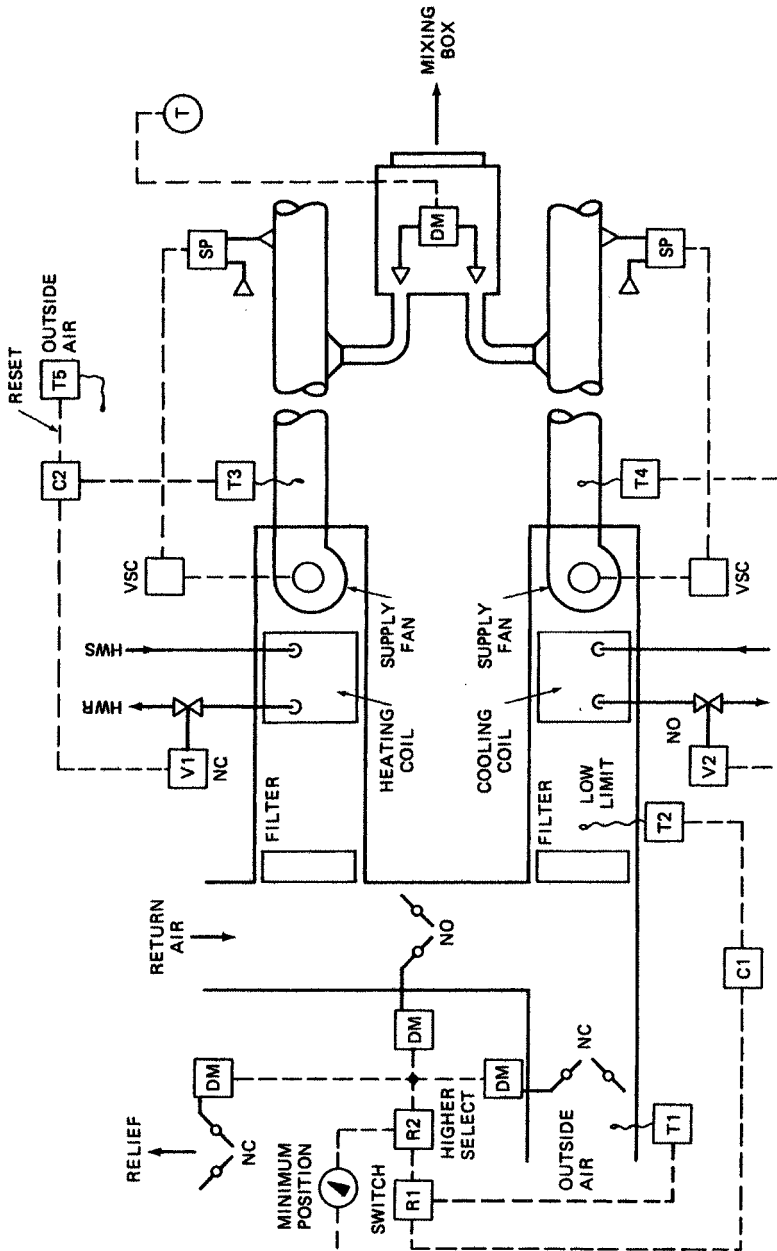
Figure 7-21. Two-fan multizone unit.

In summary, multizone and dual duct systems have the following advantages:

- Room temperature control is simple and reliable.

- One zone can be heated while another is cooled.

- Multizone systems have a relatively low first cost.

- Multizone and dual-duct systems can be more efficient than reheat systems if the heating coil is turned off during summer (and better yet if the cooling coil is turned off during winter).

Their disadvantages include the following:

- Economy cycles are difficult to implement without increasing heating energy

- Systems can be very inefficient because of the mixing of heated and cooled air.

- Hot deck and cold deck control loops require broad proportional bands in order to keep them stable. Proportional plus integral control is needed to avoid excessive offset.

- Hot deck reset (usually based on outdoor air temperature) is required to avoid excessive energy consumption.

- The most efficient control strategies require hot and cold deck reset based on the lowest and highest room thermostat signals. These control schemes are complicated, difficult to commission, and potentially unreliable.

- High-pressure dual-duct systems are no longer considered good practice, because of the high fan power requirements. However, there are many such systems in existence, installed during the 1950s and 1960s when the cost of electrical energy was very low.

Multizone and dual-duct systems were overwhelmingly popular in past designs partly because they avoided the piping and coil costs and risks of water leakage that accompany reheat systems and partly because they could be somewhat more efficient than reheat systems. They are sometimes specified in new buildings when there is a real or a perceived need for a constant-air-volume system (as opposed to the more efficient variable-volume system). Multizone systems still are sometimes used because of their low first cost or because of a designer's or an owner's prejudice against variable-volume systems.

Even though there are many multizone and dual-duct systems in use, the requirement for robust control systems was not recognized by many of their designers (PI control was rarely if ever specified, for example). Indeed, steam heating coils have been installed as part of multizone systems (steam coils are virtually uncontrollable). Many multizone and dual-duct systems that remain under stable control do so largely through the intervention of building operators who have reduced hot water temperatures, set hot deck temperatures downward, and turned heating coils off completely except during periods of very cold weather.

7.5   VARIABLE-VOLUME SYSTEMS

Variable-volume (VAV) systems provide multizone control with only a single duct. The supply air is maintained at a constant temperature, and the individual zone thermostat varies the air supply quantity to the zone to maintain the desired temperature condition. The minimum supply air quantity is usually not less than 30% of design air flow to provide sufficient ventilation. Many VAV systems do not include any heating function in the main air handling unit (except preheat when large amounts of outside air are required). Supplemental baseboard heating or reheat coils are used in exterior zones. The zone thermostat controls the VAV damper down to its minimum setting and then starts to open the heating valve if heating is required.

Varying the air flow to the space benefits system efficiency in several ways. One obvious benefit is that the total cooling load on the cooling coil diminishes more or less linearly with diminishing room loads. This is in sharp contrast to the conventional reheat system and to other constant-volume systems that add heat as the cooling load goes down.

The reduction in required cooling as the loads diminish persists until the VAV volume controls reach their minimum setting. Some minimum flow usually is required for ventilation and to maintain reasonable room air distribution. If the room VAV boxes reach their minimum position, reheating is required to keep the space comfortable. However, in contrast to a reheat system, heating is required on only a fraction of the maximum air flow rate. Because only part of the air is heated, the reheat energy requirements are substantially reduced.

Finally, the reduced air flow rates that occur when cooling loads are low or when heating is required result in substantial fan power savings.

Although reduced energy consumption is possible, VAV systems demand robust control schemes if they are to be as efficient as expected.

7.5.1 Single-Duct Variable-Volume System

Figure 7-22 shows a typical single-duct VAV system. Each zone thermostat controls its zone damper to reduce the air supply to the zone as the space temperature decreases. In zones requiring heating (exterior zones or zones that would overcool because of high minimum air volumes) reheat coils or baseboard heaters are energized on a further decrease in temperature.

Supply air temperature is controlled by a discharge temperature controller. Proportional only control, proportional plus integral control, and cold deck reset schemes are all possible. However, the control

Figure 7-22. Variable air volume control system.

scheme must perform under varying air volume flow rates, and, as with the multizone coil controll systems, stability is a concern.

For example, if the air volume flow rate over the cooling coil is only half the design maximum, then the change in valve position required to produce a given change in discharge temperature will be about half that needed at full flow. This has the effect of doubling the process gain. If a proportional only controller was used to control the cooling coil, and if the gain of the controller was set to provide stable control at full flow, the control system might be completely unstable at half flow because the overall loop gain would have nearly doubled. This problem is compounded by the more sluggish response of the temperature sensor under low-flow conditions.

The higher gains and longer time constants that result from lower flow rates require lower proportional gain settings for the coil controller. Lower gain settings mean bigger throttling ranges and a much larger droop or steady state error if proportional only, P, control is used.

Of course a large droop does not cause so drastic an increase in energy consumption with a VAV system as it does with constant-volume systems. However, the penalty can be significant, especially if the minimum flow settings for the VAV boxes are relatively high.

Proportional plus integral control for the cooling coil provides a good way of eliminating droop. Given the wider proportional band that must be used with VAV systems, PI control becomes a necessity; proportional only control may work little better than no control at all.

Economy cycle control of the outdoor and return air dampers should usually be used with VAV systems. A separate PI controller should be used for the outdoor/return air damper system.

It is also possible to reset the supply air temperature controller and the outdoor/return air damper controller from the warmest zone. Again, careful commissioning is required. The reset schedule must be set up so that the upward adjustment of the supply air temperature set point from a room thermostat takes place with the VAV box fully open and

the reheat coil valve closed. Typically, this requires that the room thermostat's throttling range be divided into three parts. The upper part is used for resetting the supply air temperature controller, the middle section modulates the VAV dampers, and the lower section modulates the reheat coil valve. In may be impractical to implement supply air reset without using digital or high-quality electronic controls.

The benefits of supply air temperature reset depend strongly on the application. If the minimum VAV box settings must be high (50% of full flow, for example) and the requirements for cooling are not dominated by zones with constant large cooling loads, then supply air temperature reset may be justified in spite of is complexity.

The variation in total air flow caused by modulating VAV boxes usually requires that one of three types of control be applied to the supply fan. The three types of control are discharge dampers, inlet guide vanes, and variable speed drives.

The goal of a pressure control system is to keep the pressure at some "representative" point in the duct system at the appropriate value. This is done to keep from bursting ductwork as the fan discharge pressure rises with decreasing flow, to keep relatively uniform pressures at the inlet to each VAV box, and to reduce the energy required by the fan.

To control the fan, a pressure sensor is placed toward the end of the main ductwork. Its signal is compared to the set point of a controller, and the output controls one of the three types of fan control devices listed above.

Of the three schemes, the discharge damper is the cheapest but poorest performer. It keeps the ductwork from bursting and helps keep the pressure constant at each VAV box. However, it simply puts a resistance to flow in the system; the fan still "rides its curve" as the VAV boxes close. Although the fan uses less energy at low flow, more dramatic energy reductions can be achieved with one of the other fan control methods.

Inlet guide vanes not only throttle the air flow, but they simultaneously change the characteristics of the fan (in part by imparting a swirling motion to the air as it enters the fan). As a result, the fan unloads more gracefully. Less power is required at reduced flow rates than when discharge dampers are used.

The most efficient fan control method is to vary the speed of the fan with a mechanical or an electrical variable speed drive unit. Basic fan laws predict that fan power will drop roughly as the cube of the volume flow rate if the duct pressure is held constant and the fan's speed is varied. That is, at half flow the fan will consume only one-eighth of the power it demands at full flow. Two basic approaches are used to vary fan speed. One allows the motor to spin at constant speed. The speed of a drive pulley is varied by varying the diameter of a pair of pulleys on an idler shaft or by a variable speed clutch. This approach is rare. The other method uses a solid-state variable speed motor controller that varies the frequency and voltage applied to the motor. Solid-state motor controllers probably are the preferred method. Their cost has dropped dramatically making them practical even for large motors.

PI control must be used to take full advantage of inlet guide vane or variable speed drive systems.

7.5.2 Dual-Duct VAV Systems

Dual-duct VAV systems are usually the result of an energy-conserving retrofit of conventional dual-duct system. They are identical to a constant-volume dual-duct system except that cold or hot air to the zone is reduced to a minimum before any blending of hot and cold air occurs. Mixing boxes must have two damper motors so that hot and cold dampers may be controlled individually. The zone mixing box dampers are controlled in sequence as shown in Figure 7-23. When full cooling is required, the cold duct damper is full open, and the hot duct damper is closed. As the cooling load decreases the cold duct damper modulates toward its closed position, but the hot duct damper remains closed until the cold damper reaches some minimum position-

usually 25% to 40% of maximum. At that point the hot duct damper begins to open while the cold duct damper continues to close. As the heating load increases, the cold damper will completely close, and the hot damper will continue to open.
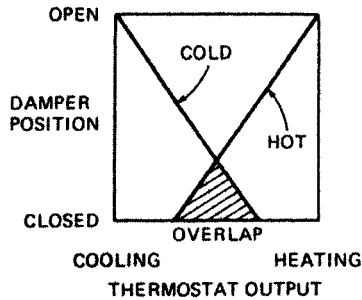
Figure7-23. Mixing box damper operation; dual-duct VAV.

7.5.3 Return/Exhaust Fan Air Volume Control

Return fans in VAV systems probably should be avoided for the same reasons as with constant-volume systems -- the positive pressure upstream of the return air dampers caused by the need to push air through the relief dampers destabilizes the mixed air control system.

Two alternatives are preferred. Recall that return or exhaust fans are needed only when economy cycles are used and when the introduction of 100% outside air would over pressurize the building. If gravity relief dampers can be provided in appropriate locations, then neither fan is needed. Also, in systems with a ceiling return air plenum, the pressure drop through the ceiling and out through the relief dampers may be low enough to avoid the need for extra fans. The first alternative, then, is to avoid either type of fan.

The second preferred approach is to use an exhaust fan (see Avery, 1986 and Avery, 1989 for details). Figure 7-24 shows the controls needed for this system. The supply fan is controlled to maintain a constant duct pressure near the end of the main ducts. Unlike the constant-volume system where the exhaust fan modulates to maintain

a constant pressure upstream of the return air damper, in VAV systems the fan modulates to maintain a constant *pressure ratio* across the return air dampers. The reason for this is best explained by an example.

Suppose that at design flow with the return dampers open the system shown in Figure 7-24 has a mixed air plenum pressure of -1 inch water gage and a pressure upstream of the return air damper of -0.75 inch water gage. Now if the outdoor air damper opens fully and the return air damper closes, the exhaust fan will have to produce -0.75 inch upstream of the return air damper to provide the required relief volume. If the total system volume now goes down to fifty percent of full flow, only -0.25 inch will be required in the mixed air plenum to bring in the required amount of air, and only one-fourth of -0.75 or -0.1875 inch of water gage will be required to exhaust half of the design return air volume.

Another feature of the control system of Figure 7-24 is the use of an outside air flow sensor and reverse-acting PI controller. This subsystem is used to guarantee that minimum outside air amounts are always delivered. If the outside air volume drops below the set point, the output from the flow controller will become the high signal and take charge of the outdoor, return, and relief dampers.

Although not preferred, return air fans are common in existing VAV systems, and several schemes have been used to control them. One scheme uses air flow measurement devices. The return fan volume is controlled to maintain an air flow rate that is less than the supply air flow rate by a constant differential. Air flow rate measurement provides accurate control, provided that the air velocities do not drop too low.

A simpler and less expensive approach is to vary the fan speed or damper position of the return fan in sync with the supply fan. This seemingly crude method may perform satisfactorily if the fans have similar characteristics, and if commissioning is done carefully.
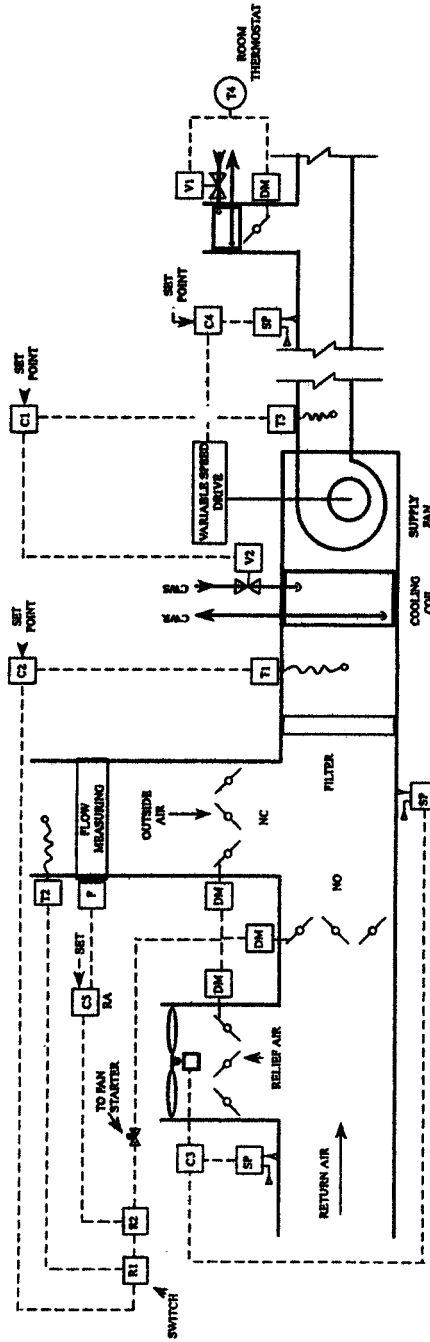
Figure 7-24. Variable air volume system with relief fan.

7.6 REHEAT SYSTEMS

Reheat systems also provide multizone control with a single duct (Figure 7-25). The air supply temperature is essentially constant at a value suitable for cooling year-round. Reheat coils for each zone are controlled by the zone thermostat to satisfy zone temperature requirements. The heating coil in the AHU may be omitted if the minimum mixed-air temperature is about 50°F. If discriminator controls are added, as in Figure 7-26, some heating and cooling energy can be saved (but see paragraph 7.3.1).

Zone reheat also may be used for controlling humidity when the dew point of the main supply air is held constant (Figure 7-27). This system is similar to the single-zone system discussed in Section 7.2.5 and igure 7-8 except that now there may be several zones, each with individual temperature control. Because the supply air dew point is the same for all zones, the actual relative humidity in a zone is determined by its dry bulb temperature and internal latent load. It follows that these parameters must be similar from zone to zone if relative humidities are to be uniform. An individual zone humidifier may be used, depending on the accuracy of control required.

7.7 HEAT RECLAIM

A heat reclaim system uses heat (or cold) that otherwise would be wasted. Heat reclaim systems may include heat pumps, runaround coil, and heat pipes. A typical heat reclaim system may use the excess heat generated in the interior zones of a building to heat the exterior zones in winter, while simultaneously using the cooling effect of the exterior zones to cool the interior. Or, where large quantities of exhaust air are required, with equivalent outside air makeup, the waste heat from the exhaust may be used to preheat (or precool) the makeup air.
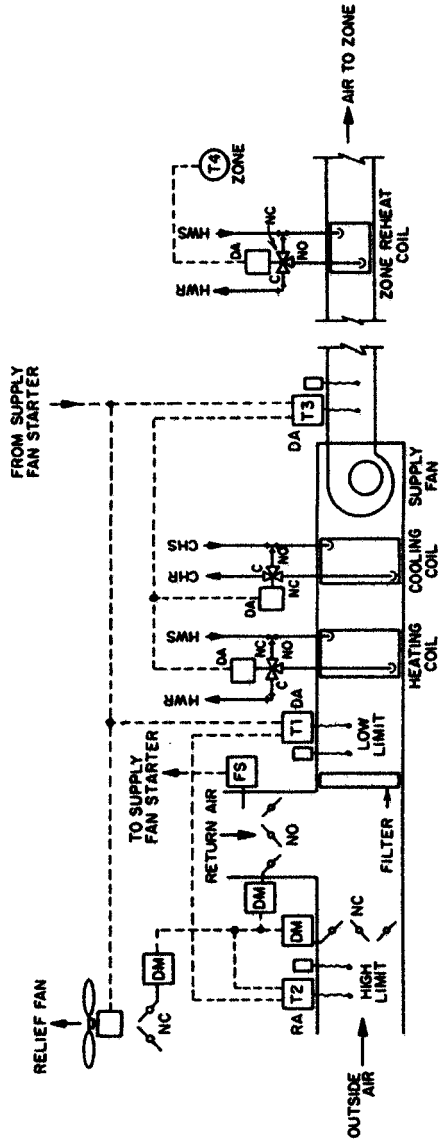
Figure 7-25. Reheat system.

Figure 7-26. Reheat system with discriminator.

Figure 7-27. Humidity control; zone reheat.

7.7.1 Heat Reclaim-Heat Pump

Reclaim systems with heat pumps move heat from interior zones that require cooling to exterior zones that require heating in winter. Figure 7-28 shows an elementary schematic of such a system. Chilled water from the heat pump cools the interior zone, the rejected heat is carried by the refrigerant to the condenser, and the condenser water is used to heat the exterior zone. As can readily be seen, this elementary idea works only when there is a balance between interior cooling and exterior heating loads; so some additional provisions are necessary.
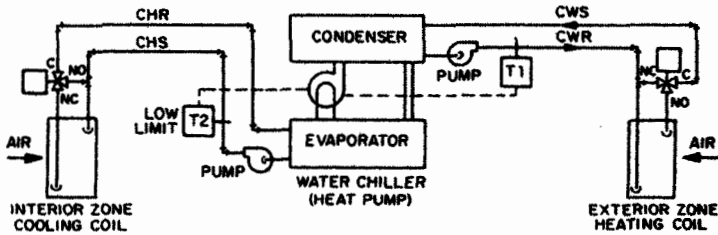
Figure 7-28. Heat reclaim; heat pump only.

One common element is a split condenser (Figure 7-29). Now, that part of the interior heat that is not needed for exterior heating can be dissipated to a cooling tower or an evaporative condenser. If a heat source, such as a boiler or an electric heater, is added for those times when exterior losses exceed interior gains, a year-round system is obtained. It is desirable, but not essential, that separate air handlers be used for interior and exterior areas.
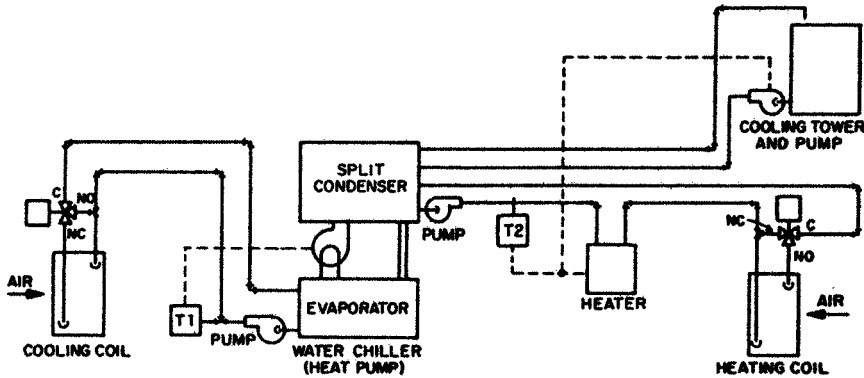
Figure 7-29. Heat reclaim; heat pump with auxiliary heat source and sink.

The interior-zone air handler, which could be any of the types previously discussed, provides cooling. The rejected heat, in the form of an increase in chilled water temperature, goes back to the heat pump, where the chilled water supply temperature is the controlled variable. The rejected heat is conveyed by the refrigerant to the condenser water, which in turn is used as a heat source for the exterior-zone air handling unit heating coil. The water temperature leaving the condenser is controlled by a thermostat that operates as follows: When the water temperature is below the thermostat setting, the thermostat calls on the auxiliary heat source for additional heating. As the water temperature increases the auxiliary source is shut down, and on a further increase, the other section of the split condenser is brought into play, dumping the excess heat to the cooling tower. In a small system this chiller-heat pump could also supply cooling for the exterior zones, but in larger systems an additional chiller (not in a heat pump arrangement) will be provided.

This arrangement is used most often in high-rise office buildings or similar structures with interior areas that comprise a large portion of the total floor area.

7.7.2 Heat Reclaim-Revolving Wheel

The revolving-wheel heat reclaim system uses a large metal wheel that revolves slowly as exhaust air passes through one side and makeup air passes through the other. Thus heat is transferred by the turning wheel. The only control required is an interlock to start the wheel drive motor when the supply fan is operating. These devices are produced by several manufacturers and are very satisfactory where the system geometry allows makeup and exhaust ducts to be run side by side.

7.7.3 Heat Reclaim-Runaround Coils and Heat Pipes

The runaround heat reclaim system uses coils in exhaust and makeup air ducts, with piping connections and circulating pumps. Although obviously more expensive than the revolving wheel, this system allows complete flexibility of system geometry and provides high efficiencies. Where freezing air is encountered, the system usually is filled with an ethylene glycol solution. Figure 7-30 shows such a system. The circulating pump is interlocked to run whenever the supply fan runs. The bypass valve and controller are used for low- limit control of the fluid temperature (35°F) to prevent frost buildup on the exhaust coil.
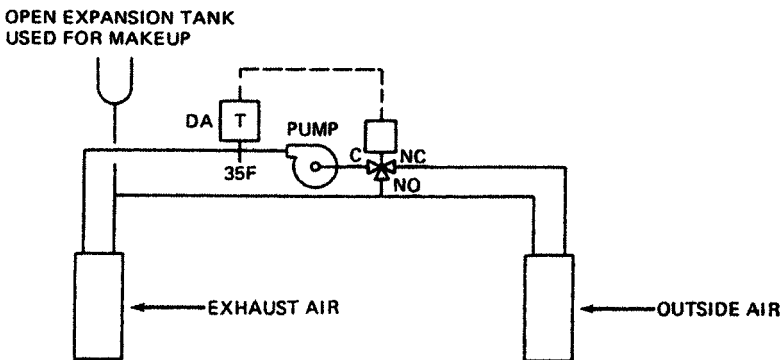
Figure 7-30. Heat reclaim; runaround coils.

A heat pipe system is similar to runaround coils except that a refrigerant is used as the heat transfer fluid. The refrigerant is boiled in the warm coil and flows to the cooler coil where it condenses, transferring heat to the cooler air stream. The fluid flows by gravity back to the hot coil. A wick is used on the inside of the tubes to help distribute the fluid over the inside of the tube by capillary action. Most heat pipe applications require that the exhaust and the makeup air ducts run side by side.

7.7.4 Heat Reclaim-Runaround Coils in Humidity Control

A special application of runaround coils for heat reclaim can be found in an HVAC system for providing low humidity in a clean room or a similar space. A typical application might be in electronic chip manufacturing where a typical space condition requirement is 75°F with 35% RH, and a high air change rate is needed to maintain a stable and clean environment. The manufacturing process also requires large amounts of exhaust air. The high air change rate means that only about a 2 to 3-degree difference is required between supply air and space temperatures; that is, the supply air temperature would be 65°F. The leaving dew point temperature required to maintain the 35% RH is 45°F. Then 20°F of reheat would be required-approximately 65% of the total cooling load. The system shown in Figure 7-31 will save a great deal of that energy by means of a simple internal runaround reclaim system. Figures 7-32 and 7-33 show the processes on psychrometric charts. For the example in Figure 7-32, a minimum of 40% outside air is assumed (for exhaust makeup), with summer design outside conditions of 95°F DB and 78°F WB. Then mixed air is at 83°F DB and 67°F WB. Cooling coil leaving air is assumed to be saturated at a 45°F dew point. To do this it may be necessary to provide low temperature brine rather than chilled water. It can be shown that a pair of ten-row runaround coils can provide about 15°F of precooling and preheating for a saving of 30,000 Btu/hr per 1000 CFM over the conventional system without runaround.

At an intermediate condition of 60°F, 80% RH outdoors the same runaround system would provide somewhat less capacity, but the

savings would still be significant (Figure 7-33). Below about 55°F outdoors the runaround system would cease to be effective and would be shut off.

In the control diagram (Figure 7-31) the runaround system is controlled by the discharge thermostat as a function of reheat requirements, with the discharge set point reset by the room thermostat. Additional conventional reheat is provided, and this will be controlled in sequence so that maximum runaround capacity is always used.

An economy cycle is provided for winter operation. The high-limit changeover point is about 56° F to maximize the mixed air temperature, as precooling furnishes the energy for reheating.

7.8 FAN-COIL UNITS

Fan-coil units are widely used in hotels, motels, apartments, and offices. Essentially, they are small single-zone air handling units with a fan, a filter, and a coil that may be used for hot or chilled water or may be split, with one row used for heating and two or three used for cooling. The former arrangement is more common because it is less expensive than the latter and requires fewer controls.

The control system may be arranged for two-pipe, three-pipe, or four-pipe supply, each of which has certain advantages and disadvantages. When a single coil is used for both hot and chilled water, hot water temperatures may be much lower than with standard heating coils. This is so because of the extra surface required for cooling at the normally low temperature difference between chilled water and leaving air. Hot water temperatures of 110°F to 140°F are typical. Fan control usually is manual, typically using a two- or three-speed switch.

Figure 7-31. AC system for low humidity using runaround coils for heat reclaim.
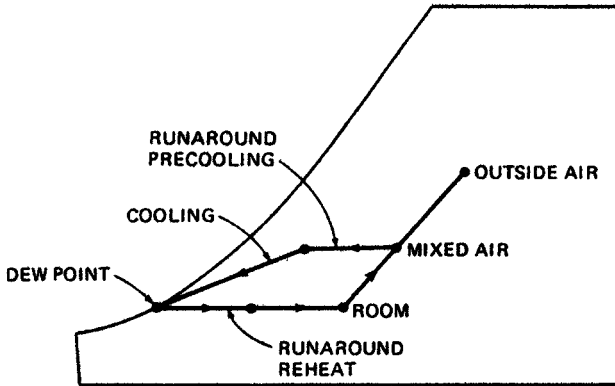
Figure 7-32. Psychrometric chart for Figure 7-33 (summer).
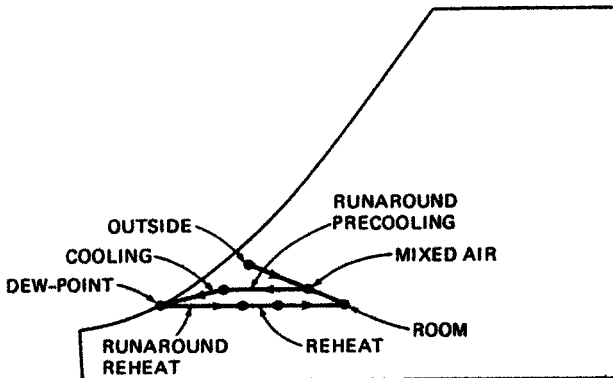
Figure 7-33. Psychrometric chart for Figure 7-33 (intermediate).

7.8.1 Two-Pipe System

Water is used, and hot or chilled water is supplied from a central plant in season. The room thermostat must be a summer-winter type; that is, it is direct-acting in winter, reverse-acting in summer when used with a normally open valve. Changeover from one to the other is done by a general signal, such as by changing the main air pressure in a

pneumatic system, by sensing the change in supply water temperature, or by a "Heating/Cooling" switch on the fan coil unit control panel. The thermostat may provide modulating or two-position control of the water flow valve.

Another approach is simply to start and stop the fan while the water flow is uncontrolled. This scheme usually is unsatisfactory during the heating season because the fan coil unit continues to act like a convector, adding heat to the room even when the fan is off.

Figure 7-34 shows a common arrangement with manual multispeed control of the fan and thermostatic control of the water flow to the coil. With chilled water being supplied and the room temperature above the reverse-acting thermostat setting, the normally open valve is open, allowing the unit to provide cooling. As the room temperature falls the valve closes. On a further fall in room temperature, no heat is provided because hot water is not available.

With hot water being supplied the room thermostat is direct-acting, and the reverse cycle takes place. Now, of course, no cooling is possible. It is apparent that this arrangement can cause loss of control in mild weather, when heating is required in morning and evening but cooling is required during the day.

A central plant for a two-pipe system includes a boiler or other heat source, a chiller, circulating pumps, and changeover controls. Changeover may be manual or automatic, but in either case involves problems.
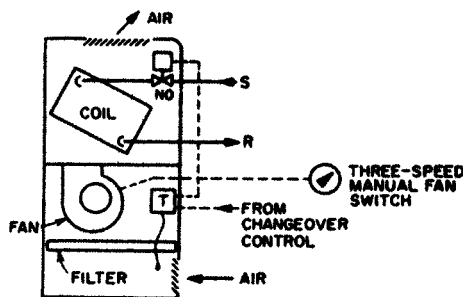
Figure 7-34. Fan-coil unit; two-pipe system, manual fan control.

The principal difficulty arises on changeover from heating to cooling and vice versa. Warm water, even at 75°F or 80°F may cause high suction pressures for the chiller compressor and overload the motor. These same temperatures can cause thermal shock and flue gas condensation in the boiler.

The recommended system shown in Figure 7-35 has heat exchangers and secondary pumps to isolate the chiller and the boiler from the distribution piping. This allows the chiller and the boiler to operate within safe limits and the distribution water temperature can be modulated through a wide range as outdoor temperature varies. This system is not very economical and is difficult to control properly in spring and fall when heating is needed in the morning, and cooling is needed in the afternoon.

