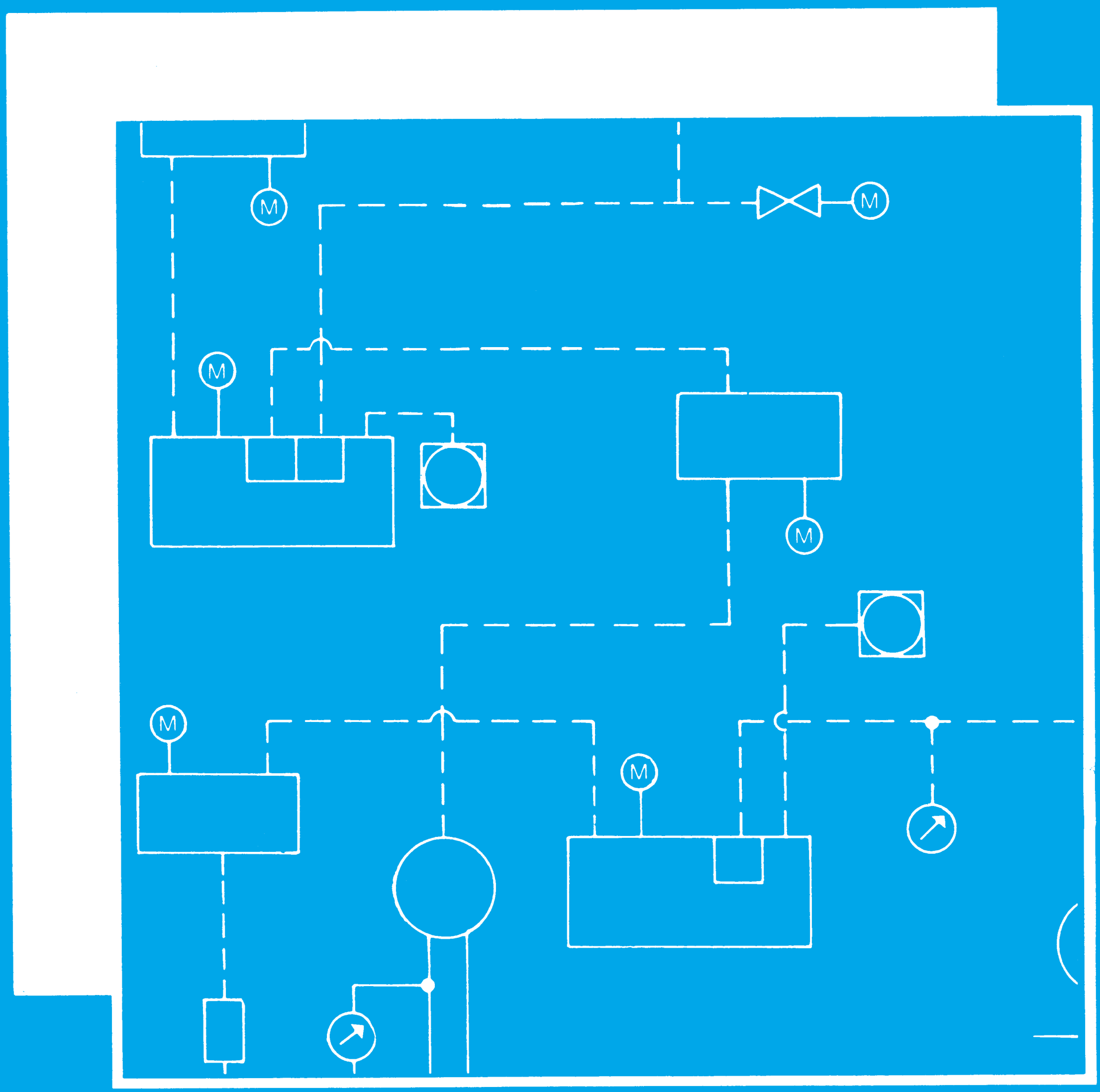
Pneumatic Fundamentals



Training Manual

TAC Training Center
www.tac.com
1354 Clifford Avenue
Loves Park, IL 61111

Pneumatic

Fundamentals

t.a.c®
by Schneider Electric

Training Manual

Examples Used in the Document

This document contains examples of HVAC equipment and applications. Many of these applications include temperatures, setpoints, throttling ranges, and other examples of conditions found in the HVAC world. When reading this document, keep in mind that these are examples only. Many factors, including equipment, climate, building codes, and local practices vary and can have a major impact on methods, procedures, and sequences used to control buildings. Also, terms and values, such as output signals, voltage or pressure requirements vary within TAC products.

This document cites examples of products manufactured by TAC. These products represent only a small portion of the TAC product offering, and the applications, features and benefits of TAC controls used in this text represent only their basic functions. TAC has a complete line of products for today's modern building, offering security, lighting, sophisticated energy management solutions, Internet-accessible reporting and many other products to support the needs of a "smart" building. For complete TAC product information, see your local TAC product distributor.

TABLE OF CONTENTS

- CHAPTER I -

CHAPTER 1

TABLE OF CONTENTS

SECTION **PAGE**

TYPES OF CONTROL SYSTEMS

In the heating, ventilating and air conditioning industry, there are six major classifications of environmental control systems. These classifications are based primarily upon the source of energy used to operate the devices within the system.

1. Self-Actuated
2. Electro-Mechanical
3. Pneumatic
4. Electronic-Electric
5. Electronic-Hydraulic
6. Electronic-Pneumatic

1. **Self-Actuated**

The source of energy, as well as the sensing element and the controlled device (usually a valve), are contained within a single instrument. A self-actuated control usually includes a bellows and a bulb connected by a length of tubing that is filled with a fluid. A change in the temperature of the variable (room temperature) surrounding the bulb causes the bellows to expand or contract with enough force to operate the valve.

SELF-ACTUATED

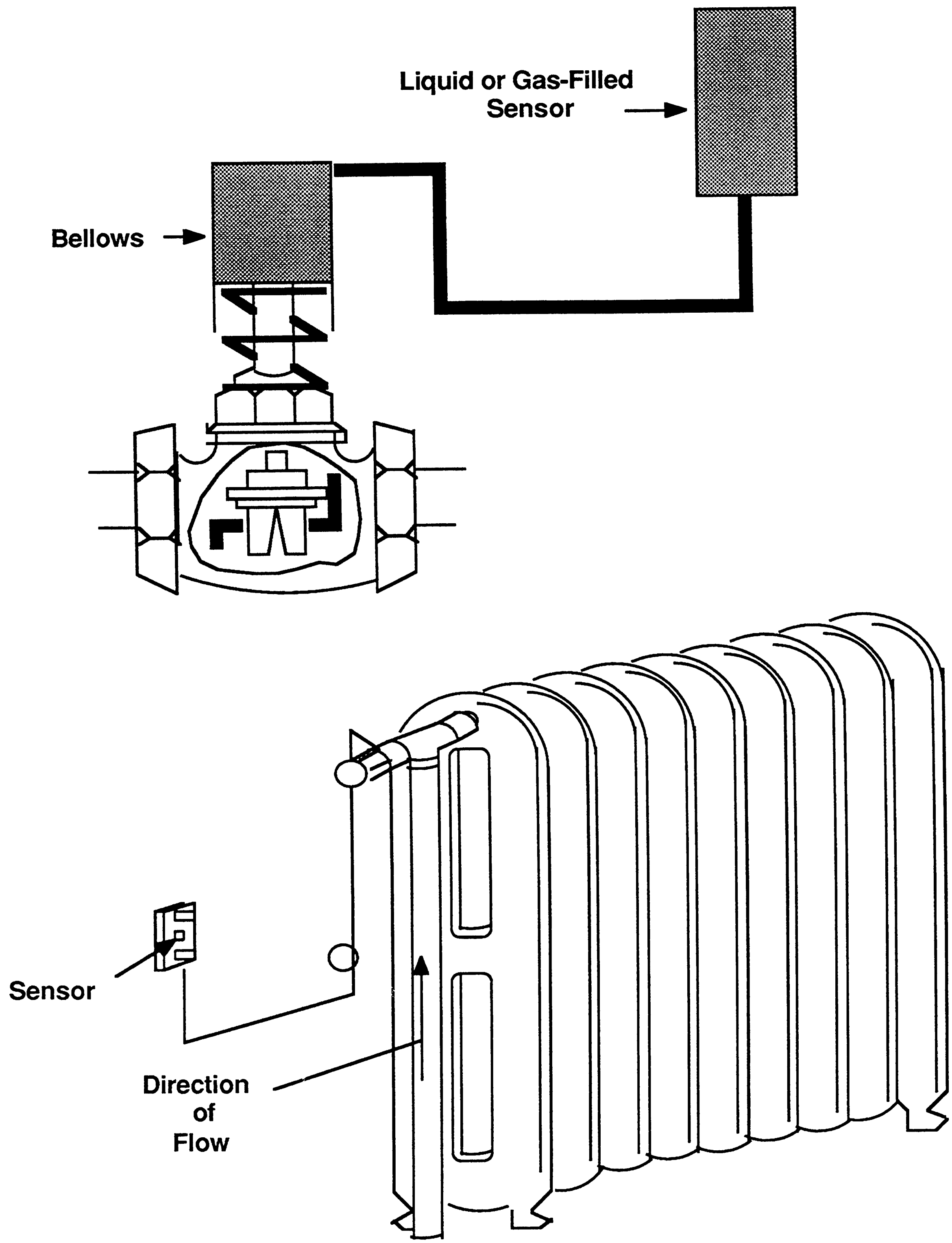


FIGURE 1-1

Figure 1-1 illustrates a simple application of a self-actuated control. The valve which controls the supply of the controlled medium to the radiator is directly operated by changes in pressure inside the bellows as the temperature of the air surrounding the bulb increases or decreases. The variations of pressure in the bellows are directly produced by changes in temperature at the bulb.

2. **Electro-Mechanical**

In electric systems the devices are powered by electric current, either line or low voltage, and control the system by starting and stopping the flow of current. The controller and the controlled device may be mounted at widely separated locations without impairing either the rate of signal transmission or the power of the controlled device.

ELECTRO-MECHANICAL

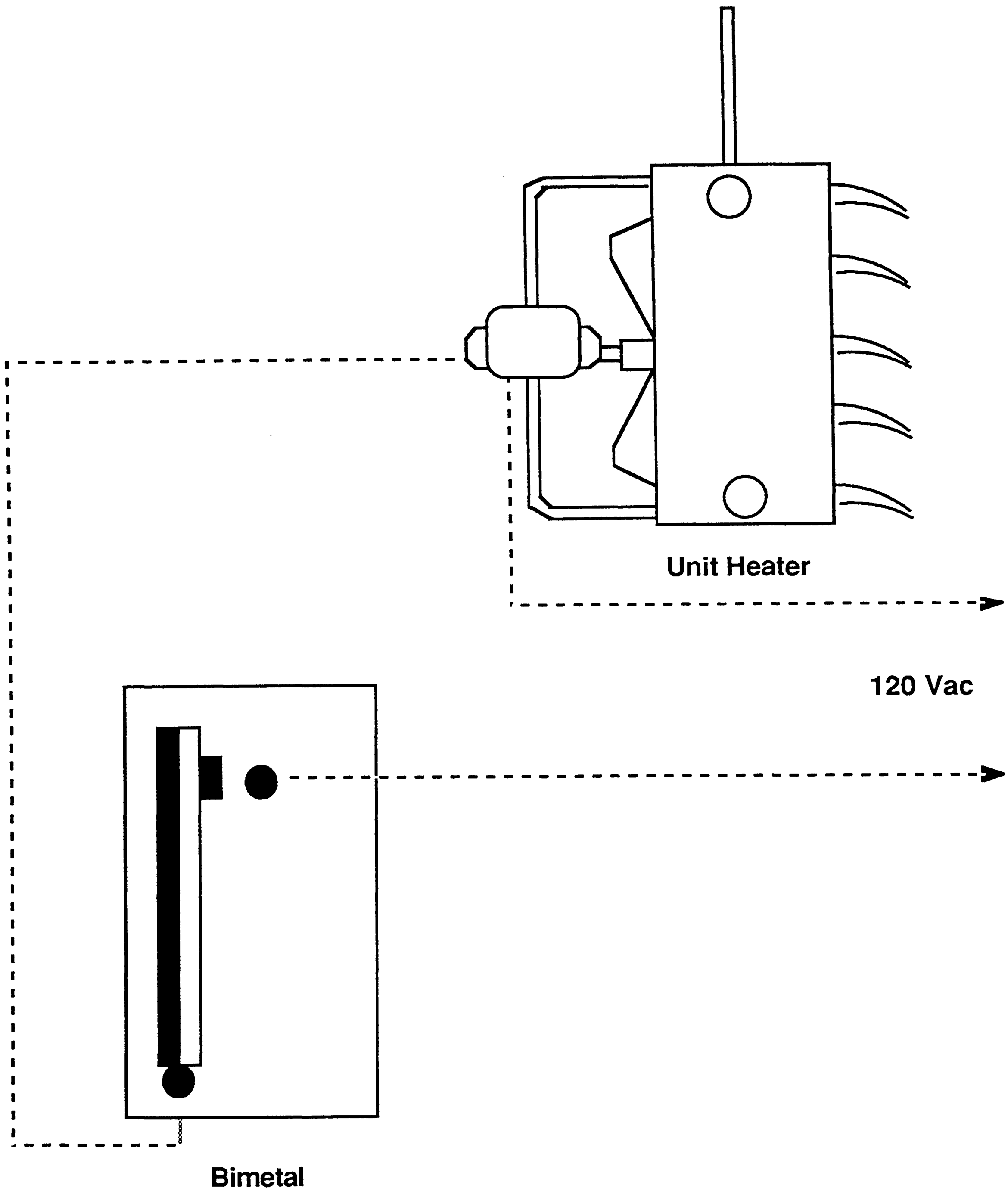


FIGURE 1-2

Figure 1-2 illustrates a simple electro-mechanical application. The electric thermostat operates the electric motor to drive the fan when an increase in room temperature is required and stops the fan when the desired temperature is reached.

3. **Pneumatic**

In a pneumatic system, compressed air is the source of energy for the controllers and controlled devices. The compressed air, usually at a constant pressure of 15 to 25 psig, is supplied to the controller. The controller regulates the air pressure supplied to the controlled device. The devices are connected by either copper or polyethylene tubing.

PNEUMATIC

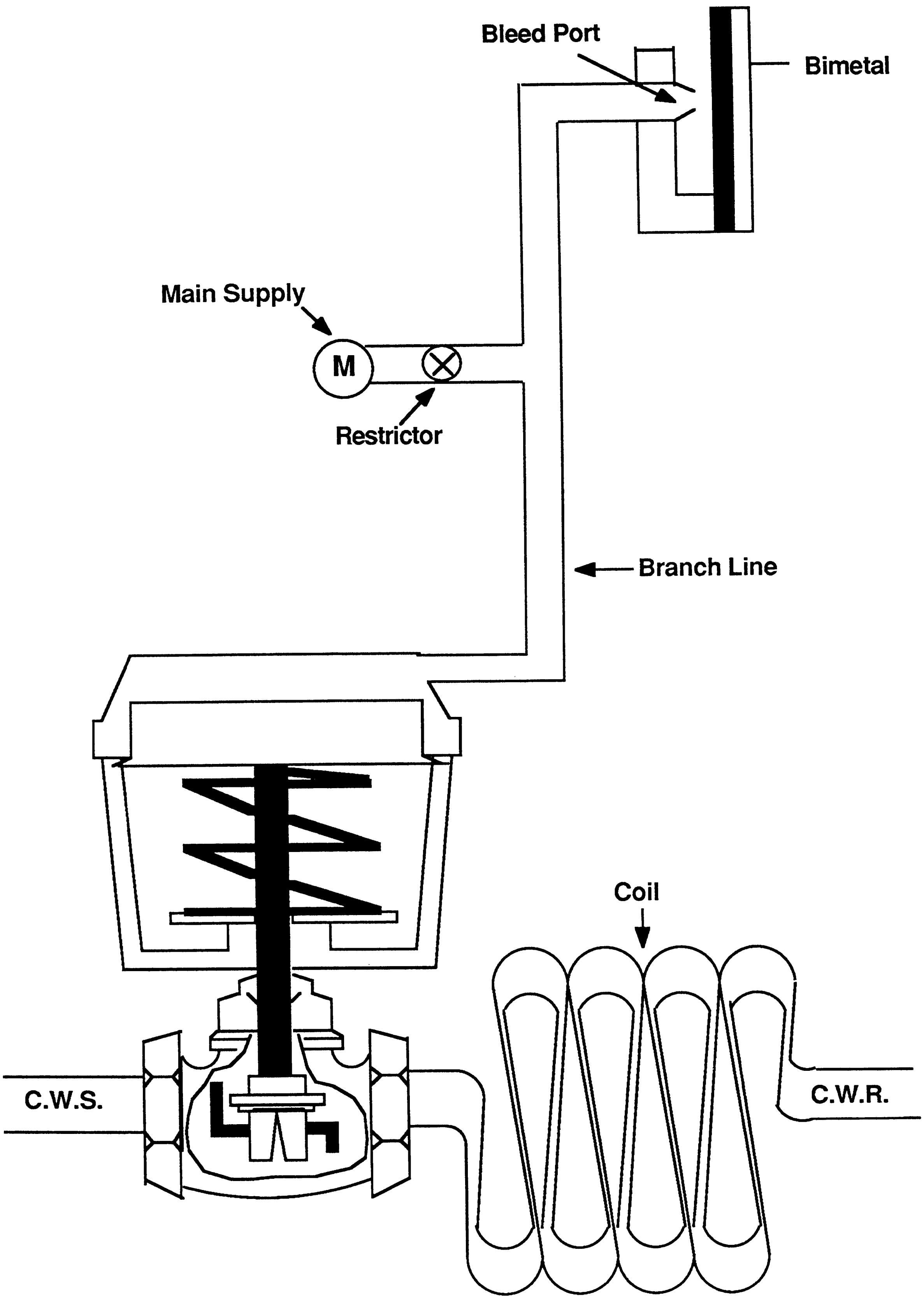


FIGURE 1-3

Figure 1-3 illustrates a simple application of a pneumatic control. The main pressure of 15 to 25 psig is supplied to the controller through a main air line. The thermostat functions as a pressure regulator. The bimetal is actuated by the change in the variable (room temperature). The line supplying the valve actuator (branch line) maintains a pressure (main or less) according to the need for cooling the room.

It should be noted that the control valve actuator does not consume air. Air flows into and out of the actuator to cause the actuator to change position. The valve assumes a position determined by the branch line pressure, the tension of spring and the force exerted by the chilled water pressure on the valve. However, the spring tension varies with the position of the valve stem. Therefore, the thermostat, by regulating the branch line pressure, effectively determines the position of both the actuator and the valve.

4. **<u>Electronic-Electric</u>**

The electronic-electric control system receives its name from its source of energy (electricity), and it makes use of electronic amplification. An electronic-electric control system consists of a resistance type sensing element, a resistance measuring circuit, an electronic voltage and an electric motor to actuate a valve or damper.

ELECTRONIC-ELECTRIC

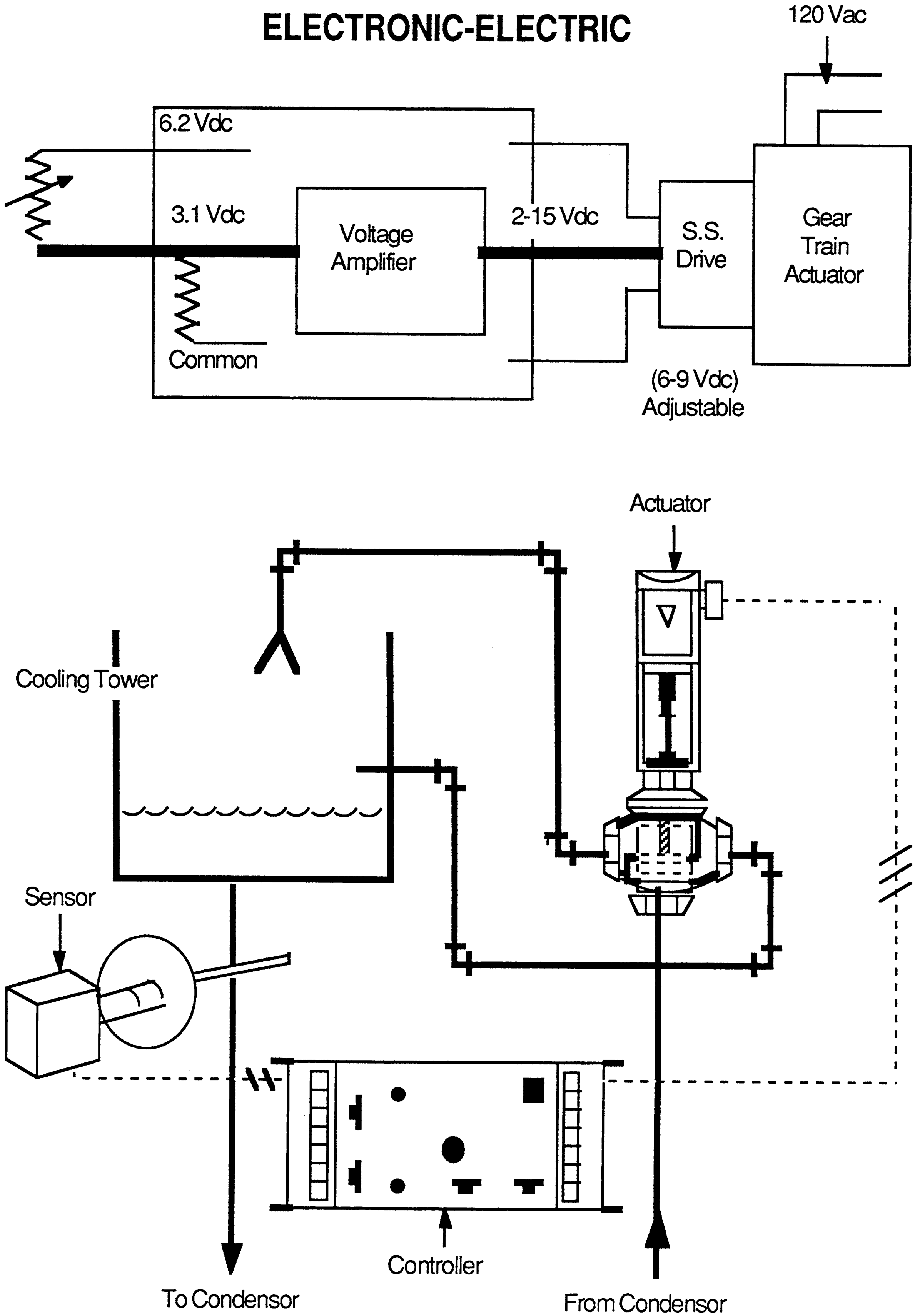


FIGURE 1-4

Figure 1-4 shows the temperature sensing element located in the condensor water return whose resistance increases with a temperature increase. This change in resistance is expressed at the measuring circuit in the controller. The amplifier is also located in the controller, and the signal produced by the controller to the drive will cause the actuator to drive in either direction, depending on whether the temperature of the water from the cooling tower is increasing or decreasing. With the rotation of the actuator in either direction, the valve will divert any portion of the water from the condensor back to the sump of the tower or up over the tower, depending on the temperature of the water returning to the condensor.

5. **Electronic-Hydraulic**

The electronic-hydraulic control system combines electricity with hydraulic pressure for control of the environment. The electronic-hydraulic system functions the same as the electronic-electric system, with the exception of the actuator. The hydraulic actuator makes use of an electric motor and hydraulic pump and piston assembly. Control of the actuator is accomplished through a hydraulic transducer controlling the hydraulic pressure applied to the piston, which causes the linear motion necessary to move the damper or valve. This approach differs from the electronic-electric method in that this linear motion of the actuator is produced by hydraulic pressure rather than driving an electric motor in either direction. If the hydraulic pressure is reduced, the spring that opposes the piston causes the actuator to retract.

ELECTRONIC-HYDRAULIC

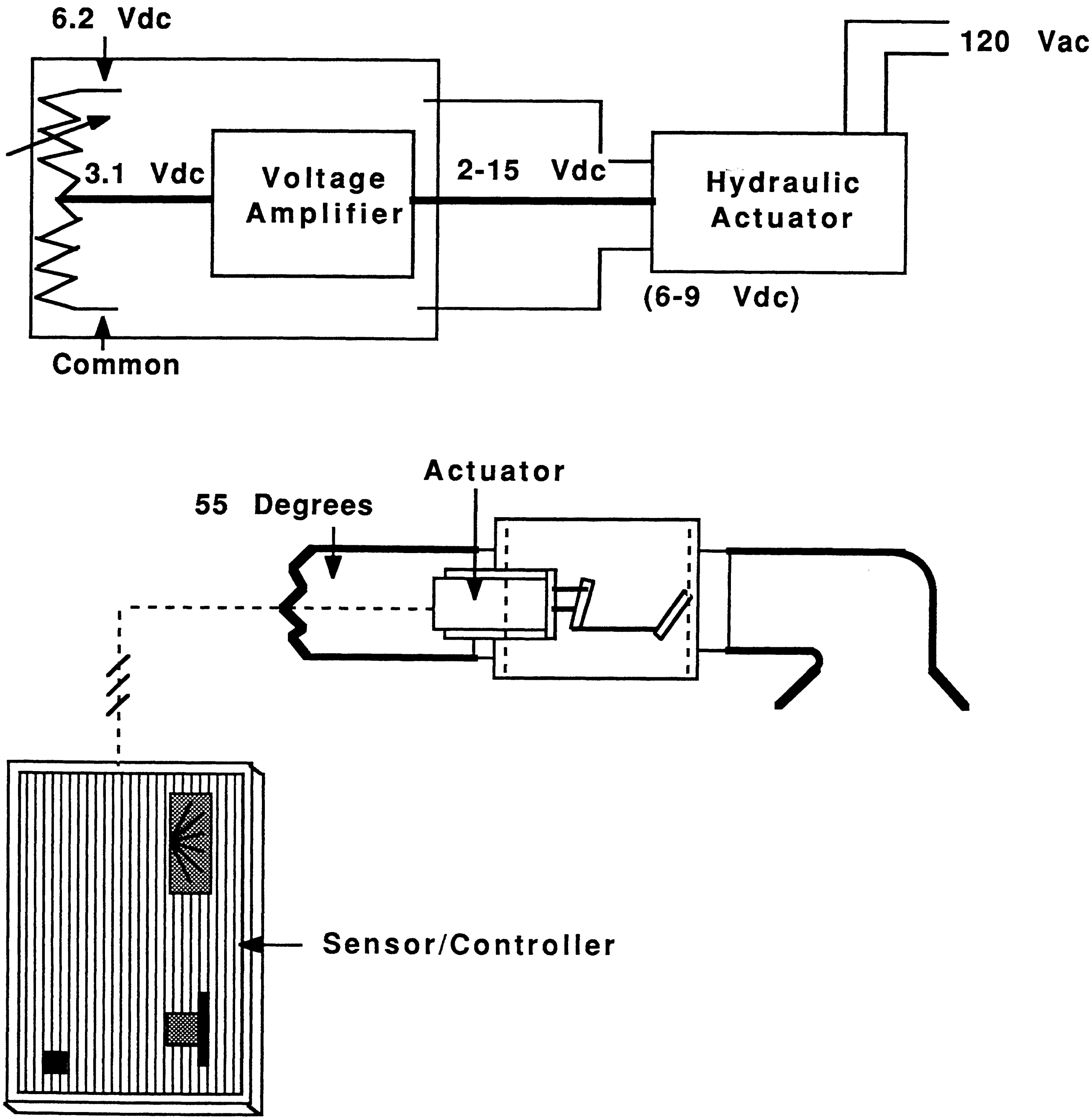


FIGURE 1-5

Figure 1-5 shows the temperature sensing and measuring circuit along with the amplifier. If the temperature were to change in the room, the resistance of the sensor would change causing the signal to the amplifier to change. This amplified signal is expressed at the transducer which will change the hydraulic pressure, which will cause the hydraulic actuator to extend or retract allowing more or less air into the room to satisfy the thermostat.

6. **Electronic-Pneumatic**

Electronic-pneumatic control systems function in the same manner as the electronic-electric and the electronic-hydraulic systems as far as the sensing and controller output is concerned. The transducer, which is a separate module, receives the signal from the controller. The transducer also receives a constant main air signal. As the electronic signal changes from the controller, this causes a change in the output pressure signal from the transducer to the actuator. The actuator consists of a pneumatic piston assembly. As the pressure from the transducer changes, a linear motion is developed to move the damper or valve.

ELECTRONIC-PNEUMATIC

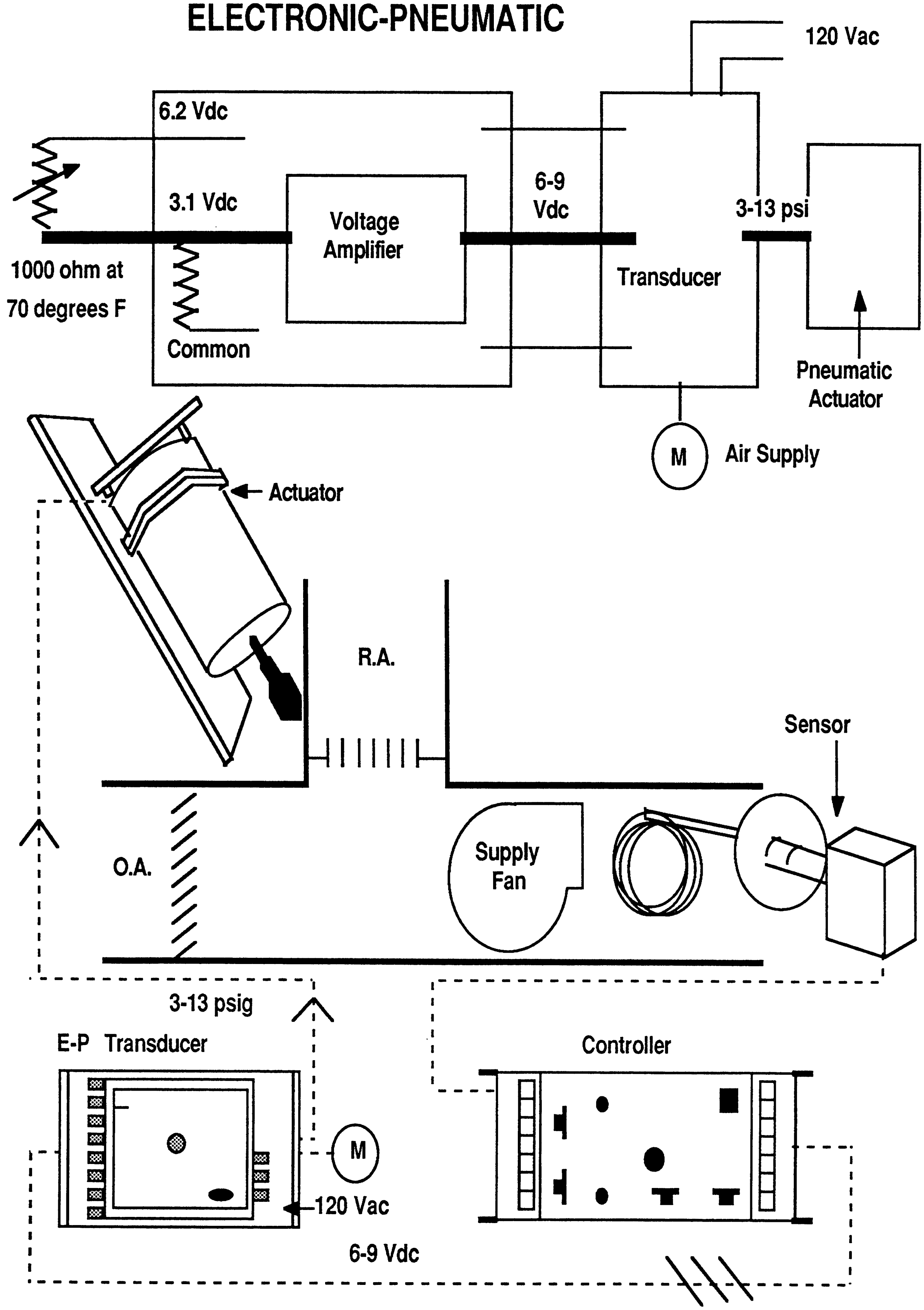


FIGURE 1-6

Figure 1-6 shows an electronic-pneumatic system. The sensor measures the variable (mixed air temperature) and the controller with its measuring circuit and amplifier, sends the electronic signal to the transducer. The E-P transducer converts the electronic signal to an equivalent 3-13 psig. pneumatic signal that is sent to the pneumatic actuator. The pneumatic actuator positions the outside and return air damper to produce the correct mixture of air to satisfy the sensor and controller.

The preceeding six types of control systems are an attempt to organize and classify the information involved. Most any installed system can be classified into any one of these categories. The particular choice of systems used is determined by several factors, such as economics or the specific application involved. For some specific applications, one type of system may be more effective, and therefore, more economically feasible.

In any heating, ventilating and air conditioning installation, the conditioned space may be controlled as a single unit divided into zones, or the rooms may be individually controlled. Single unit control is the most widely used form of control. This type of control is commonly found in a residential system or a small commercial building.

Zone control is used as the size of a building increases. This is because it becomes more difficult to provide good comfort from a single controller for an entire building, due to exposures and interior load variations. Therefore, the building is split into zones, each controlled by its own thermostat.

Individual room control is a much more accurate and satisfactory type of control for any building, whether residential or commercial. The control system of this type includes a thermostat in each room which controls some type of valve, damper actuator, unit ventilator, fan coil unit, unit heater or other source of heating and/or cooling. Thus, without regard to conditions in another room, the controller in each room regulates the amount of heating and/or cooling supplied individually to the room and maintains the room at its required temperature. This form of control, primarily because of the number of devices required throughout a building, is usually more expensive to install. However, where maximum flexibility and the most accurate control is desired, individual room control provides the most satisfactory results at minimum operating cost.

Advantages of Pneumatic Control Systems

Pneumatic control systems are commonly used for controlling heating, ventilating and air conditioning equipment in commercial applications. Pneumatic control offers a number of distinct advantages:

1. Pneumatic equipment is inherently adaptable to proportional control, yet two-position control can be provided.

2. A great variety of control sequences and combinations can be achieved by using relatively simple equipment.

3. Pneumatic equipment is normally quite free of operating difficulties.

4. Pneumatic systems are suitable where explosion hazards exist.

5. Cost is usually less with pneumatic systems in large installations due to the lower cost of pneumatic actuators compared to electric or electronic actuators and where codes require that low voltage electric wiring be run in conduit.

- CHAPTER 2 -

CHAPTER 2

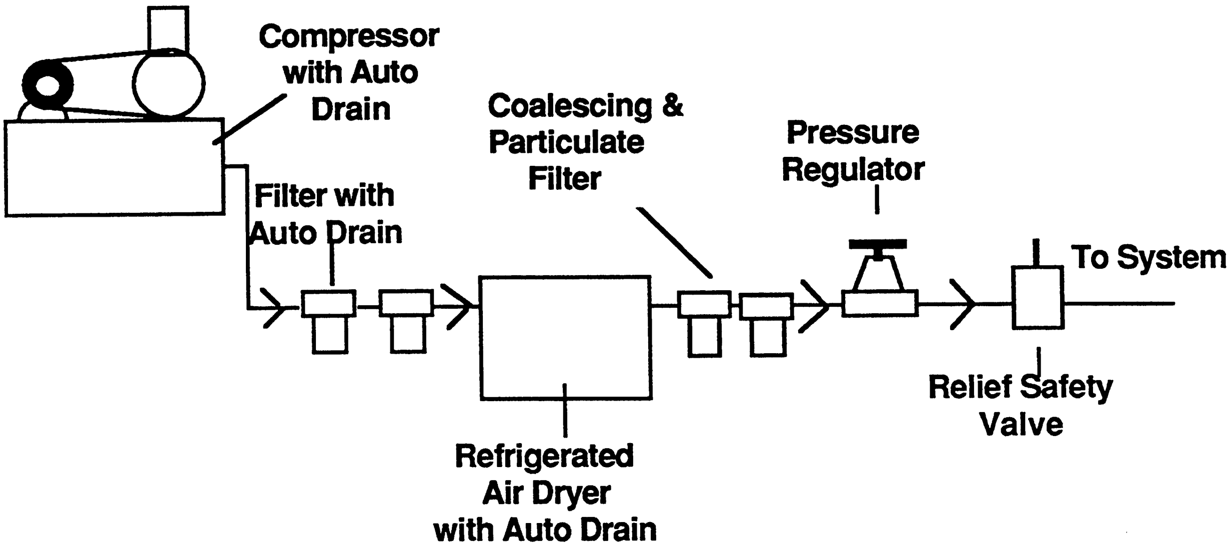
TABLE OF CONTENTS

SECTION **PAGE**

THE AIR SUPPLY SYSTEM

Pneumatic control systems use compressed air to supply energy for the operation of valves and dampers by their associated actuators, relays and other pneumatic control equipment. Consequently, the circuits consist of air lines. The air supply system is made up of the following elements:

- A compressor with its receiver tank at a pressure capable of supplying all the pneumatic devices in the system with operating energy.
- Dirt and oil removal filters to ensure clean, oil-free air downstream.
- A refrigerated air dryer to ensure dry air to the control system.
- A pressure-reducing station which reduces receiver tank pressure to a normal operating pressure of 15-25 psig.
- A pressure relief valve set at 30 psig.
- Air lines, which can be either copper or polyethylene tubing that connect the air supply to the controllers. These air lines are called mains.
- Controllers such as thermostats, humidistats, receiver-controllers and pressure controllers that are used to position the controlled devices.
- Auxiliary devices such as relays, switches, and transducers to condition the signal from controllers to obtain certain sequences.
- Air lines leading from the controllers to the controlled devices. These air lines are called branches and are also either copper or polyethylene.
- Controlled devices called actuators that position the valves or dampers.



SINGLE PRESSURE SYSTEM

FIGURE 2-1

<u>Single Pressure System</u>

The elements of a simple, single pressure system are shown in Figure 2-1.

The source of air in a pneumatic system is an electrically driven air compressor. Most pneumatic systems can be serviced with a compressor sized under 10 horsepower. The most efficient air compressors in this size range are piston style reciprocating units.

Reciprocating air compressors come in two basic designs: oil lubricated and oil-less. An oil lubricated compressor is constructed with an oil filled crankcase. As a result, oil vapor is always present in the compression chamber and the discharge air. Oil-less compressors use sealed bearings and teflon piston rings and skirts, eliminating the need for oil lubrication.

Regardless of the design of the compressor, the need for clean, dry, and oil-free air is essential to ensure that the air lines, controllers, switches, relays, restrictors and other components in the system remain clean and operate satisfactorily. For this reason, a number of related devices, such as coalescent filters and refrigerated dryers are necessary in a system to remove any oil vapors and dirt particles and to dry the air. Even the use of oil-less design air compressors does not guarantee that oil will not be present in the system. Often times the compressor intake air may contain oil vapors which could pass through the compressor and condense into harmful droplets in the system if removal devices are not employed.

Water vapor and oil vapor are natural by-products of air compression. As the hot compressed air cools, the moisture content is released. By adding an automatic drain to the compressor storage tank, the water will automatically be removed. Aerosol proportions are carried into the air stream.

Oil aerosols, water vapors and dirt particles must be removed to submicron proportions to prevent damage to Pneumatic Control components. To ensure the removal of contaminants to the degree necessary, particulate filters, oil coalescing filters and air driers must be used in proper sequence.

A filter combination before the air drier is necessary if the output from the compressor contains oil aerosols. The efficiency of the air drier will be impaired if large quantities of oil are deposited.

A refrigerated air drier removes moisture and condenses oil vapors forming aerosols. the oil aerosols must be removed and an oil coalescing filter after the air drier with automatic drain will remove 99.9% of all oil aerosols .03 microns or larger. The refrigerated drier and filters must be equipped with an automatic draintrap to remove oil and water contaminants as they are removed from the air supply.

A manual bypass valve system is recommended so that the drier and filters can be routinely serviced without interrupting the system operation.

A pressure reducing valve (PRV) downstream of these devices maintaines the operating pressure (15-25 psig) for the system. A pressure relief valve set at 30 psig is installed as the final air supply system device. The maximum safe operating pressure for pneumatic devices is 30 psig, and the relief valve prevents the pressure from exceeding this limit. If the pressure regulator fails, this device protects the controls downstream.

The compressor must be sized properly so that it does not operate continuously. Normally, compressors should not run more than one-third of the time. This extends the compressor life and allows sufficient cooling of the compressed air in the air storage tank which permits maximum condensation of water and oil vapors, so that these contaminates can be removed by the automatic drain trap.

The speed of the compressor is also an important factor to consider. As the compressor operates faster, more heat is generated. This can reduce the overall compessor life, cause premature component failures, increase the overall compressor noise level, and, in the case of oil lubricated compressors, can increase oil carry-over. Therefore, a compressor with a low revolution per minute (RPM) rating should be used.

Another consideration is the electrical power available at the compressor site, along with a working knowledge of selecting, installing or servicing starters, contactors and other electrical components within the system.

Most pneumatic devices exhaust some air to the atmosphere and this air is measured in standard cubic feet per minute (SCFM). To determine the air requirements for a system, the air usage of all pneumatic devices in the system must be totaled.

Each pneumatic controller has a specific air consumption factor expressed in units.

The following formula applies when selecting an air compressor to meet job requirements:

$$\text{SCFM} = \frac{\text{Number of Units}}{.33} \times .008$$

Where:

SCFM = Standard Cubic Feet per Minute of Compressor Capacity that is Required.