7.8.2 Three-Pipe System

Water is used, with separate hot and chilled water supply pipes and a common return pipe. Individual heating and cooling valves or a special three-pipe valve may be used. A standard direct-acting thermostat is required. Because it is possible, in season, to provide a choice of heating or cooling, this system is much more flexible than the two-pipe system. Mixing the return water, however, means that when both heating and cooling are being used the return water is at some intermediate temperature, and a false load is placed on both the chiller and the boiler, with a resultant increase in operating cost.

Figure 7-36 shows a typical three-pipe fan-coil arrangement, using a special three-pipe valve. When the room temperature is below the thermostat setting, the heating port of the three-pipe valve is open. As the room temperature increases, the hot water port closes. There is then a "dead spot" over a small temperature range, during which both ports are closed. As the room temperature goes still higher, the chilled water port opens. Chilled or hot water flows out to the common return main where it is mixed with the chilled or hot water return from other units and conveyed to the central plant.
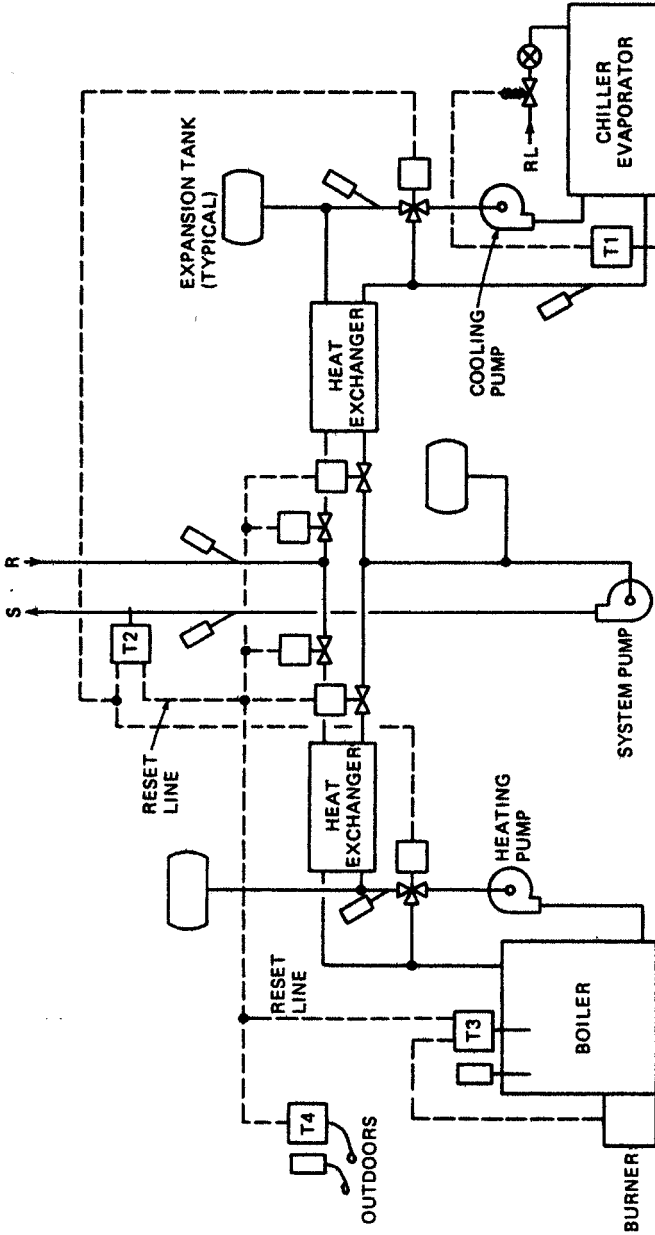
Figure 7-35. Two-pipe system; central equipment and control.
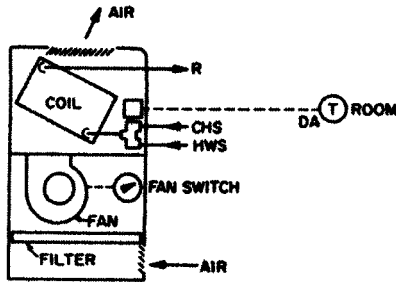
Figure 7-36. Fan-coil unit; three-pipe system, manual fan control.

A central plant for a three-pipe system includes a boiler or other heat source, a chiller, a circulating pump, and temperature controls. Because the water flows through the boiler and the chiller are determined by the sums of the individual fan-coil usages, the system is self-balancing in this regard. The return water temperature is directly related to load, increasing when heating demand is increased and decreasing when cooling demand is increased. This seeming anomaly results from the fact that return hot water is hotter than return chilled water. Because the flow through the chiller will decrease as the return water temperature rises, there is little need for the elaborate temperature-flow controls discussed under two-pipe systems. However, a sudden increase in cooling load can cause problems. Also, low-temperature water going through the boiler can cause condensation of flue gases in the boiler, with resulting corrosion. For this reason, a heat exchanger is recommended for two- and three-pipe systems requiring hot water to be supplied at temperatures below 140°F. Figure 7-37 shows a typical three-pipe central plant control arrangement. The individual water supply thermostats control the heat exchanger and chiller capacities. A flow switch on the chiller will shut off the compressor if the water flow falls below the minimum rate necessary to prevent freezing. To avoid this kind of shutdown it is preferable to add a heat exchanger and secondary pump for the chiller as shown in Figure 7-35.
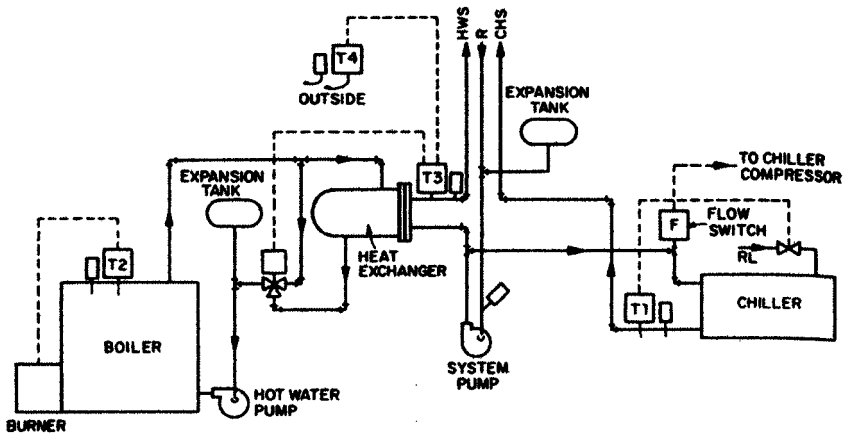
Figure 7-37. Three-pipe system; central equipment and control.

7.8.3 Four-Pipe System

Four-pipe systems provide complete segregation of the heating and cooling media, thus avoiding the problems of two- and three-pipe installations. The normal coil configuration is a split arrangement with one row used for heating and two or three used for cooling. Sometimes two separate coils are used.

Figure 7-38 shows a typical split-coil control arrangement. With the room temperature below the setting of the direct-acting thermostat, the normally open hot water valve is open, and the normally closed chilled water valve is closed. As the room temperature increases, the hot water valve closes. The controls should he adjusted to provide a "dead spot" where both valves are closed. As the room temperature rises still further, the chilled water valve opens. On a fall in temperature, the sequence is reversed.
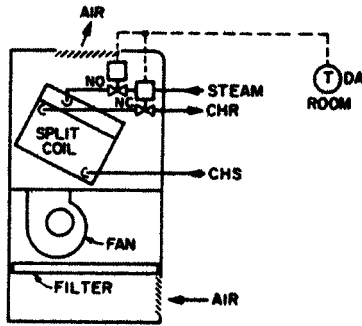
Figure 7-38. Fan-coil unit; four-pipe system, split-coil.

It is also possible to use a single coil for either hot or chilled water while keeping the piping systems segregated. This requires the use of two "three-pipe" valves. Figure 7-39 shows such an arrangement. When the room temperature is below the setting of the thermostat the inlet and outlet valves are open to the hot water supply and return lines. As the temperature increases, the inlet valve heating port may modulate toward the closed position, but the outlet valve heating port will remain full open to minimize the pressure drop. When the inlet valve is fully closed, then the outlet valve will close, using two-position action. Again, there will be a period when both ports are closed and the room temperature is at or near the thermostat setting. On an increase in room temperature above the setting, the inlet and outlet valve cooling ports will modulate toward open, or open using a two-position sequence.
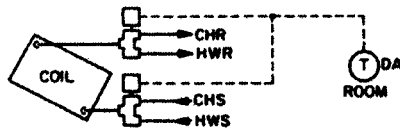
Figure 7-39. Fan-coil unit, four-pipe system, single-coil.

7.9 INDUCTION SYSTEMS

An induction unit depends for its operation upon a central source of temperature-controlled high-pressure air, which induces a secondary

flow of room air across the unit coil. This coil is supplied with hot or chilled water through a two-, three-, or four-pipe system. Thus it is necessary to control the primary air supply temperature, the temperature of the hot or chilled water, and the flow of this water.

Because condensate drains are not usually provided on induction unit coils, the chilled water supply temperature to the induction units must be kept above the dew point of the entering air. Therefore, most induction unit systems are arranged with water flowing first through the central air handling unit coils and then, in series, through the induction unit coils. Thus, chilled water may enter the central unit coil at 45°F and leave at 55°F, which is good for supplying the induction unit coil. Because a single coil is used for both heating and cooling, hot water supply temperatures can be comparatively low, on the order of 140°F or less.

Figure 7-40 shows an induction unit system with four-pipe control. Heating and cooling coils in the primary air unit are supplied with hot and chilled water from a central boiler (or other heat source) and chiller.

The primary supply air temperature is controlled at a value that varies with outdoor temperature according to some reset schedule. The chilled water leaving the cooling coil is returned to the chiller or passes into a secondary chilled water system that has its own pump and control. The supply water temperature in this secondary system is controlled at about 55°F by a thermostat that operates a three-way valve to introduce water from the primary system coil or recirculate return water from the secondary system. The secondary hot water system operates in the same way.

The individual induction unit coils provide heating or cooling as required in response to the demand of a room thermostat as detailed in the discussion of the fan-coil unit. Notice that this heating or cooling may supplement or counteract the primary air action, depending on the season and internal and solar loads.
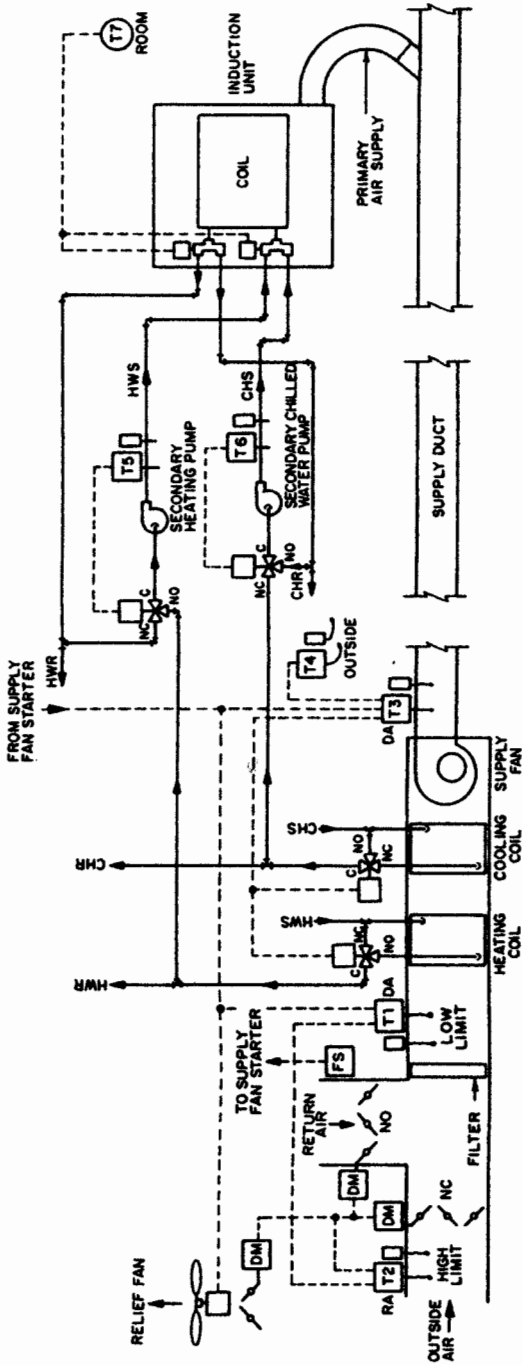
Figure 7-40. induction unit

Induction units are commonly used on exterior walls and below windows; there they can furnish individual control for small zones with highly variable solar and other loads. They are not normally used for interior zones because of their small size.

A special type of induction unit sometimes is used in interior zones. This unit is installed in a return air plenum above the ceiling and uses the heat of the return air and light fixtures to temper the primary cold air supply for interior zone control.

7.10 UNIT VENTILATORS

A unit ventilator is similar to a fan-coil unit but is usually larger and is arranged to provide up to 100% outside air through an integral damper system. This unit is designed primarily for heating, though some unit coils are sized for cooling also. Their primary use is in school classrooms or similarly heavily occupied spaces, where large amounts of ventilation air are required. Control systems for unit ventilators have been standardized by ASHRAE and are detailed in the Handbook. The principal elements of the control system are the room thermostat, the discharge air low-limit thermostat, the damper motor, and the steam or water control valve. (Electric heat sometimes is used.) When cooling is also provided, the coil control will be similar to that described for fan-coil units with the addition of outside air control. This could be as shown in Figure 7-41.
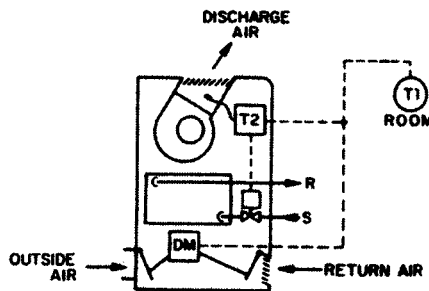
Figure 7-41. Unit ventilator.

On the heating cycle the thermostat is direct-acting. With room temperature below the thermostat setting the outside air damper is closed or open to a fixed minimum position, the return air damper is open, and the normally open water valve is open. Hot water is being provided by the two-pipe system. As the room temperature increases, the outside air damper modulates toward the open position. The discharge air low-limit thermostat will keep the valve open as required to maintain a minimum discharge air temperature of about 60°F. When the room temperature reaches or exceeds the room thermostat setting the outside air damper will be fully open, and the valve will be closed unless some heating is required to maintain the discharge air temperature low limit.

For cooling, the changeover control changes the room thermostat to reverse-acting and chilled water is supplied by the central system. Now, with the room temperature above the room thermostat setting, the outside air damper is closed or open to a fixed minimum position, the return air damper is open, and the water valve is open. As the room temperature decreases, the outside air damper modulates toward the open position, and the valve modulates toward the closed position. The discharge air thermostat must be bypassed because it would try to open the valve at low discharge air temperatures, nullifying the effect of the room thermostat. Alternatively, the discharge thermostat could be changed to reverse-acting, with the set point changed to about 75°F as a high limit.

7.11 PACKAGED EQUIPMENT

Packaged equipment, by our definition, is HVAC equipment that is factory-assembled and ready for installation in the field with a minimum of labor and material. As such it includes a complete, ready-to-function control system, usually electric but often with electronic components. Typical packaged equipment items are residential furnaces and air conditioners, rooftop units, and direct gas-fired heating equipment.

Because most of these units include the controls, they will be discussed only briefly, to point out typical arrangements.

7.11.1 Self-Contained Fan-Coil Unit

The small, self-contained fan-coil unit is widely used in motels, hotels, and small apartments. It is basically the same as a window-type air conditioner but with the form of a fan-coil unit. It contains a supply fan, a direct-expansion cooling coil, a refrigeration compressor, an air-cooled condenser with through-the-wall ducts for the air, an electric heating coil, a filter, and controls. A return air type of thermostat is often used, so that only power and outside air connections are needed to make the unit operable. Controls are simple, electric and conventional. The thermostat is direct-acting, two-position, calling for cooling as the room temperature increases above the set point and heating when it decreases below the set point. A fairly wide differential is provided to prevent short cycling of first heating and then cooling. A manual changeover summer-winter thermostat may be used. Fan control is manual, two- or three-speed.

7.11.2 Residential Air Conditioning

A typical residential central air conditioning system has a gas, stoker, or oil-fired furnace within the house. This furnace has an add-on direct-expansion cooling coil installed in the discharge. Field-installed refrigerant pipes connect the coil to an air-cooled condensing unit outdoors. A two-position summer-winter thermostat with manual changeover is provided. Typically, this thermostat starts and stops the supply fan, as well as controlling the heating and cooling. A manual fan switch usually is furnished. On the heating cycle, the fan is started and stopped by a discharge plenum thermostat, to prevent blowing cold air. A high-limit thermostat acts to shut down the burner if the high-limit temperature setting is exceeded.

Figure 7-42 shows a year-round residential air conditioning system with gas-fired furnace and air-cooled condensing unit. The thermostat has manual changeover from heating to cooling and has an "on-automatic" fan switch. In the "cooling" position, when the room temperature rises above the thermostat setting, the condensing unit and supply fan are started simultaneously, and the solenoid valve on the cooling coil is opened. The condensing unit is equipped with high- and low-pressure safety controls and thermal overload (not shown). Many units of this type and size do not use the pump-down cycle. If the pump-down cycle is not used, the solenoid valve is not needed, and the thermostat simply starts the condensing unit and fan on a call for cooling.

With the changeover switch in the "heating" position, on a call for heat the gas burner will light (provided that the pilot safety is satisfied). When the plenum temperature rises to its set point, the fan switch will start the supply fan. A high-limit thermostat (usually combined with the fan start thermostat) will shut off the burner if necessary.

Figure 7-43 shows an oil-burning furnace. Cooling controls could be provided as shown for the gas-fired furnace. On heating, a call for heat starts the oil burner motor (or, with a gravity-type burner, opens a valve) and energizes a time-delay relay in a "stack switch." This thermostat is mounted in the combustion vent stack and must be opened by the rise in temperature of the flue gas (indicating proper combustion) before the time-delay relay times out. Thus, if the oil fails to ignite, the burner will be shut off. The switch then must be manually reset. Some models of this device provide for one recycle operation to purge unburned gases and try again for ignition before finally locking out.

7.11.3 Residential Heat Pumps

In areas where natural gas is not available, and particularly in the south where winters are mild, small (2 - 5 ton) residential heat pumps are popular. A typical schematic and control system is shown in Figure 7-

44. Controls, of course, are part of the package. Peculiar to this equipment are the reversing valve and defrost controls.
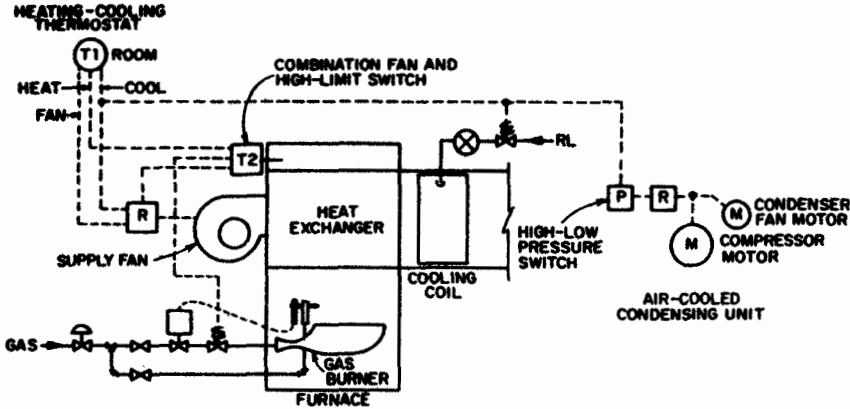
Figure 7-42. Residential gas-fired furnace with DX cooling.
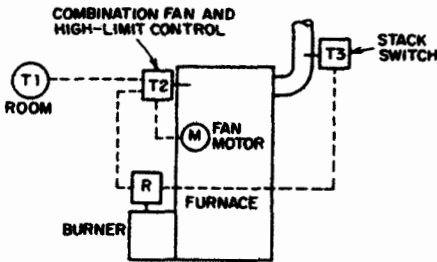
Figure 7-43. Oil-fired furnace.

The reversing valve is a four-way solenoid. It must be very carefully designed to prevent leakage between the high- and low-pressure gas streams that flow through its two sides simultaneously. The sealing problem is further complicated by the large temperature differences between the two gas streams.

When the heat pump is operating to provide indoor heating, icing of the outdoor coil is common. Therefore, some kind of defrost cycle is necessary to periodically remove the ice. Several methods are used, including the following:

CHECK VALVE

REVERSING VALVE   EXPANSION VALVE

OUTDOOR COIL   INDOOR COIL

COMPRESSOR

COOLING CYCLE

CHECK VALVE

REVERSING VALVE   EXPANSION VALVE

OUTDOOR COIL   INDOOR COIL

COMPRESSOR

HEATING CYCLE
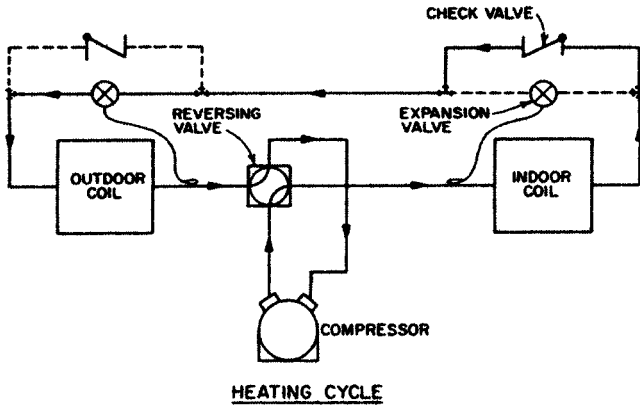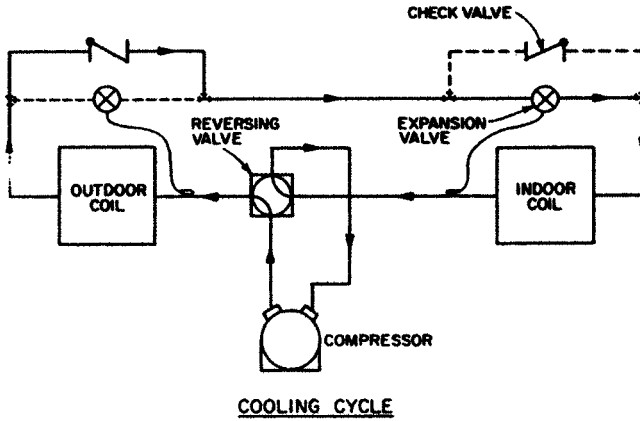
Figure 7-44. Residential heat pump

- A time clock provides a regular defrost cycle. This operates regardless of ice buildup.

- A sensor notes the increase in air pressure drop through the coil due to icing and operates the defrost cycle until normal conditions are restored.

- A refrigerant temperature or pressure sensor senses the decrease in suction pressure or temperature caused by icing.

- A differential temperature controller senses the increasing difference between outside air and refrigerant temperatures caused by icing.

In mild climates, these systems use air defrost, allowing the fan to run while the compressor is shut down. Units that may be required to operate in ambient temperatures below freezing must have electric heaters for defrost, or use occasional short periods of indoor cooling operation so that hot refrigerant will defrost the outdoor coil.

7.12 OTHER PACKAGED EQUIPMENT

"Rooftop" units are packaged heating and cooling systems (or sometimes heating and ventilating only) that combine the simplicity of direct firing with the complexity of economy cycle outdoor air control, relief fans, and even multizone supply. They are made with a variety of styles, fuels, and arrangements, and with capacities of up to 100 tons of cooling and several hundred thousand Btu's of heating. Controls are factory-furnished and operate in many of the ways discussed previously.

Boilers, and other direct-fired heating units, always include the controls as part of a complete package approved in its entirety by some national certification agency, such as the American Gas Association (AGA) or Factory Mutual (FM). Controls on the equipment cannot be changed in any way without voiding this approval.

Common features in control systems of large gas- and oil-fired boilers are: flame-failure safety shutdown; electric ignition with checking before startup; prepurge, to remove any unburned gases from the combustion chamber before startup; post-purge, to do the same after shutdown; and low-fire startup to ensure proper operation before the burner is brought to full capacity.

Controls also may include air-fuel ratio metering and measurement of O_2 or CO to improve combustion efficiency. Power boiler control diagrams typically employ ISA symbols rather than the HVAC symbols used in this book.

Boilers of any size often are interlocked to the water pump or a water flow switch, to ensure water flow before the burner is turned on.

Makeup air units are direct gas-fired units without vents. The products of combustion are mixed with the 100% outside ventilation air supplied and are not considered hazardous because of the high dilution rate. Their use is restricted to industrial applications and commercial establishments with low personnel occupancy. Control usually is achieved by means of a two-position room thermostat and gas valve, combined with a self-contained modulating valve with the sensor bulb in the discharge air. (See Figure 7-45.) It often is necessary that they be interlocked with companion exhaust systems, and in some areas they may not be used as the primary source of heating.
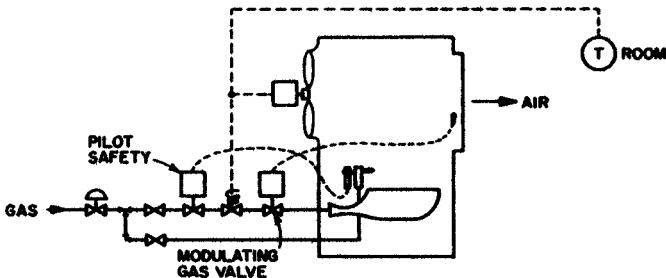
Figure 7-45. Makeup air unit; gas-fired.

7.13 RADIANT HEATING AND COOLING

By definition, radiant heating and cooling depend for their effectiveness on the radiant transfer of heat energy. There are many

varieties of radiant heating systems, ranging from low-temperature/high-mass concrete floors to high-temperature/low-mass outdoor systems with gas-fired ceramic or electric elements. No single type of control works for all radiant systems, but several distinct groupings have similar controls.

7.13.1 Panel Heating

Floor, wall, and ceiling panels are most commonly used for general heating. Floor panels with hot water pipes or electric cables embedded in concrete have a large mass, and therefore, a large time lag. A conventional room thermostat may not provide adequate control. A room thermostat with outdoor reset, or even an outdoor thermostat alone, has proved more satisfactory than the conventional thermostat. It is essential to limit the panel surface temperature to a maximum of 85°F. This can most easily be done by limiting the water supply temperature. Figure 7-46 shows a floor heating panel controlled by a room thermostat with outdoor reset and a high-temperature-limit thermostat in the water supply.

Wall and ceiling panels generally have much less mass than floor panels and so can be controlled directly by a room thermostat. Although surface temperatures up to 100°F in walls and 120°F in ceilings are acceptable, they still require supply water temperature control. The control system would look like Figure 7-46 without the outdoor reset (although reset still could be used to improve response).
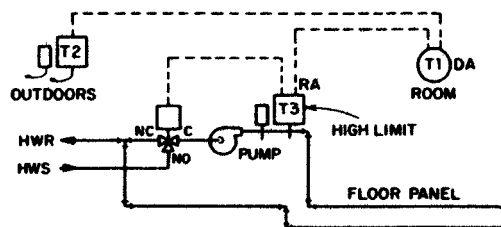
Figure 7-46. Floor panel heating; hot water.

Wall and ceiling panels also can be used for cooling. Water supply temperatures must be carefully controlled to avoid surface temperatures below the room dew point. Because this type of cooling is always supplementary to an air system, the water supply can be handled in the same ways discussed for induction units.

7.13.2 Direct Radiant Heaters

Direct-fired, high-intensity infrared heaters may use natural gas or electricity, and are used for general heating. They also are used for spot heating, indoors and out. Unless they are used for general heating, control should be manual because of the inability of a conventional thermostat to sense "comfort" when heat transfer is primarily radiant.

7.14 RADIATORS AND CONVECTORS

Convectors and convector radiators in various forms are widely used for basic heating and as supplementary heaters with air-conditioning systems. This general classification includes fin-pipe, baseboard convectors, modern European style radiators/convectors, as well as the old-fashioned cast-iron column "radiator." This equipment may use steam at low pressure or under vacuum, or hot water.

7.14.1 Low-Pressure Steam Systems

The low-pressure steam supply may be controlled by individual control valves on each radiator or by zone control valves that supply several radiators. In the latter case orifices are used at each radiator to ensure proper distribution of the steam, and two-position control is necessary. Steam pressure at the boiler is kept essentially constant by a pressure controller that operates the burner. Self-contained radiator valves, requiring no external power source, are widely used.

7.14.2 Hot Water Systems

Hot water may be controlled in the same manner as steam. Series flow through a group of radiators may be used if the design and the selection allow for a water temperature drop from the first radiator to the last.

7.14.3 Vacuum Systems

Vacuum systems (no longer common, antique!) are "controlled" by varying steam pressure and temperature to suit the building heating requirements. In a typical system (Figure 7-47) the zone thermostat modulates the steam supply valve in response to demand. The vacuum pump is operated by a differential pressure-vacuum controller, to maintain a constant differential between supply and return mains. Outdoor reset of the zone thermostat may be provided to prevent overheating. A constant boiler steam pressure is maintained by a pressure controller. Vacuums as low as 25 or 26 inches of mercury are normal in mild weather. A zone may be the entire building, or individual zone supply valves and vacuum pumps may be used.
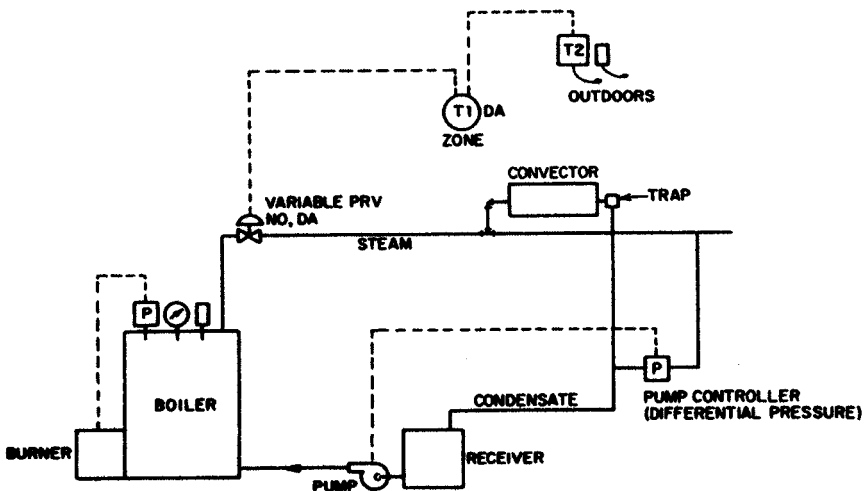
Figure 7-47. Vacuum heating system control.

7.15 HEAT EXCHANGERS

Heat exchangers as heat sources have been mentioned several times. Heat exchangers are required when a large central plant supplies high-temperature water or high-pressure steam, neither of which can be conveniently or safely used in the individual building HVAC equipment. As noted previously, heat exchangers also may be used to provide very-low-temperature heating water and thus protect the boiler from thermal shock and/or condensation of flue gases. Obviously, the control system varies with the function of the heat exchanger.

7.15.1 Heat Exchanger, Low-Pressure Steam

Figure 7-48 shows a simple heat exchanger control system, suitable for producing water in the 100°F to 200°F range using low-pressure steam or medium-temperature (up to 260°F) boiler supply water as a heat source. The valve is controlled by the low-temperature supply water thermostat, which may be reset by an outdoor thermostat if so required. An interlock from a water pump or a water flow switch keeps the valve closed when water is not flowing. Modulating or two-position control may be used.
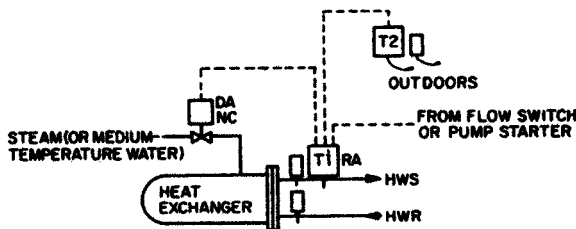
Figure 7-48. Hot water heat exchanger.