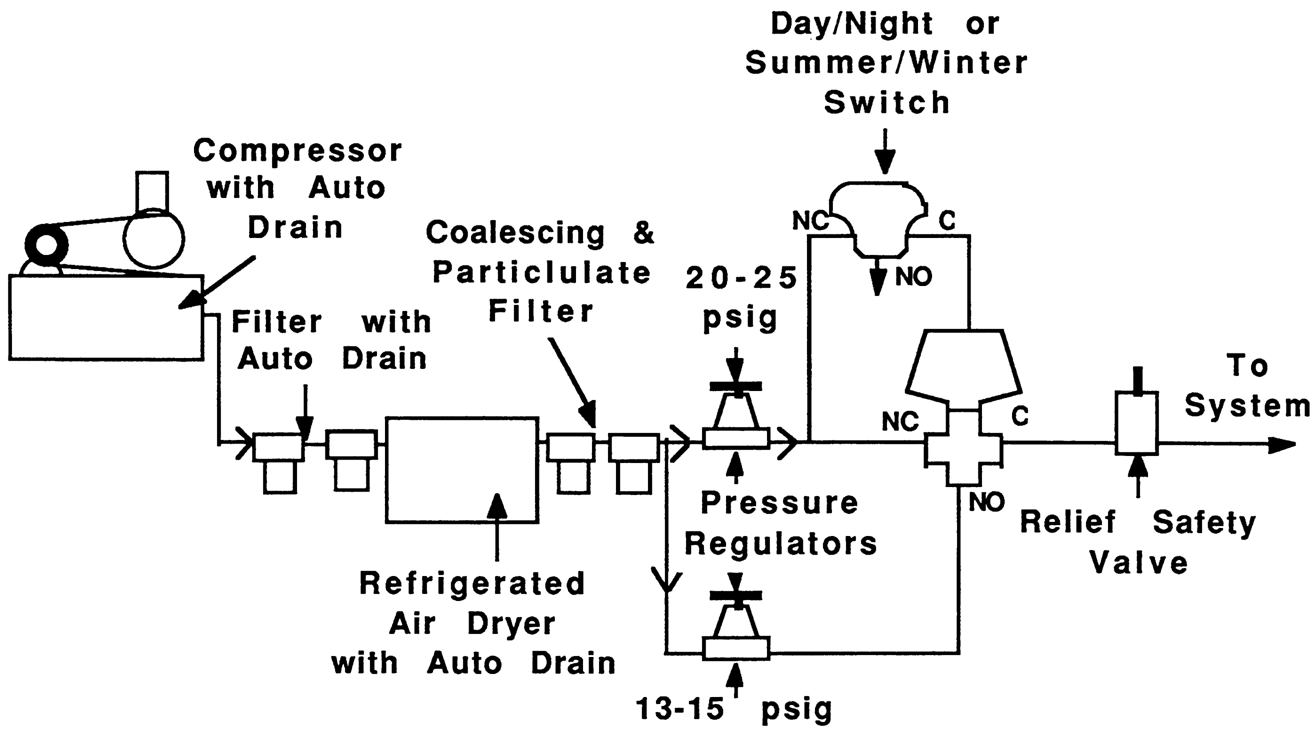
Number of Units = The Total Number of Equivalent Units of Air Consumed by the Devices.

.008 = 1 Unit = .008 SCFM

.33 = One-Third Run Time

<u>Dual Pressure System</u> (Day/Night or Summer/Winter)

In pneumatic control systems, there are two applications that require two different main air pressures in order to function. These are summer/winter and day/night systems. The air station configuration for a dual pressure system is the same as the single pressure system shown in Figure 2-1, up to the pressure reducing valve. Because two distinct main air pressures are necessary, there are two separate pressure reducing valves, each at the system operating pressure necessary for one mode of operation.



DUAL PRESSURE SYSTEM

FIGURE 2-2

In the typical dual pressure system shown in Figure 2-2, one pressure regulator reduces tank pressure to between 13 to 15 psig, and the other to between 20 to 25 psig. The lower pressure is normally supplied to the controller such as a summer/winter thermostat, when the demand calls for cooling (summer cycle). The higher pressure is supplied to the thermostat when the demand calls for heating (winter cycle). In day/night applications, the lower pressure is usually day and the higher for night operation.

The two main air signals from the PRV station are supplied to a three-way air valve before going on to the thermostats. There is also a two-position switch (either manual or automatic). The function of the switch is to supply pressure to the three-way air valve actuator to cause the normally closed port to open and the normally open port to close. Pressure on the valve actuator permits the valve to allow either one of the operating pressures out the common port to the thermostats.

- CHAPTER 3 -

CHAPTER 3

TABLE OF CONTENTS

SECTION **PAGE**

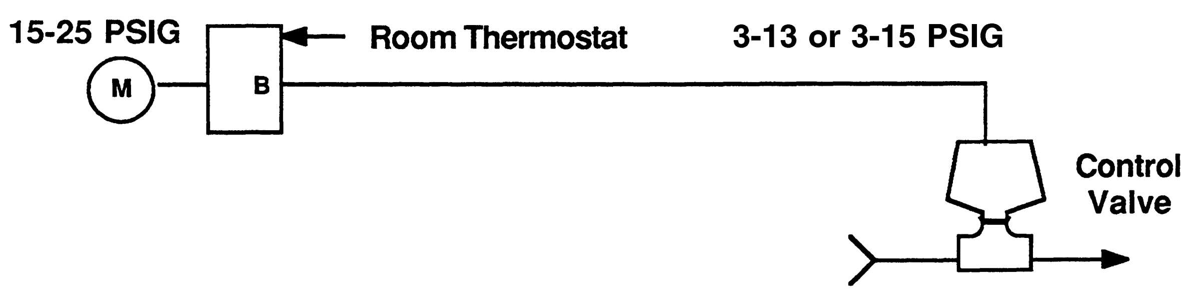
PNEUMATIC CONTROLLERS

Pneumatic controllers are inherently proportional. This means that they are capable of regulating an air signal between 0 psig and the main air pressure supplied to a controlled device, such as a valve or damper actuator. Systems require either direct or reverse acting control. These devices include thermostats, humidistats, pressure controllers and receiver-controllers.

A **direct acting** controller increases its branch line pressure as the variable it is sensing increases. Conversely, a direct acting controller decreases its branch line pressure as the variable it is sensing decreases.

A **reverse acting** controller works oppositely to a direct acting controller and decreases its branch line pressure as the variable it is sensing increases. And visa/versa a reverse acting controller increases its branch line pressure as the variable it is sensing decreases.

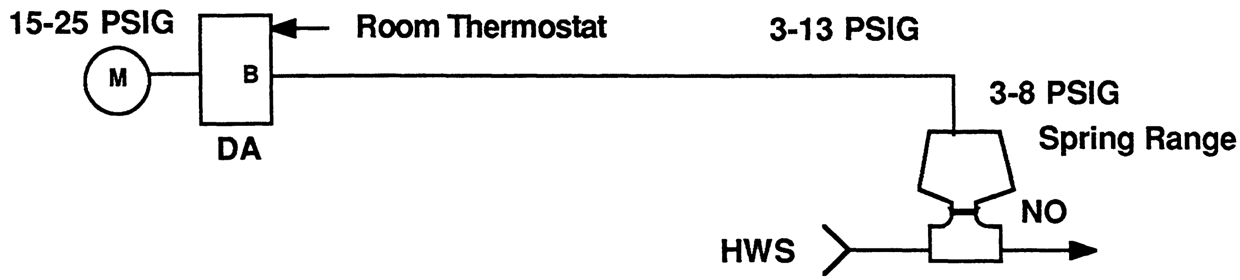
The change in value of the variable which must occur to create a 3-13 psig output change is called **throttling range.** Although controllers are capable of producing a signal change from 0 to main pressure as the variable changes a considerable amount, a small change in the variable is used to create a 3-13 psig change in the branch line. For example, a standard thermostat has a temperature range of 55-58°F. Within that span, a throttling range can be selected between 2 and 10°F. Assuming that a 4°F throttling range is selected, the branch line output of the controller will vary from 3 to 13 psig or 3 to 15 psig over a 4° change in room temperature.



TEMPERATURE CONTROL APPLICATION

FIGURE 3-1

Figure 3-1 illustrates the basic application of a pneumatic thermostat. As illustrated, the thermostat is governing the operation of a pneumatic valve. The controller could be either direct acting or reverse acting depending upon the requirements of the system as illustrated on the following applications.

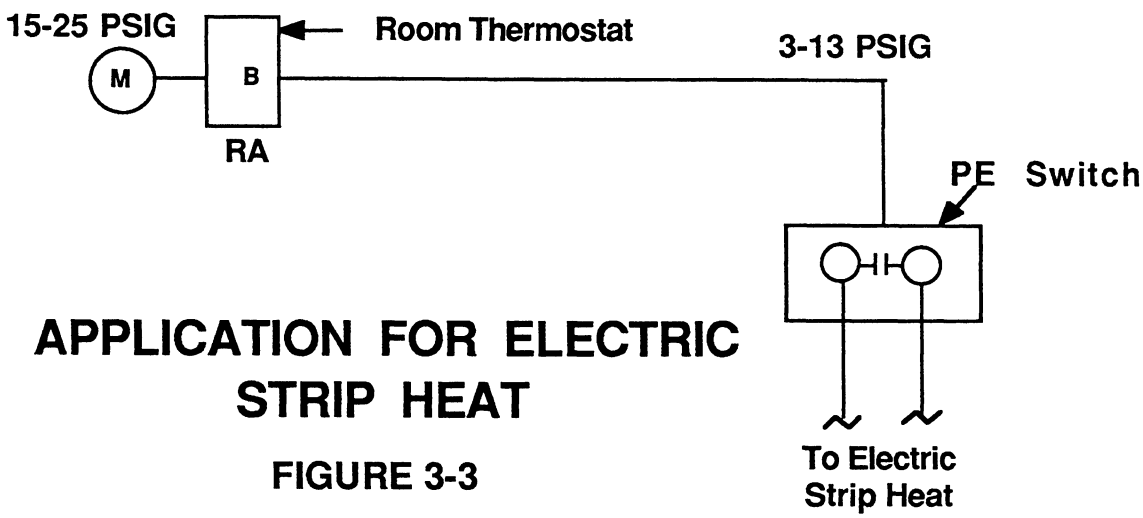


TYPICAL HEATING APPLICATION

FIGURE 3-2

In Figure 3-2 the thermostat is identified as being direct acting and it is controlling a normally open valve with a spring range of 3-8 psig. This combination is commonly used for controlling the flow of a heating medium such as hot water or steam. With respect to the spring range, the valve will begin to close at a pressure from the thermostat of 3 psig, and will be completely closed when the output signal of the branch line is 8 psig. In the event of a system failure, such as a loss of main air, the valve would assume a fully open position. In colder climates this allows the heating medium to continue flowing through the coils to prevent coil freeze-up and provide heating to the conditioned space.

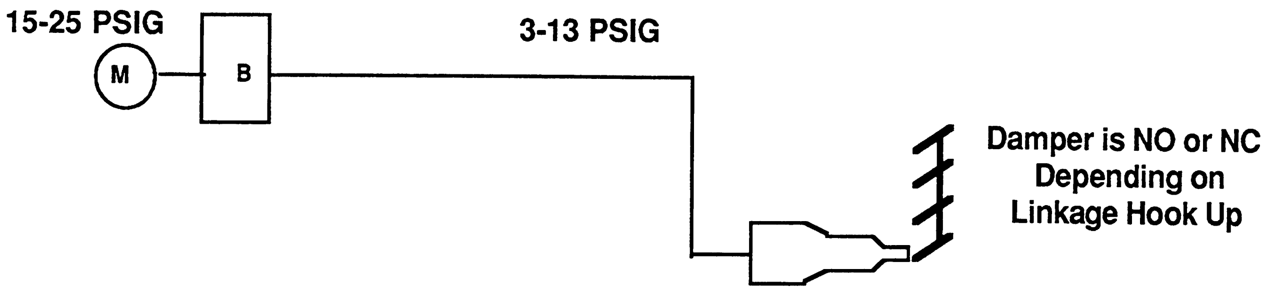
For the above normally open valve in figure 3-2 to operate properly with chilled water, a reverse acting thermostat is required. As the temperature sensed by the reverse acting thermostat increases, its output pressure decreases causing the valve to open further.



APPLICATION FOR ELECTRIC STRIP HEAT

FIGURE 3-3

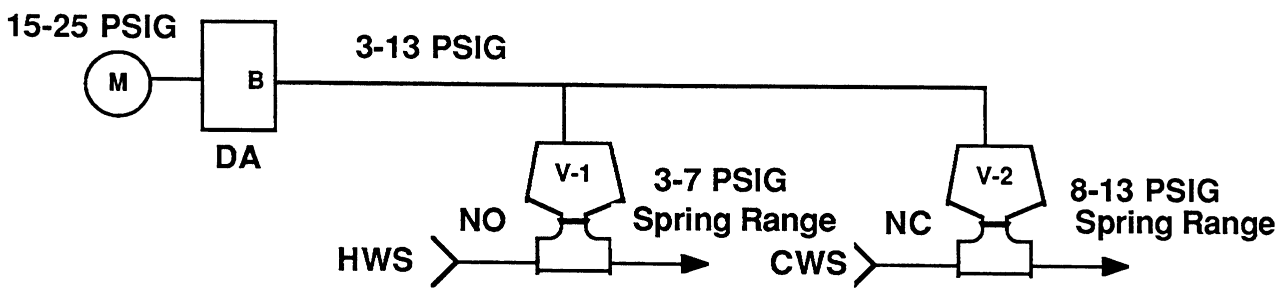
Another application using a reverse acting thermostat is shown in Figure 3-3. In this case, the thermostat is transmitting its air signal to a normally open pneumatic-electric switch (P-E) which, in turn, activates the electric strip heater element when a decrease in temperature occurs.



BASIC DAMPER CONTROL

FIGURE 3-4

Figure 3-4 illustrates a combination of a controller and a damper actuator for controling air flow. The controller sends an air signal to the damper actuator which positions the damper to control air flow across a coil in a duct or into a conditioned space. However, normally open or normally closed applies to the damper rather than the damper actuator. If the damper is located in the outside air stream, it is usually specified as normally closed. In other words, in the event of a signal loss, the damper closes.



VALVES IN SEQUENCE TO PROVIDE HEATING AND COOLING

FIGURE 3-5

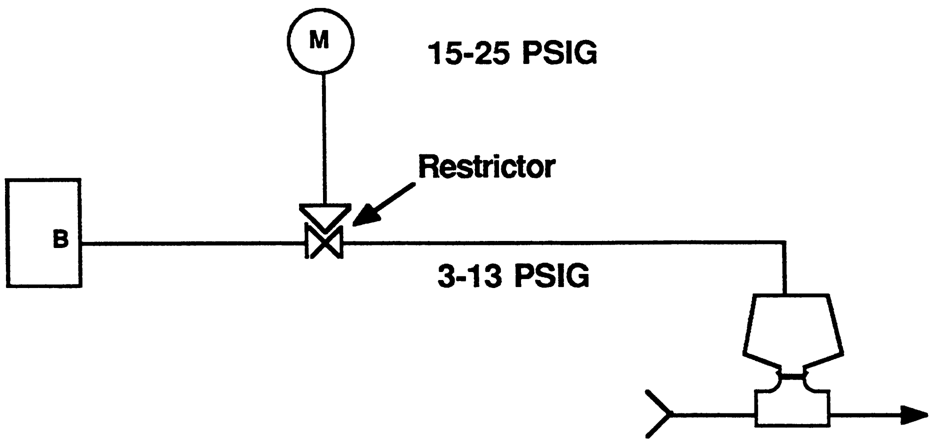
A single pressure room thermostat controlling two valves in combination is shown in Figure 3-5. This arrangement is often used to control both heating and cooling in a single pressure pneumatic system. For this sequence of operation, it is necessary to know the pressure at which the valves are fully open and fully closed. In addition, the valves must be identified as being normally open or normally closed. The thermostat must be direct acting for this application to operate properly. Valve V-1 is normally open and Valve V-2 is normally closed.

The different spring ranges insure the sequential operation of the valves. Valve V-1 is normally open with a spring range of 3-7 psig. Valve V-2 is normally closed and has a spring range of 8-13 psig.

With this arrangement, when the output pressure of the direct acting thermostat is between 3 and 7 psig, Valve V-1 is throttled and Valve V-2 remains fully closed. When the output pressure of the thermostat reaches 8 psig, V-1 is fully closed and V-2 is starting to open. When the output pressure is between 8 and 13, valve V-2 is throttled and V-1 is fully closed. Above 13 psig, Valve V-1 remains closed and Valve V-2 is fully open. The deadband between 7 and 8 psig eliminates the possibility of simultaneous heating and cooling.

Bleed and Relay Thermostats

Thermostats are either bleed type (one-pipe) or relay-type (two-pipe) devices. A **bleed type** thermostat utilizes one connection to the thermostat, hence, the common name "one-pipe". As Figure 3-6 illustrates, the main air supply is fed through a restrictor into the branch line between the thermostat and the controlled device. The restrictor may be the tee type as shown, or the in-line type.



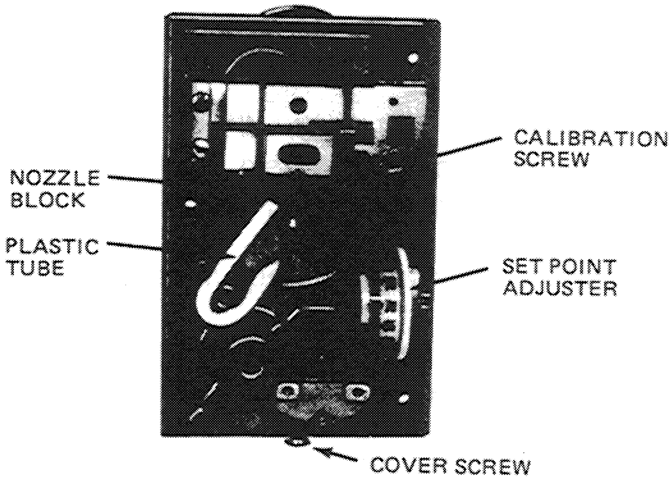
ONE-PIPE (BLEED) THERMOSTAT PIPING

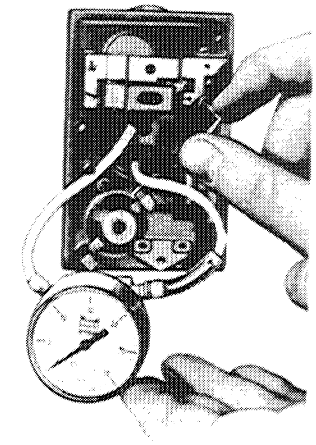
FIGURE 3-6

The restrictor allows only a fixed volume of air through its orifice. The thermostat allows more or less air to flow through the bleed port in response to changes in space temperature.

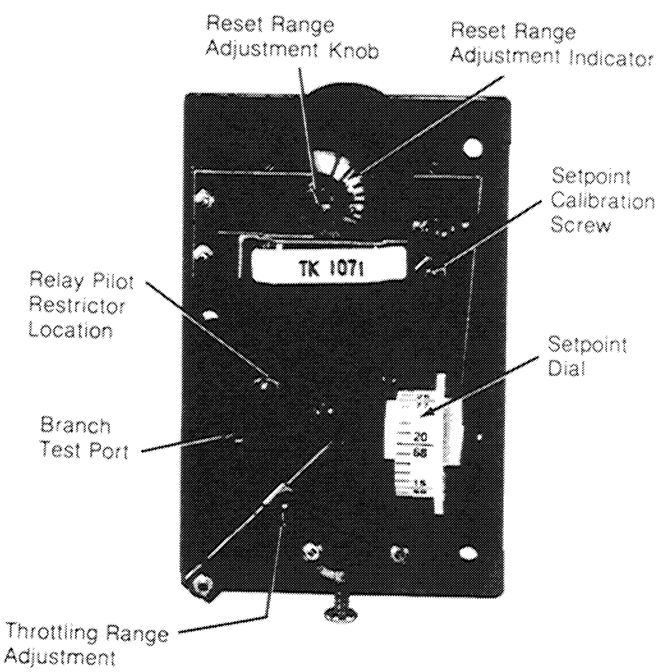
A **relay type** thermostat receives main air directly. Through relay action it will feed air to the branch line and/or exhaust it through its vent port in response to temperature changes. It provides a greater volume of air to the branch line than the bleed-type thermostat because the air that is delivered to the branch line does not pass through the restrictor. This provides a faster system response to a change in the space temperature. The piping would be the same as shown in Figures 3-1, 3-2, 3-3, 3-4 and 3-5. With the covers on, the appearance of bleed-type and relay-type thermostats is the same. With the covers off, a distinct difference can be seen as shown in Figure 3-7.

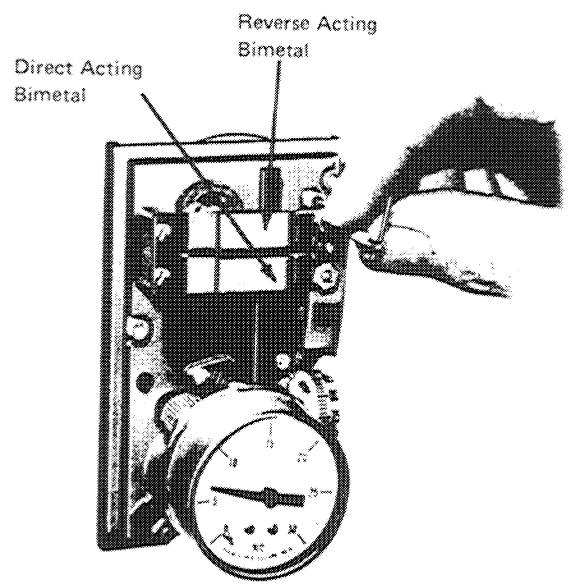
One-Pipe (Bleed)





Two-Pipe (Relay)





ONE-PIPE AND TWO-PIPE THERMOSTATS
FIGURE 3-7

The methods for sampling branch pressure of bleed and relay type thermostats are different. Notice that you must tee into the branch line with the bleed-type, and there is a branch test port provided on the relay type.

The following procedure can be used to calibrate the single pressure pneumatic thermostats:

1. Measure the ambient temperature at the thermostat with an accurate thermometer.

2. Adjust the setpoint of the thermostat to match the ambient temperature.

3. Adjust the calibration screw until the branch line pressure reads 8 psig (midpoint of 3-13 psig) or (the midpoint of the spring range of the actuator being controlled).

4. Adjust the setpoint to the value that is desired for comfort or system requirements.

It is good practice to verify that a main air pressure of 15 to 25 psig is present before the calibration process begins. The importance of an accurate thermometer must be emphasized. If an inaccurate thermometer is used, it is quite possible that a calibrated thermostat may be uncalibrated when following the above procedure. Also, it should be noted that after completing the first two steps of the process, and the gauge reads 8 psig, do not complete step 3. It is best to complete step 4 and then investigate the HVAC system for possible problems.

Service Notes:

Turning the setpoint dial from one extreme to the other will illustrate the following:

1. Turning the dial all the way up and hearing air escape (the stat bleeding off) indicates a direct acting thermostat.

2. Turning the dial all the way down and hearing air escaping indicates a reverse acting thermostat.

3. Turning the dial all the way up and down and not hearing air escaping indicates:
   a. No air supply to the thermostat - check compressor.
   b. Plugged restrictor - remove and clean or replace

Deadband Thermostats

A deadband thermostat is a two-pipe device that is used when it is desirable to have a temperature span in which the HVAC system uses no energy for heating or cooling between the selected heating and cooling setpoints.

The thermostat utilizes two bimetals, one heating and one cooling, to interrupt the deadband pressure. The deadband pressure is the output pressure of the thermostat at which no heating or cooling take place. The heating bimetal modulates the pressure between zero and the deadband pressure, and the cooling bimetal modulates the branch pressure between the deadband pressure and main air pressure, in response to the space temperature. The deadband pressure is adjustable. The desired deadband is determined by the selected heating and cooling setpoints.

The deadband thermostat operates in the same manner as a single pressure, single temperature thermostat. The bimetal assembly controls a single bleed port which prohibits the two individual setpoints from overriding one another, which would result in simultaneous heating and cooling.

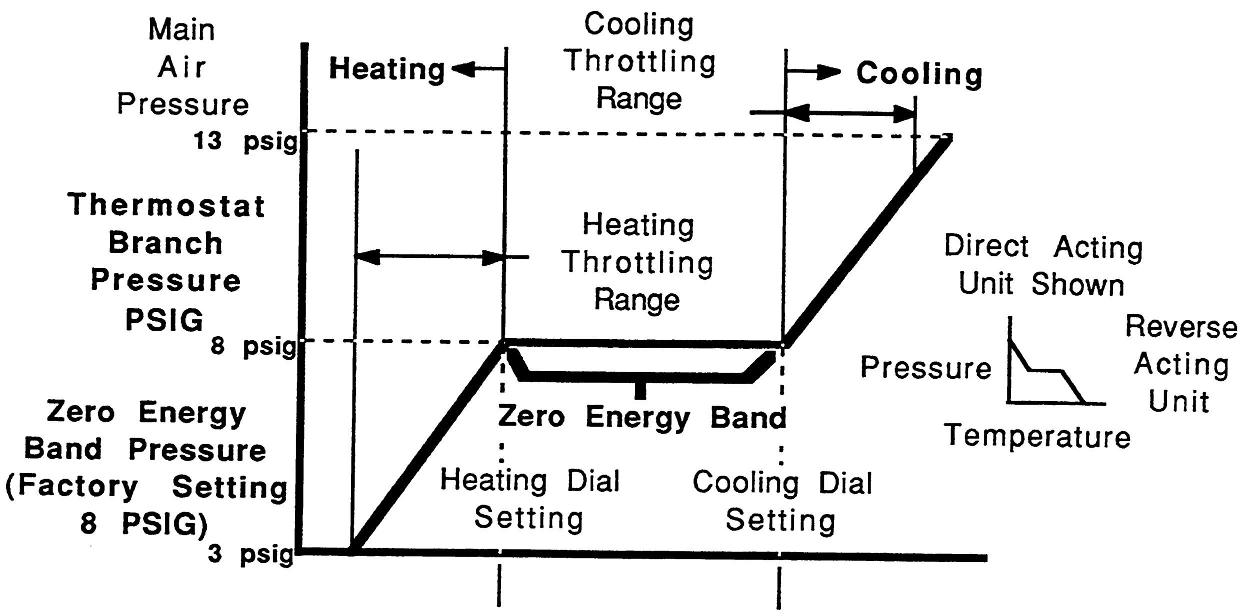


FIGURE 3-8

Figure 3-8 shows the output of a direct acting thermostat rising to the deadband pressure. In this example, it is 8 psig. The deadband pressure is then maintained until the cooling setpoint is reached. At this point, should the room temperature continue to rise, the branch pressure will rise from 8 psig to main air pressure.

Dual Pressure Thermostats

Summer/Winter

A **summer/winter** system is one which provides the seasonal requirements for heating or cooling to the system. The cooling or heating medium is supplied to the system via one supply line and one return line. Depending on the season, hot water or chilled water is provided. Since the valve controlling the flow of water remains the same (either normally open or normally closed, but not both), then the system must have a thermostat which can be either direct or reverse acting. This function in the thermostat is accomplished by changing the main air pressure depending on whether heating or cooling is desired.Figure 3-9 on page 33 illustrates the various components for a typical summer/winter thermostat.

The switching action of the thermostat is an internal, automatic function. There is no manual adjustment to be made at the thermostat to change it from direct to reverse acting. The summer/winter switch, positions a three-way air valve to change the main air pressure supplied to the thermostat. Refer to Figure 2-2 to review the dual pressure system used with summer/winter control.

Day/Night

The **day/night** system is designed for applications requiring separate setpoints due to the occupied and unoccupied time periods. Day/night control is particularly adapted to buildings such as schools and office buildings which are occupied for time periods of 10 to 12 hours and unoccupied for time periods of 12 to 14 hours. The purpose of the day/night application is to control temperature at different setpoints during the day than at night. A day/night thermostat is similar to the summer/winter thermostat. The major difference is that both bimetals are either direct or reverse acting. The two bimetals may have separate setpoint adjustments to provide the set-back or set-up function, or they may have two calibration adjustments with one setpoint.

When the main air pressure is 13-15 psig, the day bimetal controls the output of the thermostat. With 20-25 psig main air pressure, the night bimetal controls the output pressure. For example, a school may desire a 70^{o}F space temperature in the day time, but at night when the building is unoccupied, it needs only to maintain control at 60^{o}F. The main air pressure is changed from 15 to 20 psig. This brings the bimetal in control which has a setpoint of 60^{o}F.

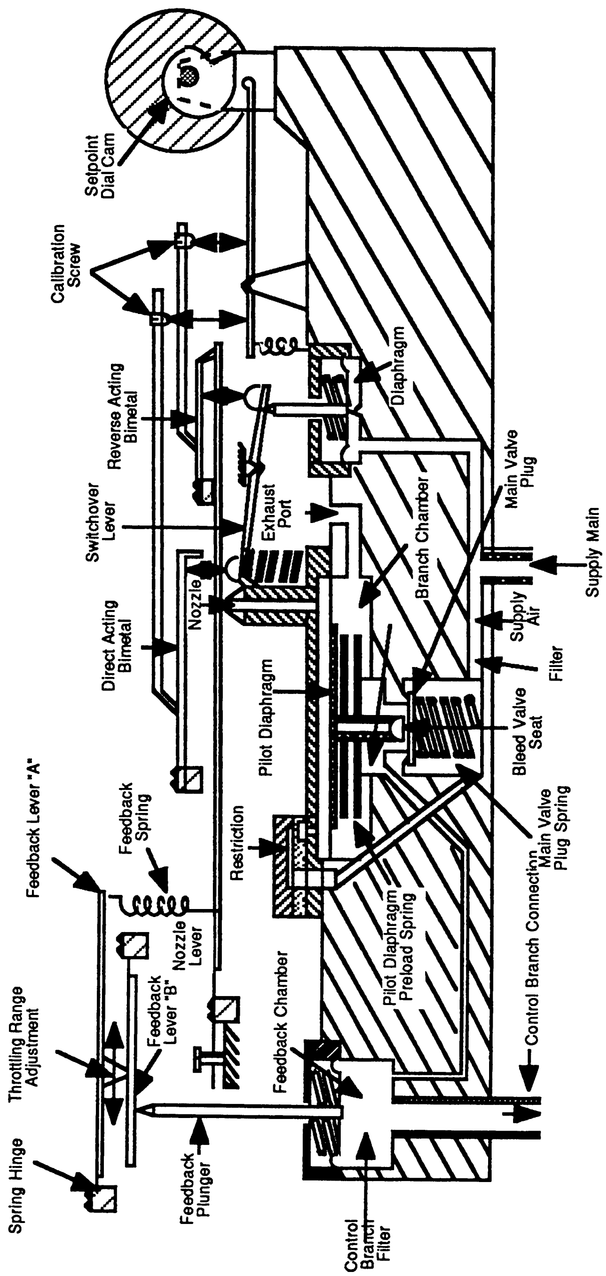

Setpoint Dial Cam
Calibration Screw
Reverse Acting Bimetal
Switchover Lever
Direct Acting Bimetal
Feedback Lever "A"
Feedback Spring
Nozzle
Exhaust Port
Diaphragm
Pilot Diaphragm
Restriction
Branch Chamber
Main Valve Plug
Supply Main
Supply Air
Filter
Bleed Valve Seat
Main Valve Plug Spring
Control Branch Connection
Pilot Diaphragm Preload Spring
Nozzle Lever
Feedback Lever "B"
Throttling Range Adjustment
Spring Hinge
Feedback Chamber
Feedback Plunger
Control Branch Filter

Figure 3.9

The day/night thermostats have a manual override feature. This is a lever on the thermostat which allows individual thermostats to be changed to the day cycle, while other thermostats within the system are operating on the night cycle from the central switch-over station. This feature is often utilized in schools where a limited number of classrooms or an office are occupied at night.

The thermostat can be returned to the night cycle by resetting the lever to its original position. Otherwise, the thermostat will stay in the day cycle. When the main air pressure returns to 15 psig, the lever returns to the day position continuing the day setpoint. When the system indexes to 20 psig that evening, the thermostat will change to the night setpoint.

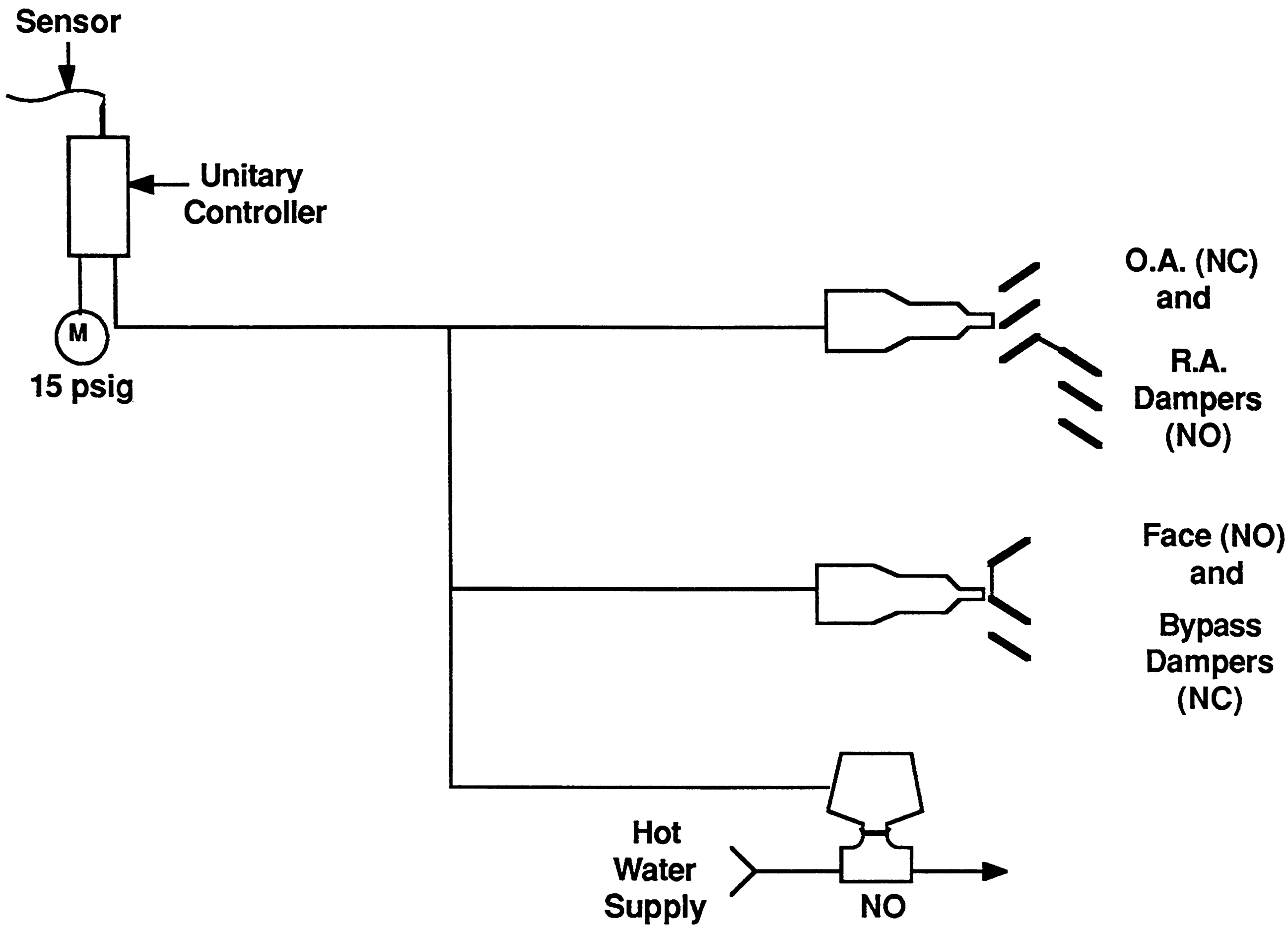
Unitary Temperature Controllers

Unitary temperature controllers are used primarily as return air controllers in induction units, fan coil units and unit ventilators. These are bleed-type remote bulb thermostats.

The physical appearance of these devices is similar to other thermostats, however, the bimetal for temperature sensing is omitted. A remote bulb-type sensing element is used in which the bulb is filled with a liquid which expands and contracts with temperature changes. The remote bulb temperature sensor for these devices is generally installed in the return air flow of the system. The sensed temperature is fed back to the controller which, in turn, provides the branch line signal to the controlled device.

Relay-Type Remote Bulb Controllers

These are similar in capacity and function to standard wall-mounted relay-type thermostats except a remote bulb and capillary is used instead of an internal bi-metal to sense temperature. The typical application for these devices is the remote sensing of the temperature of air in ducts and liquids in pipes and tanks.

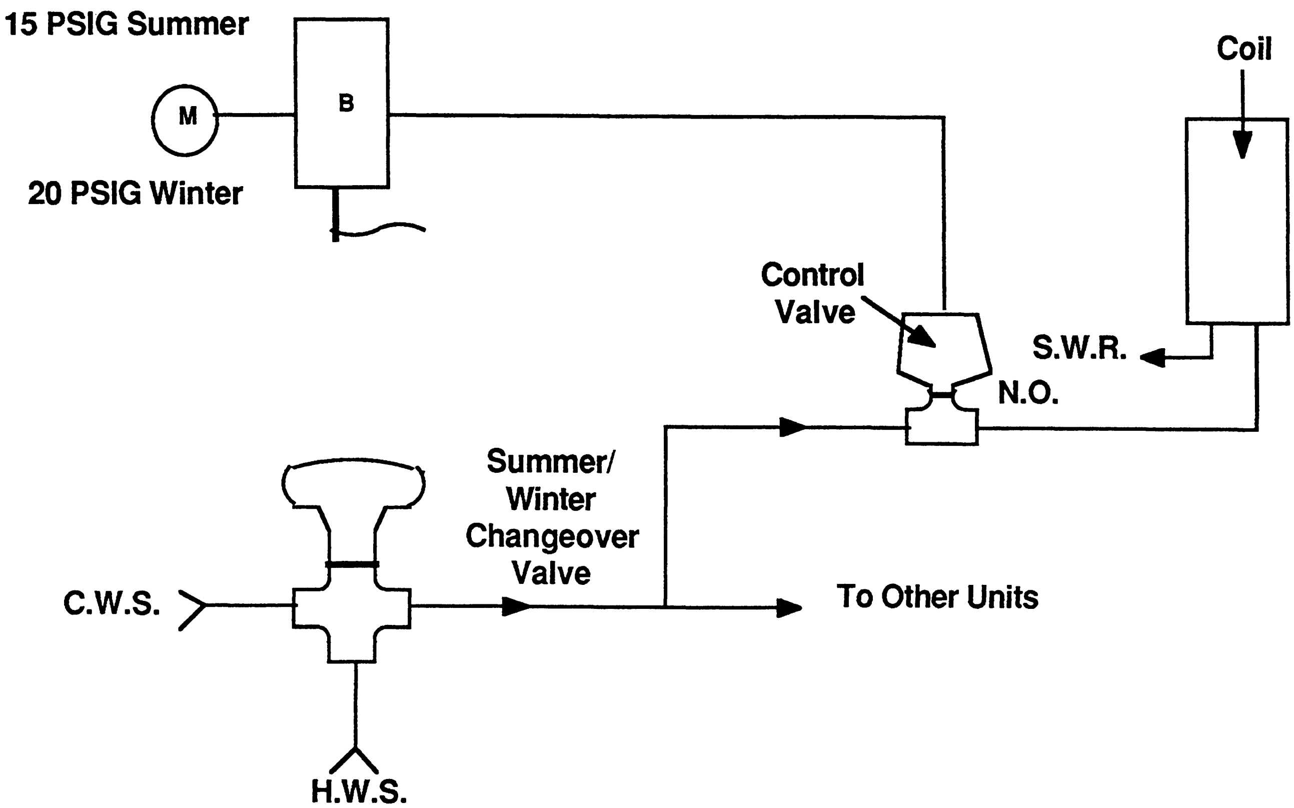


SINGLE PRESSURE UNITARY CONTROLLER

FIGURE 3-10

There are different types of unitary temperature control devices available to be used for single or dual pressure applications. A control system is illustrated in Figure 3-10. The controller, in this case, is direct acting. The unit ventilator is of the heating and ventilating type. When the return air temperature drops below the setpoint of the controller, the unit valve will open further to permit an increase flow of heating medium to flow through the coil and balance the increased load. The same branch signal that controls the valve also controls the dampers. As the signal to the valve decreases, to provide an increase input of heat to the space, the signal also decreases to the damper actuators which reduces the amount of cold outdoor air that is brought in and opens the face damper, further allowing more air to flow across the coil.

With an increase in space temperature, the opposite conditions will occur. That is, the branch signal will increase, closing the valve further, and reducing the flow of air across the coil, and increasing the amount of outdoor air that will be brought in to the space.



DUAL PRESSURE UNITARY TEMPERATURE CONTROLLER

FIGURE 3-11

Figure 3-11 illustrates a system utilizing a dual pressure unitary temperature controller.

The controller can be either reverse or direct acting, depending on the main air pressure supplied to it, and it is used to sense return air temperature. When the heating medium is supplied to the coil, the controller will normally be in the winter cycle, or direct acting. When the return air temperature drops below the setpoint of the controller, the unit valve will open further to permit the heating medium to flow through the coil at an increased rate which will balance the load. When the cooling medium is supplied to the unit coil, the controller will normally be in the summer cycle or reverse acting. When return air temperature exceeds the setpoint of the controller, the branch line presssure will decrease allowing the unit valve to open further, permitting the cooling medium to flow through the coil at an increased rate which will balance the load.

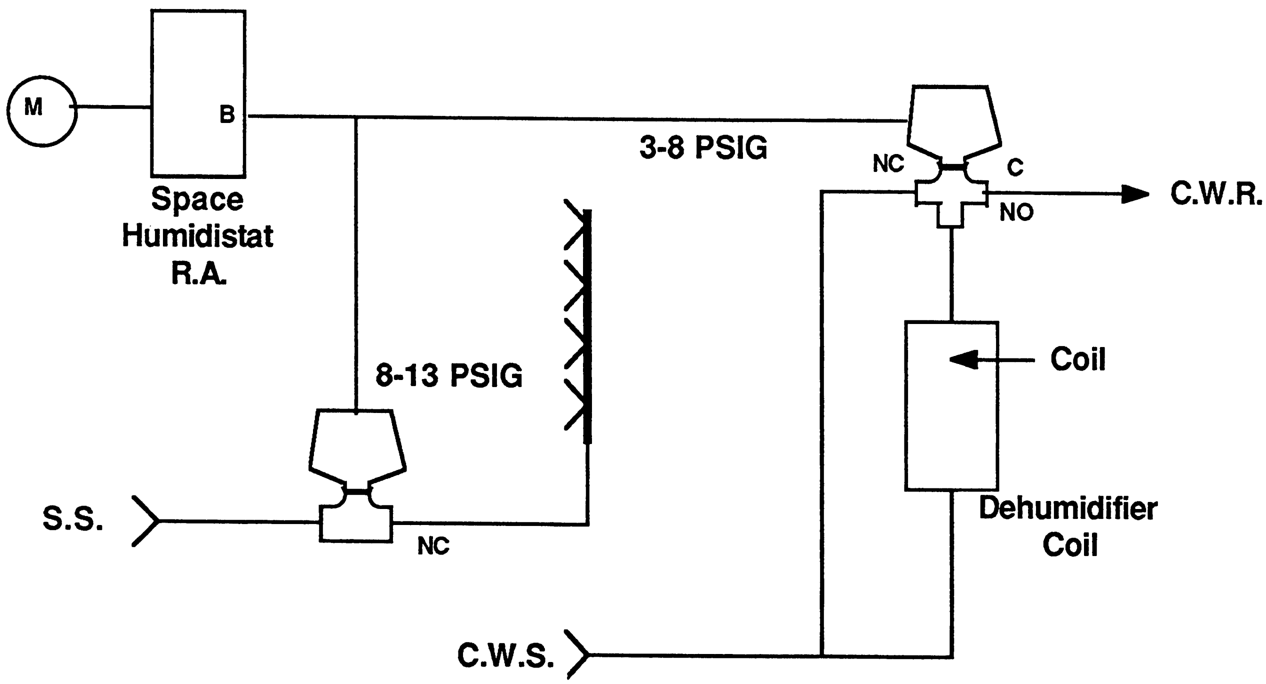
A significant point about unitary controllers, which should be noted, is that when the controller is being used to control flow through one coil used for heating and cooling, the controller should normally be capable of providing both direct and reverse action. When separate coils are used for cooling and heating (each coil having its own control valve), either a direct acting only, or reverse acting only unitary controller can be used.