7.15.2 Heat Exchanger, High-Pressure Steam

When high-pressure steam is the heat source some additional considerations are necessary. The control valve must be made of materials suitable for the temperatures and the pressures encountered. Relief valves should be provided on both the steam and water sides of the exchanger, and a high-limit water controller is required. (See Figure 7-49.) Two-position control is not satisfactory because the high temperature differentials. Modulating control must be used, and even then there is a tendency to instability and hunting unless a wide throttling range is used. Some problems may be minimized by using a pressure-reducing valve in the steam supply line. This makes the flow rate easier to control, but the large temperature differences remain. Care must be exercised in designing and applying these control systems.
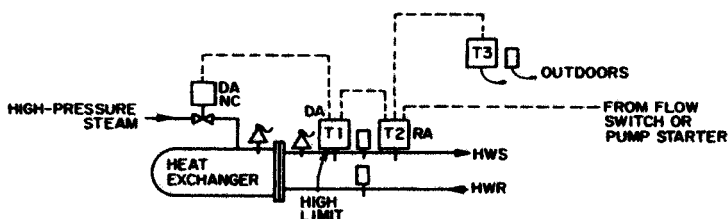
Figure 7-49. Heat exchanger; high-pressure steam.

7.15.3 Heat Exchanger, High-Temperature Water

High-temperature water (HTW) poses even greater problems than steam to the designer of heat exchanger controls. A typical HTW central plant produces water at 400°F. To keep the water liquid at this temperature, the system must be pressurized to 350 psi or higher. These temperatures and pressures require careful selection of the control valves and piping materials.

An additional consideration is that, to save pumping and piping costs, the high-temperature water must have every possible Btu extracted from it at the point of use. That is, it must be cooled as much as possible, 250°F being a typical return water temperature. Heat exchangers thus are selected for small final temperature differentials

between the high-temperature water and the fluid being heated. "Cascading" is common, with low-temperature heat exchangers using the HTW last in a sequence that may start with low-pressure steam generators and go to building heating water exchangers.

Relief valves are required, and control valves are commonly applied on the leaving side of the exchanger to minimize operating temperatures except for domestic hot water generators, where safety requires the control valve to be on the upstream side. Modulating control is used.

Figure 7-50 shows a typical set of HTW exchangers for a large building. The HTW temperatures illustrated are those that will exist at design loads and water flows. In operation they will, of course, vary somewhat. Notice the "cascade" effect from the steam generator to the building heating converter. So long as the steam generator is at full capacity, enough water will flow through valve V1 to provide full capacity to the convertor. If the convertor is at part load, some water will be bypassed through three-way valve V3, and the HTW return temperature will be greater than 215°F. If the convertor load is high and the steam generator load is low, then T2 will sense the falling building hot water supply temperature and open valve V4, providing additional HTW. Valve V4 is selected with a high pressure drop at design flow to match the pressure drop through the steam generator and valve V1.

The domestic hot water generator is not in the cascade, because its loads are so different from those of the other devices. The manual bypass valves may be used in case of difficulty with the automatic controls.

Control of expansion and maintenance of system pressure are critical. A drop in system pressure results in a lower water temperature, creating a cycle that causes shutdown of the plant. For a full discussion of this phenomenon and methods of control refer to the ASHRAE Handbook, Systems volume. One method of controlling expansion, pressure and make up is shown in Figure 7-51. The expansion tank (or two or more tanks in parallel) is sized to suit the system, with a wide

differential between high-level drain control and low-level makeup control. The steam boiler provides a steam cushion to maintain the required pressure. The makeup pump or pumps will have a small flow rate at a high head. The makeup water meter is useful for noting unusual makeup requirements that would show system leaks or malfunctions. A backflow preventer (not shown) is needed on the makeup water.
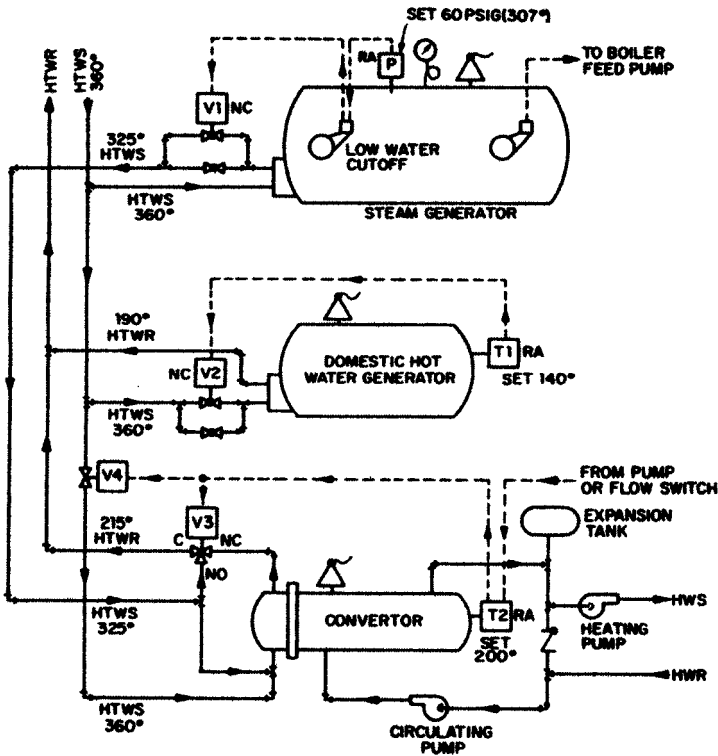
Figure 7-50. High-temperature end-use controls.

7.16 SOLAR HEATING AND COOLING SYSTEMS

Solar energy systems come in a variety of arrangements and usages but all have three essential elements in common-collector, storage, and distribution.
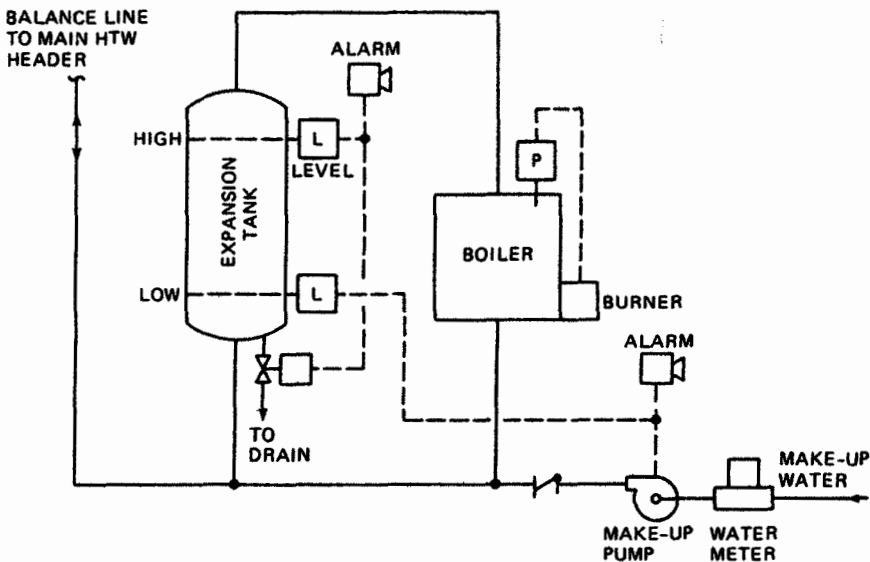
Figure 7-51. High temperature water expansion-pressurization system.

7.16.1 Elementary Solar Heating System

The elementary solar heating system is shown in Figure 7-52. A water-to-water system is shown, but air-to-air and water-air systems also are used. The differential thermostat allows the collection pump to operate only when the collector temperature is at least 4 to 5 degrees warmer than the storage. In some water-to-water systems the collector loop will use a propylene glycol solution to prevent nighttime freezing.

The space thermostat operates the distribution pump as required to maintain space temperature, and the low-limit thermostat operates the auxiliary heating system when usage exceeds storage.
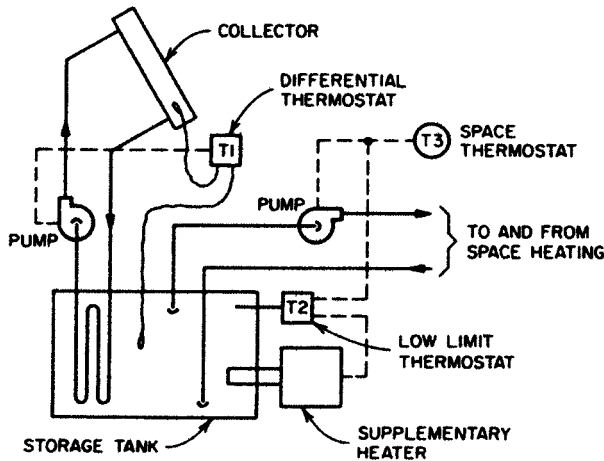
Figure 7-52. Elementary solar heating system.

7.16.2 Self-Draining Collector

The self-draining collector shown in Figure 7-53 is frequently used for domestic water heating, using a heat exchanger as shown and eliminating the need for a glycol loop with its possible cross-contamination. When the collector pump stops, all the fluid in the collector loop drains into a space in the drain back tank, thus eliminating the risk of freezing. The differential thermostat is the only control needed. Another implementation of the design eliminates the storage side pump and works by natural convection flow on the storage tank side. This leads to good storage tank stratification and exceptional thermal performance.

7.17 SUMMARY

In the preceding discussions the energy source generally has been ignored. Obviously, many functions, such as direct control of motors

and solenoid valves require electrical energy. The next chapter considers the methods of describing electric control circuits and interfacing those circuits with other types of energy.
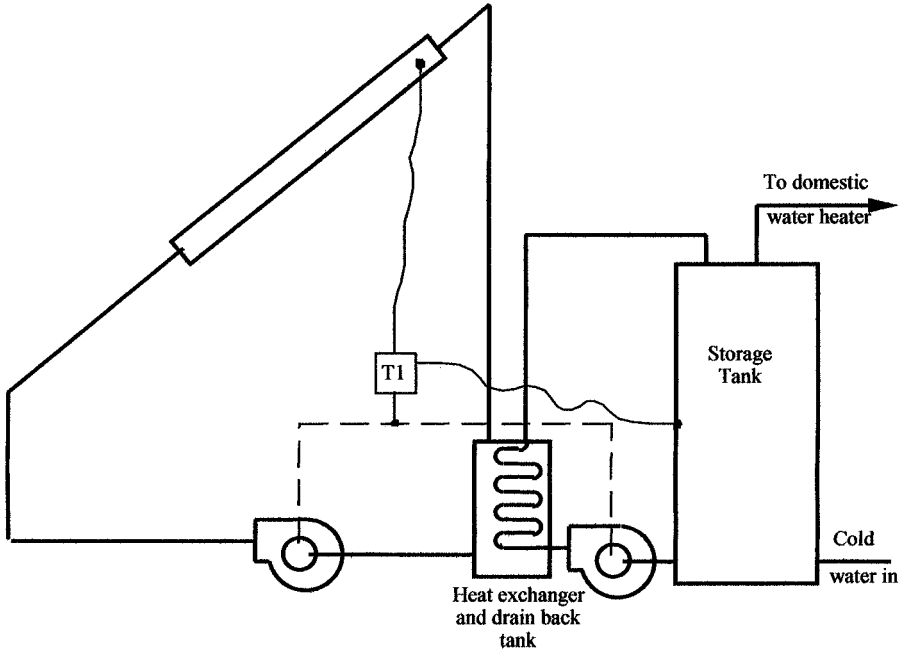
Figure 7-53. Drain back water heating system.

8

Electric Control Systems

8.1 INTRODUCTION

Control of HVAC systems includes starting and stopping electric motors for fans, compressors, boilers, pumps, and accessories. All too often the system designer thinks that motor control is the responsibility of the electrical engineer.

Obviously, the electrical engineer must size and select the wire, conduit, starters, disconnects, and switchgear necessary for supplying power and control to the motor; and this information must appear on the electrical drawings for the benefit of the electrical contractor who is to install it. However, control designers must specify the necessary interlocks to the HVAC control system. So we must learn how to express electrical equipment and wiring diagrammatically using standard electrical symbols and interfacing between the electrical and temperature control systems with proper relays and transducers.

8.2 ELECTRIC CONTROL DIAGRAMS

Consider some conventions and symbols used to convey electrical information. Figure 8-1(A) is a point-to-point or graphic schematic of a motor starter with momentary contact pushbuttons for manual start-stop control. The same schematic is shown in ladder form in Figure 8-1(B). Either of these forms is acceptable, but the ladder schematic is more frequently used and is easier to follow.

The symbol 1M over the circle in Figure 8-1(B) designates the solenoid coil in the starter, which, when energized, actuates the power and

auxiliary contacts. The related contacts are identified by the same number, especially when spread out in the ladder schematic. The symbol –| |– shows a normally open contact. That is, when the coil is not energized, the contact is open; when the coil is energized, it closes. The symbol –|/|– shows the opposite or normally closed contact. The symbol ⌇ indicates the heater portion of the thermal overload relay in the starter. Next to each in the graphic schematic is a normally closed contact. When excessive current is drawn by the motor, for any reason, the heater heat ups and opens the contact. After the trouble is found and corrected, the contact may be closed (reset) manually. Various types of overload relays are available, depending on the size and the voltage of the motor.
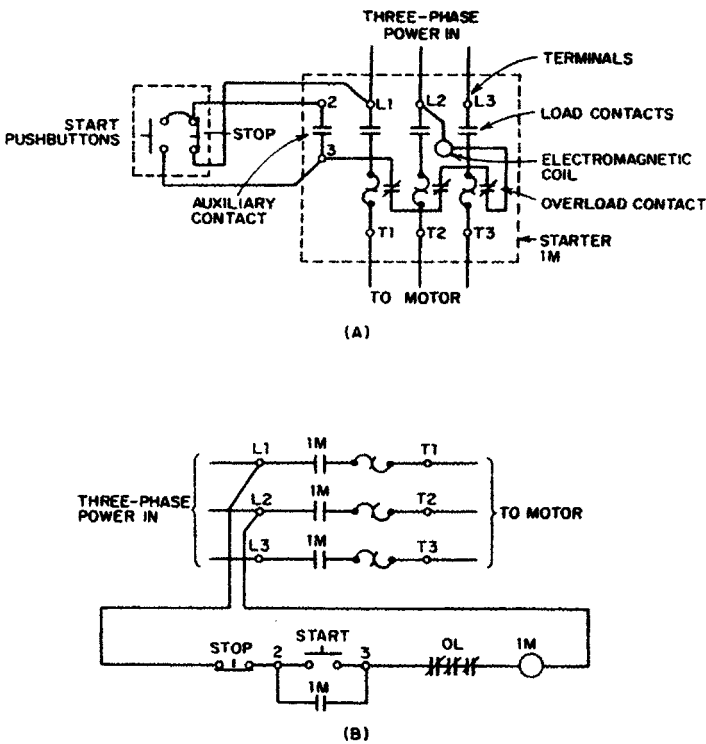
Figure 8-1. Motor starter with pushbuttons. (A) Pictorial diagram. (B) Ladder diagram.

Notice that the auxiliary contact in the starter is wired in parallel with the start pushbutton. In this relation it serves to maintain the circuit after the start button is released, and is therefore called a maintaining or a holding contact. When the motor stops for any reason, the holding circuit opens, and the motor will not restart until the start button is pushed.

In this and the other diagrams in this chapter the overload contacts have been shown wired on the hot side of the coil according to recommended practice. Most starters, however, are factory-wired with the overload contacts on the ground side of the coil. This is satisfactory if there is no ground, as may happen with some three-phase circuits. When there is a ground side to the control circuit, then the wiring should be changed to follow the diagram.

Figure 8-1 showed a typical manual system, with control power taken directly from the motor power source. Because the motor power source frequently is high-voltage (230 V or 480 V for pumps and fans, sometimes much higher for large centrifugal compressors), it is often considered desirable to provide a low-voltage source for control power. This may be 120 V or 24 V and may come from a separate source or from a control transformer in the starter. The advantage of the control transformer is that opening the disconnect switch interrupts all power, whereas with a separate control power source it is necessary to open two switches to interrupt all power. With complex, interlocking control systems, a separate power source is sometimes necessary.

Figure 8-2 shows a typical starter with a control transformer for supplying low-voltage control power. This is almost identical with Figure 8-1(B), except that the transformer has been added.

The motor also may be controlled by a hand-off-automatic switch (Figure 8-3). Thus it may be started and stopped manually as needed, but is normally operated by a pilot-device contact in the auto circuit. This arrangement is used where motors are interlocked with other motors, or with temperature, pressure, or flow controls.
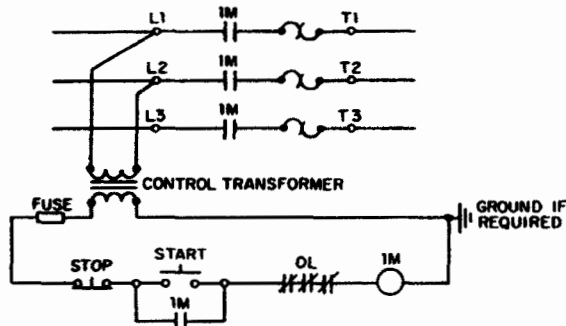
Figure 8-2. Motor starter with low-voltage control circuit.
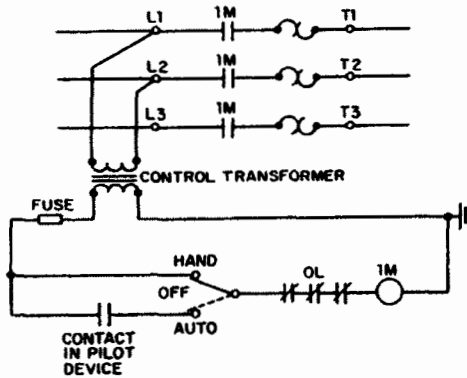
Figure 8-3. Motor starter with hand-off auto switch.

Pilot or indicating lights are often used to show the on or off condition of a device. For "on" indication it is desirable to use a flow or a pressure switch in the circuit to give a positive assurance of operation, as the mere fact of power being delivered to the starter does not necessarily prove that the motor is running, or, if running, is effectively moving a fluid.

Figure 8-4 shows the same control as Figure 8-3 but with the addition of a running pilot light. This figure also shows the fused disconnect switch and the motor.
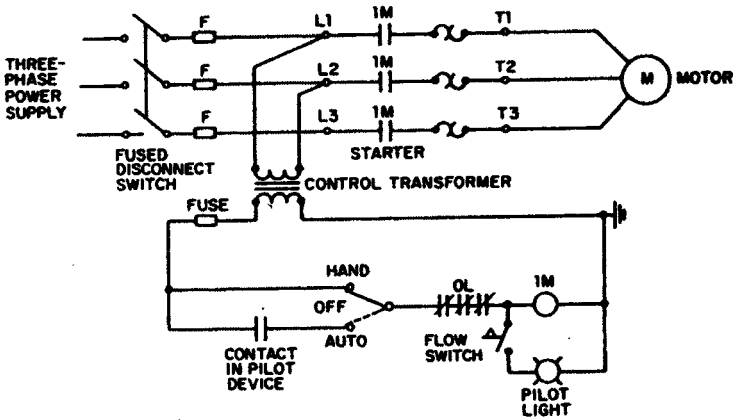
Figure 8-4. Motor starter with pilot light.

8.3 ELECTRICAL CONTROL OF A CHILLER

To illustrate a basic interlock system consider the diagram in Figure 8-5. A water chiller, a compressor, a circulating chilled water pump, a condensing water pump, a cooling tower fan, and the necessary relays and safety controls are shown. The power for the control circuit is obtained from a control transformer in the compressor starter, as this is the critical piece of apparatus. To place the system in operation the chilled water pump is started manually with a start pushbutton. As the water flows in the system, its temperature leaving the chiller is measured by a two-position thermostat. When the water temperature is above the thermostat setting, the thermostat contact closes, opening the solenoid valve in the refrigerant liquid line to the evaporator. The resulting rise in the suction pressure will close the low-pressure switch, starting the condensing water pump. If water is flowing in both chilled and condensing water circuits (flow switches closed) and all safety controls are closed, then the compressor motor will start.

The cooling tower fan is started and stopped by a thermostat in the condensing water supply, so that the condensing water will not get too hot or too cold.
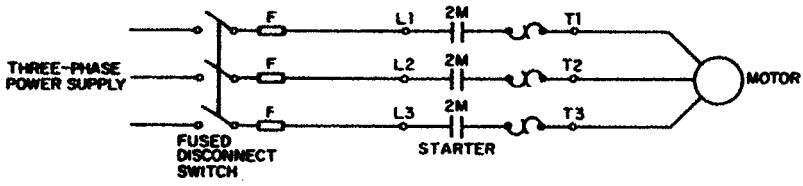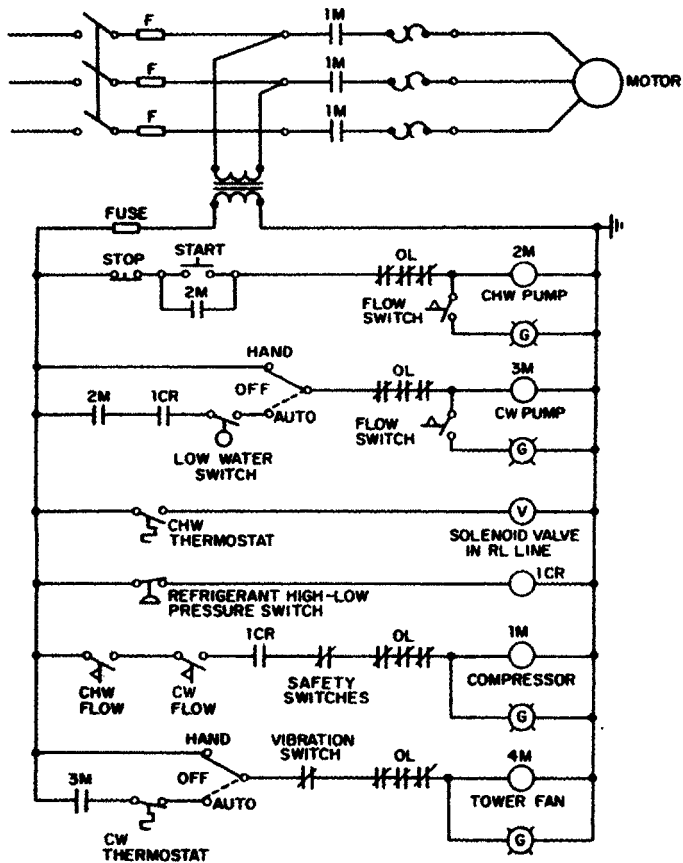
Figure 8-5. Chiller and accessories.

When the chilled water thermostat is satisfied, its contacts will open, closing the solenoid valve, but the condensing pump and compressor will continue to run until the system pumps down, that is, until the decreasing suction pressure opens the low-pressure switch. This pump-down cycle is controlled by the pump-down relay (1CR).

Many safety controls are provided to protect the equipment against operating under adverse and potentially damaging conditions. The oil pressure switch contains a heater that is energized when the compressor starts. It is necessary for increasing oil pressure to open a pressure switch in the heater circuit before the heater opens a thermally delayed contact in the compressor starter circuit.

The compressor also may be stopped by too high a condensing pressure, too low a suction pressure, a high refrigerant temperature, a low chilled water temperature, or inadequate flow of chilled or condensing water-or, of course, by the thermal overloads in the motor starter.

A float switch in the cooling tower sump will stop the condensing water pump if the water makeup system fails and the water level gets too low. A vibration switch will stop the cooling tower fan if the fan blades become damaged or get out of alignment.

The control system just described contains several conventional symbols for various types of operating and safety switches. These and many other standard symbols for electrical devices are in common use, but not everyone uses the same standards. It is necessary to define exactly what each symbol means somewhere in the contract documents.

8.4   ELECTRICAL CONTROL OF AN AIR HANDLING UNIT

Figure 8-6 shows a simple air handling unit electrical control sequence. Control power is obtained from a control transformer in the supply fan

starter. The supply fan is started manually. Interlocks provide for operating the return fan and supplying power to the temperature control system whenever the supply fan is running. The diagram shows an air solenoid for supplying compressed air to appropriate control components (usually damper actuators), but this also could be an electric relay for supplying power to electric or electronic controls.
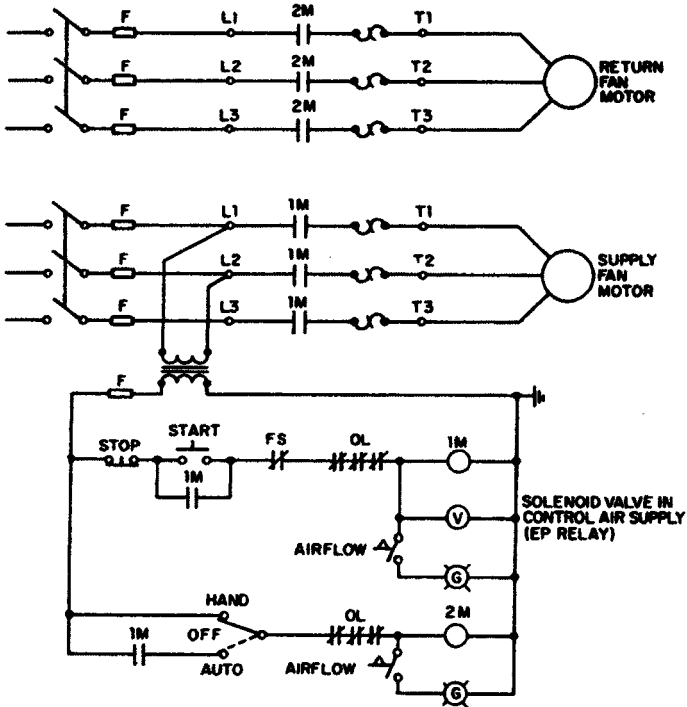
Figure 8-6. Air handling with return fan.

8.5 EXAMPLE: A TYPICAL SMALL AIR-CONDITIONING SYSTEM

Consider now that we are designing the controls for a small commercial-type building air-conditioning system, and we need to communicate with the electrical engineer.

The first step is reverse communication: We ask the engineer to provide the characteristics of the power system being supplied-voltage, phase, cycles. The answer may be that the engineer is providing 120-208 V three-phase, 60-cycle four-wire, which gives us a choice for our motors of either 120 V single-phase or 208 V single- or three-phase. It is common practice in the industry to use the higher voltage and three-phase when available for all large motors. Large generally means 1 hp and larger but could include smaller motors. Using higher voltage and three-phase power improves motor efficiency and thus lowers operating cost.

For this example our system will include an air handling unit with a 5 hp motor, a 2 hp return air fan, a 1/8 hp toilet exhaust fan, a 15 hp water chiller, a 5 hp air-cooled condenser, a boiler, two 1 hp pumps, and a 1/4 hp air compressor for the pneumatic temperature control system.

The schematic ladder diagram would appear as in Figure 8-7. In the diagrams that follow, the chiller safety and operating controls are shown by the notation "See Manufacturer's Wiring Diagram," since all package chillers are factory-wired and the arrangement varies from one manufacturer to another. We might even have to correct this diagram to show any changes required to suit the installed chiller. All the other equipment layouts are similar to those previously discussed. The air-cooled condenser fan is interlocked with and to the chiller in much the same relationship as the condensing water pump in Figure 8-5. The boiler is assumed to be a small, low-pressure type with an atmospheric burner and a built-in factory-wired control system, for which we provide 120 V power.
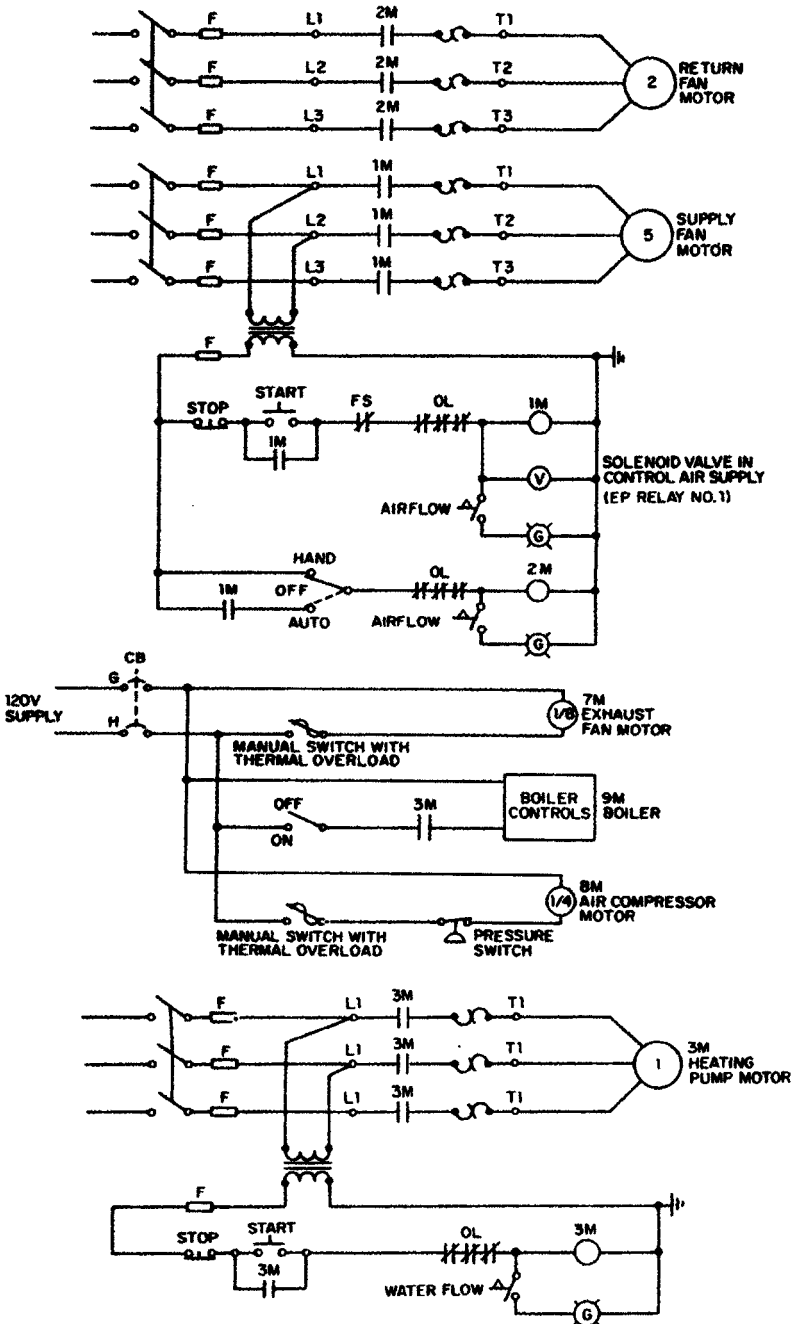
Figure 8-7. Air conditioning system.
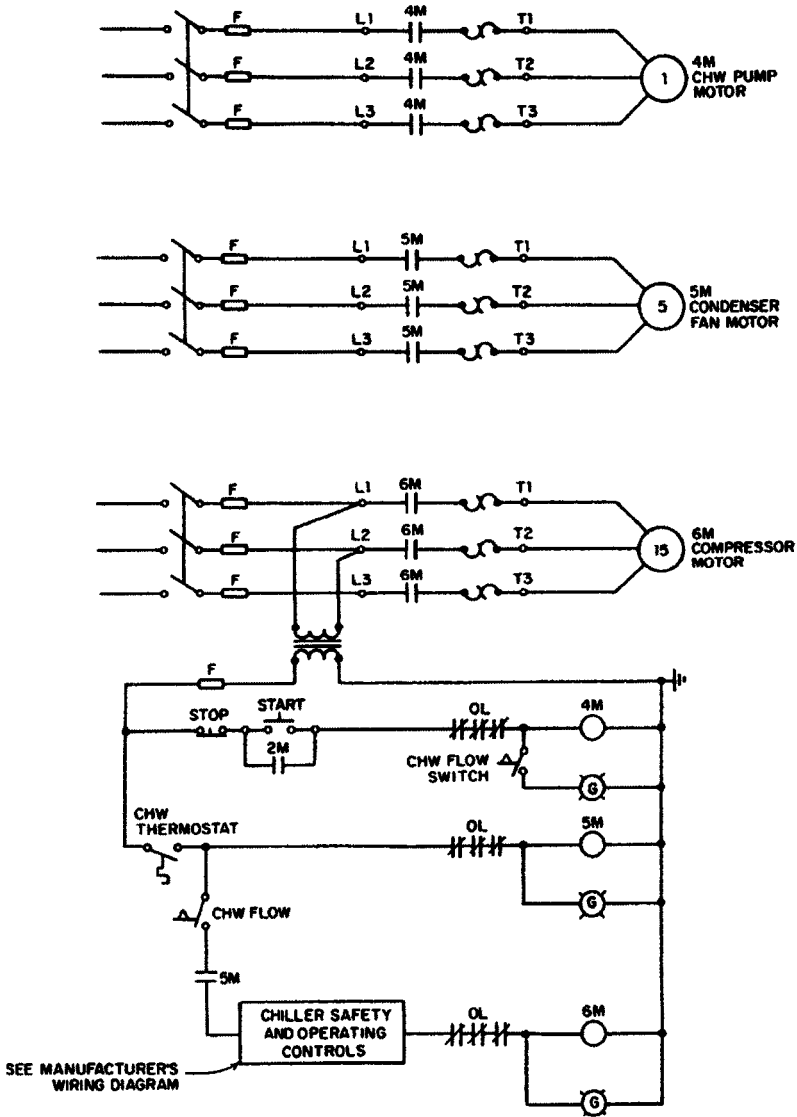
Figure 8-7 Continued. Air-conditioning system.