In this case, the action of the controller chosen depends on how the unit valves are piped to the coils (normally open or normally closed).

Humidistats

A pneumatic humidistat is a proportioning type device designed to control pneumatic valves or damper actuators associated with cooling coils, humidifiers and air washers to maintain relative humidity control.

Room humidistats are similar in appearance to room thermostats. However, the humidity is sensed with a material which is hygroscopic rather than a bimetal. A hygroscopic element is one which absorbs moisture and increases in size. Human hair, nylon, silk, wood and leather are all hygroscopic elements. Nylon is commonly used in humidistats today because it is manufactured and its uniformity can be controlled.



HUMIDITY CONTROLLER APPLICATION

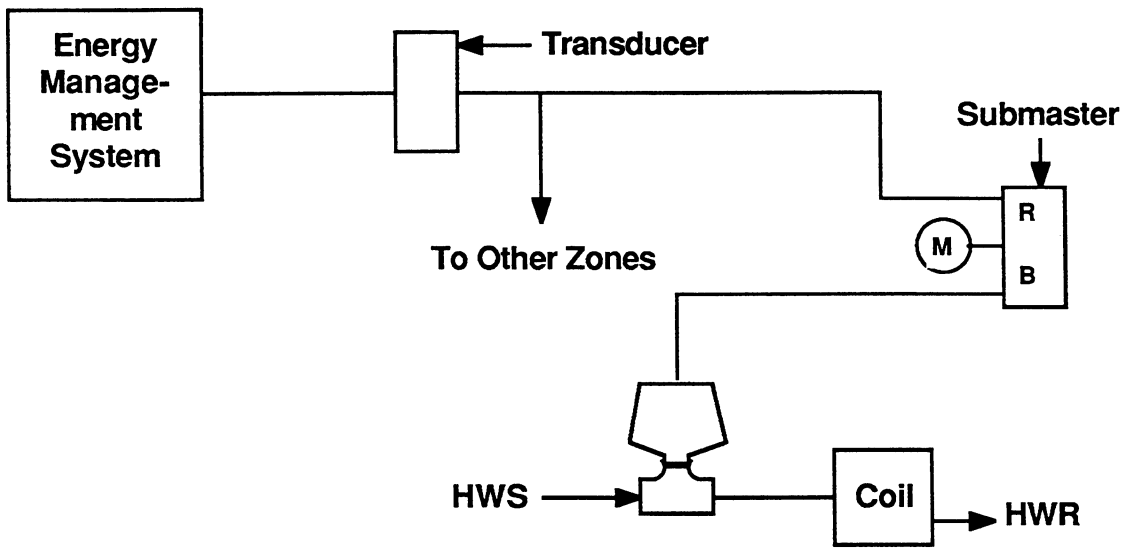
FIGURE 3-12

Figure 3-12 illustrates a humidistat application. The reverse acting humidistat is controlling a normally closed steam valve and a three-way chilled water valve. Hence, when the humidity is low, the output pressure of the humidistat is between 8 and 13 psig, and the normally closed steam valve is open to some value and adding humidity to the system.

The chilled water mixing valve modulates open between 3 and 8 psig. Thus, as humidity in the space increases, the output signal decreases and allows the chilled water flow through the coil to increase, providing dehumidification for the space.

Submaster Controller

Some control objectives can be met by transmitting an output signal to a second controller. The second controller, or submaster, is similar to a standard type thermostat. The significant difference is that the setpoint of the submaster changes as the signal input changes. The typical application for this device is an alternate method of day/night control.



RESETTING SUBMASTER CONTROL

FIGURE 3-13

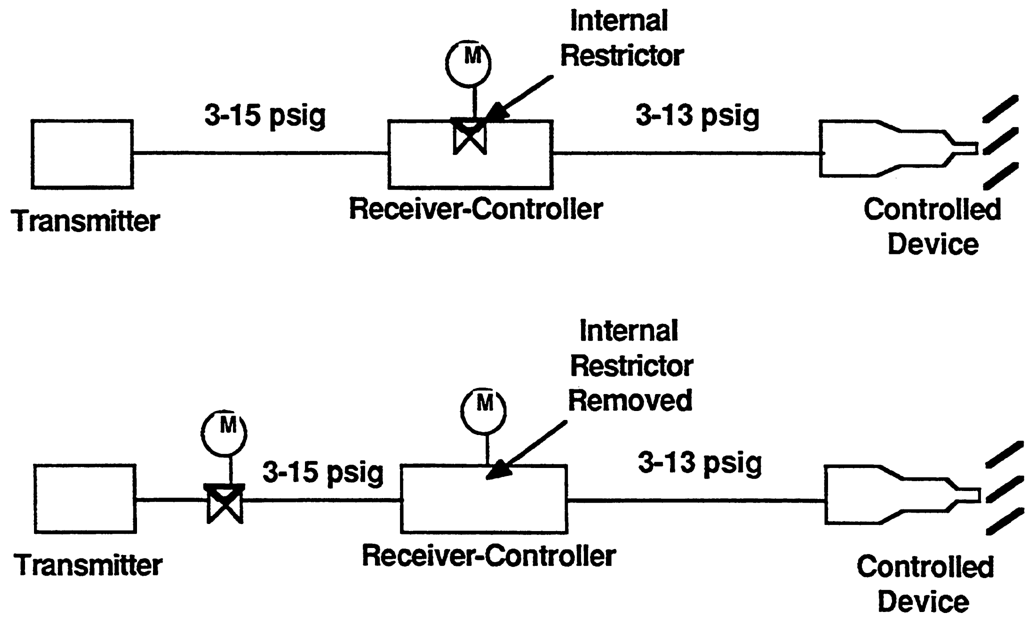
Figure 3-13 shows the resetting of the submaster thermostat to control at a different setpoint during the unoccupied hours. The advantage to this approach is that a dual pressure system, as shown in Figure 2-2, is not required to operate the day/night switchover. The submaster type thermostat makes it possible to interface with "state of the art" building management systems.

Transmitters and Receiver-Controllers

A **receiver-controller** is the controlling device in most modern pneumatic control systems. The sensing device used in conjunction with a receiver controller is called a **transmitter.**

The pneumatic transmitter senses changes in the variable (temperature, pressure, humidity) and varies the air signal in the connecting or transmission line to the receiver-controller. The receiver-controller then varies the branch line pressure to the controlled device in response to the transmitter signal.

A transmitter is a one-pipe, direct acting, bleed-type device which utilizes a restrictor in the receiver-controller to help maintain the proper volume of air between the transmitter and receiver controller. The restrictor may be removed from the receiver-controller and placed in the line between the transmitter and the receiver-controller. Figure 3-14 shows the two variations that are described here. The distance between the transmitter and the receiver controller may be greater with the restrictor placed between the transmitter and receiver-controller.



TRANSMITTER RECEIVER-CONTROLLER SYSTEM

FIGURE 3-14

There are different types of transmitters available for various applications. Transmitters are available in a variety of spans or ranges. Transmitters are direct acting, so there is a direct relationship between the span of the transmitter and its pressure output. For example, a 50-150°F range transmitter has a 100°F span. All transmitters produce a 3 to 15 psig output over their entire span. The output span of all transmitters is 12 psig (15 minus 3).

The psig change per engineering unit (degree, % R. H., etc.) is different with the various transmitters because they have different spans. As previously stated, all transmitters produce a 12 psig change over their span. Therefore, if the span is different, the pressure change per engineering unit will be different. Figure 3-15 shows the relationship of range, span and psig change. The psig change is calculated by dividing the total pressure change (12) by the span. For example, if the transmitter, which was described earlier, has a range of 50° to 150°F, and, therefore, a span of 100°F (150° minus 50°). You merely divide the total pressure change by the span ($12 \div 100 = .12$). Thus, for each degree change in the sensed temperature, the output pressure from the transmitter will be varied .12 psig.

Temperature Transmitters

Range	Span	PSIG Change 1 Degree F
50-100°	50°	.24
0-100°	100°	.12
50-150°	100°	.12
- 40-160°	200°	.06
40-240°	200°	.06

Humidity Transmitters

Range	Span	PSIG Change 1% R. H.
10-90%	80%	.15

Pressure Transmitters

Range	Span	PSIG Change 1" WC
- .25-6" *	2"	6

* This is actually the setpoint range of the transmitter. With any given setpoint between -.25" and 6", the span will be 2". This fixed 2" span will shift up and down the setpoint range as the setpoint is adjusted.

TRANSMITTER PRESSURE CHANGE PER DEGREE/PER % R.H./PER " WC

FIGURE 3-15

Gauges

These gauges are used in conjunction with the various Pneumatic Transmitters. They are precision 3 to 15psi pressure gauges that have face mounted scales that indicate temperature, humidity or pressure. Every transmitter has a receiver gauge with a corresponding scale.

Since all transmitters have a direct acting 3-15psi output, the proper receiver gauge will accurately indicate the condition of the variable as measured by the transmitter. These gauges are available as stem mounted models for mounting on the receiver controller or for flush mounting in panels.

Throttling range refers to the change in the variable (number of degrees, % R. H., etc.) in order for a receiver-controller to change its branch pressure output between 3 and 13 psig. **Proportional band** is the throttling range expressed as a percent of the span of the primary transmitter.

For example, a room type temperature transmitter has a range of 50^o to 100^oF, and, therefore, a span of 50^oF and a pressure change per degree of .24 psig. The output change of the transmitter to the receiver-controller causes the receiver-controller to change its output to the controlled device. Figure 3-16 shows this in more detail. A typical throttling range for room control would be 3^oF. That means as the load in the room were to change from 0% to 100%, the temperature in the room will change 3^o. That means that the pressure output of the transmitter to the receiver-controller would change .72 psig (3 x .24 = .72). This .72 psig change into the receiver-controller causes the receiver-controller to produce the 3 to 13 psig change. Therefore, the receiver-controller is an amplifier.

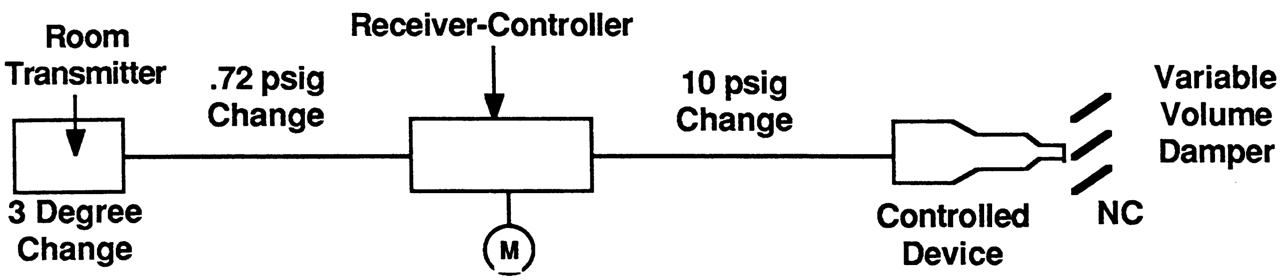
In this example, for the receiver-controller to produce the 3 to 13 psig when it receives a .72 psig change, the proportional band setting must be on 6%. The proportional band setting is determined by dividing the throttling range by the span and converting this decimal answer to a percent. **[PB = (T.R. ÷ Span) x 100]**

It can easily be seen that 3^oF is	**[PB = (3 ÷ 50) x 100]**
6% of the entire span (50^oF)	**[PB = 6%]**

For another example, perhaps a $40\text{-}240^oF$ transmitter is selected and a 10^oF throttling range is desired.

By using the previous formula	**[PB = (T.R. ÷ Span) x 100]**
You arrive at the	**[PB = (10 ÷ 200) x 100]**
Proper proportional band setting	**[PB = 5%]**

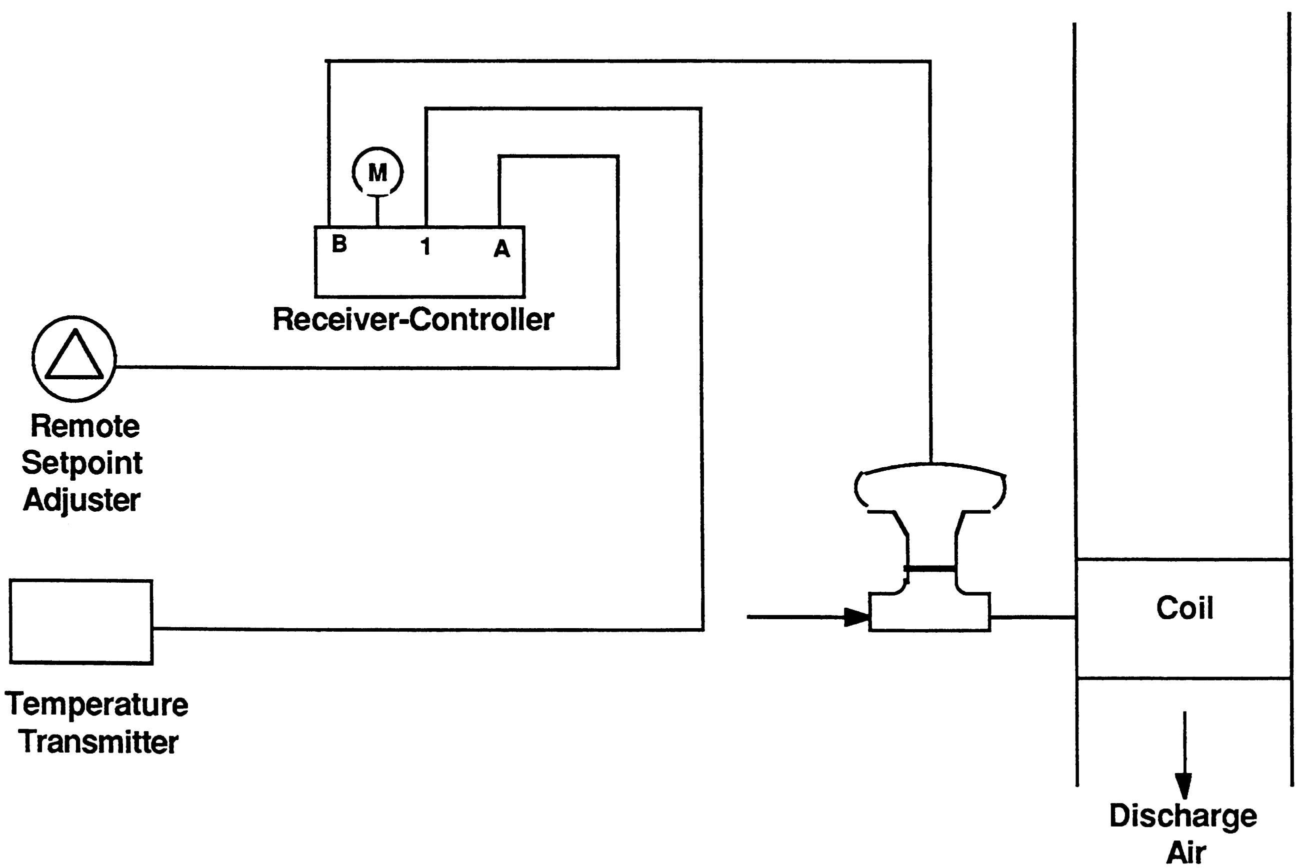
You can see, from this example, that the proportional band setting is 5%. A greater throttling range in this example requires a narrower setting than the previous example, although a wider throttling range is required. The reason is the smaller pressure change per degree (12 ÷ 200 = .06). The output of the transmitter will be .6 psig (.06 x 10 = .6). With the 200^oF span transmitter, it takes 10^oF to produce a .6 psig change. With the 50^oF span transmitter, it took 3^oF to produce a .72 psig change. The only method to accommodate the various spans of transmitters that transmit their signals to the same receiver-controllers is to convert the throttling range to a percent of the span of the transmitter. This percent of the span of the transmitter is called **proportional band.**



TRANSMITTER RECEIVER-CONTROLLER ROOM CONTROL

FIGURE 3-16

The correct throttling range is, of course, determined by the requirements within each system. If the setting is too wide, a large deviation from setpoint will occur during load changes. A throttling range which is too narrow causes the system to hunt. Therefore, these parameters must be considered when selecting the throttling range.

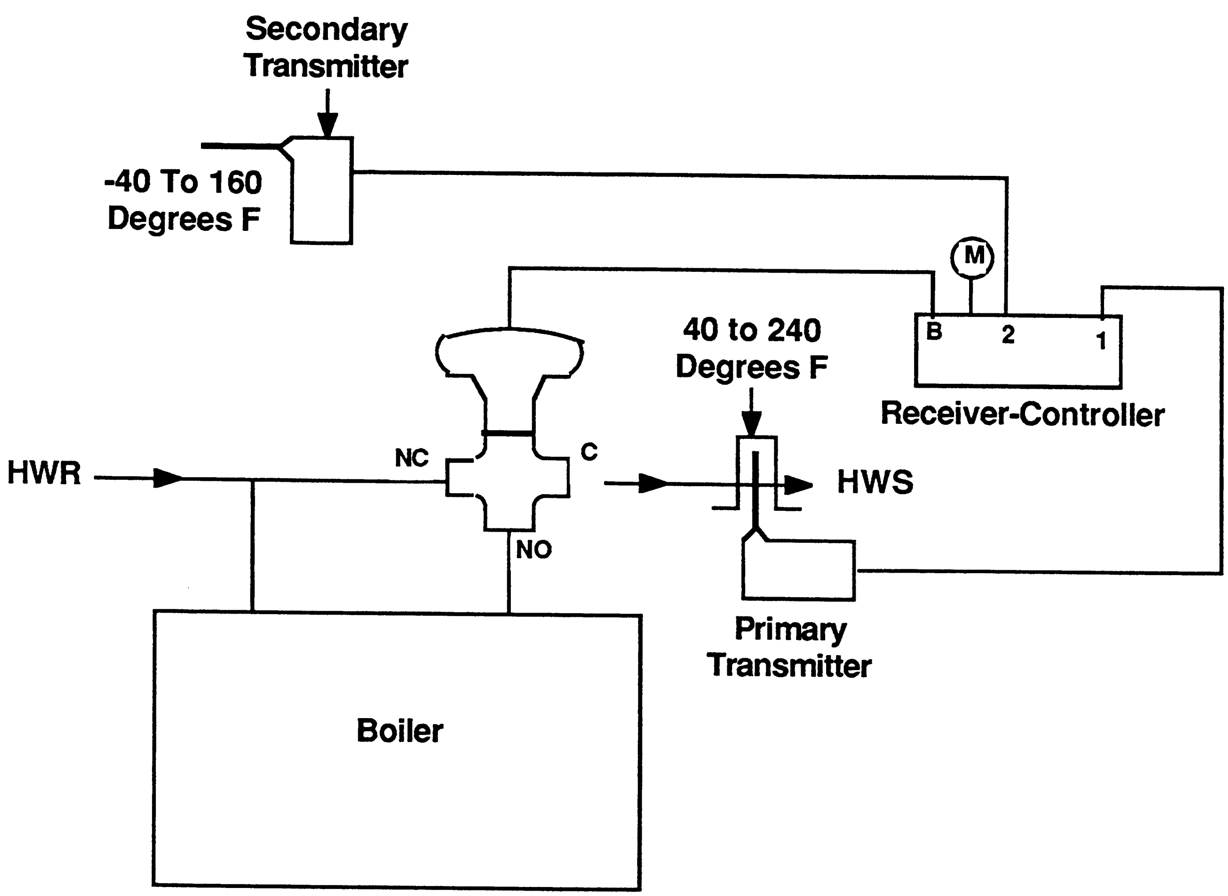


SINGLE INPUT RECEIVER WITH REMOTE SETPOINT

FIGURE 3-17

Figure 3-17 shows another single input application. There is a local setpoint adjustment and a setpoint scale. The setpoint scale is installed on the receiver-controller at the time of installation and must match the range of the transmitter being used.

This variation of the single input model being used has the capability of remote setpoint. The setpoint adjustment on the receiver-controller is also the calibration adjustment. The remote setpoint adjustment uses a gradual switch to provide a means of manually raising or lowering the setpoint of the receiver-controller without recalibrating the device. Varying the input at Port "A" from 3 to 15 psig varies the setpoint ± 10% of the span of the primary transmitter.



DUAL INPUT RECEIVER-CONTROLLER APPLICATION

FIGURE 3-18

Figure 3-18 illustrates an application that resets the setpoint of the receiver controller controlling the hot water valve in response to variations in the outside air temperature.