		1M	2M	3M	4M	5M	6M	7M	8M	9M
Mechanical Symbol		1M	2M	3M	4M	5M	6M	7M	8M	9M
	Mechanical Equipment Served	Supply fan	Return fan	Heating pump	CHW pump	Condenser fan	Compressor (refrig.)	Exhaust fan	Air compressor	Boiler
MOTOR DATA	Horsepower	5	2	1	1	5	15	1/8	1/4	—
	Voltage	480	480	480	480	480	480	120	120	120
	Phase	3	3	3	3	3	3	1	1	1
	K.W. input									0.20
	Windings									
STARTER DATA	Type						Controls by manufacturer	none	none	none
	NEMA size	1	1	1	1	1		none	none	none
	Number of speeds	1	1	1	1	1	1	1	1	—
	Control volts	120	120	120	120	120				
	Pushbutton	S-S		S-S	S-S	—				
	Selector switch		HOA			—				
	Pilot lamp	G	G	G	G	G				
	Normally open contact	2		2	1	1				
	Normally closed contact	—	—							
REMOTE CONTROL	Control volts						120	120	120	120
	Pushbutton									
	Selector switch							on-off	on-off	on-off
	Pilot lamp							—	Press switch	—
RELAYS	PE relay number	1								
	EP relay number									
	Cont. relay									
INTERLOCK FROM		1M				5M CHW flow				3M

Figure 8-8. Motor starter and control schedule.

It is also desirable to have a schedule of motor starters and controls, such as Figure 8-8. We fill in part of this, as shown, and give it to the electrical engineer along with the schematic. (The NO and NC contacts are the auxiliary contacts required in the starter.) The engineer then can fill in starter sizes, make other electrical decisions, and reproduce everything on the electrical drawings. Adequate coordination assures that the finished job will perform to the requirements of our design.

Although this illustration is simple, the principles can be applied to a system of any size and complexity.

8.6 ELECTRIC HEATERS

In Chapter 6 the control of electric heaters was dealt with in very general terms. Figure 8-9 shows a detailed electric control and power circuit diagram for a single-phase, single-stage heater, with manual and automatic reset high-limit controls, an electric thermostat, an airflow switch, and a relay-contractor. Figure 8-10 shows controls for a three-phase, two-stage heater, with high-limit controls, two-stage thermostat and contactors. There are many possible variations to these basic arrangements, depending on the heater size and sequencing requirements.
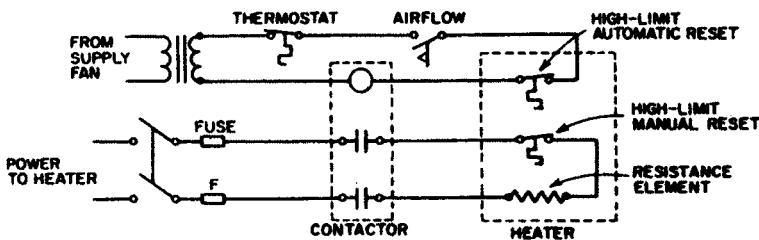
Figure 8-9. Electric heater; single-phase, single stage.
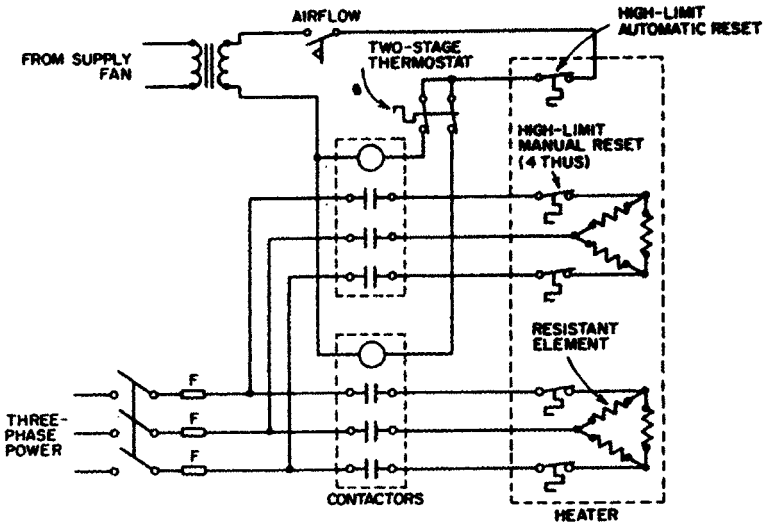
Figure 8-10. Electric heater, threre-phase, two phase.

8.7 REDUCED-VOLTAGE STARTERS

So far in this chapter only small motors and across-the-line starters have been considered. The inrush or starting current on any motor is three to six times the normal running current. As motors get larger, the local electric utility requirements and good engineering practice require that we take steps to reduce this inrush current. This is done by means of reduced voltage starters. The motor size above which reduced-voltage starters are used depends on the voltage (higher voltages mean less current for the same horsepower) and the size of the building electrical service. The electrical engineer can advise you about the requirements for any particular job.

Reduced-voltage starters are of several types. Some require special matching motors, others work with any standard motor. Remember that any reduction of inrush current also reduces starting torque.

8.7.1 Part-Winding Starter

Perhaps the simplest reduced-voltage starter is the part-winding starter. This must be used with a special type of motor - a part-winding motor. Figure 8-11 shows the circuit schematically, including the special motor winding. When the start button is depressed, coil 1M1 is energized, and the contacts serving part of the motor windings are closed. The motor starts and accelerates for a preset period, as determined by the timing relay (1TR). After this period the 1TR contact closes, and all the windings are energized. Inrush current will be about 60% of across-the-line inrush. This starter also provides closed transition, which is discussed later in this chapter.
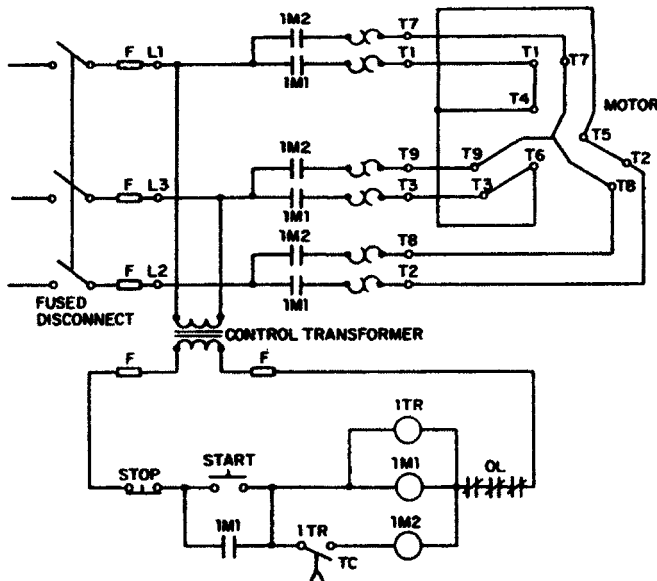
Figure 8-11. Part-winding starter.

8.7.2 Auto-transformer and Primary Resistor Starters

These starters may be used with any standard squirrel cage motor. Inrush may be limited to as little as 50% of maximum.

Figure 8-12 shows a primary resistor starter. When coil 1M is energized, closing contact 1M, the resistors reduce the current flow during the startup and acceleration period. When 1TR times out, coil 1A is energized, closing contact 1A. This shunts out the resistors and applies full voltage to the motor. This system may be furnished with two, three or more steps of acceleration. This is also a closed-transition circuit.
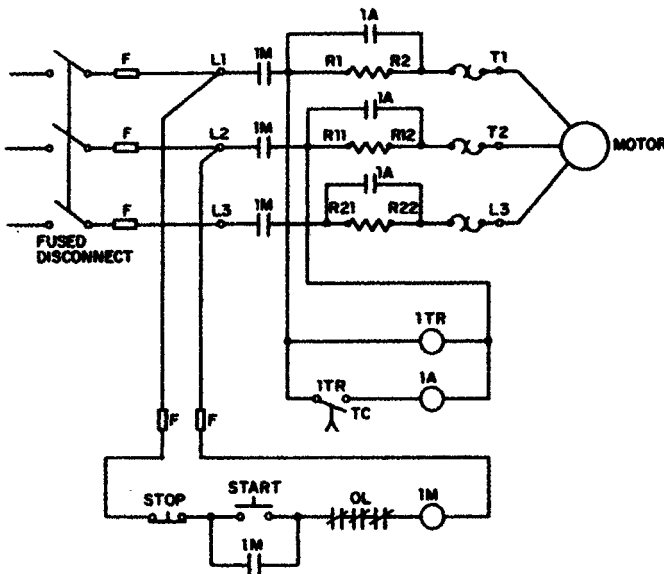
Figure 8- 12. Primary resistor starter.

Figure 8-13 shows an auto-transformer starter. This arrangement has an excellent ratio of starting torque to power input. The start button is depressed and energizes relay 1S, which in turn energizes relay 2S.

This provides current through the auto-transformer coils in the starter, starting the motor at reduced voltage and current. After an acceleration period the 2S timed opening contact opens, de-energizing 1S, and then the 2S timed closing contact closes, energizing relay R. This shunts out the transformer coils and applies full voltage to the motor windings. Relay 2S is de-energized, restoring those contacts to their normal position with coil R held in by a maintaining contact. Note that coils 1S and R are mechanically interlocked so that only one can be energized at a time. Note also that the transformer coils are tapped for various percentages of line voltage, usually in the range from 50% to 80%. The tap selected governs the inrush and starting torque.

Figure 8-13. Auto-transformer starter.

8.7.3 Wye-Delta Starters

Wye-Delta starters require specially wound motors, but have excellent starting characteristics, especially with very large motors. They are available with both open-and closed-transition arrangements.

Figure 8-14. Wye-Delta starter.

Figure 8-14 shows a closed-transition arrangement. A pilot control circuit (not shown) activates control relay 1CR. Contact 1CR closes and energizes relay 1S. This in turn energizes 1M1, and the motor windings are energized in a Wye configuration. This allows the motor to start with low current and voltage draw. After an acceleration period determined by the setting of 1TM, contact 1TM closes, energizing relay 1A. This first shunts a portion of the current through the resistors

and then the late-opening 1A contact disconnects 1S, opening the contacts in the Wye. The NC contact in 1S closes, energizing 1M2 and the circuit is now in running configuration with motor windings connected in a delta arrangement. The pilot control circuit will usually include an anti-recycle timer that prevents restarting the motor if it is stopped within 20 minutes of the original start. Too frequent starting may damage the starter.

8.7.4 Solid-State Starters

A new development in reduced voltage starters is the solid-state starter, a typical arrangement of which is shown in Figure 8-15. The contactors found in electro-mechanical starters have been replaced by silicon controlled rectifiers (SCRs), which provide proportional control of current flow to the motor. Current is sensed by current transformers (CT) (which replace the overloads). Current and voltage data are fed into a controller that drives the SCRs. Current during startup can be held to a desired maximum. When the SCRs are fully on, they offer essentially no resistance to current flow and therefore act like closed contacts. Overcurrent and low-voltage protection during operation are inherent in the control logic.
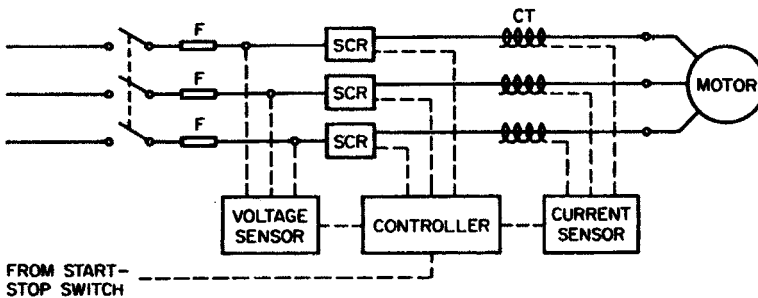
Figure 8-15. Solid state starter.

8.7.5 Open and Closed Transition

All the starter arrangements shown in the preceding paragraphs had closed-transition arrangements; that is, there was no break in the current flow during the starting sequence. The switching relay contacts were arranged so that the steps overlapped.

Open-transition starters are also available. The switching is somewhat simplified because there is a momentary break in current flow at the changeover from start to run configuration. This momentary break can cause a transient current of a very high value, sometimes called a spike. With small motors or on large electrical distribution systems, these spikes may be unimportant; but often a transient of this nature can disrupt an electrical distribution system, or cause a malfunction of a sensitive electronic device, such as a computer. Check this out with an electrical engineer.

8.8 MULTISPEED STARTERS

Two-speed fans are frequently encountered on air distribution systems or cooling towers. The motors commonly used in these applications are either variable-torque or two-winding motors. There are other types of multispeed motors, but they are seldom encountered in HVAC systems.

Figure 8-16 shows a variable-torque motor with simple two-speed control. Pushing the slow or fast pushbutton will start the motor at that speed. Besides the electric interlock contacts, the two coils are usually interlocked mechanically to prevent damage to the motor.

Figure 8-17 shows the different starter arrangement used for a two-winding motor. The control circuit can be the same for either type of motor. The advantage of a two-separate-windings motor is that the starter circuit is a bit simpler than that of a single-winding motor, and the motor will continue to function in an emergency on only one winding if the other is damaged. Also, two windings allow the use of any combination of speeds desired, whereas a single-winding variable-

torque motor must operate with high speed at some fixed ratio to that of low speed, usually 2:1.
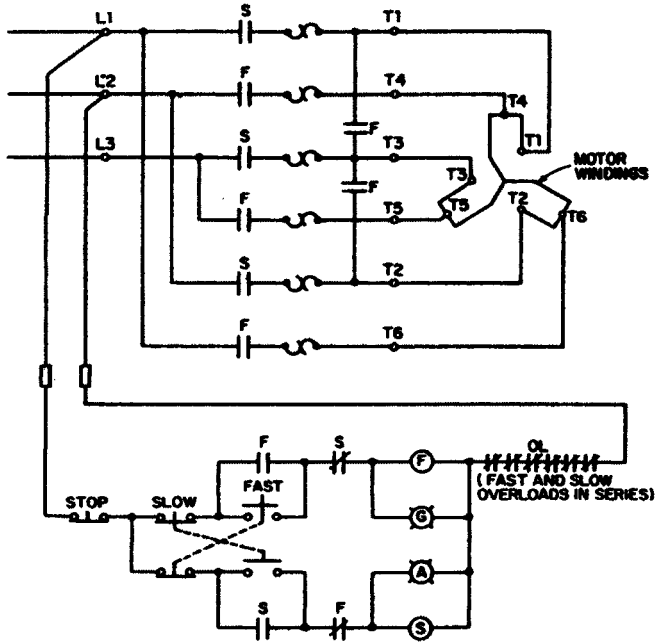
Figure 8-16. Two-speed starter, variable torque motor.
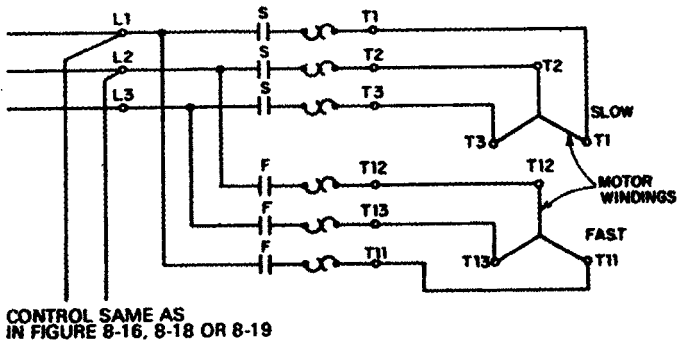
Figure 8-17. Two-speed, two-winding starter.

The control circuit shown in Figure 8-16 is satisfactory for small motors. However, the same starting problems occur as previously discussed under reduced-voltage starting. A progressive control, as shown in Figure 8-18, has the same effect as reduced-voltage starting. When the slow button is pushed, the motor simply starts on slow speed. When the fast button is pushed, the motor starts first on slow speed, then after a time delay for acceleration, switches to fast. This is an open-transition starter because it is necessary to disconnect the slow coil before connecting the fast coil.
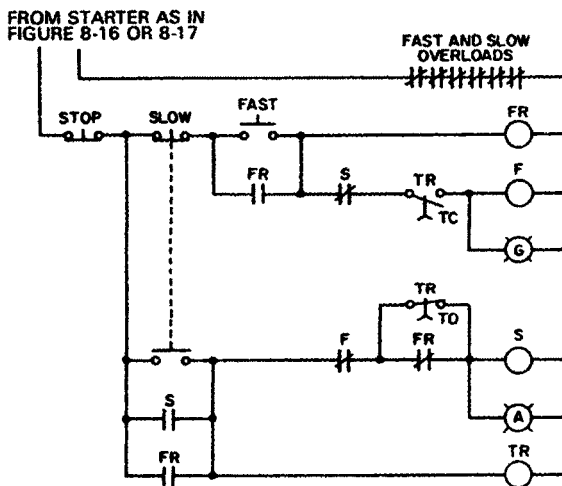
Figure 8-18. Two-speed; progressive start.

An additional problem arises in switching from fast to slow. An abrupt change will cause a dynamic braking effect, as the motor wants to run slow, but its momentum opposes this slowdown. The resulting currents can damage the motor. It is therefore desirable to disconnect the fast coil and allow the system to decelerate to near the slow speed before connecting the slow coil.

Figure 8-19 shows a two-speed, progressive-start, timed-deceleration control circuit. When the fast button is pushed the motor starts on slow and after a time delay switches to fast. When this occurs, relay 2TR is energized and the 2TR contact in the slow circuit opens. Now, when the slow button is pushed, the fast coil and 2TR are de-energized, but the 2TR contact provides a time delay to allow the motor to decelerate

before the slow coil is energized. Relay SR provides a holding circuit during the delay period.
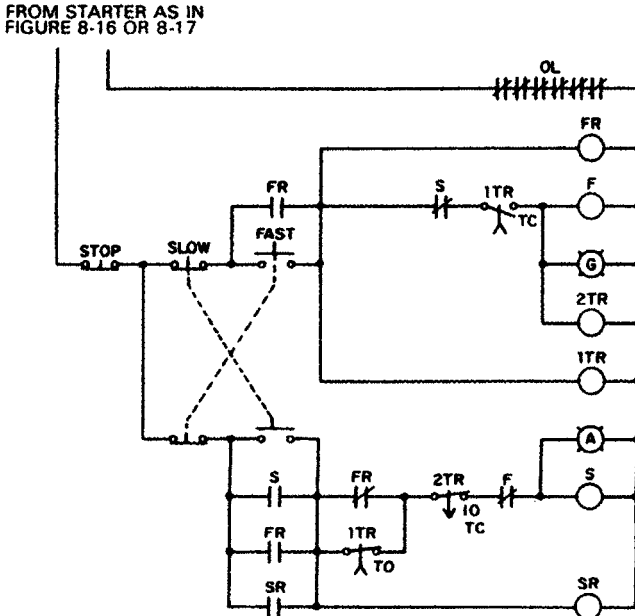
Figure 8-19. Two-speed, progressive start, time delay from fast to slow.

8.9 VARIABLE SPEED CONTROLLERS

Variable speed motors have been in use for many years. The direct current (DC) motor is inherently a variable speed device because speed depends on current flow. However, DC motors and controllers are more expensive to purchase and install than the more common AC squirrel cage motors.

Standard AC motors may be speed-controlled by varying the input frequency and voltage to the motor. This is now done by solid state motor speed controllers. The frequency and voltage can be varied from the standard 60 Hz and line voltage to any lower values. The controller responds to a varying electrical input signal and usually has an

adjustable low-limit setting because damage to the motor may occur at very low speeds. The input signal may come from an electrical control system or from a pneumatic system, through a transducer as shown in Figure 8-20. This is a typical VAV system, with duct static pressure control of motor speed.

Variable speed controllers are available in capacities up to several hundred horsepower. They replace conventional starters because current flow is automatically limited to full load current, and no inrush current is experienced.
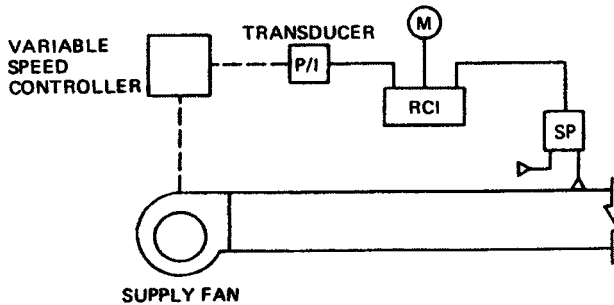
Figure 8-20. Variable speed motor controller for VAV.

8.10 SUMMARY

The discussions in this chapter have been necessarily short. It is not the intention to present a detailed treatise on electrical engineering. It is hoped, however, that you now have a better idea of the electrical problems inherent in the design of any control system and how to cope with some of them.

Most important, electrical control diagrams, preferably ladder diagrams, are an important part of the HVAC control system documentation. It is the control system designer's responsibility to be sure that correctly configured diagrams are part of the construction documents for any HVAC project.

9

Special Control

9.1 INTRODUCTION

The control systems discussed up to this point might be considered conventional. They are intended to provide adequate occupant comfort and reasonable efficiency. In some applications, however, very precise control of temperature and/or humidity is required. This chapter will discuss some of these control challenges in detail.

9.2 CLOSE TEMPERATURE AND/OR HUMIDITY CONTROL

Requirements for extremely close control occur in many end uses, ranging from machine shops to hospitals to computer rooms. High-efficiency filtration often is an additional requirement. Let us consider some of these applications.

9.2.1 Standards Laboratory

A standards laboratory, for either primary or secondary standard comparison, always requires extremely close control of temperature to maintain dimensional stability in the standard gage blocks. Close control here means specified system differentials of 0.1°F to 0.2°F.

Although the absolute accuracy of available temperature sensors is probably not better than plus or minus 0.5°F, the system sometimes can be designed and calibrated to control within the specified differentials.

Many special features required are to make such a system controllable. Besides a high air change rate at a low differential between room and supply air temperatures, it is necessary to maintain the interior wall, floor and ceiling surfaces at or near the room temperature. This is sometimes done by using a ventilating ceiling, with a hollow wall serving as a return plenum, all heavily insulated. One excellent method involves the use of a constant cold air supply with reheat as close to the room as possible. Reheat can be accomplished by means of hot water or electric heating coils; steam cannot be used because of its high temperature. The hot water supply temperature cannot be greater than 80°F, as a higher water temperature causes too great a rise in the supply air temperature, even though the control valve may be only cracked open. Electric heaters must be modulated by saturable core reactors or solid-state controls such as silicon-controlled rectifiers.

Figure 9-1 shows a system for temperature control in a standards laboratory using hot water reheat. A special heat exchanger in required to provide 75°F to 80°F supply water, using central hot water, steam, or electricity as the heat source. This low temperature is necessary to avoid a high system gain, which would make the system uncontrollable. The temperature controller must be an electronic industrial type or a high speed digital controller to achieve the necessary sensitivity. Because there will be some temperature variation throughout the room, the temperature sensor must be mounted as close to the gaging work station as possible. The diagram shows zone control for one of a group of standards rooms.

Most standards rooms require only a small amount of outside air, as the rooms are normally unoccupied.
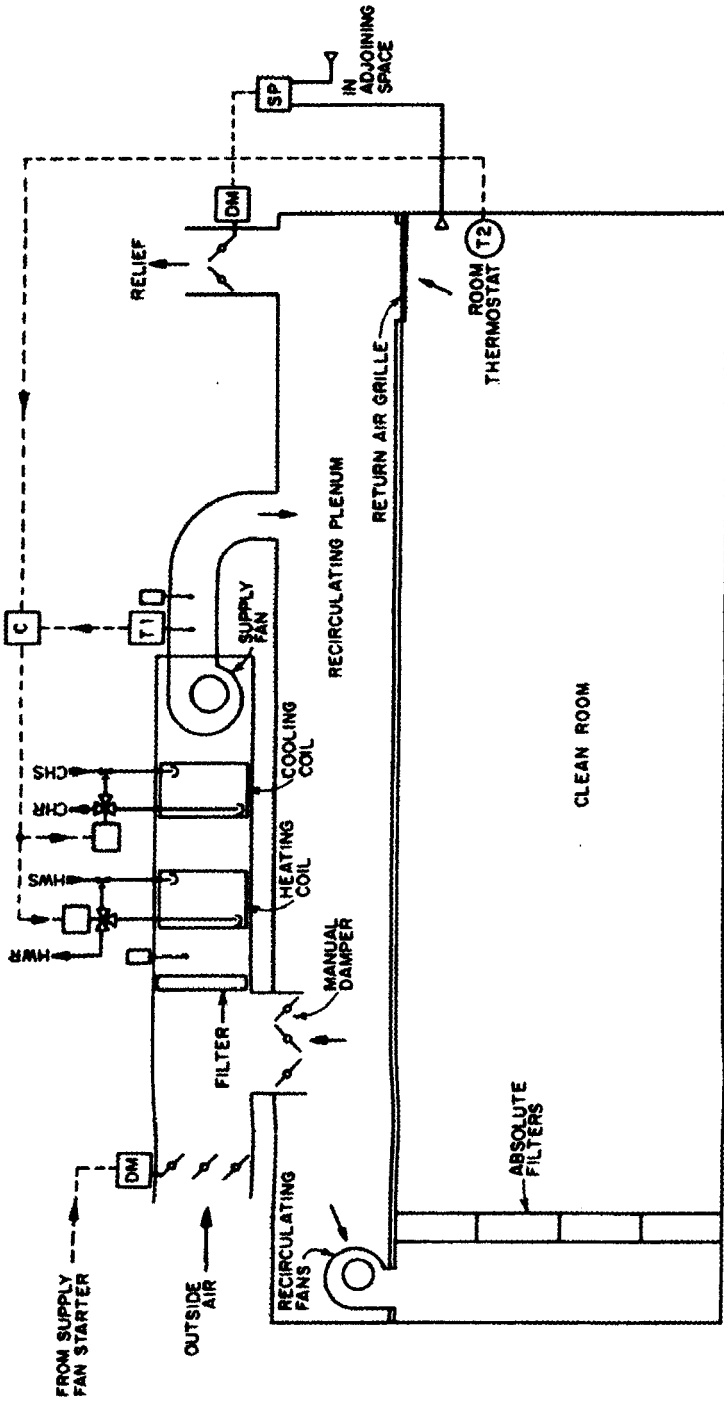
Figure 9-1. Control for a standards laboratory.

9.2.2 Clean Rooms

Clean rooms generally require high air flow rates to maintain the clean conditions, with temperature differential requirements in the order of plus or minus one to two degrees Fahrenheit. Humidity, too, is often controlled. Because the airflow rate required for cleanliness is much higher than that required for temperature control, a double air-handling system may be used. Figure 9-2 shows the system and control arrangement for a horizontal laminar flow clean room. A high-flow-rate fan and filter system circulates the air from a plenum to the room and back. Air temperature and ventilation rate are governed by a small air handler with heating and cooling coils and a fixed minimum outside air damper. Relief dampers are modulated by a static pressure controller. Temperature controls are conventional, with a room or return air thermostat as the principal controller. A humidistat and humidity control equipment could readily be added.

9.2.3 Hospitals

Several areas in hospitals require close control of temperature, humidity, cleanliness or all three. These areas include especially: surgery and delivery rooms, nurseries, intensive care units, and laboratory and research areas. Because many present codes do not allow recirculation of air from these spaces, 100% outside air often must be used. Although the laboratory area should have its own separate supply and exhaust system, the other areas can be supplied by a single air system with heat reclaim, humidity control, and zone reheat. Figure 9-3 shows such a system. The outside air passes through a heat reclaim coil that is transferring heat to or from a similar coil in the exhaust air. The air is further preheated as required and passed through a cooling coil with the discharge temperature corresponding to the lowest design dew point required for any of the controlled zones. Zone thermostats control reheat coils, and zone humidistats control humidifiers in the branch sucts. The psychrometric charts in Figure 9-4 show the cycles for summer and winter design conditions.
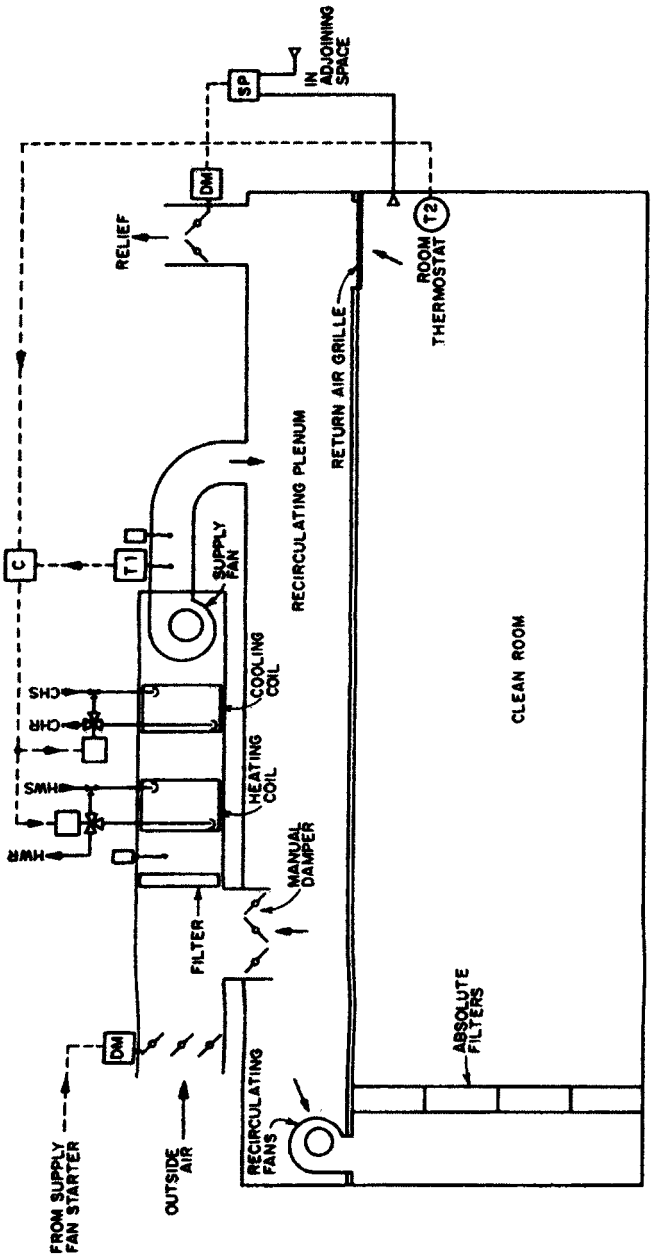
Figure 9-2. Temperature control for a clean room.
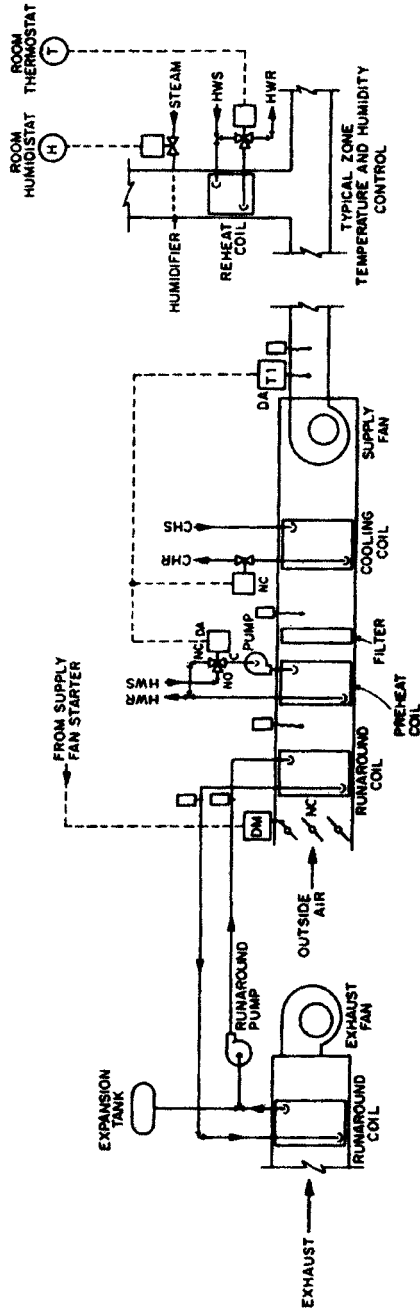
Figure 9-3. Air system with heat reclaim, 100% outside air, zone reheat. humidifiers in the branch ducts.

The psychrometric charts in Figure 9-4 show the cycles for summer and winter design conditions.
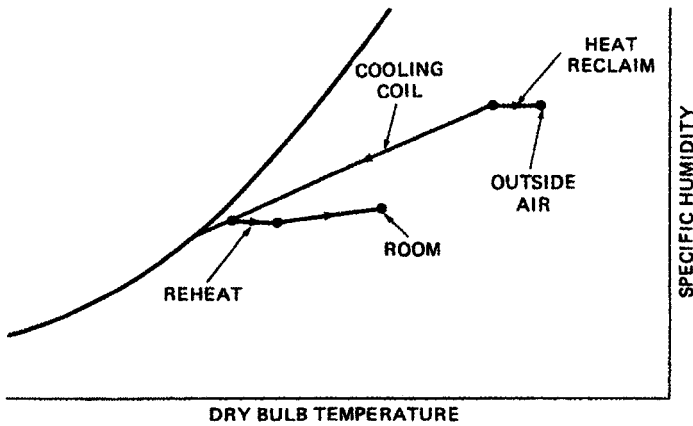
Figure 9-4(A) Summer cooling cycle for Figure 9-3.
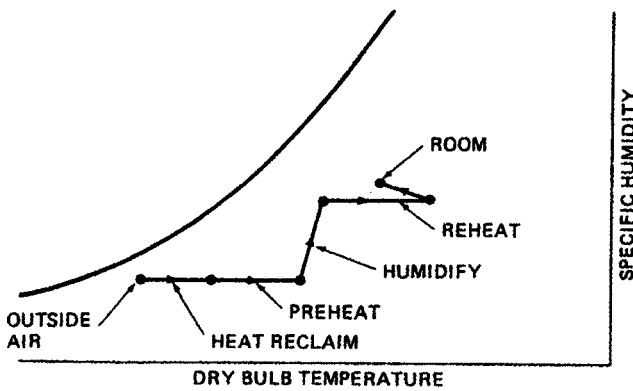
Figure 9-4(B)

9.2.4 Computer Rooms

Computer rooms have many similarities to clean rooms, but here high airflow rates are the result of high heat gains from the computer equipment. Cleanliness standards are lower, and humidification often is

required. Temperature control should be fairly accurate (plus or minus 1 to 1 1/2 °F or less), because some solid-state devices are sensitive to rapid changes in temperature. Location of the room thermostat and air distribution patterns become very important because some components have very high heat emission rates. Some computer equipment items require direct air supply from an under-floor plenum at a controlled temperature. Other items have a complete environmental control system within them, requiring only chilled water from an outside source. Figure 9-5 shows a hypothetical system with all of these elements included.

It is common practice to use packaged air-handling units-available in capacities from 2 to 20 tons-located within the computer room or rooms. Supply air is discharged downward into the space below the raised floor. This space becomes an air distribution plenum, from which air is fed into the room through floor registers or special perforated floor panels. Where it is required, computer cabinets may be supplied with air directly from the plenum. Chilled water piping is located in the under-floor space for connection to those computer elements that require it. For prevention of damage due to water leaks, floor drains and moisture sensors are required. Direct expansion systems are also common. Water-cooled and remote air-cooled condensers both are used. Ventilation air is provided by an outside air connection to one or more package units or by a special ventilation system. The packaged units include all controls.

9.3 CONTROLLED ENVIRONMENT ROOMS FOR TESTING

Environmental test rooms are used in industry for testing products. Prefabricated chambers complete with controls may be purchased from several manufacturers.

Environmental test rooms for physiological testing of humans usually are custom-designed as part of a laboratory complex. The air-conditioning system is required to provide a variety of environments, for example, from 32°F to 120°F and simultaneously from 20% to 80% relative humidity, with close control of differentials at any set point. Obviously, a complex system is required.

Figure 9-6 shows an air-conditioning and control system that will satisfy the requirements of a physiological environmental test room. The small amount of outside air required for ventilation is mixed with return air and then flows through a cooling coil. This is a direct-expansion coil, with its own air-cooled condensing unit, and with widely spaced fins to minimize the effect of ice buildup at low temperatures. The coil is provided with an automatically adjustable back-pressure suction valve that is operated by the room thermostat to control the coil suction pressure and thus the minimum temperature of the air leaving the coil.

The air then passes through the supply fan and to a chemical dehumidifier that is provided with a bypass and a regeneration air duct and fan. The dehumidifier includes circular trays of a chemical absorbent that rotate through the supply air and regeneration air in series. The supply air is dried, and the heated regeneration air removes the moisture and carries it away. An electric heater and a duct thermostat are provided for controlling the temperature of the regeneration air.

The humidistat controls the proportions of air flowing through the dehumidifier and bypass because direct control of the dehumidifier is not easy to achieve. The supply air leaving the dehumidifier is hot because the heat given up by the hot absorbent and the heat due to the chemical reactions. Therefore, it must be recooled, and a second cooling coil and condensing unit is provided for this purpose. An electric reheat coil provides final temperature control when high temperatures are needed. A steam humidifier provides high-level humidification.
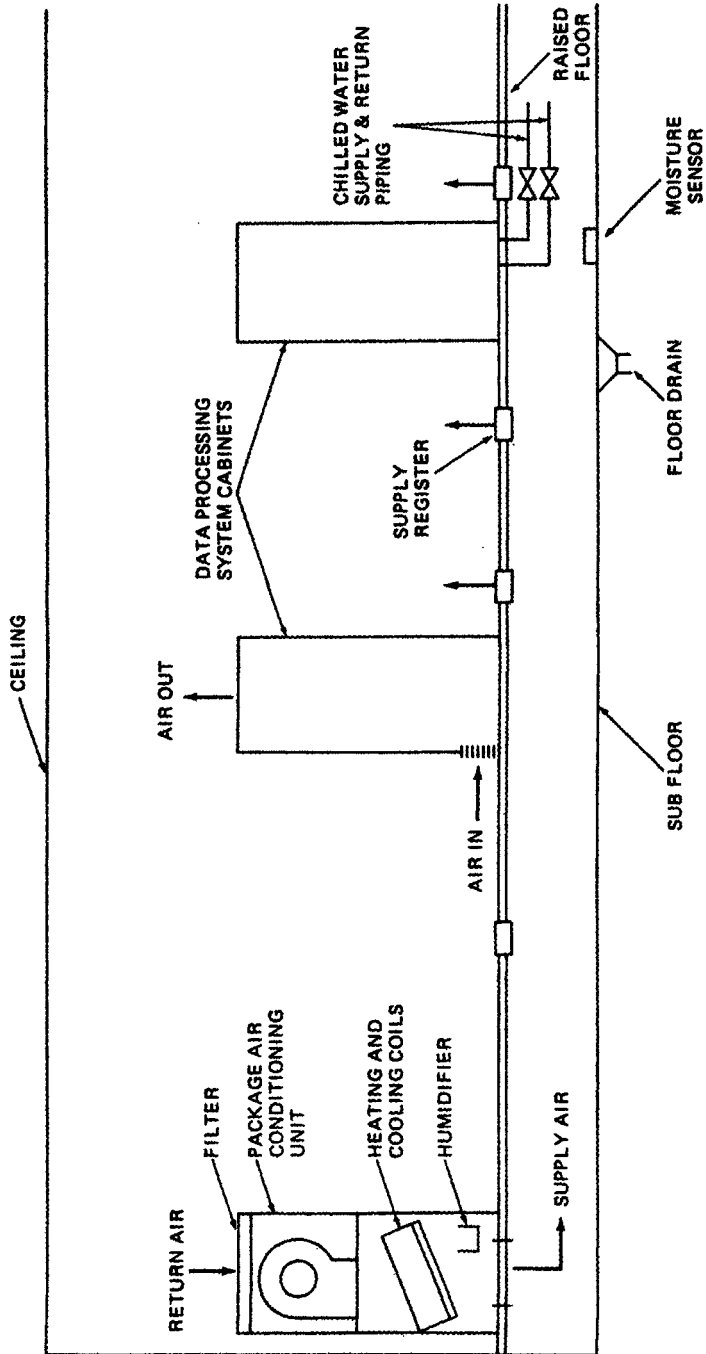
Figure 9-5. Section through computer room.
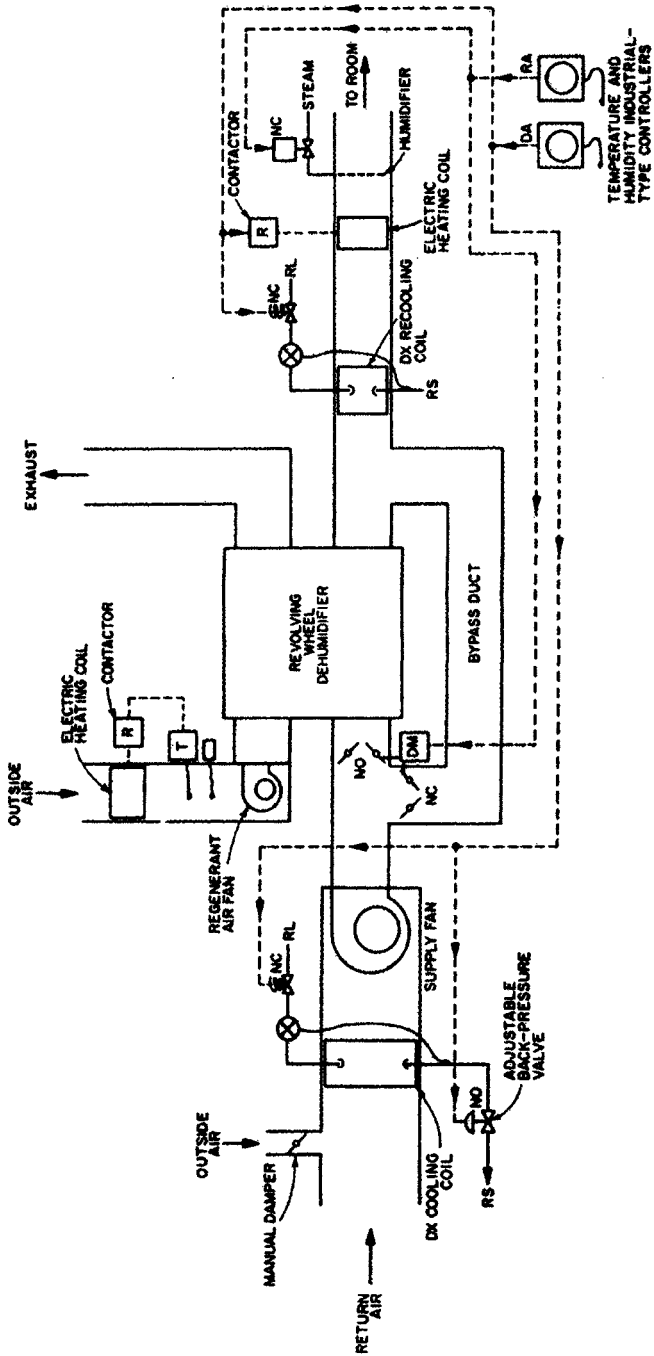
Figure 9-6. Environmental test room.

Temperature and humidity controllers are industrial electronic or digital instruments with remote sensors, so that remote monitoring and adjustment of room conditions are available.

9.4 SUMMARY

Designing the special purpose control systems described above requires thought and experience. It is especially important to make the HVAC and control systems stable by keeping the HVAC process gains constant and low enough to be controllable. Proportional plus integral control action is always required to achieve tight control.

10

Digital and Supervisory Control Systems

10.1 INTRODUCTION

Nothing has changed more since the last edition of this book than the information technology (IT) side of HVAC control. Direct digital control (DDC) has penetrated new construction projects to a level that might not have been predicted. One designer remarked on questioning that: "I would guess that 90% of our jobs are DDC." That designers are embracing this technology is a result of a transformation in the approach of control system manufacturers'.

When HVAC control companies first approached digital technology, many developed their own hardware, even manufactured their own computers. Each had its own programing language, graphical interface, and communications protocol. This kept costs high and kept many away from DDC.

To be fair, it is not clear that HVAC control companies were behind the state of the art when they began to implement digital building automation systems (BAS). However, few would argue that personal computer (PC) and Internet developments did not quickly surpass the developing DDC and BAS systems in the HVAC industry. This is hardly surprising since, reportedly, the information technology industry spends more than three time the gross revenue of the HVAC control industry on research.

Many in the industry realized that piggybacking on the PC/Internet technology development made DDC and BAS less costly and

potentially much more profitable. Users also sought interoperability, the ability to interchange brands while maintaining communication capabililty between components.

In this chapter we will briefly describe the older supervisory system and then discuss the most modern direct digital control systems.

Please note that digital control technology is evolving rapidly; so do not rely on this chapter as an exclusive source of information. Manufacturers' literature and trade journals might be good sources for the latest information.

10.2 HARD-WIRED SYSTEMS

Hard-wired central systems were the first used. As the name implies, they are simply the extension of the conventional individual control wires to a central point. Each element in the control system, whether start-stop control of a motor, indication of a temperature, reset of a control point or malfunction indication, requires one or more wires running all the way to the control panel, plus a separate indicating device on the panel. Obviously, a building need not be very large before this equipment becomes too cumbersome to be useful. Many such early systems were built, though few are still in use.

The transmission of signals from distant air-handling units and other equipment to the central control point poses some problems. Two-position or digital signals are simple; they are used for starting and stopping motors and indicating on-off or out-of-control conditions. Analog signals for measuring flow, pressure, temperature, and the like are more difficult. These signals may be transmitted as low-voltage direct-current signals, or as air pressure signals that can be translated and read at the receiving end by means of a transducer. Electrical line losses become important, and careful calibration and compensation are necessary.

10.3 MULTIPLEXING SYSTEMS

Multiplexing provides a way to share central control panel components. In its simplest form, a selector switch was used to connect individual sensors to a common meter. Alternatively, a relay at each sensor or group of sensors allows the sensors to be selected for monitor and control through common signal wires. When the data for one group are completed, another group can be selected. The relays used are called multiplexing relays.

Later systems of this type used coding signals to energize the relay. This requires only a few wires to address most relays. An automatic projector and slides were used to give the operator a visual picture of the system being monitored.

Where the equipment rooms are remote from the central control panel, as is usually the case in a large complex, the monitoring systems may include an audio or audiovisual communication system. With this available, the actions of a maintenance worker doing troubleshooting on the equipment may be guided by the supervisor at the central location. Closed-circuit TV can also improve security by providing visual observation of unoccupied equipment rooms.

Although a few systems that use simple multiplex relay schemes are still in use, they are obsolete today, having been replaced by computer-based systems.

10.4 COMPUTER-BASED SYSTEMS FOR MONITORING AND CONTROL

The term "computer-based" refers to a control system that uses some type of programmable digital computer. The computer often is referred to as intelligent because it can be programmed to make decisions. Some of the most promising developments are in the area of adaptive

control. A computer with adaptive control capabilities will learn from experience which decisions are best and will modify its operating programs accordingly.

The once clear-cut distinctions among various types and sizes of computers have become blurred. Increasingly, small devices have much greater capabilities. Some local loop controllers use programmed chips to increase their sophistication and provide better communication to supervisory computers. Modern systems are hierarchical with intelligence at all levels.

10.4.1 Functions of older Computer-Based Systems

The computer-based system can provide monitoring, intervention control, or direct control of local loop elements. It also can provide historical data summaries, data analysis and maintenance scheduling.

Monitoring means frequently looking at the status or the value of the various sensors or contacts connected to the computer. It usually includes comparing the status with some norm and providing an audible, visual, and/or printed display of off-normal (alarm) conditions.

Intervention control is used when the computer has no direct control of the local loop but may provide reset of set points, start/stop commands for motors, or open/close commands for dampers or valves. Intervention control is used primarily for energy conservation and efficiency in HVAC system operation.

Direct control of the local loop means that the computer provides all the control functions, interfacing directly with the sensors and controlled devices. This is usually called direct digital control (DDC). Direct digital control generally requires a computer dedicated to controlling no more than a few control loops. When a remote supervisory computer is used to perform DDC, reliability and accuracy may be poor. Reliability decreases because of the possibility that failure of the computer or the communication system would lead to loss of control. Performance decreases as a function of the number of points

interfaced to the computer. As the number of points increases, the frequency with which the computer can scan (look at) any specific point decreases. At some frequency this leads to loss of control.

Historically, energy management and control systems have performed the monitoring and intervention control functions only. Depending on the size of these systems, a supervisory computer can be connected to several intelligent field interface devices or FIDs.

Sensors are connected to a FID and scanned using electronic multiplexing. The FID usually does the intervention control -- it might adjust a pressure regulator that sends a set point signal to a pneumatic controller for example. Some FIDs store data and are periodically polled by the supervisory computer so that data can be transferred for analysis. FIDs also are programmed to detect alarm conditions and interrupt the supervisory computer when alarms occur.

Trend logging and alarm reporting occur at the supervisory computer. The keyboard, monitor, printer, and disc drive(s) also are supported by the supervisory computer. In newer systems, computer graphics replace the slide shows of older schemes.

10.4.2 Modern Direct Digital Control and Monitoring Systems

The elements of a computer-based system are shown in Figure 10-1. Modern systems communicate using a standard peer to peer network protocol. BACnet (developed by an ASHRAE committee) and LonWorks (trade mark Echelon Inc.) seem to be the most popular. Standard chips from the computer industry have replaced electronics built by HVAC control companies.