The primary transmitter senses the hot water supply temperature, and its signal is transmitted to Port "1" of the controller. The secondary transmitter, which is used to provide the reset effect, is piped to Port "2". The restrictors are within the receiver-controller.

As the outside temperature decreases, the setpoint of the receiver changes to increase the supply water temperature to offset the increase demand created on the building system. The opposite occurs on a rise in outside air temperature. The heating load decreases, thus the setpoint decreases to save energy costs.

In order for this system to operate properly, the correct range transmitters must be selected. Additionally, a reset schedule, throttling range and **percent authority** must be determined. For this example, an outside air transmitter with a -40 to 160°F range and a hot water supply transmitter with a 40 to 240°F range were selected.

Second, a throttling range must be determined. For this example, a throttling range of 10°F is used. Based on a primary transmitter span of 200°F (240 minus 40°F), the proportional band setting would be:

[PB = (T.R. ÷ Span) x 100]
[PB = (10 ÷ 200) x 100]
[PB = 5%]

The third step is to calculate the **percent authority**. In order to make this calculation, it is necessary to refer to a reset schedule. The reset schedule shows at what temperature the water should be maintained as the outside air temperature varies. Without a reset schedule, the controller cannot be calibrated properly. In this example, the system requires 180°F water when the outside air temperature is -10°F and 90°F water when the outside air temperature is 65°F.

Primary H.W. Temp.	**Secondary O.A. Temp.**
180 Degrees F	-10 Degrees F
90 Degrees F	65 Degrees F

RESET SCHEDULE
FIGURE 3-19

The reset chart shown in Figure 3-19 shows the relationship of outside air temperature to the hot water discharge temperature throughout the schedule. Thus, a properly calibrated system should be within the parameters of this chart.

To calibrate the **percent authority**, the following formula can be applied:

$$[PA = (\Delta T \text{ Primary} \div \Delta T \text{ Secondary}) \times (\text{Span of Secondary} \div \text{Span of Primary}) \times 100]$$

$$[PA = (90 \div 75) \times (200 \div 200) \times 100]$$

$$[PA = 120\%]$$

The various items placed in the equation are defined as follows:

ΔT Primary (from reset schedule); 180° minus 90°F = 90°F

ΔT Secondary (from reset schedule); 65° minus -10°F = 75°F

Span of Secondary (from range of outside air transmitter); 160° minus -40°F = 200°F

Span of Primary (from range of hot water transmitter); 240° minus 40°F = 200°F

The spans of the transmitters are used in the formula because they will have an effect on the amount of reset when they are of different spans. Refer again to Figure 3-15. Remember the different spans of the transmitters will cause them to have different pressure changes per engineering unit. This portion of the formula accounts for the different signal values that would occur with all the different transmitters. In the example shown, both transmitters were of equal spans. Therefore, these two values cancelled out. In other instances where the two transmitters have different spans, this portion of the formula will make this correction.

The percent authority adjustment for this example should be set at 120%. The controller can now be calibrated.

Two-Input Receiver Controller Calibration

There are two methods used to calibrate these controls; the first is manually using the transmitters that are already piped in the controller. The second method employs the use of a precise calibration box. This device with its very exact adjustments allows the technician to not only calibrate very accurately but to also check the effects of the proportional band and percent authority adjustments.

The manual method is as follows:

1. Check to be sure 18 psig minimum is available to main air connection "M".
2. Select and apply proper setpoint decal provided with receiver-controller to setpoint slider. Decal must match primary transmitter range.

3. Calculate and adjust proportional band and percent authority.

4. With an accurate measuring instrument, measure the condition at the transmitter, ie. temperature, relative humidity, pressure. This must be stable.

5. Adjust all permanent gauges to match the conditions as measured.

6. If remote setpoint is used, adjust RSPA to obtain 9 psig at port "A".

7. Rotate setpoint knob CW or CCW until mid-spring valve of controlled device is read on branch gauge (or 8 psig if unknown).

8. Slide setpoint scale until measured value of primary transmitter is read at setpoint indicator.

9. Rotate setpoint knob until condition for primary transmitter is indicated on setpoint scale for current condition at secondary transmitter.

10. Slide setpoint scale to minimum condition of primary transmitter as shown on reset schedule.

- CHAPTER 4 -

CHAPTER 4

TABLE OF CONTENTS

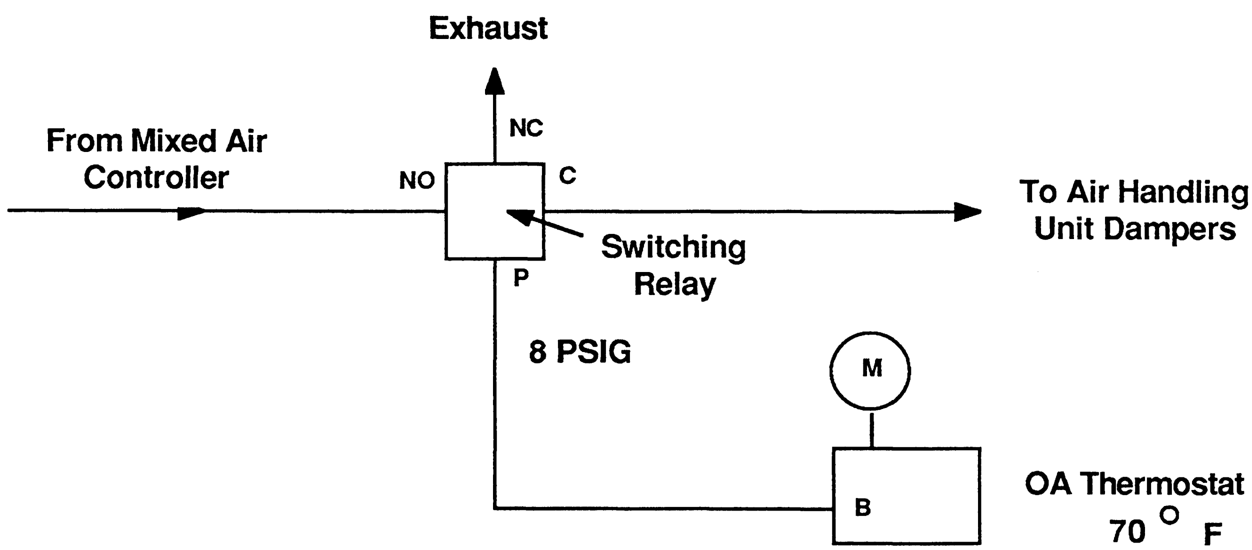
SECTION **PAGE**

PNEUMATIC RELAYS

Pneumatic relays are used to provide a multitude of functions that can be summarized into three functions. They are (1) switching, (2) volume amplifying and (3) selecting.

Switching Relay

A switching relay is a three-way air valve which is designed to provide a variety of switching and interlocking functions in the pneumatic system. Its primary function is to convert a pneumatic signal, at a predetermined setting, to a final control device.



CHANGEOVER SYSTEM USING SWITCHING RELAY

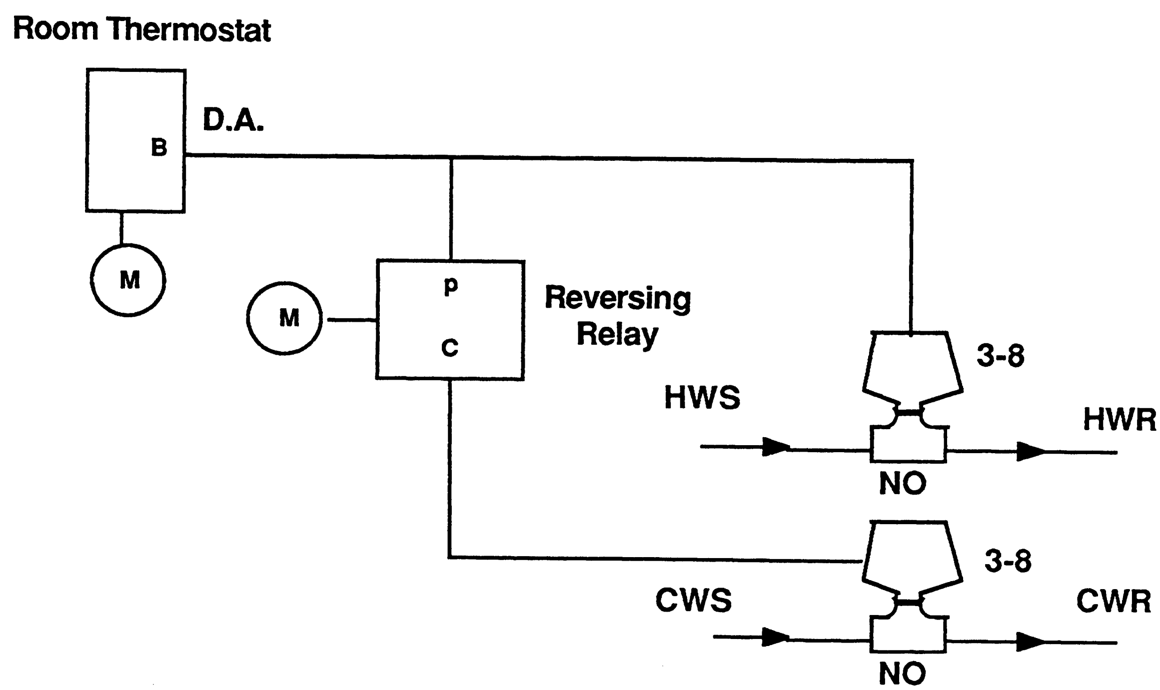
FIGURE 4-1

In the illustration shown in Figure 4-1, the switching relay is being utilized to execute temperature changeover to an economizer application. The switching relay allows the mixed air controller to control the air handling unit dampers up to the setpoint of the outside air thermostat, typically 70^{O}F. The signal from the outside air thermostat goes to the pilot port. When the pressure from the outside air thermostat reaches the setpoint of the switching relay (8 psig for this example), the normally open port closes and the normally closed port opens. At this point (8 psig), the switching relay switches and blocks the signal at the "NO" port and exhausts the "NC" port to allow the air handling unit dampers to close or go to minimum position. This application is considered switching from a high or increasing signal.

This relay can also be used to switch from a low or decreasing signal by piping the signal from the controller to the "NC" port instead of the "NO" port, and exhausting out through the "NO" port. The signal from the thermostat that causes the switching to take place would still go to the pilot port.

Reversing Relay

This device is designed for use in pneumatic control systems where the application requires reversing a signal from a controller. The branch line pressure output of the relay decreases in direct proportion (1:1) to an increase in input signal pressure from the controller.



REVERSING RELAY APPLICATION

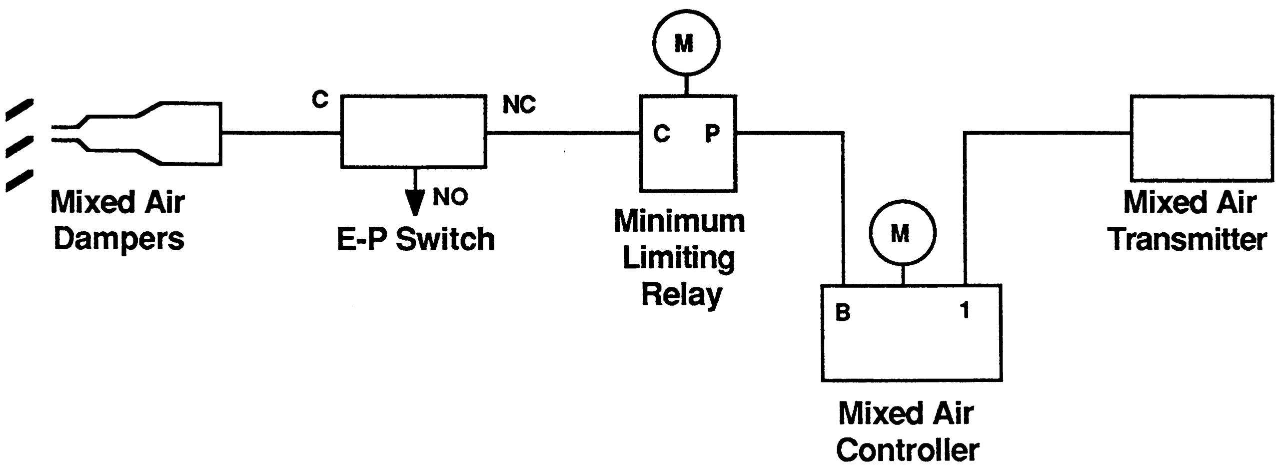
FIGURE 4-2

Figure 4-2 shows the typical way in which a reversing relay is used. The direct acting space thermostat is controlling both a heating and cooling valve. Each valve is normally open. In this example the branch signal from the thermostat is piped to the heating valve and to the reversing relay. The signal from the reversing relay goes to the cooling valve. There is also main air piped into the reversing relay. As the space thermostat senses a decrease in temperature, its branch pressure decreases, which causes the heating valve to open to some value and permit an increased flow of hot water.

The same signal goes to the reversing relay where it is reversed to an increase in pressure to the cooling valve. This closes the cooling valve. As the thermostat senses an increase in temperature, its output pressure increases, closing the heating valve. The signal to the reversing relay is reversed, causing its output pressure to decrease and open the cooling valve to some value so that the chilled water will flow at a rate to balance the load.

Minimum Limiting Relay

This device is commonly used in an economizer system. The main requirement for this system is to use outside air for cooling purposes whenever the temperature is low enough to make this possible.



MINIMUM LIMITING RELAY

FIGURE 4-3

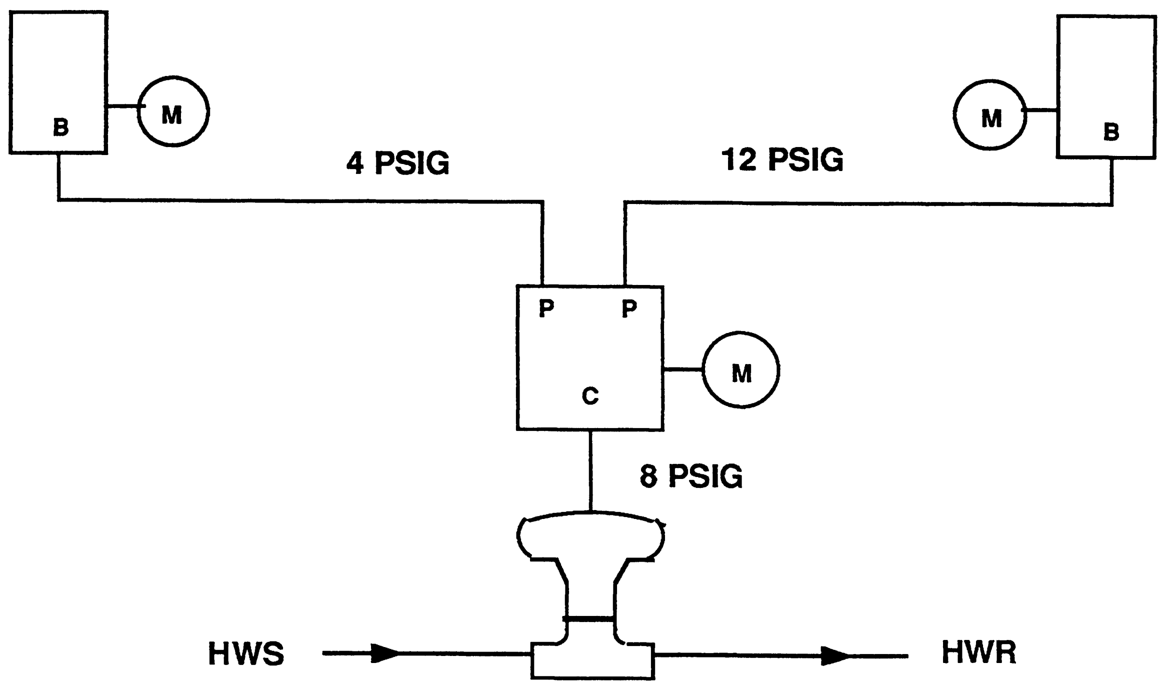
Figure 4-3 shows a minimum limiting relay used to position the outside air damper.

The controls are activated when the unit fan is on through the electric-pneumatic relay (E-P Switch). The transmitter senses the mixed air temperature and sends a signal to the receiver-controller. The receiver-controller develops the signal to position the dampers. The minimum limiting relay has a setpoint which is the pressure required to maintain a minimum position of the outside air damper. Whenever the receiver controller output (branch signal) pressure is less than the minimum limiting relay setpoint, the relay will provide the minimum pressure to the damper actuator. This allows for a minimum volume of outside air required by local codes.

When the fan motor stops, air is exhausted through the normally open port of the electric-pneumatic relay. The damper actuator returns to its normal position, closing the outside air damper.

Averaging Relay

This type of relay is designed for use in pneumatic control systems where the application requires operation of a final control device by the average signal from two pneumatic controllers.



AVERAGING RELAY APPLICATION

FIGURE 4-4

In the example, shown in Figure 4-4, the two space thermostats send their respective branch signals to the averaging relay. The relay, in turn, sends the average of the two signals to control the heating valve.

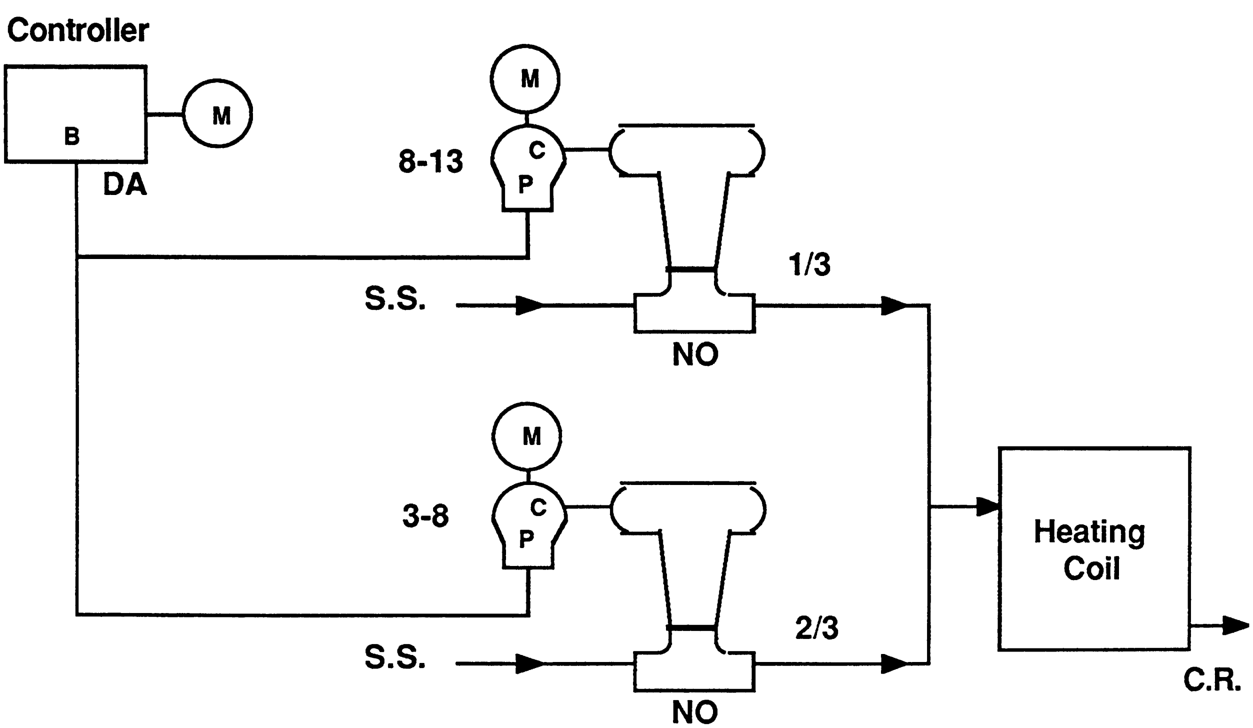
The typical application for this approach is with a single zone air handling unit where the load in the zone may be very uneven. With one thermostat, the occupants would be uncomfortable in extreme areas of the zone. By averaging these extremes, the occupants would be much more comfortable.

Positive Positioner Relay

The positive positioner relay is used to achieve accurate positioning of a valve or damper. The relay is mounted on the actuator.

The accurate positioning occurs because the maximum force that is available (which is main air pressure) can be supplied at any point of the actuator stroke. In a regular arrangement where the controller sends its signal directly to the actuator, the pressure to the actuator will be at some value between 3 and 13 psig. For example, if the pressure were to increase from 3 to 4 psig, that may not be enough force, under some circumstances, to cause the actuator to reposition the valve or damper. With the positive positioner, the actuator will have up to the main pressure applied to it to move the damper or valve to the position that the branch pressure of 4 psig requires. A feedback mechanism tells the positive positioner when the actuator reaches the positions that the branch pressure calls for.

Also the pressure to the actuator may be decreased to zero psig by the positive positioner on the return stroke to give the actuator the maximum spring force. For example, if the branch pressure decreases from 8 to 7 psig and the actuator was unable to move the damper or valve, the pressure to the actuator could decrease to 0 psig until the actuator reached the position that 7 psig requires. The feedback from the actuator to the positive positioner tells the positive positioner when the actuator has reached the position that the branch pressure calls for.



POSITIVE POSITIONER

FIGURE 4-5

The positive positioner has two adjustments that make the device very useful beyond providing highly accurate positioning.