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┌──────────┐   ┌──────────┐   ┌──────────┐
│ Security │   │Supervisory│  │ Actuators│
│ Sensors  │   │ Computer │   │          │
└────┬─────┘   └────┬─────┘   └────┬─────┘
     │              │              │
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  ▲       ┌─────────┴────┐ ┌───────┴──────┐
 ╱        │Air handling unit│ │   Room       │
╱         │controller(s)  │ │ controller(s)│
          └──────────────┘ └──────────────┘
LonWorks
or BACnet
network
```
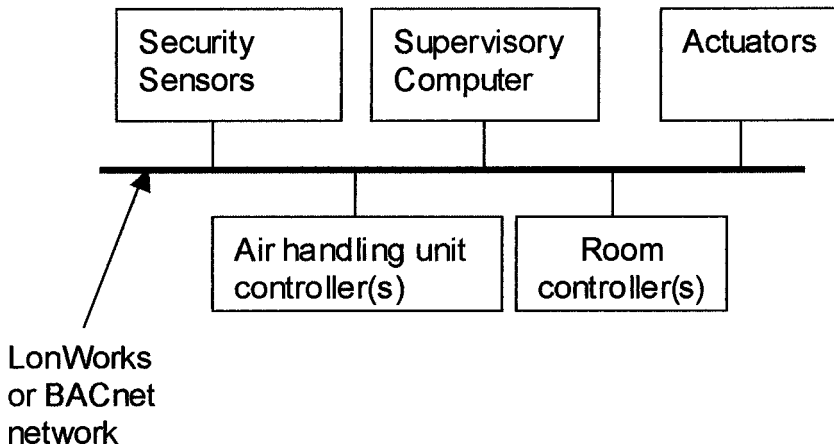
Figure 10-1. Digital control and monitoring system.

Figure 10-2 shows the functional parts of a DDC unit. Analog inputs are electric signals (voltages) coming from circuits similar to those used for electronic control. For example, a resistance temperature detector might be connected to an appropriate bridge circuit, and the varying voltage would be the analog input to the multiplexer. The multiplexer is a solid-state switching device that sequentially connects each analog input to the analog-to-digital converter (A/D converter). This device converts the analog signal to binary digits that are stored in the memory of the microcomputer.

The microcomputer compares the digital value of the sensed variable to a set point stored in memory. The resulting error is manipulated by software algorithms to produce a digital output value. Usually the algorithms are digital versions of the proportional plus integral control schemes described in Chapter 1.

The controlled device (a valve or damper actuator, for example) often needs an electronic or pneumatic analog signal as its input. The digital output signal can be converted to an analog electronic signal via the digital-to-analog converter (D/A converter). The analog electronic

signal can be further transformed to a pneumatic signal through an electronic-to-pneumatic transducer (see Chapter 3). Some controllers produce a varying pneumatic signal by sending electronic pulses to a stepping motor attached to a pressure regulator.
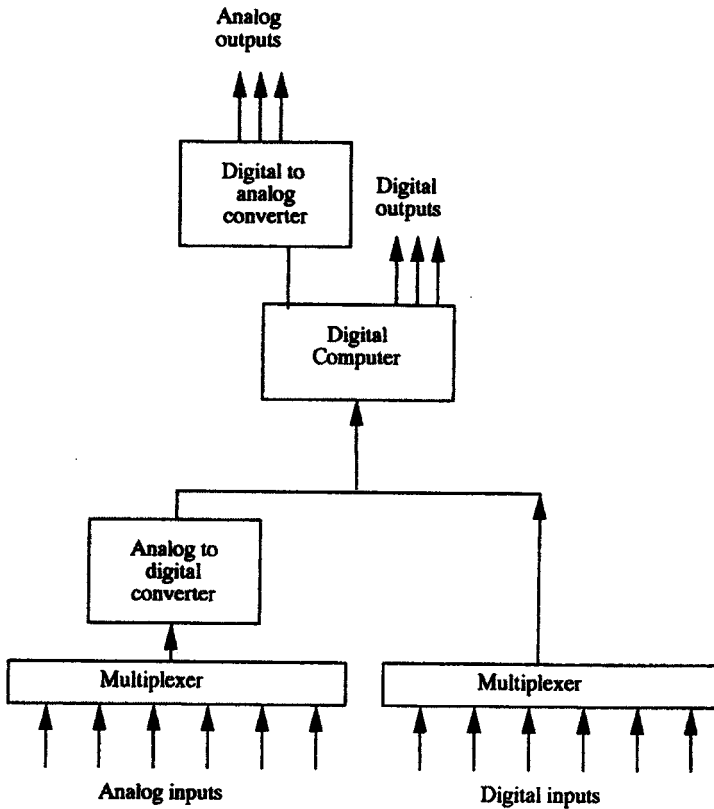
Figure 10-2. Digital controller functional diagram.

Digital inputs and output also can be part of a direct digital control system. Digital input (DI) signals are contact closures or openings, showing status or alarm conditions in a two-position mode (on/off, open/closed). Digital output (DO) signals are two-position signals (full voltage or zero volts) used to implement commands (start/stop,

open/close). They can also be used to pulse a stepping motor or to reposition an electric actuator by timing the length that the output is "on."

Modern digital controllers vary in size and complexity, depending on their intended application. They can be categorized as adjustable, configurable, or programmable.

The simplest controllers are adjustable only. They are often single-loop controllers (modulating a variable-air-volume box to control room temperature, for example). The set point of the controller and sometimes the controller gain settings are the only variables that can be adjusted by the user. The software implementing the control algorithms cannot be altered.

Configurable controllers usually are designed to control several components in a system (all the dampers and valves in a variable-volume air handling system, for example). Although the software cannot be altered by the user, the controller can be "configured" by determining which of the programmed functions are to be used and by specifying certain control parameters. For example, an air handling unit configurable controller might have an economy cycle as a user-selectable option. Other options might include cold deck reset and preheat coil control. Configurable controllers are more powerful than adjustable controllers but the software is application-specific (a configurable air handling unit controller cannot be used to control a chiller, for example).

Programmable controllers are the most flexible but require more user skill and training than other types. Many of the first DDC units used for air-conditioning system were programmable-type controllers. Users (usually control company engineers) wrote software for each application. Programming languages varied with vendors - some being completely unique, others resembling BASIC or C. Today, programmable controllers are selected when unique control functions are desired, or when a configurable controller does not fit the application. Modern units can be programmed by using a graphical

user interface implemented on a personal computer (PC). Software then is downloaded to the programmable controller.

In smaller systems, programmable or configurable digital control units may be all that is required. In more complicated applications, a supervisory computer, a PC, is used to start and stop air handling units, monitor system status, sound alarms, adjust set points, and perform other monitoring and intervention functions. The supervisory computer also supports the operator interface, allowing the user access to all system data.

In extremely large systems, more than one supervisory computer may be needed. Here, an additional computer may be provided to perform data management functions so that operators are not overwhelmed with data. Trend logging and automatic troubleshooting might be programmed as functions for the data management computer.

In modern systems, all digital control and supervisory computers communicate on a common computer network. This means that data are available to all levels of the hierarchy. This approach can simplify programming. For example, adjustable VAV box controllers can provide room temperature data to be used by a programmable air handling unit controller to reset the discharge air temperature. A supervisory computer can use the same room temperature data to monitor system performance, sounding an alarm when a room becomes too hot or too cold.

Figure 3 shows a DDC device that combines many features. This device is a combination controller/actuator for a variable volume box. The actuator is bolted over the shaft of the VAV box. The room temperature sensor is connected to the unit. The two tubes at the top of the unit are connected to the flow sensor in the VAV box –a pressure sensor is build into the controller/actuator. The unit also accepts other inputs such as signals from occupancy sensors or CO_2 sensors. An output signal to drive a heating coil is also provided.

The unit is connected to the network so that it and the other units on the system can communicate with an air handling unit controller (see

Figure 10-4). Inputs and outputs from the controller/actuator can be used by the air handling controller logic to adjust air flow or temperatures. The supervisory PC, also on the network, can monitor temperature, damper position, and other inputs and take actions as needed.
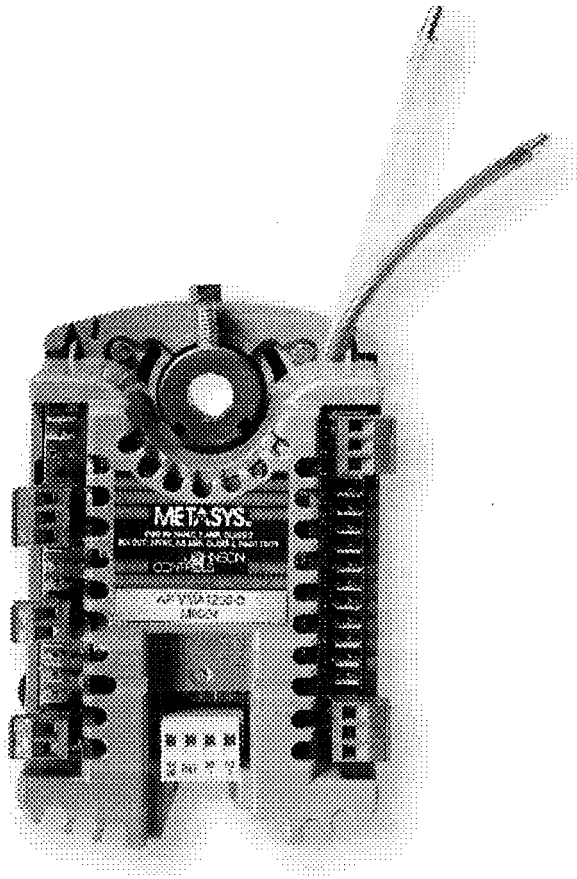
Figure 10-3. Combined controller actuator (Courtesy Johnson Controls)
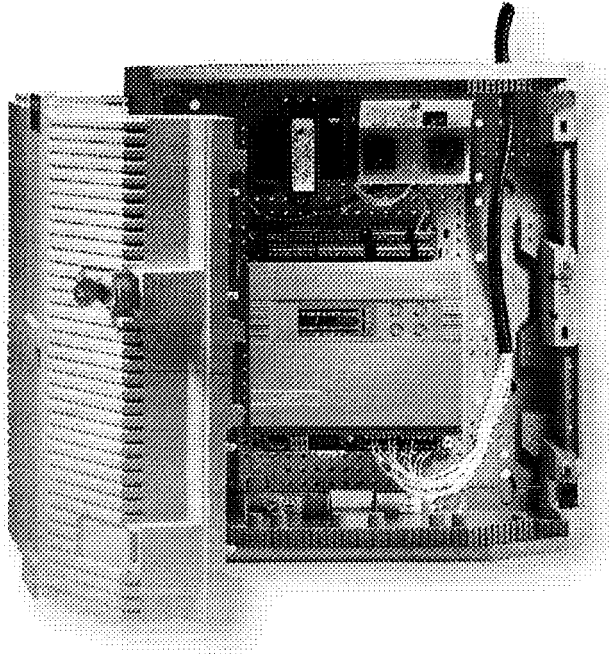
Figure 10-4. Air-handling unit controller (Courtesy Johnson Controls).

For example, damper position signals from the controllers/actuators could be used by the VAV air handling unit controller to modulate fan speed to meet actual space requirements.

10.4.3 Security and Fire Reporting

Security and fire-reporting functions are fundamentally the same monitor-control-alarm functions used in HVAC supervision. The systems that are used for HVAC control thus can easily be adapted to these additional functions. In some localities code authorities require the use of proprietary and UL-labeled systems for fire reporting. The basic HVAC system with a few special features will meet this need.

Fire reporting systems, particularly in high-rise buildings, require a special control panel for fire department use. The computer-type system is well fitted for this arrangement, with the special panel protected against unauthorized use by a key lock or a special code. Many HVAC control manufacturers have a version of their HVAC supervisory control adapted especially for fire and/or security use.

10.5 BENEFITS OF THE COMPUTER SYSTEM

A computerized system has some interesting and valuable side benefits.

10.5.1 Optimization

One side benefit is a program that will continuously analyze performance of the various systems, compare actual performance with some ideal standard, and make adjustments to approach this standard and thus improve efficiency.

10.5.2 Data Management

Because the computer can gather lots of data about operating conditions, it can be programmed to analyze and summarize the data in ways that are useful for operators, owners and tenants. Some of the data may be very valuable for evaluating the existing design and improving future designs.

10.5.3 System Commissioning

Direct digital control and building automation systems can be used to great benefit in commissioning a new HVAC system. In some installations, significant parts of the commissioning activity can be automated such as sequence verification. Some components can also be

commissioned from a central console, using data provided by the components while comanding the component to "go through its paces."

Some controllers are now being offered with self tuning capability, providing better operating performance over the whole range of operation of a control loop.

10.5.4 Maintenance Schedules

The regular preventive maintenance programs can be stored in the computer memory and printed out on schedule to remind the operator that they must be performed.

Maintenance sensors can be made a part of the monitoring equipment. Such sensors will detect overheating of motors or bearings, excessive vibration, loud and extraneous noises, and numerous other malfunction indicators.

10.5.5 Energy Consumption

Where a central chilled water or heating system serves many tenants, it may be necessary to meter the services supplied to each tenant. Special meters with outputs that can be read by the computer may provide an automatic measuring and billing service. Further, any sudden change from normal in a tenant's consumption can be detected, and cause a warning message to be printed for investigation by operating personnel.

10.6 TRAINING FOR MAINTENANCE AND OPERATION

One of the most important considerations with a computer-based control system is the need for properly trained and motivated personnel for operation and maintenance. Computer control systems generally

require skills and training in electronics and software programming as well as knowledge of the HVAC system being controlled.

The operating people preferably should be involved during design and construction of the system, with training furnished by the contractor. Training should continue after the system is installed and working.

10.7 SUMMARY

Digital control of HVAC systems eventually may replace electronic and pneumatic systems. If there is a risk with this technology it is that designers may try to implement control schemes that are unnecessarily complicated, confusing the building operator.

A promising aspect of digital control is the possibility of developing adaptive configurable controllers that can improve system stability and performance. These configurable controllers can be thoroughly tested and debugged as part of their development.

11

Psychrometrics

11.1 INTRODUCTION

Psychrometrics is the branch of thermodynamics devoted to the study of air and water vapor mixtures. Because moist air is usually the final transport medium used in the air-conditioning process, psychrometrics is of great interest to the HVAC system designer and operator.

This chapter will present a simple discussion of psychrometric charts and their uses with a minimum of theory. For a detailed theoretical discussion and complete psychrometric tables, see the ASHRAE *Handbook, Fundamentals* volume, current edition.

11.2 PSYCHROMETRIC PROPERTIES

The properties of the air/water vapor mixtures that are used in HVAC design include dry bulb, wet bulb, and dew point temperatures; relative humidity; humidity ratio; enthalpy; density; and atmospheric pressure.

11.2.1 Temperature

Dry bulb temperature (DB) is measured with an ordinary thermometer. "Temperature," when not otherwise defined, means dry bulb temperature. Wet bulb temperature (WB) is measured using a thermometer with a wet cloth sock wrapped around the bulb. The air

being measured is blown across the sock (or the thermometer is moved through the air), allowing moisture to evaporate. Evaporation has a cooling effect that is directly related to the moisture content of the air. Wet bulb temperature thus will be lower than dry bulb temperatures unless the air is saturated (100% relative humidity). The difference between the two temperatures is called the wet bulb depression. Dew point temperature (DP) is the temperature to which a given sample of air must be cooled so that moisture will start condensing out of it. When the air is saturated, the dry bulb, wet bulb, and dew point temperatures will all be equal.

11.2.2 Relative Humidity

Relative humidity (RH) expresses the relationship of the amount of moisture in the air to the amount the air would hold if saturated at that dry bulb temperature. It is the ratio of the partial pressures of the water vapor at the two conditions. (See the ASHRAE *Handbook*.) Percent humidity is not the same as relative humidity, and is not used in HVAC design.

11.2.3 Humidity Ratio

Humidity ratio, sometimes called specific humidity, designated by the symbol "w," is the amount of water in the air expressed as a ratio of the mass of water per unit mass of dry air (pounds per pound). Sometimes grains of water per pound of dry air is used. One pound equals 7000 grains.

11.2.4 Enthalpy

Enthalpy, designated by the symbol "h," is the sum of the internal energy plus the pressure-specific volume product ($h = u + pv$), in Btu's per pound of dry air. As used in psychrometrics it is not an absolute value, and relates to an arbitrary zero, usually at 0°F . For this reason, differences between enthalpies are used but not ratios.

11.2.5 Density and Specific Volume

Density refers to the mass of the moist air per unit volume, with units in this text of pounds of air per cubic foot. Specific volume is the reciprocal of density.

11.2.6 Atmospheric Pressure

Variations in atmospheric pressure due to elevation above or below sea level have an important effect on the values of the various properties. This is so because the total pressure of the mixture varies with atmospheric pressure whereas the partial pressure of the water vapor in the mixture is a function only of dry bulb temperature. High-altitude tables and charts are available for elevations to about 7500 feet above sea level, and the Bureau of Mines provides a low-altitude chart for depths to about 10,000 feet below sea level.

11.3 PSYCHROMETR1C TABLES

Tables of psychrometric properties are available from several sources, including the ASHRAE *Handbook, Fundamentals* volume. For elevations to about 2000 feet above or below sea level the "standard pressure" tables may be used. For higher and lower elevations, new tables can be calculated using basic psychrometric equations.

11.4 PSYCHROMETRIC CHARTS

A basic tool used in HVAC design is the psychrometric chart. This chart is simply a graphical representation of the properties described above. The ASHRAE chart, Figure 11-1, is used for illustration in this

text and is very helpful in illustrating HVAC cycles and finding state points at various stages of each cycle.

Figure 11-1. ASHRAE psychrometric chart. (Reproduced by permission.)

11.4.1 State Points

Any point on the chart can be a state point. It can be defined and located by the values of any two properties. Once the point is located, the values of all other properties can be read (Figure 11-2).
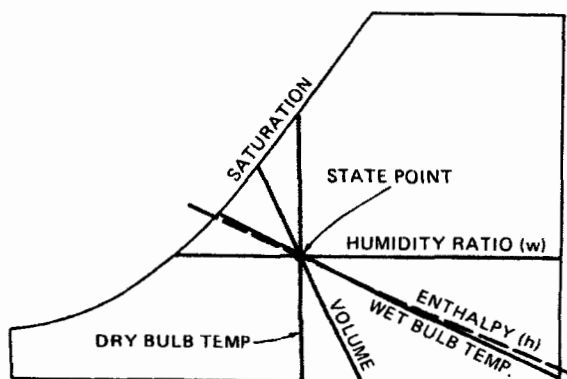
Figure 11-2. Relationships of lines of properties on the psychrometric chart.

Notice the relationship of the various lines on the chart. The basic coordinates on which the chart is drawn are enthalpy (h) (lines sloping down from left to right) and humidity ratio (w) (horizontal lines). Dry bulb temperature lines are nearly vertical and not quite parallel, but are uniformly spaced. Wet bulb temperature lines slope down from left to right and are not parallel to the enthalpy lines or to each other. They are not uniformly spaced; intervals between lines increase with temperature.   A series of curving lines represents relative humidity, with the saturation curve equal to 100% RH. Saturation means that the air is holding all the moisture it can at that temperature and pressure. RH lines are almost but not quite uniformly spaced.

Lines of constant specific volume slope down from left to right and are parallel and uniformly spaced.

As Figure 11-2 shows, any point represents an intersection of all six

properties, and values can be read by interpolation. For clarity, most of the enthalpy lines are shown only at the edge of the chart and may be read in the body of the chart by using a straightedge.

Computer algorithms for psychrometric calculations have been published, and several microcomputer programs are available that incorporate psycrometric calculation.

11.5 PROCESSES ON THE PSYCHROMETRIC CHART

Many HVAC processes can be represented as straight lines connecting two or three state points on the psychrometric chart.

11.5.1 Mixing

Mixing of two air streams can be shown on the chart as a straight line connecting the state points of the individual air streams. The state point of the mixture will fall on the line, dividing in into two segments that are proportional to the volumes of the two air streams. For example, Figure 11-3 shows mixing of return air and outside air, with proportions of 80% return air and 20% outside air. Note that the mixture point is closer to the state point representing the larger air quantity. It is 100%-80% or 20 % of the distance from point 1 to point 2 and 100%--20% or 80% of the distance from point 2 to point 1.

Mixtures of three or more air streams require that any two points be used first, with the resulting mixture combined with the third point, and so on.

11.5.2 Sensible Heating and Cooling

The term "sensible" applied to heating or cooling means that no moisture is added or removed as the air temperature is increased or

decreased. These processes are therefore represented on the chart as horizontal lines with a constant value of specific humidity (w). (See Figure 11-4.)
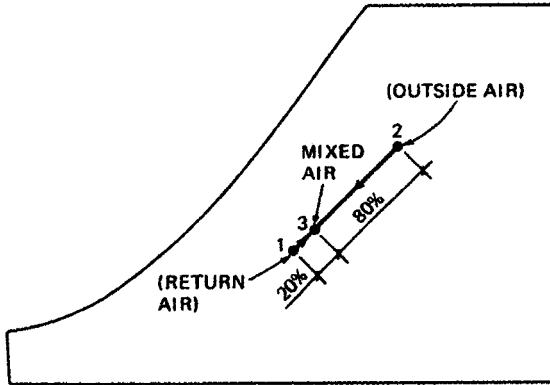
Figure 11-3. Mixing of two air streams. (Example.)
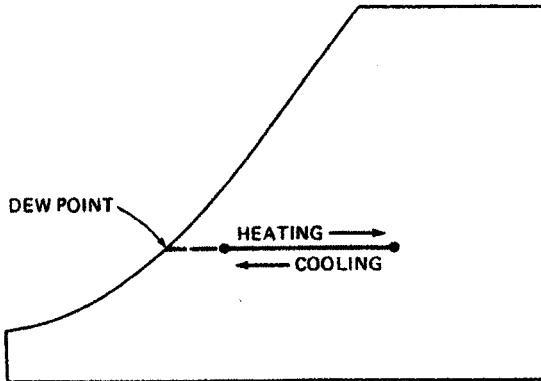
Figure 11-4. Sensible heating and cooling.

11.5.3 Cooling and Dehumidifying

If the process of sensible cooling is continued (in Figure 11-4) until the process line intersects the saturation curve, the air is said to have been cooled to its dew point temperature. Any further cooling requires the removal of moisture. This process would follow the saturation curve down and to the left.

The typical extended-fin-and-tube cooling coil does not have the capability to cool all the air passing through it to saturation and beyond. Some percentage of the air stream will pass through the coil without any contact with tubes or fins. This is known as the coil bypass factor and is typically 5% to 10%, with smaller factors resulting from increased numbers of rows and closer fin spacing.

The result is actually a mixture, with some air cooled to the apparatus dew point and some unchanged. This is shown graphically in Figure 11-5.
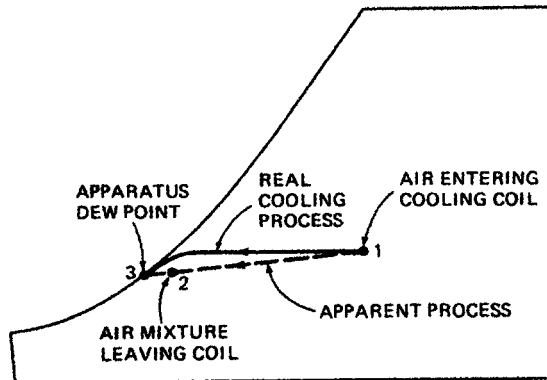
Figure 11-5. Cooling and dehumidifying.

11.5.4 Chemical Dehumidifying

Chemical dehumidification is used to obtain very low humidities. It is usually an adsorption process, using silica gel or some similar moisture adsorbent. The process is an indeterminate curve on the chart, with the final state point determined from the equipment manufacturer's data. The air temperature always increases. (See Figure 11-6.)
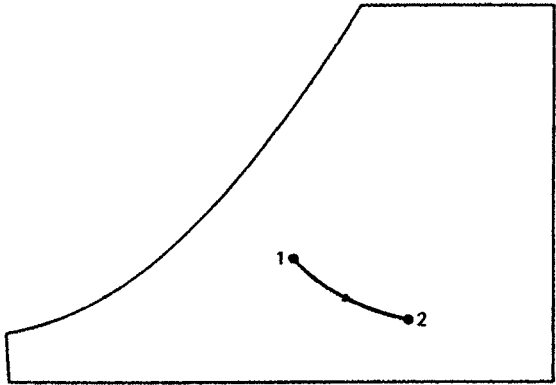
Figure 11-6. Chemical dehumidification.

11.5.5 Humidifying

Humidity may be added to the air stream in various ways, including evaporative cooling, as described in Section 11.5.6. If humidity is added by a steam grid or a heated pan, the process may be shown on the chart as in Figure 11-7. The process line slopes up (humidity increase) and to the right (heat added).

Adding humidity by means of unheated evaporator pans or atomizing sprays is equivalent to evaporative cooling.
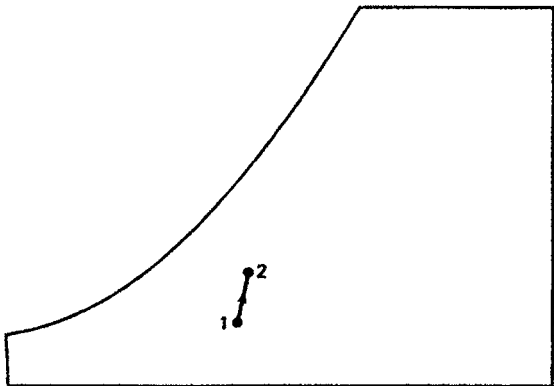
Figure 11-7. Humidification.

11.5.6 Evaporative Cooling

Evaporative cooling is a process in which water vapor is added to the air stream by adiabatic evaporation. That is, no heat is added to or subtracted from the system. The heat required to evaporate the water (latent heat) is obtained by cooling the air. On the chart, Figure 11-8, this shows as a constant wet bulb temperature process, sloping up from right to left (increasing specific humidity, decreasing dry bulb temperature). If the evaporative cooling system were 100% efficient, the final state point would be on the saturation curve. In practice, efficiencies range from 80% to 95%.
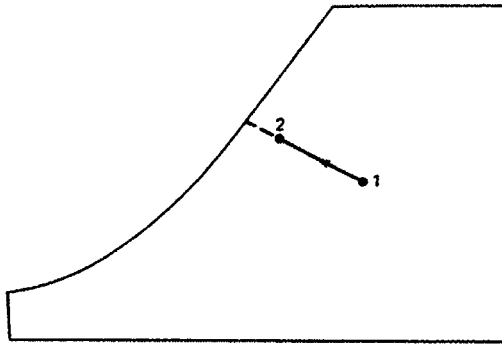
Figure 11-8. Direct evaporative cooling

11.5.6 Direct and Indirect Evaporative Cooling

It is possible to used indirect evapoative cooling to provide sensible cooling prior to evaporatively cooling the air stream. A cooling tower and a cooling coil or other indirect system might be used. This has the effect of moving point 1 on Figure 11-8 to the left, alowing point 2 to be at a colder final temperature thus achieving more evaporative cooling.

11.6 HVAC CYCLES ON THE CHART

A complete HVAC cycle on the chart will include several of the processes described above. A typical cooling cycle is shown in Figure 11-9. Room and outside air conditions are determined from design criteria. For simplicity, return air is assumed to be the same as room air. The mixture condition is based on the minimum outside air required by the design. Point E represents the condition leaving the cooling coil and is determined from the desired room condition, the design air volume flow rate, and the design latent and sensible cooling loads. Some reheat may be required or may occur in the distribution duct system, and it is shown as the line from points E to F. Point F is the condition of the air entering the room.

A typical heating cycle with humidity control is shown in Figure 11-10. At winter outside air design, humidity must be added.
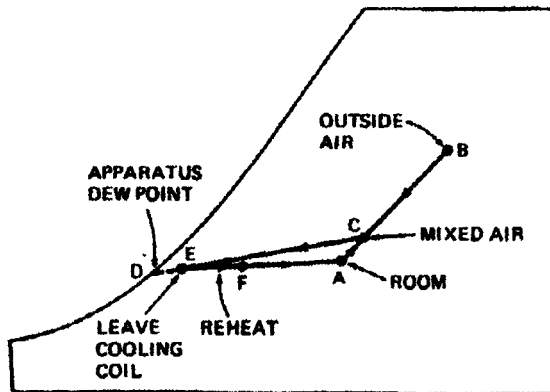
Figure 11-9. Complete HVAC cycle (cooling)

The chart is also useful for investigating the cycle at intermediate (off design) outside conditions. Figure 11-11 shows a cycle based on outside conditions of about 60°F with high relative humidity. This shows that the mixed air is nearly 100% outside air under economy cycle control, but some cooling is necessary to provide dehumidification, and there is considerable reheat.
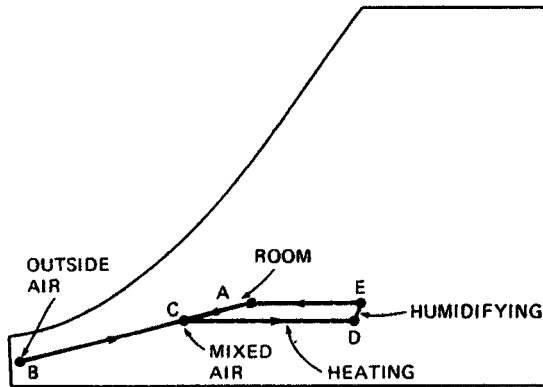
Figure 11-10 Complete HVAC cycle (heating).
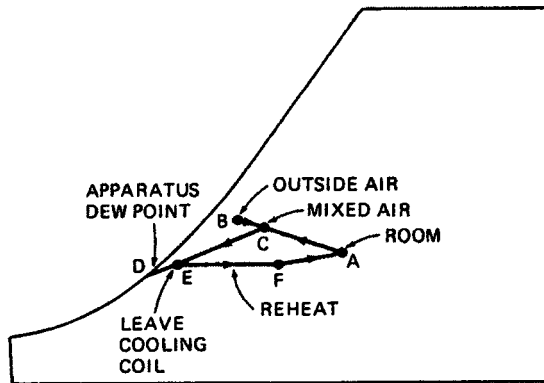
Figure 11-11. Complete HVAC cycle (intermediate).

11.7 IMPOSSIBLE PROCESSES

Sometimes it is not possible to go directly from one state point to another. That is, although a line can be drawn on the chart, the process is thermodynamically impossible. Drawing the cycle on the chart will clearly show the impossibility and suggest alternative ways to accomplish the desired result.

A common problem is the "missing ADP" (apparatus dew point). The process of cooling air through a cooling coil requires that there be an equivalent coil surface temperature that is on the saturation line and in line with the entering and leaving coil conditions. If the specified coil entering and leaving air conditions lie on a line that does not intersect the saturation curve, as in Figure 11-12, then the process is physically impossible. The required result can be obtained by moving the leaving coil condition to the left until an ADP is possible, and then adding a reheat process to get to the final state point (Figure 11-13). Of course, the refrigerant or chilled water in the cooling coil must have a temperature below the ADP.
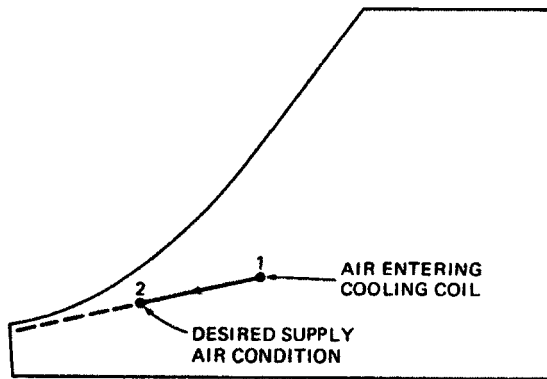
Figure 11-12. Missing apparatus dew point.
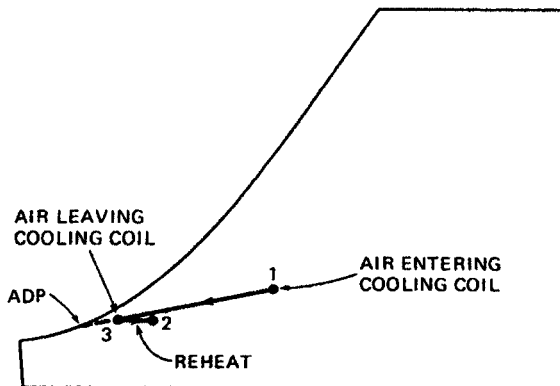
Figure 11-13. Correcting for missing ADP.

11.8 EFFECTS OF ALTITUDE

Many locations where air conditioning is used are at altitudes of several thousand feet above sea level. As noted above, the difference in pressure due to altitude becomes significant above about 2000 feet above sea level. Data obtained from a standard pressure chart may be in error at higher altitudes. High-altitude charts are available and should be used when appropriate.

The effect of decreasing atmospheric pressure is to "expand" the chart (Figure 11-14). That is, given an unchanged coordinate grid of enthalpy and humidity ratio, as total pressure decreases the chart is affected as follows:

- Dry bulb lines are unchanged.
- The saturation curve and RH curves move upward-and farther apart.
- Wet bulb lines move farther apart.
- Volume lines move to the right and upward.
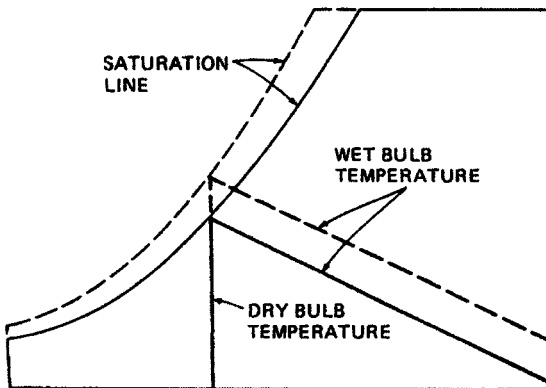
Figure 11-14. Effect of altitude on the psychrometric chart. Atmospheric pressure decreases with elevation above sea level. Solid lines – lower altitude (higher pressure). Dotted lines -- higher altitude (lower pressure).

11.9 SUMMARY

Psychrometric chart applications to control design are used extensively in this text and psychrometrics an essential tool in the design of both HVAC systems and their controls.

Several computer programs for providing psychrometric properties and solving psychrometric problems are available. (see Klein,1992).

12

Central Plant Pumping and Distribution Systems

12.1 INTRODUCTION

The term "central" plant refers to a grouping of one or more chillers and/or boilers supplying chilled and hot water (or steam) to HVAC units located at various points in a building or a complex of buildings. The plant is not necessarily centrally located with respect to the HVAC units. Some plants may have only one or two elements, whereas others may contain a dozen or more chillers and boilers. Some large plants, which have grown over a period of years as a campus developed, have been spread out in two or three locations as the originally assigned spaces have become too small.

The discussions in this chapter are concerned with plants of two or more chillers and boilers circulating normal-temperature hot and chilled water to a scattered group of HVAC units. Steam and high-temperature hot water are excluded. The typical plant has chillers, boilers, circulating pumps, expansion/pressurization equipment, makeup equipment, chemical feeders, controls and accessories. Normally the chilled water supply temperature is between 40°F and 45°F with return water design temperature 12 to 15 degrees higher. Hot water system supply temperatures are usually 160°F to 200°F with return temperatures 40 degrees lower. The hydraulic principles and energy conservation procedures are the same regardless of system size. Diversity becomes a greater factor as system size increases.

12.2 DIVERSITY

Diversity is very important in the design and operation of a central plant, for two reasons:

- An HVAC unit seldom or never operates at peak design conditions.

- In a group of HVAC units, all units do not reach peak load at the same time.

Therefore, central plant capacity need not be equal to the sum of the HVAC unit design loads. The amount of allowable reduction is known as the diversity factor and is expressed as a decimal ratio of the connected load to the central plant capacity, connected load being the sum of the design loads of all the HVAC units served by the central plant. For example, typical diversity factors for a college campus are 0.60 to 0.70.

12.3 CONSTANT FLOW SYSTEMS

Almost all small central plants and many larger ones are operated as constant flow systems (Figure 12-1). The same quantity of water is circulated at all times, regardless of load. At light loads the difference between supply and return water temperatures may be only one or two degrees. Chillers and boilers may be turned off, but water continues to circulate through all equipment to maintain flow.

Three-way valves are used on HVAC unit coils. Flow through the coils varies with load, but total system flow stays more or less constant as supply water is bypassed to the return. Attempts to save energy by turning pumps off will usually result in inadequate flow and/or pressure at the end-of-the-line HVAC units. Turning off a chiller (or boiler) while continuing to circulate water through it results in a mixed supply temperature that is usually higher than design.
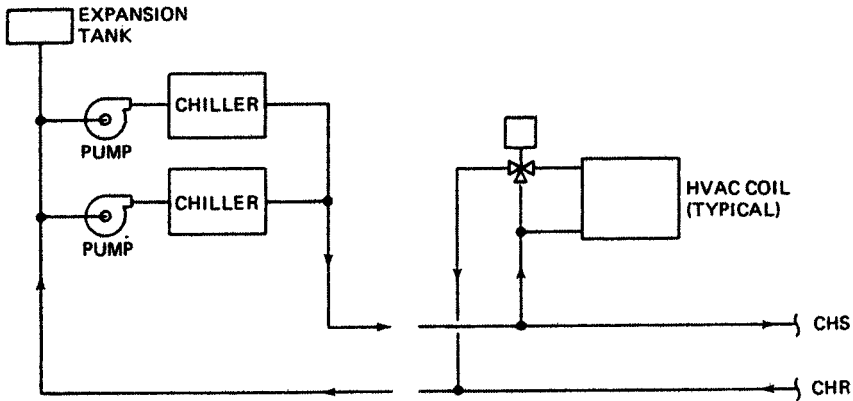
Figure 12-1. Central plant; constant flow arrangement.

Controls alone cannot improve the performance and decrease the energy consumption of such systems. System redesign is required to arrive at one of the variable flow methods described below.

12.4 VARIABLE FLOW SYSTEMS

One reason for using constant flow systems is the need to maintain a constant water flow rate through any on-line boiler to a chiller to prevent damage and maintain efficiency. A variable flow system is designed to do this while allowing the flow in the distribution system to vary with load. This also will maintain the system design temperature difference between supply and return, which allows more efficient operation of chillers and boilers.

Two general arrangements are used, as discussed below.

12.4.1 Bypass Control

The bypass arrangement in Figure 12-2 uses a pressure-operated bypass valve and throttling (two-way) valves at the HVAC unit coils. As the HVAC unit control valves modulate, the distribution system flow will vary. This results in a change in differential pressure between supply and return mains that is sensed by the pressure differential controller. The controller then modulates the bypass valve to compensate. Flow through the chillers remains essentially constant.
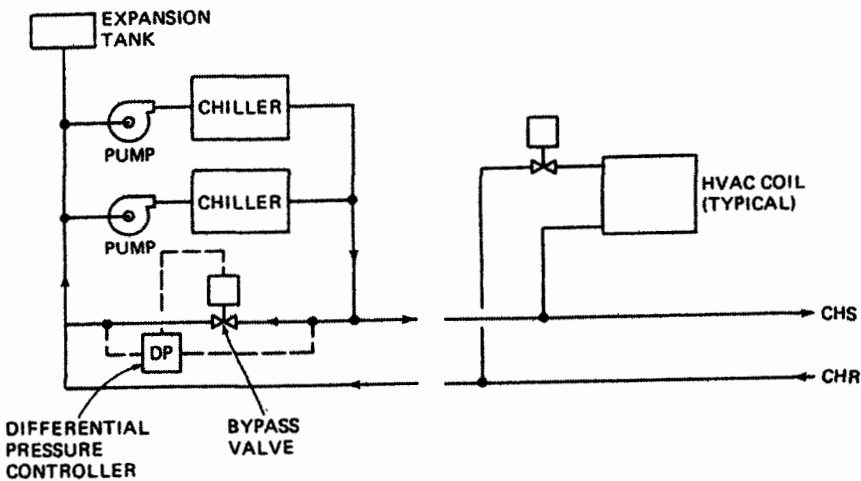
Figure 12-2. Central plant; bypassflow arrangement.

The bypass valve should be sized for the flow rate of one chiller. Limit switches on the bypass valve should be used to show 10% and 90% open. If the plant is started at light load with one chiller and its pump running, the bypass will be in some modulated position. As the HVAC load increases, the bypass will modulate toward the closed position. Limit switch closure can be used to start a second chiller and pump automatically, or alert the operator to do so. Further load increases may bring on more chillers. As the load decreases, the opposite sequence takes place.

12.4.2 Chiller Pumps plus System Pumps

Figure 12-3 shows what may be considered an optimum central plant arrangement. Here the chiller pumps circulate at a constant rate through a closed loop that includes only the chillers and loop piping. Any number of chillers can be on- or off-line, depending on the load. System pumps are used for the distribution, with system supply and return connections to the chilled water loop located close together (points A and B). Distribution flow will vary with load. This short section of piping, common to both pumping circuits, is known as a hydraulic isolator. It must have a low resistance to flow compared to the whole system (keep it short and full-line size) and have no restrictions or valves. When the two circuits are hydraulically isolated, changes in flow in one circuit have no effect on flow rates in the other circuit (it may help to see this by imagining that the short pipe section is a water tank with inlets and outlets from the chiller loop and system loop -- clearly changes in flow in one loop do not effect flow in the other). System pumps may be sequenced to maintain the desired minimum pressure differential between supply and return mains. Ideally, one system pump will be a variable speed type so that a constant differential pressure may be maintained.

Designers must carefully select the pump motors for variable flow, multiple pump systems. Under part load, if one or more pumps are turned off, the lower friction in the system may subject the remaining pumps to lower head and increased flow when compared to the design condition. This condition will require more power for the pumps that are on than is needed under full load conditions.

12.5 DISTRIBUTION SYSTEMS

Piping distribution systems connect the central plant to the HVAC units. There are three arrangements in use: out-and-back, reverse return, and loop.
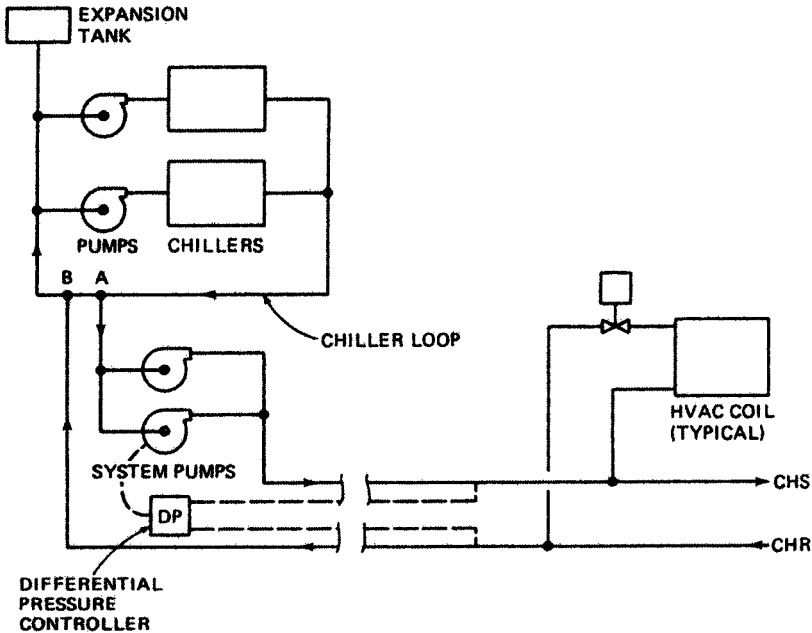
Figure 12-3. Central plant; chiller pumps plus system pumps.

12.5.1 Out-and-Back Distribution

This arrangement, shown in Figure 12-4, is the oldest and most common system. It has the virtue of simplicity because supply and return mains decrease in size equally as branch takeoffs for HVAC units occur. The pumps must be sized to overcome the pressure drop in the entire line plus the losses in the HVAC unit at the end. Thus, pressure differential from supply to return mains may be excessive at HVAC units near the central plant, making them more difficult to control. To clarify this, refer to Figures 12-5 and 12-6. Figure 12-5 shows the hydraulic profile at design water flow rate with balancing valves set properly. In Figure 12-6 at 50% of design flow rate the pressure drops through coils and balancing valves are 25% of design, and the control valve must make up the difference.
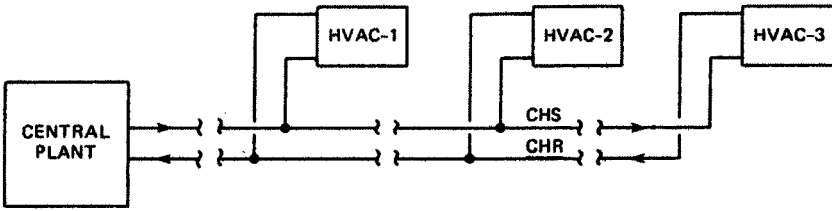
Figure 12-4. "Out and back" distribution.

It is very difficult to add another HVAC unit on this system. Some main piping may become too small, and the entire system must be rebalanced.

(NO BALANCING VALVE ON END-OF-LINE UNIT)
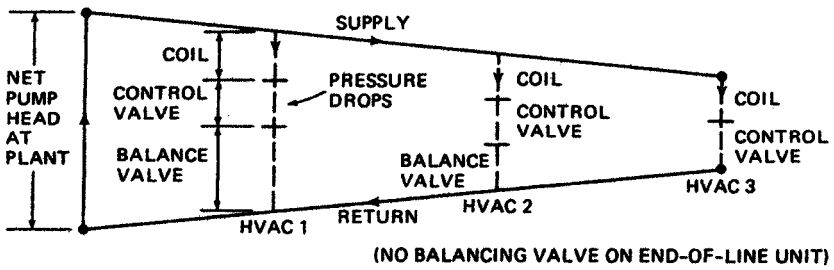
Figure 12-5. "Out-and-back" hydraulic profile; design flow rate.
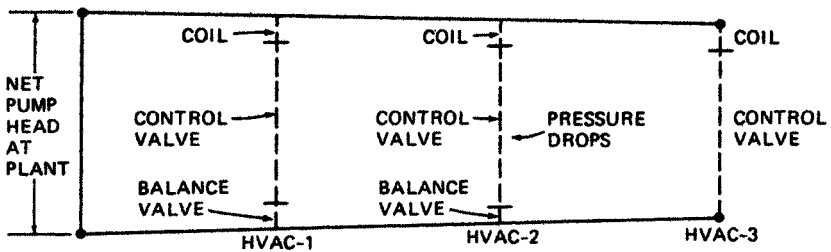
Figure 12-6. "Out-and-back' hydraulic profile; 50% of design flow.

12.5.2 Reverse-Return Distribution

The reverse-return system is designed to equalize distribution pressure drops throughout the system. As shown in Figure 12-7, the return line starts with the first HVAC unit on the supply line and flows parallel to the supply line until end of supply is reached. Then the return line reverses and runs back to the central plant. Thus the total length of distribution piping (and pressure drop) is approximately equal for all HVAC units. This system is more expensive than the out-and-back system but is easier to balance and control. The hydraulic profiles at design flow and 50% of flow are shown in Figures 12-8 and 12-9. Again, it is difficult to add HVAC units to this system.
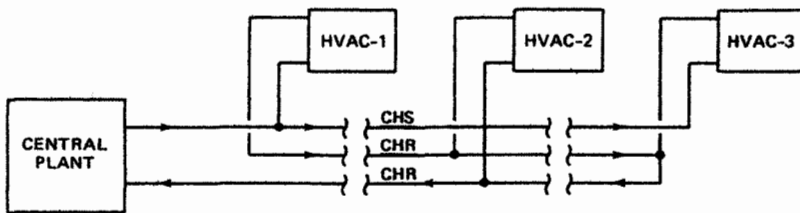
Figure 12-7. Reverse-return distribution.
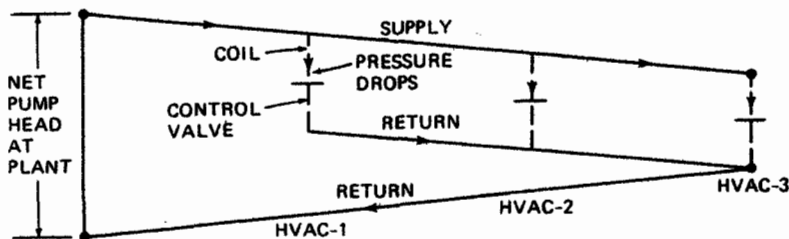
Figure 12-8. Reverse-return hydraulic profile; design flow rate.
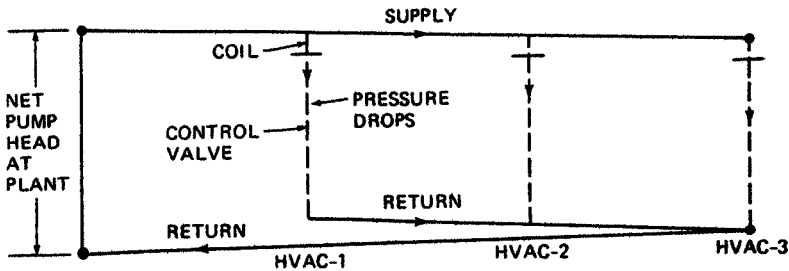
Figure 12-9. Reverse-return hydraulic profile; 50% of design flow.

12.5.3 Loop Distribution

For a large campus, or even a large building, and especially where changes and additions to the HVAC units are expected, a loop distribution system is preferred. The basic scheme is shown in Figure 12-10. The loop mains are of one uniform size throughout. This size is approximately 40% of the initial main size in a comparable out-and-back system for the same design pressure drop. Each loop is hydraulically self-balancing. Flow takes place in both directions, with pressure drop resulting, and at some point there is an equal pressure from both sides, resulting in zero flow at that point. The point of zero flow/balanced pressure will automatically adjust to any changes in load, including addition and deletion of HVAC units or buildings.

The hydraulic profile in Figure 12-11 assumes that a fixed differential pressure is maintained at the chiller plant, like the bypass control shown in Figure 12-2. If the system pump arrangement shown in Figure 12-3 is used, with variable speed pumping and a differential control point out in the system away from the central plant, then system pump head will be decreased at part load, saving energy (See Figure 12-12).
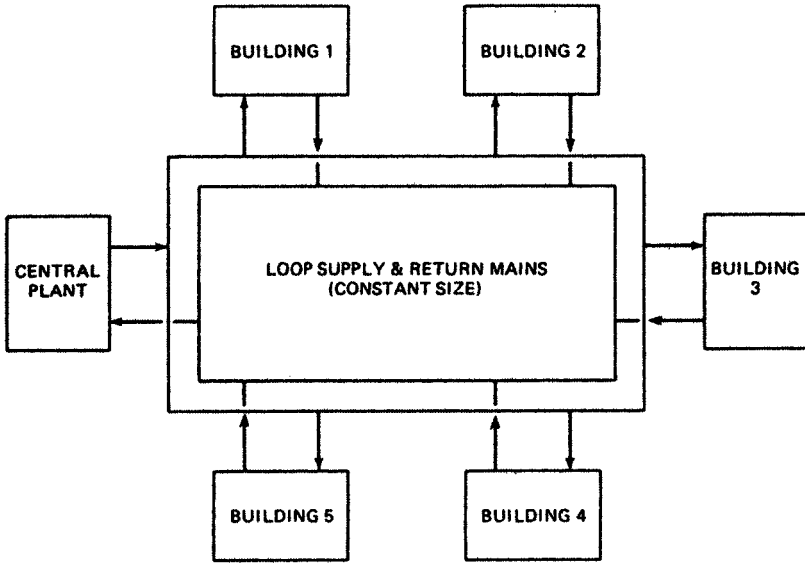
Figure 12-10. Loop distribution.
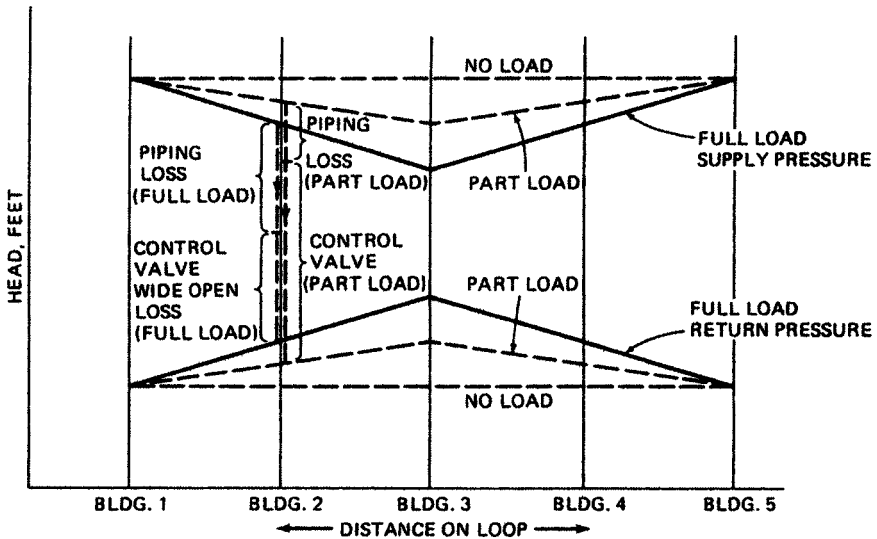
Figure 12-11. Loop distribution; hydraulic profile, plant bypass control.
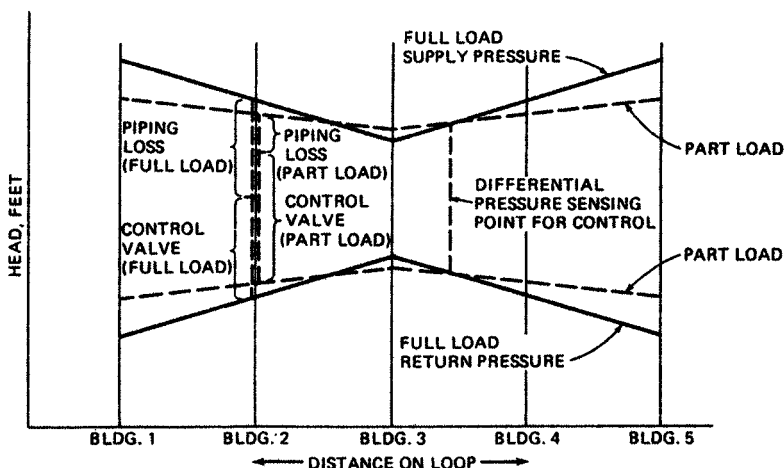
Figure 12-12. Loop distribution; hydraulic profile, variable speed pumping.

12.6 BUILDING INTERFACES

When a central plant serves several buildings, those buildings may vary greatly in size, load, and internal pressure losses. If the system pumps are required to provide sufficient pressure to overcome the building losses, some severe energy penalties will result. It is typical, therefore, to provide secondary pumps, at least at the larger buildings, to avoid this penalty. The important criterion when connecting secondary pumps is to provide hydraulic separation. That is, the operation of the secondary pump should not influence the pressure differential in the primary distribution system. Additionally, it should be possible to control the return water temperature from the building to maximize the water temperature differential in the primary distribution. There are a number of secondary connections in common use, and not all of them meet the above criterion. One that does is shown in Figure 12-13. The thermostat modulates the control valve to maintain a constant return water temperature as the load varies. The control valve "sees" only the pressure differential between primary supply and return because the bypass line has no restrictions. The

bypass line provides hydraulic isolation between primary and secondary systems.
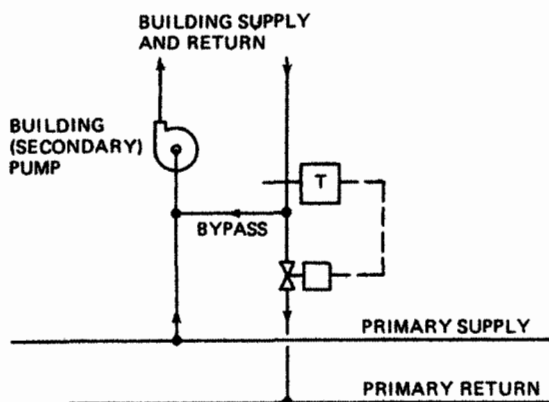
Figure 12-13. Building interface with secondary pump.

With this arrangement the building water temperature will increase as building load decreases. Where a low dew point is required for humidity control, this may not be satisfactory. The return water temperature can be reset as a function of building humidity to satisfy this requirement.

12.7 SUMMARY

Existing central plant systems use many arrangements and control methods. Not all of these are satisfactory in performance and most use excessive pumping and equipment energy. Obviously, there is ample opportunity for improvement in system design and control. This chapter is by no means a comprehensive study and the reader should refer to other more detailed references such as the ASHRAE *Handbook, Systems* volume, Bahnfleth, 1976 and Haines, Jan. and Dec., 1981, in the bibliography.

13

Retrofit of Existing Control Systems

13.1 INTRODUCTION

The "energy crisis" of the 1970s created an awareness of the inefficiency of many existing HVAC systems and their controls. Most systems installed in the fifties, sixties and early seventies were concerned only with control for comfort, and minimization of energy consumption was seldom considered. Also, many existing systems have failed to perform well with respect to comfort or economy.

This chapter will consider some of the more common existing HVAC systems and show how they can be most easily retrofitted to save energy while still providing adequate comfort control. Because most existing control systems are pneumatic, some illustrations will use pneumatic devices. Positive positioners for valves and dampers are recommended, though not always shown.

13.2 ECONOMIC ANALYSIS

It will often be necessary to make an economic analysis of the cost-effectiveness of an HVAC system/control retrofit. Methods for making such analyses are described in many references, including the ASHRAE *Handbook*.

13.3 DISCRIMINATORS

Many of the following diagrams suggest the use of discriminators. When using discriminators, keep in mind the problems that may be encountered, as described in Chapter 7.

13.4 CONTROL MODES

Most existing systems will be using proportional controllers, and it is usually beneficial to replace them with proportional plus integral controllers. An improvement in both energy efficiency and control accuracy should result. See Chapter 1 for a discussion of control modes.

13.5 ECONOMY CYCLE CONTROLS

A surprising number of HVAC systems are operating with fixed outside air quantities. In some special situations or because of geometry this may be necessary, but, in general, the addition of economy cycle controls will save energy. Remember that relief must be provided for outside air in excess of that needed for exhaust and pressurization.

Economy cycle with reset of the mixed air temperature will usually conserve the most energy over the season. Other factors may require modification or limitation of this approach. This scheme is discussed in Chapter 7 and included in the descriptions that follow.

13.6 SINGLE-ZONE SYSTEMS

A single-zone HVAC system with a traditional economy cycle (Figure 13-1) is typical of many existing systems. Energy consumption is low

compared to that of multizone or reheat systems. The energy consumption can be decreased by any or all of several improvements shown in heavy lines in Figure 13-2. These include:

- Sequencing of heating and cooling control valves to avoid overlap (simultaneous heating and cooling). Most, but not all, systems are designed in this way. Sequencing is done by adjusting or replacing springs in the valves. The heating valve should be normally open (NO) with a 3-8 psi spring. The cooling valve should be normally closed (NC) with an 8-13 psi spring range. Then as the thermostat output goes through the range from full heating to full cooling (0-13 psi) the heating valve will gradually close and, in sequence, the cooling valve will gradually open. If a digital control system is used sequencing will usually occur in software and pneumatic or electronic actuators can be used.

- Add reset from room temperature to the mixed air controller. This will allow the mixed air temperature to rise or fall with space heating or cooling demand, and, in particular, will minimize heating energy use.

- Add supply fan variable air volume (VAV) control using inlet dampers as shown or motor speed control. Fan speed should be maximum at maximum cooling load, reducing to a minimum of about 40 to 50 percent at "thermostat satisfied" condition. The minimum should be retained for heating, and must be large enough to provide adequate heating.

13.7 REHEAT SYSTEMS

A typical reheat system is shown in Figure 13-3. Reset of supply temperature from outside air is sometimes, but not always, provided. Reheat systems are inherently wasteful of energy, but much can be done to improve them, as shown in Figure 13-4.
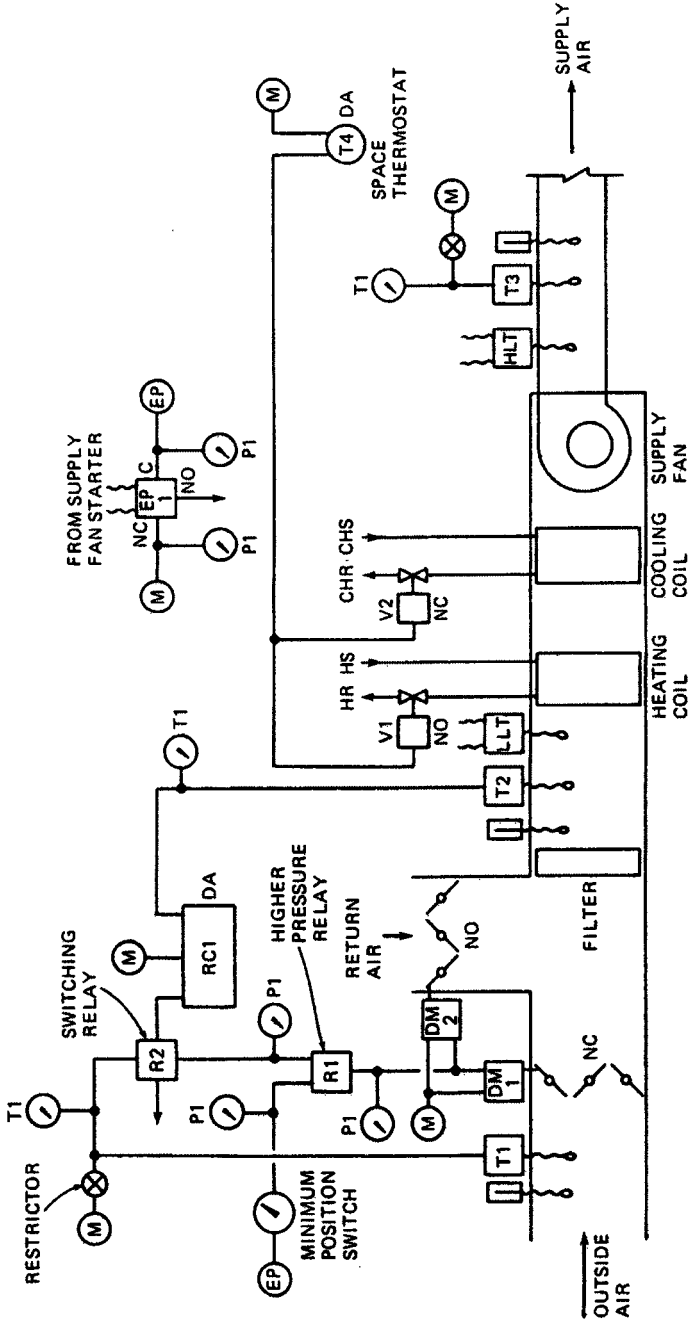
Figure 13-1. Single-zone HVAC System.
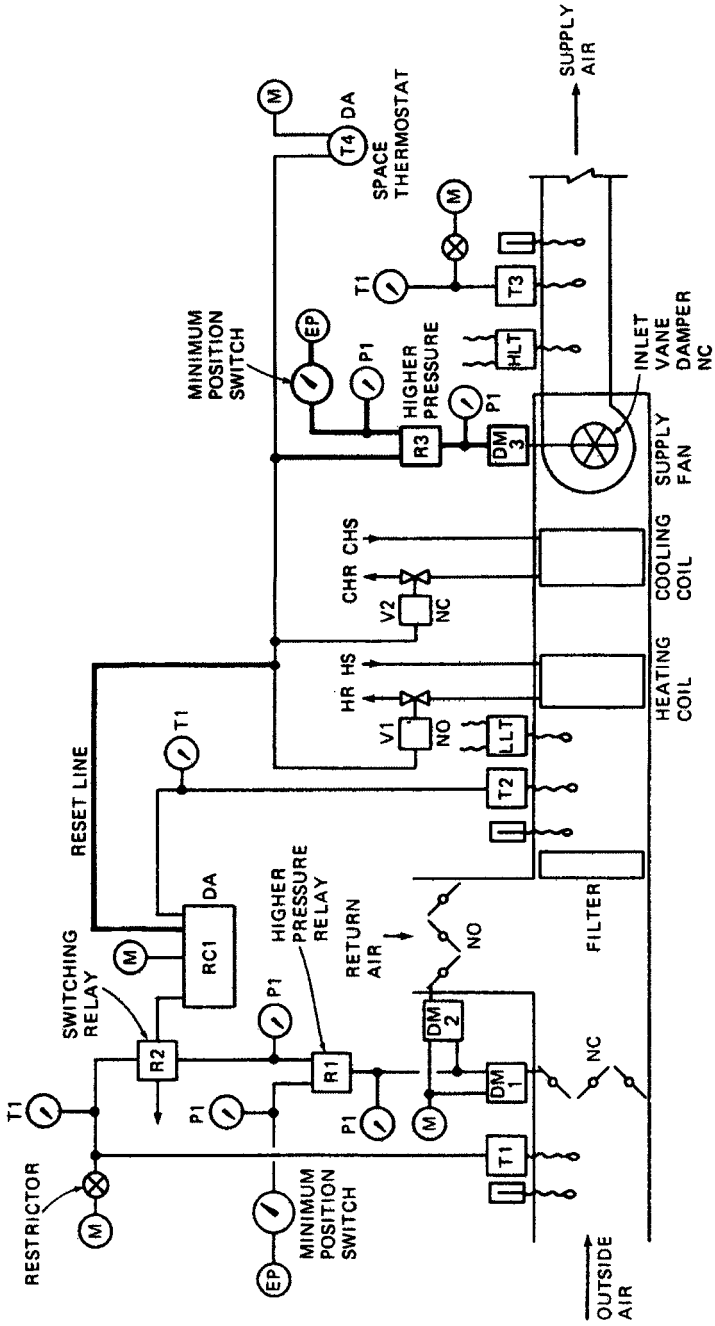
Figure 13-2. Single-zone HVAC system; retrofit.

- Replace existing thermostats with dead-band thermostats. This will minimize reheat but will not, by itself, minimize cooling energy use.

- Provide a discriminator relay (output equal to highest of several input pressures) to reset the supply air temperature to satisfy the zone with the greatest cooling demand. This will minimize cooling and heating energy use. Obviously, there will still be reheat, with energy waste, in some zones.

- Reset the mixed air temperature controller from the discriminator relay. This will minimize preheating and save some cooling as well, especially in mild weather (50-70°F outside).

- Remodel the system to become a VAV system. This will require adding variable volume boxes or dampers at all zones, with a duct static pressure control to control the supply fan. A larger supply fan and motor and higher fan speed may be needed to provide the extra pressure needed for the VAV boxes. There will still be a net saving in fan motor energy use over the conventional system, but the cost-benefits of this revision requires careful study.

13.8 MULTIZONE SYSTEMS

A typical multizone system with economy cycle is shown in Figure 13-5. Space temperature control is obtained by mixing hot and cold duct air through zone mixing dampers. Because one motor drives both dampers simultaneously (constant volume), the use of dead-band thermostats would tend to position the dampers in the 50-50 mixing position and would not conserve energy. The use of summer-winter thermostats is better here. Typically, thermostats are reset manually from heating to cooling as the seasons change.
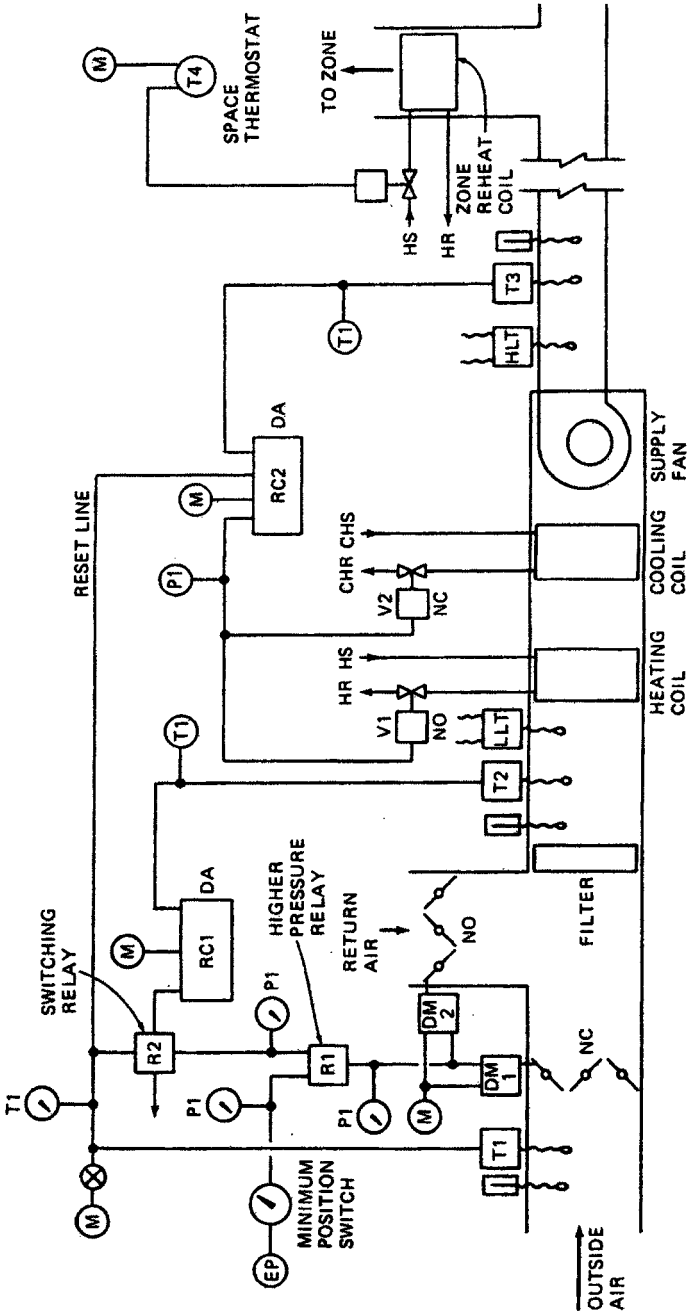
Figure 13-3. Zone reheat HVAC system.
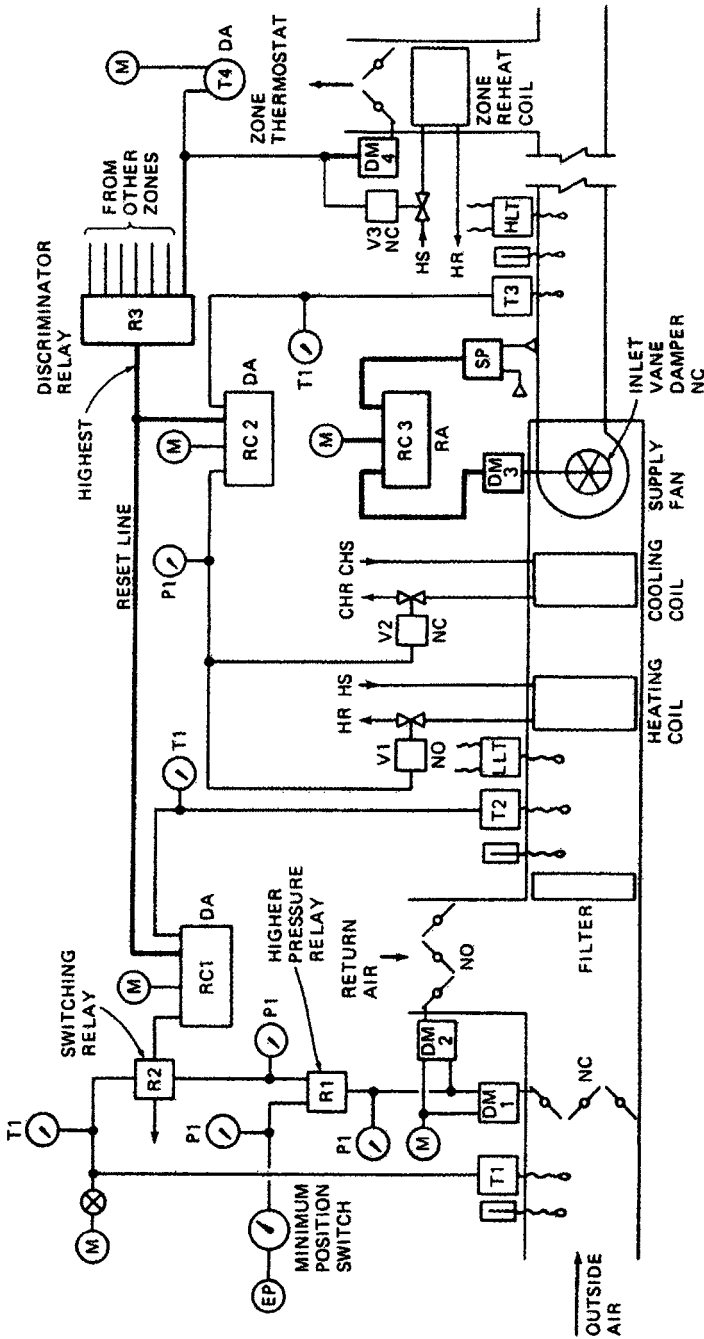
Figure 13-4. Zone reheat HVAC system; retrofit

As discussed in Chapter 7, one way to improve efficiency of multizone systems is to shut the heating coil off in the summer and the cooling coil off in the winter. However, it may be necessary to run both coils in the spring and the fall.

Under these conditions, and especially with the fixed mixed air and cold plenum control set points, multizone systems are notorious energy wasters. Some procedures are helpful:

- Add discriminator relays for reset of hot and cold plenum set points (Figure 13-6). Notice that the lowest pressure output is used for hot plenum reset. This satisfies the zone with the greatest heating demand. The highest pressure output is used for cold plenum reset to satisfy the zone with greatest cooling demand. There will still be reheating and wasted energy but it will be limited to that required to meet zone control requirements.

- Reset the mixed air controller as a function of cooling demand (Figure 13-6). This will minimize reheating requirements.

- Alternatively, reset the heating coil controller based on the outdoor temperature.

- Remodel the system for variable air volume, as described below for dual-duct systems. This is not a simple task, and may not always be cost-effective.