One is a start point adjustment, which means the actuator can be started at a selected psig value different from the start point of the spring of the actuator. Also, there is a span adjustment.

By combining these two adjustments, an actuator can be made to operate over an entirely different range than the spring of the actuator would dictate without the positive positioner.

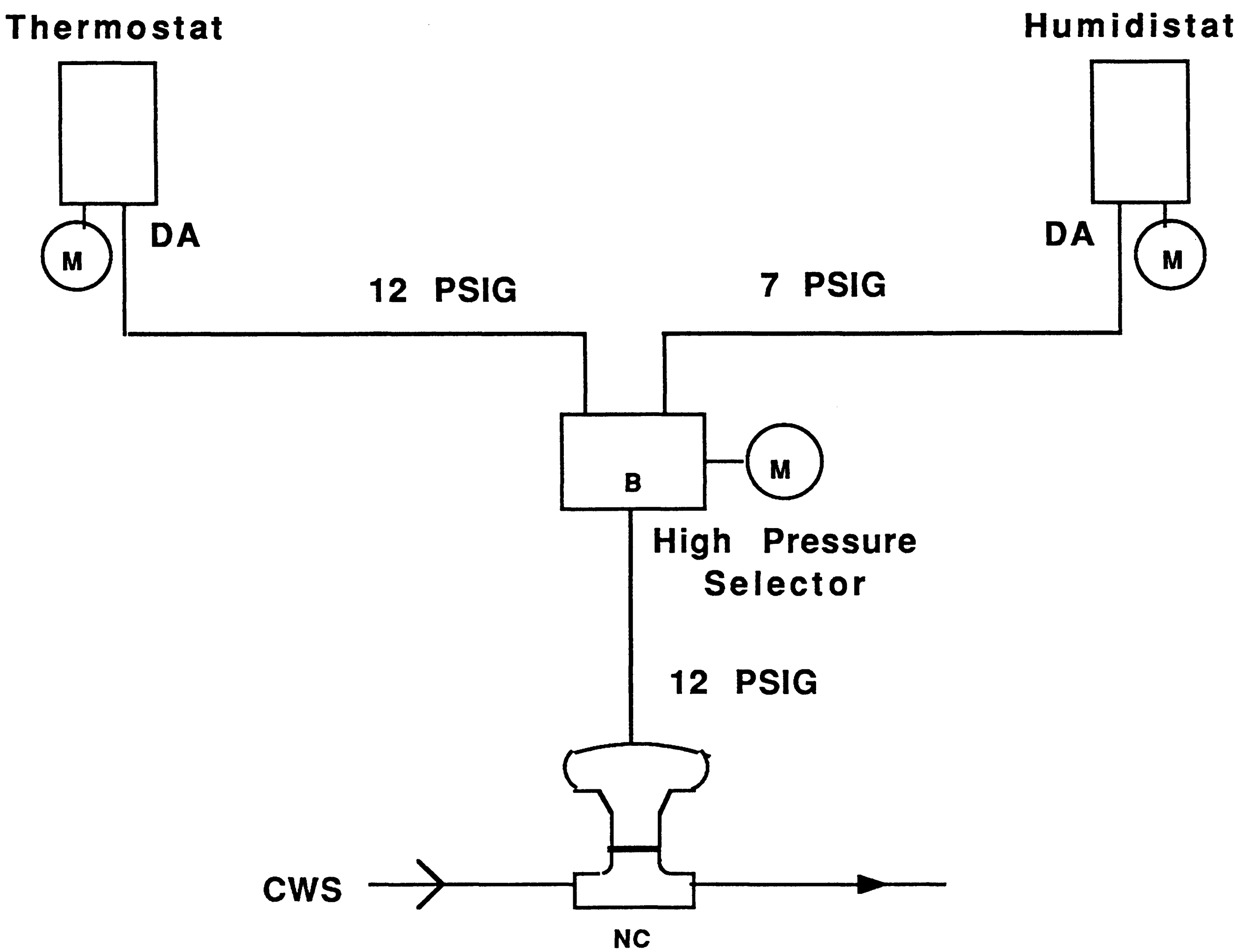
Figure 4-5 shows a common application of the positive positioner. Commonly, it is used to sequence two actuators with the same spring range. This application requires that we sequence the 1/3 valve open first and the 2/3 valve last. In order to have enough force to obtain adequate close off pressure, both actuator springs must have the 3-8 psig range. Therefore, the adjustments on the positive positioner are used to obtain the 3-8 and 8-13 sequence required.

It is a good practice to use positive positioners on large valves (2 inch and larger), outside air dampers, inlet vane dampers, and any other applications where accurate positioning is necessary.

The last four relays that have been described, which are the reversing, minimum limiting, averaging and positive positioning, are volume amplifying relays, in addition to performing the specific functions. That means that the signal from the controller enters a pilot chamber of the relay. The change in pressure at the pilot chamber manipulates a mechanism to divert the main air supply from the output of the relay to the controlled device. With the volume of air that is available from the main and the size of the air passages of the valving mechanisms, a high volume of air can pass to the controlled device. This is especially important if a bleed type controller is used. The low volume of air that is delivered from the branch line of a bleed type controller to the pilot chamber of a relay is more than enough to activate the mechanism to divert the main air to the controlled device to give rapid system response.

Pressure Selector Relays

These devices are used in pneumatic control systems where the application requires the comparison, selection and transmission of one of two or more proportional signals supplied to the relay. Units are available as either a low pressure selector or high pressure selector. Both units allow for two inputs or multiple inputs. The low pressure selector receives input signals, compares, selects and transmits the **lower** signal. The high pressure selector receives input signals, compares, selects and transmits the **higher** signal.



DEHUMIDIFYING OR COOLING FROM HIGHEST SIGNAL

FIGURE 4-6

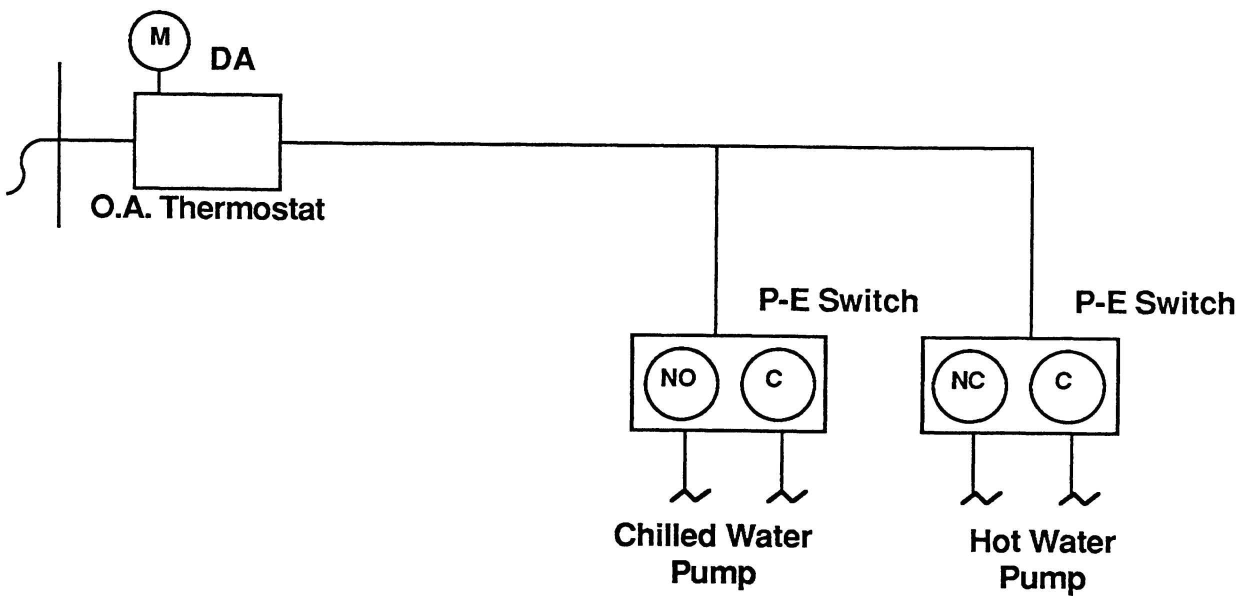
The arrangement shown in Figure 4-6 for a high pressure selector is often used for cooling because either the thermostat or the humidistat can control the cooling valve. In this manner, the cooling valve is controlled by either high temperature or high humidity.

In appearance the low selector relay looks the same as a high selector. However, functionally it works in the opposite manner. It passes on the lowest of the signals it receives. If it had been used in the illustration and example set forth in Figure 4-6, the branch pressure to the valve would be 7 psig.

Pneumatic-Electric Switch

Pneumatic-electric switches, or P-E switches, are used in control systems requiring conversion of an air pressure change to a positive electrical switching action. Typical applications are starting and stopping unit ventilator fan motors, fan coil motors, unit heaters, air handling unit fans, chillers and chilled and hot water pumps.

Several models are available; the major difference is the switching action and differential adjustment. Most P-E switches utilize single pole-double throw (SPDT) snap-acting switches. However, there are also double pole-single throw (DPST) models and double pole-double throw (DPDT) models available which open either on a decrease in pressure or on an increase in pressure. The differential is adjustable so that either a wide or narrow differential may be obtained for the application involved.



PNEUMATIC-ELECTRIC SWITCH

FIGURE 4-7

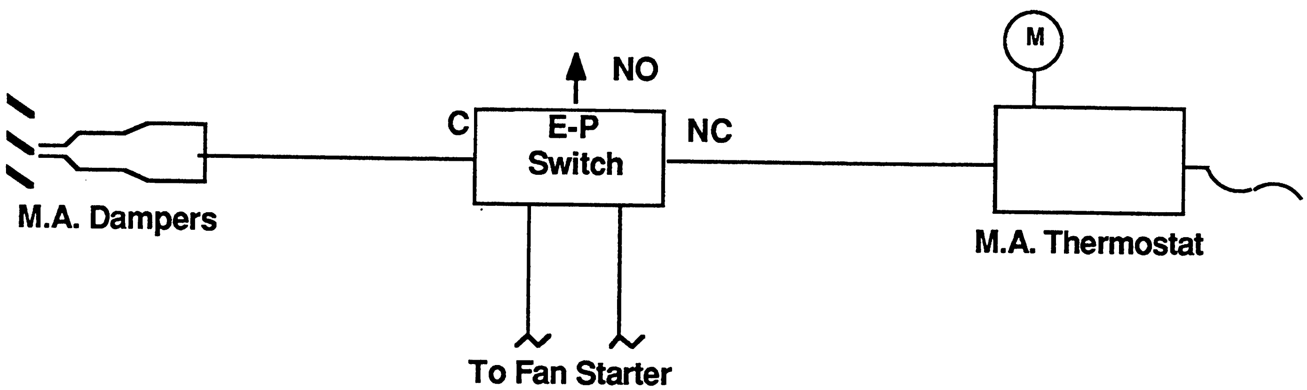
The application of a P-E switch is shown in Figure 4-7. A change in the signal pressure from the outside air thermostat is used to open and close the contacts of the P-E switch, which, in turn, energize or de-energize the pump starter. Separate P-E switches can be used to control the operation of the chilled water pump starter and the hot water pump starter. Each P-E switch is set to either open or close its contacts at a predetermined pressure signal from the outside air thermostat. Since the P-E switch in this example is a SPDT switch, it can be wired for either normally open or normally closed operation. In this example, the chilled water pump starter is wired to the normally open and common terminals of the P-E switch. An increase in the pressure signal from the outside air thermostat causes the contact to make and energize the pump starter. The hot water pump starter is wired to the common and normally closed terminal of the P-E switch.

As the outside air thermostat signal decreases below the setpoint of the P-E switch, the contacts will make and energize the hot water pump starter.

Electric-Pneumatic Switch

This device is a solenoid air valve for two position action. It has three connections which are marked normally open, normally closed and common.

It is designed for applications when an electrical circuit is used to control a pneumatically operated device. It may be used to direct supply air to a pneumatic device when the coil is energized or de-energized, depending on supply and exhaust connections.



ELECTRIC-PNEUMATIC SWITCH

FIGURE 4-8

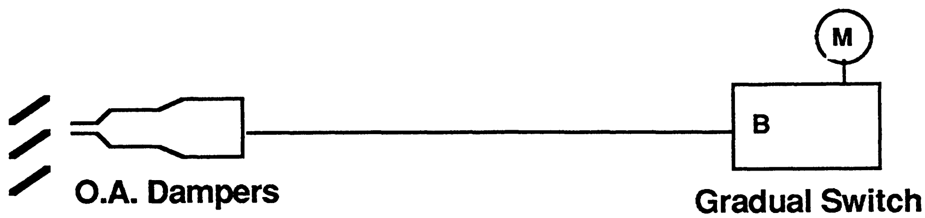
Electric-pneumatic relays, or E-P switches, are commonly used as interlocking devices between an electrical circuit and a pneumatic circuit. Figure 4-8 utilizes an E-P switch to control air handling unit dampers. It is wired electrically to the fan starter circuit so that when the fan motor is running, the proper ports are connected to allow the outside and return air dampers to modulate.

Should the fan stop running (due to a time clock function or because of system malfunction), the circuit in the E-P switch will de-energize. When de-energized, the signal controlling the outside and return air dampers is blocked, the control air is exhausted to the atmosphere and the dampers return to their normal position.

Gradual Switch

This is a manually operated device designed to deliver a selected air pressure from 0 to main pressure. Therefore, the device requires a main input, and the variable signal output is the branch line pressure to the controlled device.

This switch is widely used in pneumatic control systems for manually positioning dampers or valves.



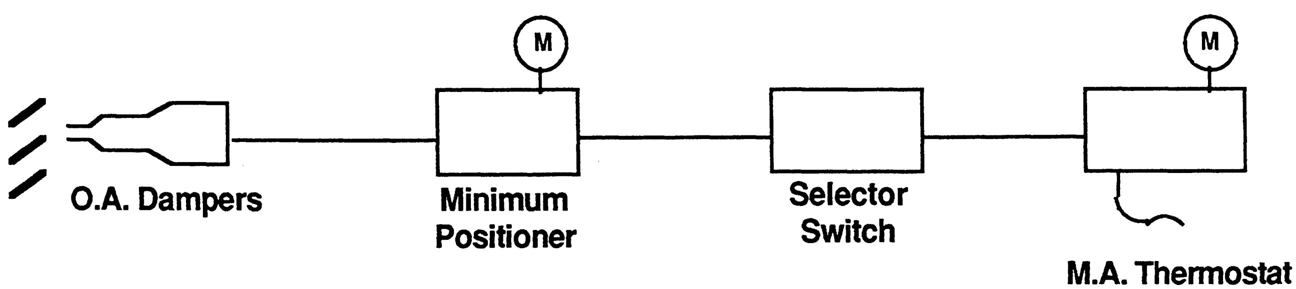
MANUAL CONTROL OF OUTSIDE AIR DAMPERS

FIGURE 4-9

In the example shown in Figure 4-9, the building operator may control the amount of ventilation by adjusting the manual positioning switch. For this system, the outside air may be used for cooling, in addition to ventilating. Therefore, the temperature can be controlled manually.

<u>Selector Switch</u>

This is also a manually operated device designed to deliver different pneumatic signals to controlled devices. Selector switches may select one, two, three or four different signals to be passed on one at a time to accomplish a given function.



MANUAL CHANGEOVER SYSTEM

FIGURE 4-10

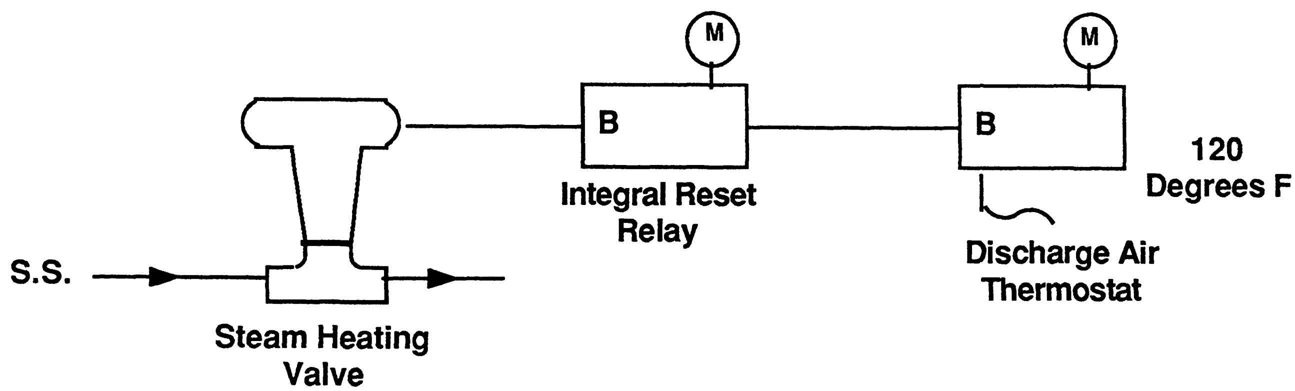
In Figure 4-10 the selector switch is used to accomplish changeover of a winter ventilation system to the summer cycle. During the winter cycle, the signal from the mixed air controller goes to the outside air dampers in order to cool and ventilate the space.

For the summer cycle, the switch is positioned to prevent the signal from reaching the dampers. Now the dampers are positioned by the minimum positioner and the mechanical cooling is in operation.

Integral Reset Relay

This device is designed to add integral reset to proportional controllers. It can be used with any proportional controller, and its function is to minimize control point offset. Control point offset is the deviation from the setpoint of the controller that frequently occurs within the throttling range of the controller as significant load changes occur.

The integral reset relay continuously increases or decreases its output as necessary to maintain the condition of the variable to match the setpoint of the proportional controller.



INTEGRAL RESET CONTROL

FIGURE 4-11

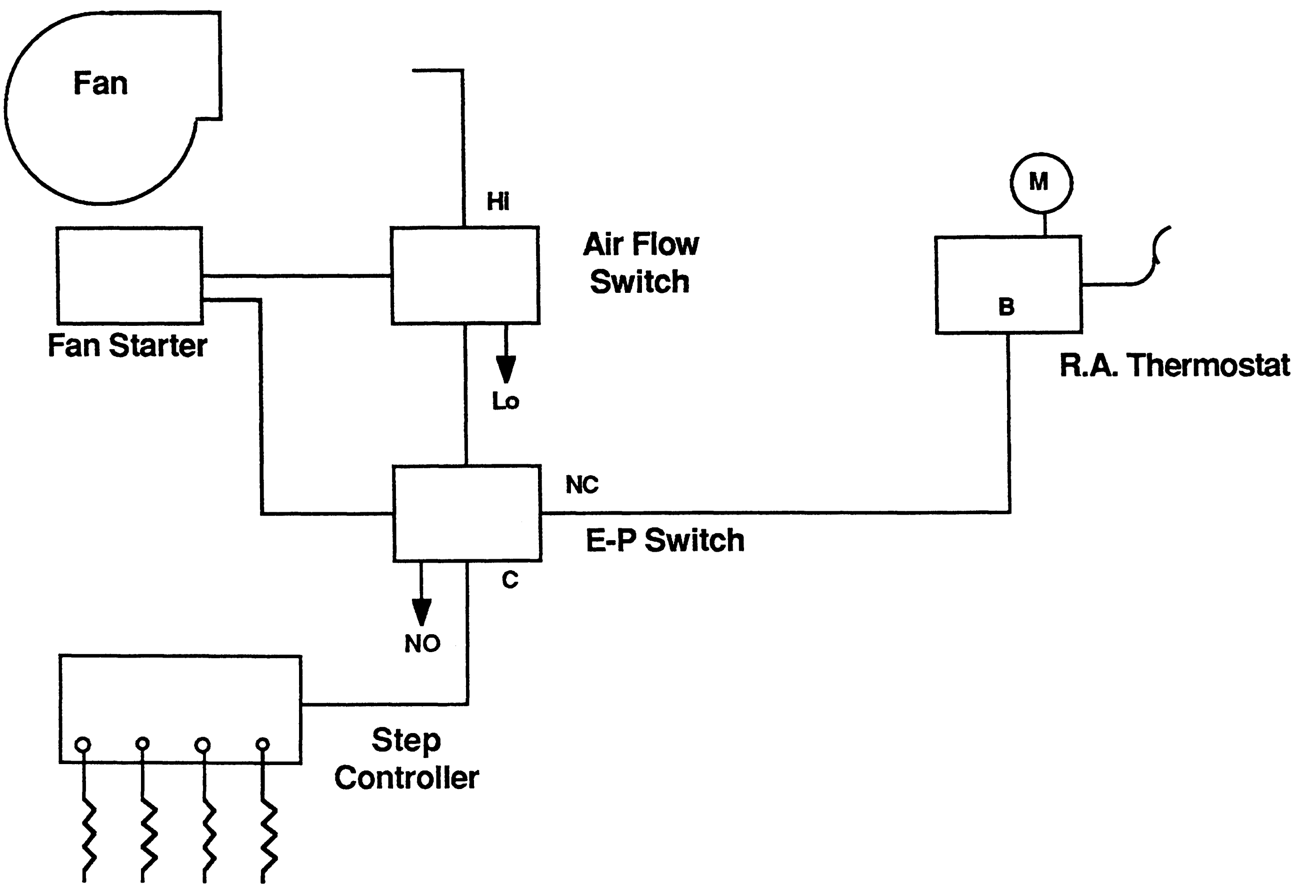
In the example, shown in Figure 4-11, it is desired to maintain the supply air temperature at a constant 120°F. Because of the potentially large differential between the entering air temperature and leaving air temperature, a wide throttling range is required on the thermostat for system stability (prevention of hunting or cycling). The offset that results from a wide throttling range would be objectionable (typically plus or minus 10°F). The addition of the integral reset relay can substantially reduce or eliminate the offset, and allows stable and precise control.