13.9 DUAL-DUCT SYSTEMS

Dual-duct systems are simply multizone systems with hot and cold plenums extended and mixing dampers located near the zone. They have the same problems and solutions as multizone systems:
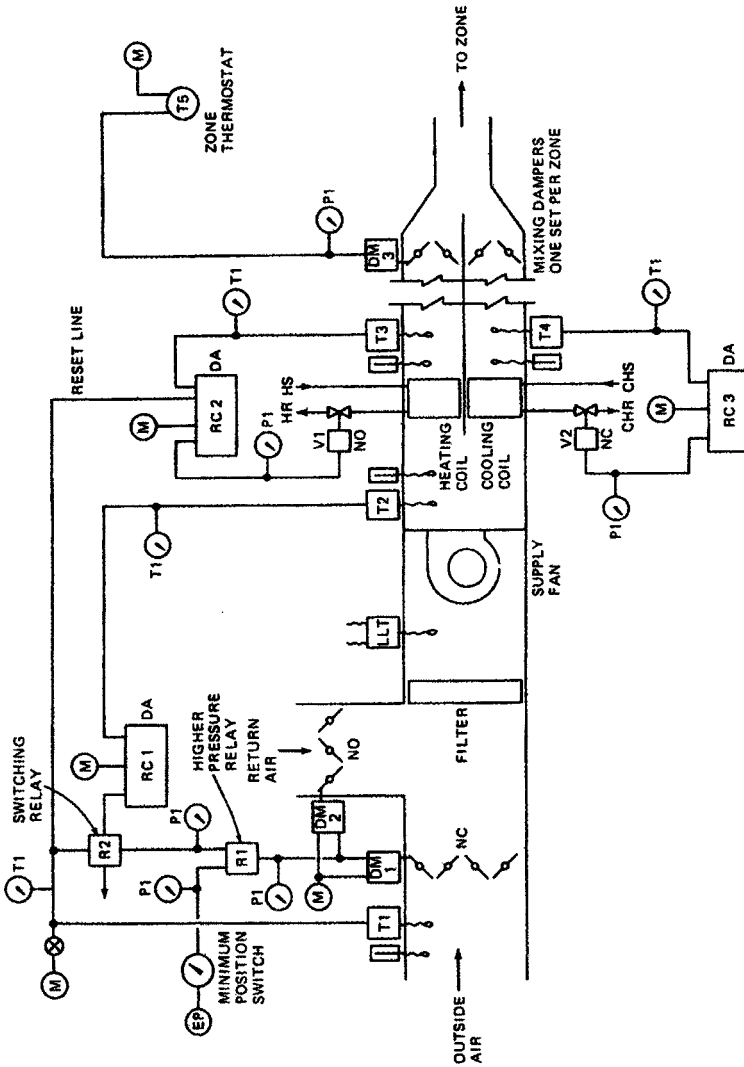
Figure 13-5. Multizone HVAC system.
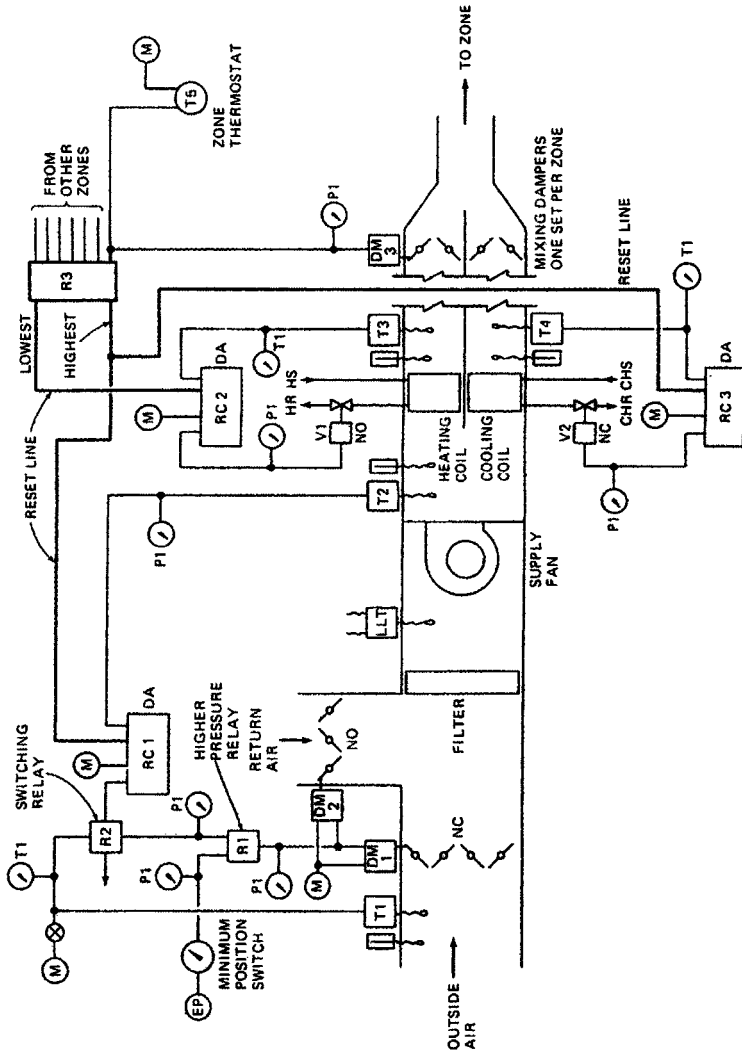
Figure 13-6. Multizone HVAC system; retrofit.

- When discriminator controls are added, it is not usually practical to provide signals from all zones to the discriminator relay. A few typical zones may be selected.

- Dual-duct systems may be converted to variable-volume as shown in Figure 13-7. Most existing systems use single motor mixing boxes. It is necessary to provide separate motors for hot and cold dampers. Then the dampers are controlled in sequence as shown in Figure 13-8. At maximum cooling the cold damper is full open, and the hot damper is closed. As cooling demand decreases, the cold damper modulates toward the closed position while the hot damper remains closed (variable volume). At some minimum position of the cold damper-usually 25% to 30% open-the hot damper begins to open. As the heating demand increases, the cold damper continues to close and the hot damper modulates toward the open position. When the hot damper is 25% to 30% open, the cold damper is closed. There is some overlap and mixing of hot and cold air, but it occurs at minimum air flow. Supply fan volume controls must be provided (either dampers or motor speed controls) using the lower of the static pressures in hot and cold ducts.

- Dual-duct systems also may be converted to "standard" VAV systems, by using the hot and cold ducts in parallel as though they were a common supply duct. The mixing boxes will be replaced with VAV boxes (with reheat coils if needed), and the hot plenum coil will be replaced with a cooling coil. A preheat coil may be required. This is a major remodeling project but is very effective where the air supply might otherwise be inadequate.
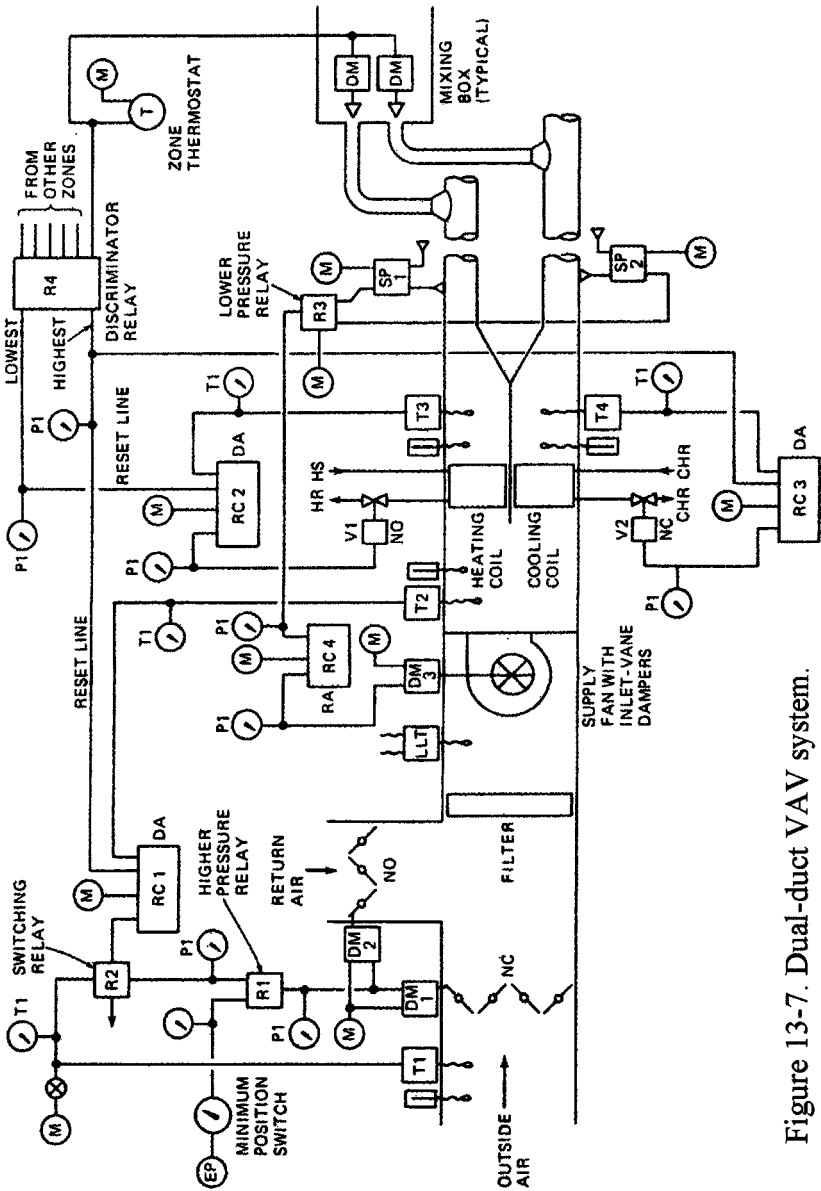
Figure 13-7. Dual-duct VAV system.
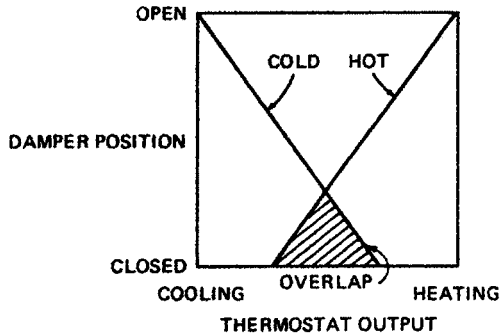
Figure 13-8. Damper relationships; dual-duct VAV.

The typical dual-duct system is designed with a high pressure drop, and a change to VAV may produce dramatic savings in fan power requirements.

13.10 SYSTEMS WITH HUMIDITY CONTROL

Systems serving spaces requiring close control of relative humidity do not easily lend themselves to energy conservation. Close control of space temperature always goes along with humidity control. The simplest way to achieve humidity control is by controlling the dew point temperature at the HVAC unit, and this, inevitably, requires reheat.

Chemical dehumidification requires heat energy for regeneration and additional cooling to remove the heat added to the air stream by the dehumidification process.

There is one method of saving energy: heat reclaim at the HVAC unit. This system was described in detail in Haines, Aug., 1980 and is shown in Figure 13-9. The runaround system provides some or all of the required reheat by precooling the mixed air. This limits the outside air economy cycle somewhat, but overall results in considerable saving in energy use. (See Section 7.7.4.)

13.11   CONTROL VALVES AND PUMPING ARRANGEMENTS

Many chilled and hot water piping distribution systems are designed using three-way control valves at the HVAC units. In a small system with only one chiller (or boiler) normally in use, this is satisfactory. When more than one chiller (or boiler) is needed, then pumping energy can be saved by using throttling control valves and taking advantage of diversity as described in Chapter 12 and Haines, Jan. and Dec., 1981.

In the typical retrofit situation, the coil and valve are piped as shown in Figure 13-10(A) (cooling) or Figure 13-10(B) (heating). The bypass valve shown will not always exist, and in some systems the bypass line may be extremely short. There are three ways of changing the system to throttling control:

- If the bypass valve exists, simply close it off.
- Remove the bypass piping and plug the bypass port of the control valve.
- Replace the three-way valve with a straight-through valve.

The change from three-way to straight-through valves may create some system pressure problems unless a pressure-controlled bypass is provided at the chillers or boilers, and some provision is made for sequencing pumps with load. It is also useful to retain the three-way valve on the HVAC unit at the end of the piping system to provide a bypass to keep the line cold (or hot).
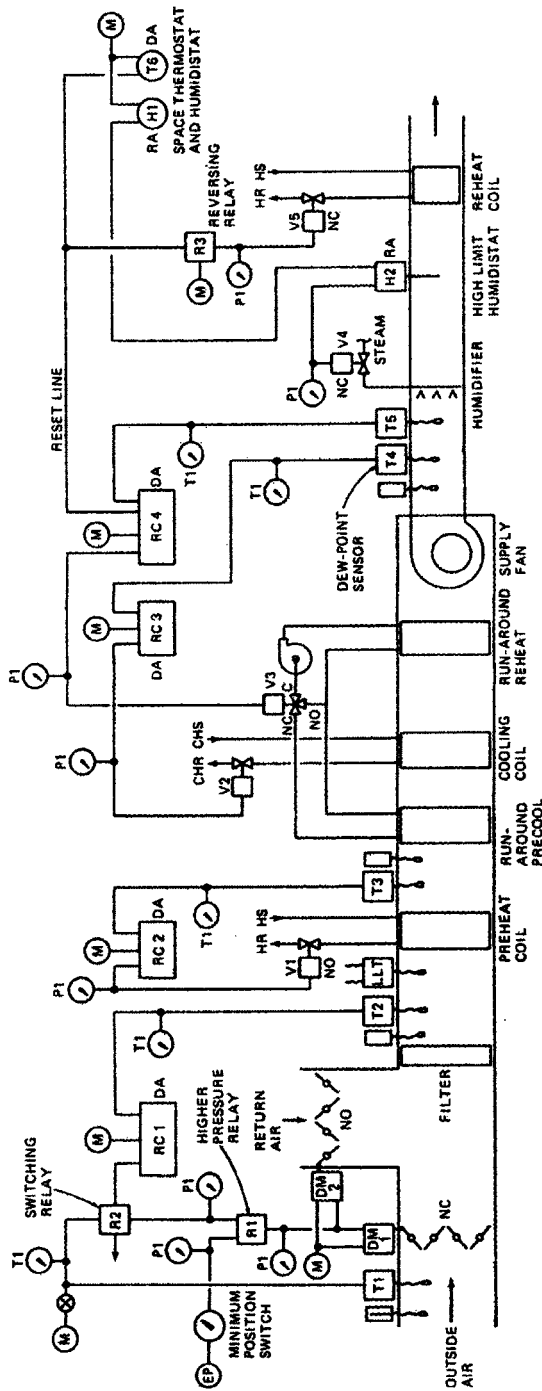
Figure 13-9. Humidity control HVAC system with run around for reheat.
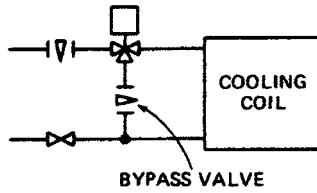
Figure 13-10(A). Three-way valve; cooling coil.
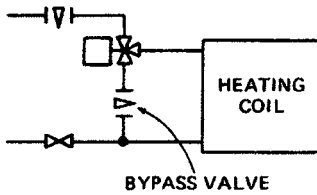
Figure 13-10(B). Three-way valve; heating coil.

13.12   SUMMARY

As noted at the beginning of this chapter, these are suggestions for solving some more common retrofit problems. One should not undertake any modification without carefully applying one's knowledge of HVAC and control fundamentals to figure out the economic and technical feasibility of any proposed changes.

14

Dynamic Response and Tuning

14.1 INTRODUCTION

HVAC system control systems can be especially difficult to adjust or tune because the process gain of the control loop varies with the control point and with a range of other variables. For example, if the air flow rate over a hot-water heating coil is cut in half (as in a multizone application), the gain of the coil (the ratio of the change in outlet temperature to the change in valve position) will increase significantly.

The impact of valve and damper characteristics and authority were discussed in Chapter 5 where we showed that when certain variables are fixed (i.e., air flow rate or available pressure drop), careful matching of components can help give the system a linear characteristic. However, even if the controlled variable changes roughly linearly with controller output, the gain or the slope of this line can change with changing temperatures and flow rates. An obvious example of this is the mixing of two air streams when economy cycle control is used. If the outdoor air is cold, a small change in damper position will result in a large change in the mixed air temperature. If the outdoor and return air are nearly at the same temperature, then gross changes in damper position will have almost no effect on the mixed air temperature. Luckily, good dynamic response probably is not needed in the later case because the outdoor air damper merely needs to go to the full open position and stay there.

In this chapter we will discuss the dynamic response of HVAC subsystems and how this response affects control system tuning.

14.2 DYNAMIC RESPONSE

Figure 14-1 shows the response of typical HVAC subsystems to a sudden change in input. In this example the controlled variable is the temperature leaving a heating or a cooling coil; so we have plotted temperature change on the Y-axis. The X-axis is time. A sudden change in the output from the controller that modulates the coil valve is assumed to have occurred at the point where we start recording system response (at t = 0). The new controller output remains constant after that. As the changing controlled variable (temperature) does not affect the output from the controller, the response over time is called the open loop response.
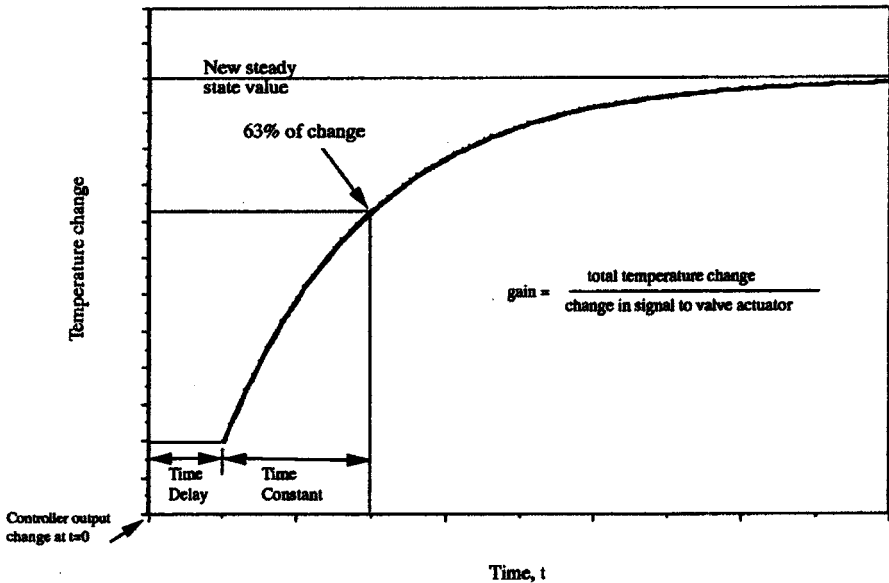
Figure 14-1. Time delay, time constant and gain for a first order system with time delay.

This and many other HVAC systems are characterized by a time delay and what is called first-order response. By this we mean that there is a

delay after the controller output signal changes before the controlled variable changes, and that once the controlled variable starts to change, it asymptotically approaches its new value exponentially with time. One exponential term serves to describe the system response after the end of the time delay; hence the characterization of the system as first order.

Figure 14-1 serves to help define several terms used by control specialists. The time delay is obvious. It is simply the difference in time between when the controller output (the input to the coil/valve system) is changed and when a response is observed. Sometimes the time delay is hard to pick off a system response graph because of a lack of a sudden transition from no response to exponential response. Therefore, some practitioners recommend drawing a straight line tangent to the steepest part of the exponential response curve and noting where this line intersects the initial value of the controlled variable. The time corresponding to this intersection is approximately the time delay.

The time constant is the time it takes for the controlled variable to reach 63% of its final value, not including the time delay.

The system gain is simply the total change in the controlled variable divided by the change in the controller output. For example if a 10% change in the controller output to a coil valve produced a 4^OF change in leaving coil temperture, then the gain would be 4/10 or $0.4^OF/\%$.

Carrying out an open loop experiment in the field to measure the time delay, time constant, and gain is one way to get the information needed to tune a control loop.

14.3 TUNING HVAC CONTROL LOOPS

The primary criterion for a well-tuned controller on an HVAC subsystem is that it be stable over the expected range of operating conditions. To get the appropriate closed loop response without a

broad throttling range, proportional plus integral (PI) control is required for most control loops. Recall that the formula for the output from a PI controller is:

$$O = A + K_p e + K_i \int edt \qquad (14\text{-}1)$$

where :

O=controller output

A=a constant equal to the value of the controller output with no error

e= the error, equal to the difference between the set point and the measured value of the controlled variable

K_p = proportional gain constant

K_i = integral gain constant

We therefore must select two control constants, the proportional gain and the integral gain, to tune the controller. One approach is the so-called quarter wave or Ziegler-Nichols tuning method (Ziegler and Nichols, 1942). This produces a response to a step change in the set point similar to that shown in Figure 14-2.

There is considerable overshoot and some oscillation before the new steady state value is reached. The term 'quarter wave reponse' refers to the fact that the amplitude of the first overshooting wave is four times the amplitude of the second. Tuning HVAC systems to achieve this response is probably unnecessarily aggressive; so we will forgo the details of the method.

A less oscillatory response has been judged to be more appropriate for HVAC systems. Figure 14-3 show three catagories of alternatives. A critically damped response is one with no overshoot but on the verge of

overshooting. Increasing the gains will cause a critically damped system to become underdamped and lead to some overshoot such as the response shown in the figure. Decreasing the gains of a critically damped system produces an overdamped system (again see the figure). An overdamped response is slower to respond to set point changes or other disturbances than critically- or underdamped.

Time
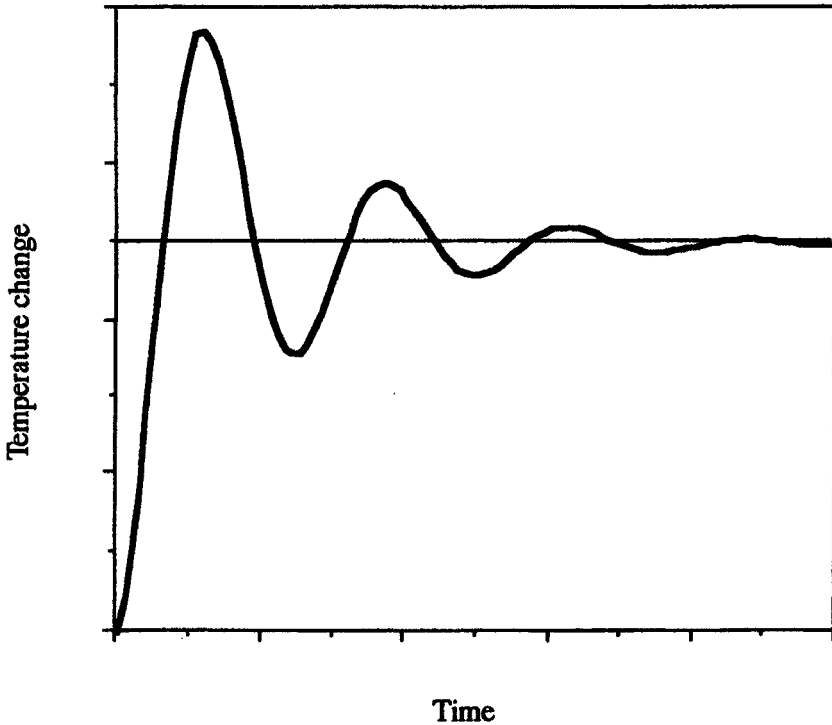
Figure 14-2. Underdamped quarter wave response.

Nesler (1986) and others have suggested that most HVAC control loops should be tuned to be critically damped or slightly underdamped with the system in its highest gain state. If the system gain changes to a lower gain, then the response will be overdamped. We will describe two methods of achieving this response.
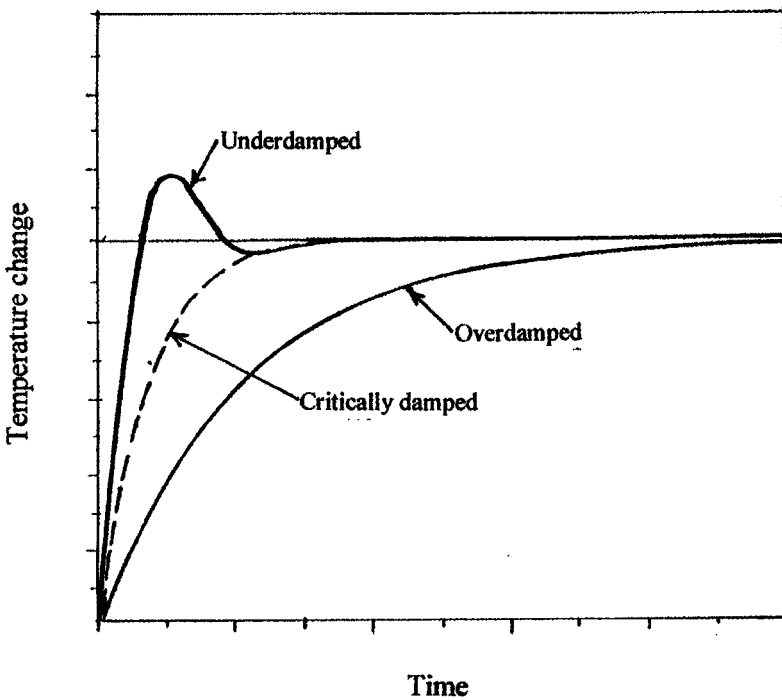
Figure 14-3. Typical control loop responses.

14.3.1 Open Loop Tuning

Bekker 1991 suggests an open loop tuning method designed to acheive a critically damped response. The procedure is to put the HVAC subsystem in an operating condition that represents the highest system gain condition expected under normal operation. For example, a heating coil subsystem would be in a high gain state with the air flow rate at its lowest expected value, with the hot water temperature at its hottest value and with the coldest expected inlet air temperature. Next the open loop response is determined by causing a sudden change in the controller output and measuring the change in the controlled variable over time.

The proportional and integral gains to produce a critically damped closed loop response are:

$$K_p = \frac{\tau}{T_d K_s} e^{-1} \tag{14-2}$$

$$K_i = \frac{K_p}{\tau} \tag{14-3}$$

where :

K_p = proportional gain constant

K_i = integral gain constant

τ = system time constant

T_d = system time delay

K_s = system gain

$e^{-1} = \exp(-1) = 0.368$ (e is not the error here)

After implementing closed loop control with these gain settings, it is a good idea to check the response to a sudden set point change to be sure it is not oscillatory. If it is, both the proportional and the integral gains should be reduced by the same proportion until a satisfactory closed loop response is achieved.

14.3.2 Closed Loop Trial and Error Tuning

Nesler in Stoecker (1989), presented a method for tuning control loops by observing the closed loop response to a sudden change in set point.

Again, the system should be put in a high gain condition. Next the integral action of the controller is set to zero or some very low value. The proprotional gain is adjusted by observing the response to step changes in the set point until a small amount of overshoot is observed in the closed loop response. Figure 14-4 show what the proper response might look like. Note that with proportional only control, there is some steady state error between the control point and the set point.

Next the integral gain is increased until the closed loop response is centered around the new set point. Figure 14-5 shows a typical response.
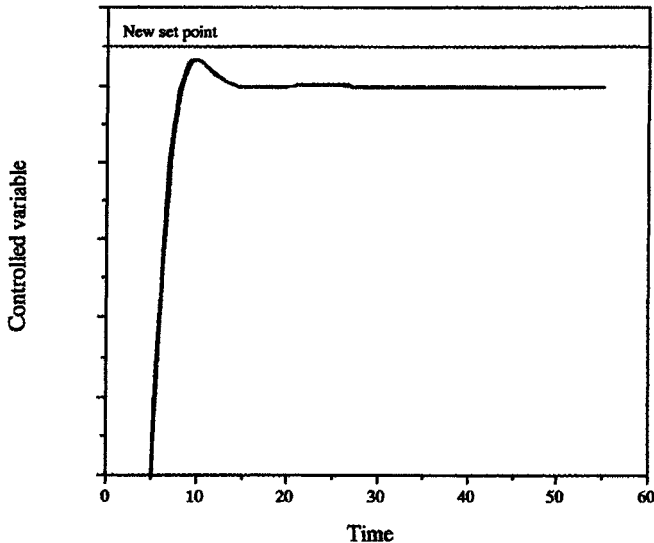
Figure 14-4. Proportional only response.

14.3.3 Reset Rate and Digital Implementation of Integral Control

Many control manufacturers define the integral gain in terms of the reset rate as follows:
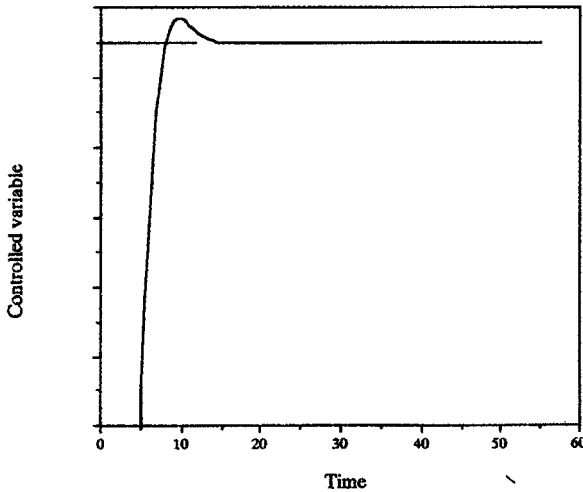
Figure 14-5. Proportional plus integral response.

$$O = A + K_p (e + \frac{1}{T_i} \int e \, dt \qquad\qquad (14 - 4)$$

where T_i is the integral time or reset rate in seconds and is equal to $\dfrac{K_p}{K_i}$.

Notice that increasing the reset rate results in a decrease in the integral gain and that increasing the proportional gain without changing the reset rate produces a higher integral gain. Obviously, it is important to know how the integral control function is implemented on any given controller.

With digital controllers, the integral control action is implemented in software as a summation over time. The output at time t + 1 is computed as follows:

$$O_{t+1} = K_p \left(e_{t+1} + \frac{1}{T_i} \sum_{j=0}^{t} e_j \Delta t \right) \qquad\qquad (14-5)$$

where Δt=controller sampling interval.

Because Δt is constant it can be brought outside the summation:

$$O_{t+1} = K_p \left(e_{t+1} + K_i^* \sum_{j=0}^{t} e_j \right) \qquad (14\text{-}6)$$

where $K_i^* =$ integral gain constant $= \dfrac{\Delta t K_i}{K_p}$.

Notice that the new integral gain constant, K_i^*, depends on the sampling rate. When tuning a digital controller using one of the methods described above, Nesler recommends using the fastest possible sampling rate. However, even though a fast rate may be possible during tuning, limited computer resources of the controller may require a slower rate during operation; K_i^* must be adjusted accordingly. The sampling interval during operation should not be longer than one-fifth to one-tenth of the rise time. The rise time is the time it takes to reach the new set point after a sudden set point change.

So-called integral windup can be a problem with the digital PI algorithm above. If the controlled variable cannot be controlled (a damper is wide open, for example) an error may persist causing the integral summation to increase or to decrease to unreasonable values. One way to avoid this is to use an incremental control algorithm. This scheme computes the change in output rather than the output itself.

$$\Delta O_{t+1} = O_{t+1} - O_t \qquad (14\text{-}7)$$

$$= K_p \left(e_{t+1} + K_i^* \sum_{j=0}^{t} e_j \right) - K_p \left(e_t + K_i^* \sum_{j=0}^{t-1} e_j \right)$$

$$= K_p (e_{t+1} - e_t + K_i^* \, e_t)$$

The change in output is used to reposition the controlled device. For example, ΔO_{t+1} might be used to switch an electric actuator on for a

specified time depending on the value of ΔO_{t+1}. In systems using pneumatic actuators ΔO_{t+1} might be the number of counts to send to a stepper-motor-driven pressure regulator. Notice that because the summation is not present, and because the actuator or the regulator have limited arcs of rotation, integral windup is not a problem.

A slightly different strategy is used when the change in output cannot be used to drive the controlled device. Here, the new output is obtained by adding the computed change to the old output:

$$O_{t+1} = O_t + \Delta O_{t+1} \qquad (14\text{-}8)$$

This algorithm requires that a stored initial value for O_{t+1} be used on start up. O_{t+1} must also be constrained between some minimum and maximum value to avoid integral windup.

14.4   SUMMARY

Properly tuning HVAC control loops is the last step in successfully implementing the control system. Formal tuning methods tailored to HVAC applications have only recently become available. With the increased use of digital controllers connected to personal computers, tracking and plotting the dynamic performance of control loops is easier, making tuning easier.

Self-tuning controllers and controllers that adapt to changing system dynamics are likely to be part of the next generation of HVAC control hardware.

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
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Abbreviations Used in This Book

A   Amperes
AC   Alternating current
Btu   British thermal units
C   Common
CB   Circuit breaker
CHR   Chilled water return
CHS   Chilled water supply
CHW   Chilled water
CWR   Condensing water return
CWS   Condensing water supply
D   Derivative control mode
DA   Direct-acting
DC   Direct current
DB   Dry bulb temperature
DP   Dew point temperature
DPDT   Double-pole, double-throw
DX   Direct-expansion
EP   Electric-pneumatic
ft   Feet
h   Enthalpy
HOA   Hand-off-auto
hp   Horsepower
HTWR   High-temperature water return
HTWS   High-temperature water supply
HVAC   Heating, ventilating, and air conditioning
HWR   Hot water return
HWS   Hot water supply
IC   Instantaneous closing
in.   Inches
IO   Instantaneous opening
KW   Kilowatt
MAX   Maximum
MIN   Minimum

min   Minutes
NC   Normally closed
NO   Normally open
OL   Overload
P   Proportional control mode
PB   Pushbutton
PD   Pressure drop
PE   Pneumatic-electric
PI   Proportional plus integral control mode
PRV   Pressure-reducing valve
PSIA   Pounds per square inch, absolute
PSIG   Pounds per square inch, gage
RA   Reverse-acting
RG   Refrigerant gas (discharge)
RH   Relative humidity
RL   Refrigerant liquid
RS   Refrigerant suction
SCFM   Standard cubic feet of air per minute
SCIM   Standard cubic inches of air per minute
sec   Seconds
SP   Static pressure
SPST   Single-pole, single-throw
SS   Start-stop
TC   Timed closing
TO   Timed opening
TR   Tower return
TS   Tower supply
V   Volts
VAV   Variable air volume
WB   Wet bulb temperature
w   Humidity ratio or specific humidity

Symbols Used in This Book

—||— Capacitor

⊙ Centrifugal fan

—⋏— Check valve

CB
—o⌒o— Circuit breaker

Ⓥ Coil for solenoid valve

——▶ Contact, or point of force application

—⫲— Contact, NC

—||— Contact, NO

[C] Contactor or controller

RC   Receiver-controller

M   Control air supply

EP   Control air supply from EP relay

X   Restrictor in control air supply

Control valve, three-way

Control valve, two-way

DM   Damper motor

Diaphragm

DP   Differential pressure sensor or controller

EP   Electric-pneumatic relay

(1M) Electromagnetic coil in starter or relay with identifier. All contacts actuated by this coil have the same identifier.

FS   Fire safety switch

Float switch
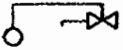
Float valve

F   Flow switch
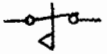
Flow switch, NC
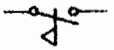
Flow switch, NO

F
Fuse

Gas pilot flame with thermocouple

Globe valve

—‖ᵢ Ground connection

HAND
OFF ⟩₀ Hand-off-auto (HOA) switch
AUTO

—⋀⋀⋁— Heater (heating element) or resistor
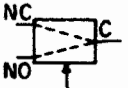
Ⓗ Humidistat, room

[H] Humidistat, duct

[◯] Industrial-type recording controller

✳—[DM] Inlet vane damper (for centrifugal fan)

—o⊂o— Limit switch

NC
⌐ ⌐ C Logic relay
NO

⊘ Manual switch

Manual switch with thermal overload

Motor

Motor field coil

Motor starter

Multipole switch (disconnect switch)

Needle valve

Opposed-blade damper (for modulating control)

Overload actuator

Overload contact

Parallel-blade damper (for two-position control)

Pilot light, color indicated by initial

Plug valve

Point of solid contact, as to a device case or baseplate

Pressure gage

Pressure indicator at control panel

Temperature indicator at control panel

Pressure regulator (pressure-reducing valve)

Pressure switch or sensor

Pressure switch, NC

Pressure switch, NO

Propeller fan and motor

PC Proportioning controller, solid-state

Pump

Pushbutton, normally closed (PB, NC)

Pushbutton, normally open (PB, NO)

R Relay

OR Relay coil

Relay or starter contact, NC

Relay or starter contact, NO

Relief valve

Resistor

Resistor, variable

RR   Reversing relay

SC   Sequencing controller

SD   Smoke detector

Solenoid valve; solenoid valve, two-way

Solenoid valve, three-way

SPDT switch

SPDT switch with center-off position

Spray nozzle

Spring (where identified as such)

SPST switch

Static pressure controller

Steam trap

Thermal expansion valve, thermostatic expansion valve

Thermal switch, NC

Thermal switch, NO

Thermometer, remote-bulb or insertion type

Thermostat or temperature sensor, insertion type

Thermostat or temperature sensor, remote-bulb, duct or pipe, or insertion type

Low temperature safety cutout
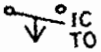
(T) Thermostat, room

Three-pipe control valve

IO
TC Time-delay switch, NC, instantaneous open after energizing, timed close
after deenergizing

TO Time-delay switch, NC, timed open after energizing, instantaneous close
after deenergizing

IC
TO Time-delay switch, NO, instantaneous close after energizing, timed open
after deenergizing

TC Time-delay switch, NO, timed close after energizing, instantaneous open
after deenergizing

Transformer

Transformer coil

VSC Variable-speed controller

L1
o Wiring terminal with identification

Index