The relay will output any pressure between 0 psig and full main pressure as necessary to maintain the temperature of the discharge air from the heating coil to match the setpoint of the thermostat.

Air Flow Switch

This switch is used to sense suction and/or discharge pressures across a coil or fan and control pneumatic actuators. By the use of sensing lines located at the fan suction and discharge and piped to the low and high ports of the air flow switch, the device is able to know whether or not the fan is operating.

The air flow switch performs this function with an E-P switch. It allows the control system to operate only when the fan is operating. If only an E-P switch was used, the fan circuit could be energized, and the control system would function even if the fan motor itself was to fail. Using an air flow switch, the fan must be moving a volume of air, therefore, producing enough air pressure necessary to actuate the air flow switch.



AIR FLOW SWITCH

FIGURE 4-12

When using the air flow switch as shown in Figure 4-12, there must be enough differential pressure to cause the switch to operate properly. In this example, the low pressure sensing port is left open to atmosphere, and the high pressure sensing tap sensing the fan discharge pressure is piped to the high pressure port.

When the fan is operating, the switch senses the discharge pressure of the fan, causing the contacts to close. This allows the E-P switch to be energized and allow the signal from the return air thermostat to control the electric resistance heaters. When the fan is not operating, the result of no pressure across the diaphragm of the air flow switch will allow the contacts to fall away and de-energize the E-P switch. Any pressure at the step controller will then exhaust; therefore, the resistance heaters cannot be energized.

- CHAPTER 5 -

CHAPTER 5

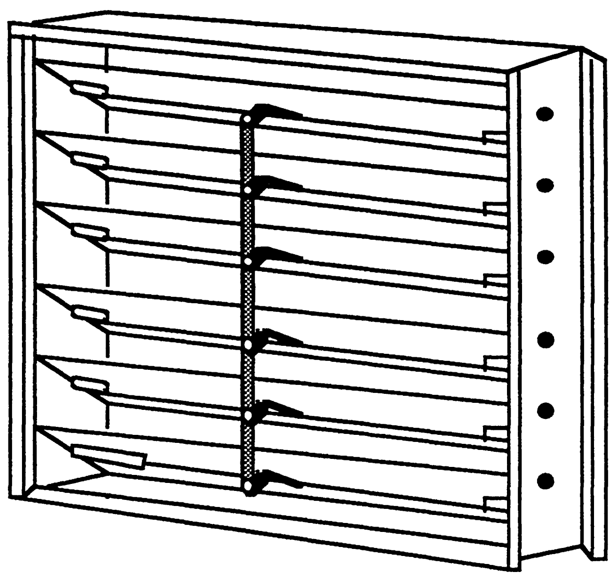
TABLE OF CONTENTS

SECTION **PAGE**

FINAL CONTROL DEVICES

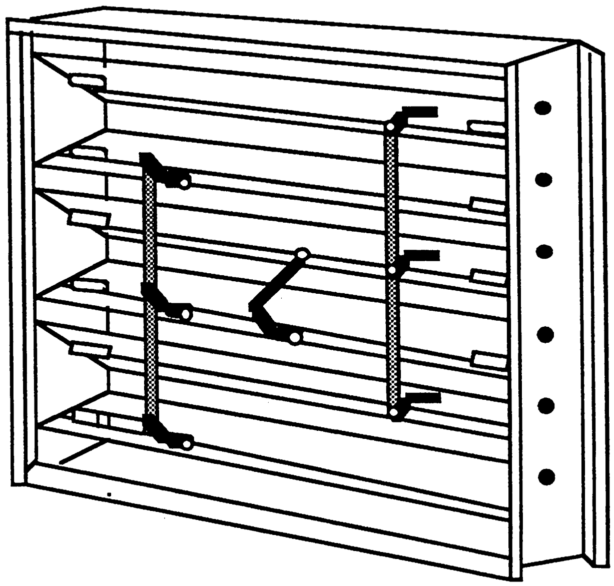
Dampers

Air flow in an HVAC system is regulated by dampers which are installed in air-carrying ducts. There are various types of dampers and blade designs, including butterfly, flap and louver. The blade-type damper is the most commonly used for controlling inlet or exhaust flow because it provides better control.



PARALLEL BLADE DAMPER
FIGURE 5-1

A parallel blade damper, shown in Figure 5-1, is constructed so that all of the blades rotate in the same direction. Air flow, through parallel blade dampers, results in increased turbulence, thus better mixing. This type of damper is best suited for most ventilation applications.



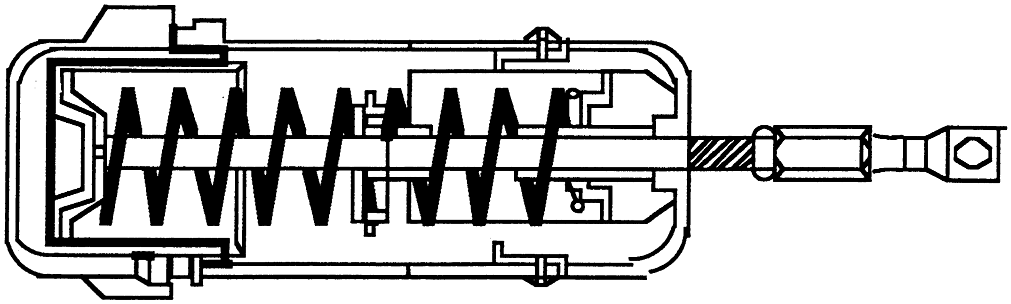
OPPOSED BLADE DAMPER

FIGURE 5-2

An opposed blade damper, as shown in Figure 5-2, is constructed so that alternate blades rotate in opposite direction. They have an equal percentage air flow characteristic which allows close control at low air flow. Because they are more expensive than parallel blade dampers, they are usually used only where accurate control at low air flow is required.

Damper Actuators

Pneumatic damper actuators position dampers according to the signal provided by a controller. The movement of the piston of the actuator varies proportionally with air pressure applied to the diaphragm. This air pressure applied to the diaphragm forces the piston to move against the force of the spring. If the air pressure from the controller decreases, the spring force may be greater and the actuator will retract to a new position. When air pressure is removed from the actuator, the spring returns the device to its normal position. See Figure 5-3 for a diagram showing the parts of a damper actuator in detail.



DAMPER ACTUATOR

FIGURE 5-3

The spring range of the actuator restricts the movement of the piston to set limits. For example, an actuator with a 3-8 psig spring range is in its normal position when the air pressure applied to the device is 3 psig or less. Between 3 and 8 psig, the stroke is proportional to the air pressure supplied. Above 8 psig the actuator remains at maximum stroke.

Damper actuators are rated by the damper area (in square feet) which they can operate. The diaphragm area of the actuator determines the output torque that the actuator can deliver. For a comparison of the damper actuator sizes and the square footage of dampers they can operate effectively, refer to Figure 5-4.

Actuator Diaphragm Area	Nominal Damper Area	
Square Inches	Parallel Square Feet	Opposed Square Feet
3	1.25	1.6
6 *	1.25	1.6
8	11.6	15.0
11 **	4.4	5.6
20	37.5	48.2

* Left arm arrangement with 6 square inch diaphragm produces same torque as 3 square inch actuator.

* * Left arm arrangement with 11 square inch diaphragm produces less torque as 8 square inch actuator.

FIGURE 5-4

If the damper area is greater than the rating of the actuator, a larger, more powerful damper actuator must be used. As an alternative, two or more smaller dampers with an actuator on each can be used; or two or more actuators may be linked to the same damper.

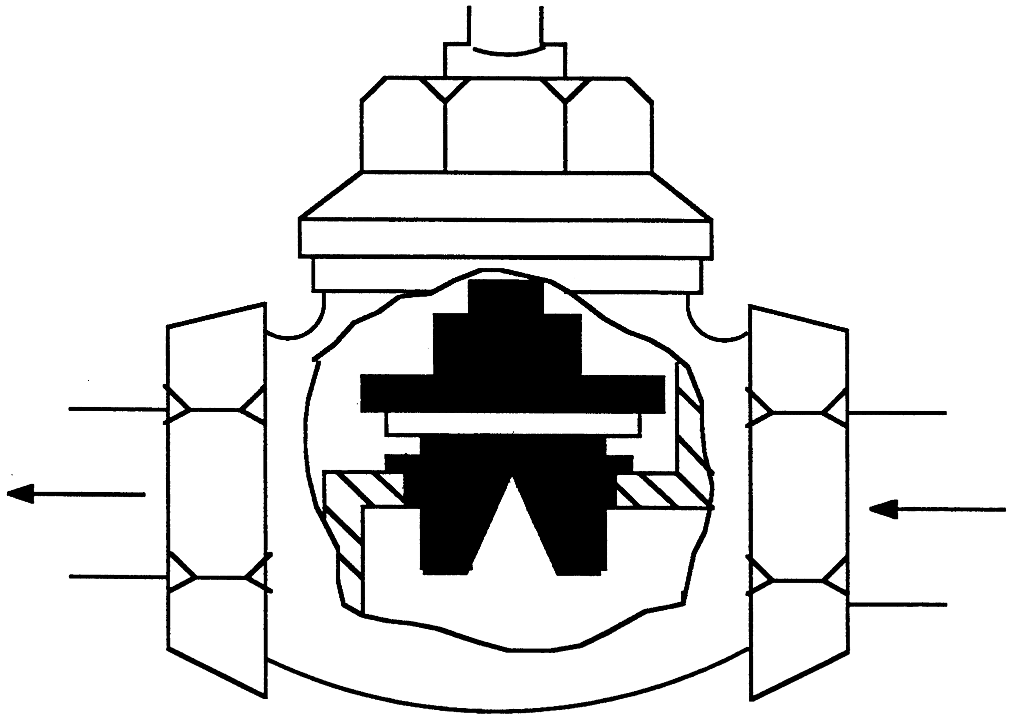
Valves

The valve is used as the means for regulating the flow of the heat transfer medium. The heat transfer medium is steam, hot or chilled water, and must be controlled in order to match the capacity of the heat transfer device to the load on the system. The proper design and sizing of these valves is essential to obtain the desired results from an air conditioning system.

Valves are classified according to the design of the valve body. They are known as two-way valves and three-way valves. When valves are identified according to the control action, they are called normally open, normally closed, and three-way mixing or diverting. Normally open refers to a valve that is constructed to remain open when no air pressure is applied to the actuator. A normally closed valve is designed to remain closed when no air pressure is applied. When these terms are used to describe a three-way valve, they refer to the ports rather than the entire valve.

In northern climates, normally open valves are commonly used on heating applications such as unit ventilators, radiators and heating coils. Normally closed valves are commonly used on humidifiers, chillers, or cooling coils. In southern climates a normally open valve may be used on cooling coils. Normally open and normally closed valves are often used in the same control scheme in sequence for a heating and cooling application.

Two-Way

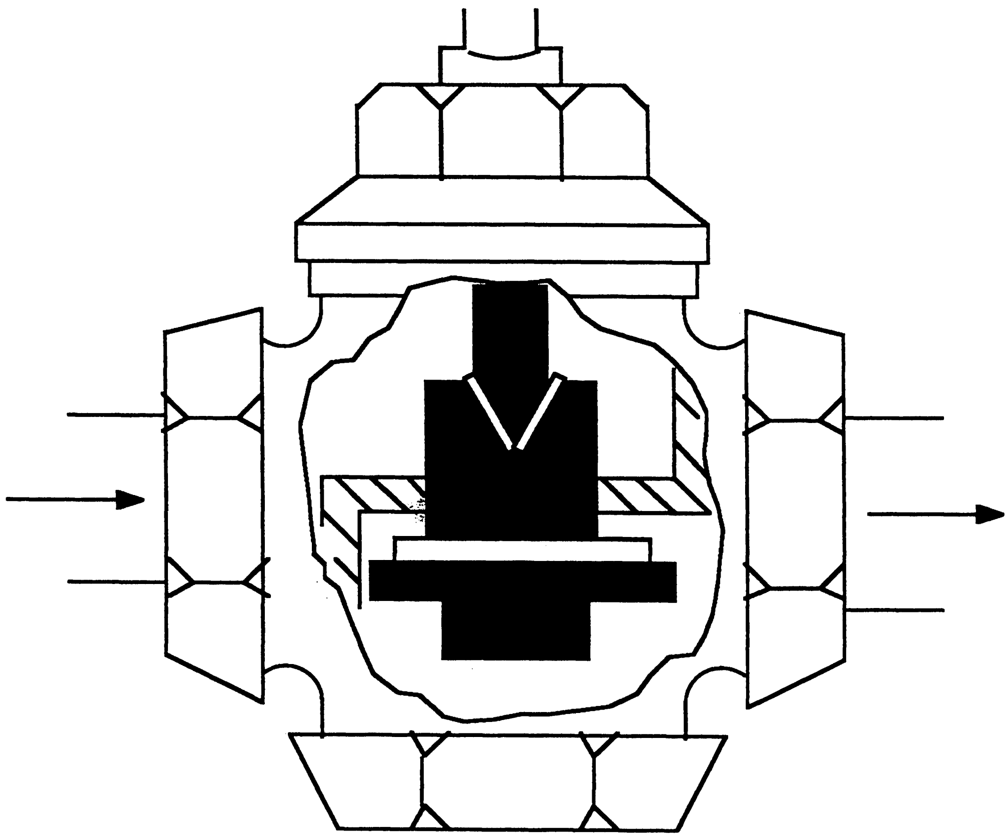


NORMALLY OPEN (N.0.)
STEM UP OPEN

FIGURE 5-5

Figure 5-5 shows a two-way normally open valve. Because there is nothing in a single seated valve to balance the force exerted by the fluid pressure against the disc, the single seated valve always requires more force to close than to open. Notice the stem must be moved down in order to close the valve.

Two-Way

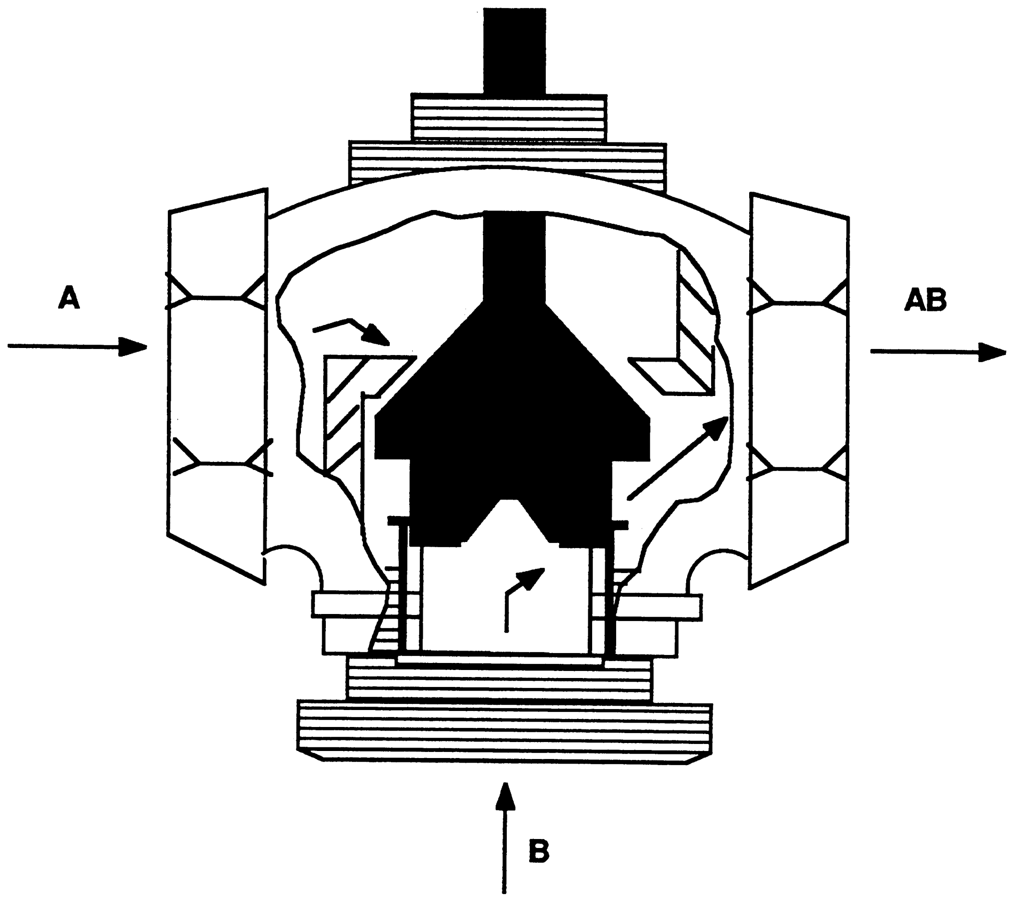


NORMALLY CLOSED (N.C.)
STEM UP CLOSED

FIGURE 5-6

Figure 5-6 shows a two-way normally closed valve. With this seat and disc arrangement, the stem must be moved up in order to close the valve. The fluid pressure exerts its force against this movement. Therefore, the greatest force is required to close against this system pressure. With both valve configurations (N.O. and N.C.), the actuator must exert the greatest force to obtain close-off. This arrangement is made so that good proportional control is accomplished in either case. If the system pressure exerts its force on the opposite side of the disc by piping the heat transfer medium in the opposite direction, disfunctional characteristics would occur, such as water hammer; or perhaps the actuator might be unable to open either type of valve against the system pressure.

Three-Way

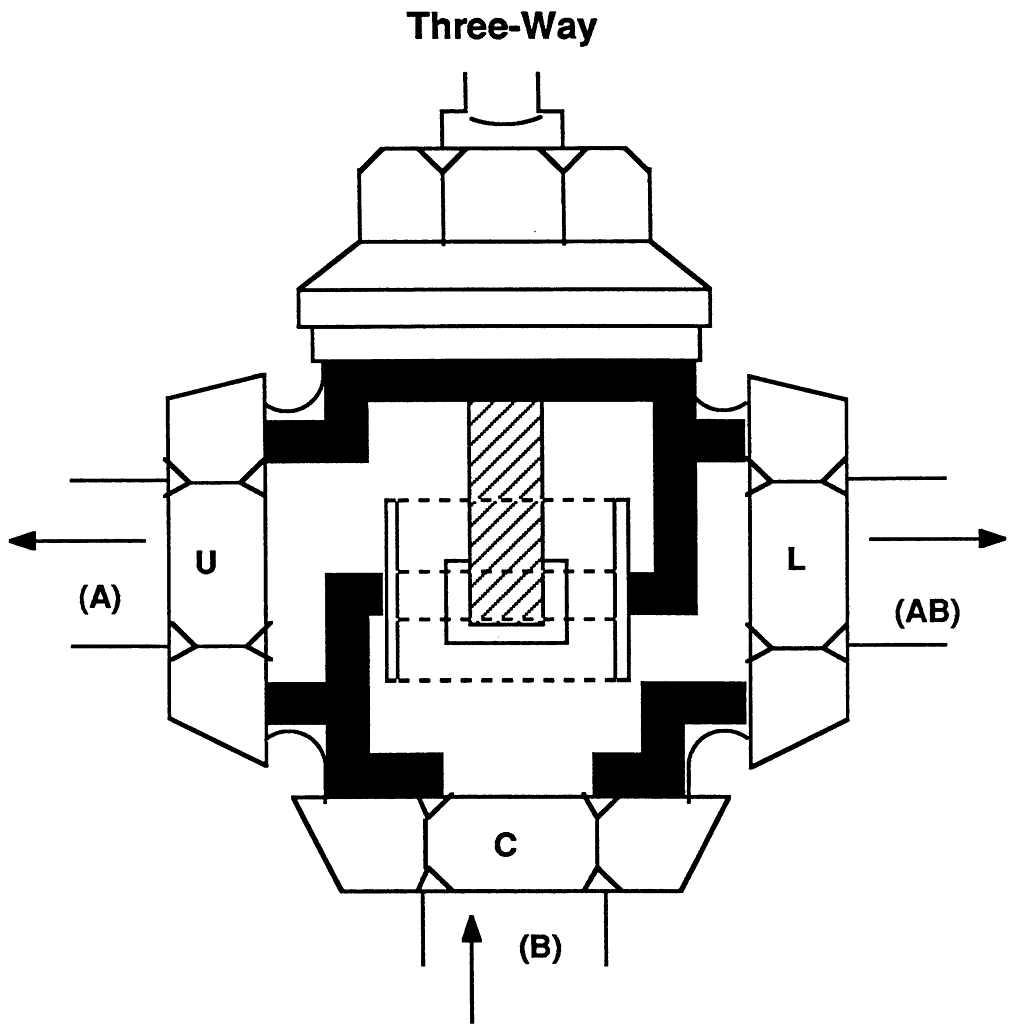


THREE-WAY MIXING

Stem Up B to AB
Stem Down A to AB
Normally Open (N.O.) B to AB

FIGURE 5-7

Figure 5-7 illustrates a three-way mixing valve. There are three openings. There are two inlets and one outlet. The proportion of fluid entering each of the two inlets can be varied by moving the valve stem. Valves designed for mixing service are not suitable for diverting service. This is because the mixing valve has only one disc and two seats. If the mixing valve was piped for diverting service, the inlet pressure would slam the disc against the seat when it nears closing. This is called water hammer and is the same disfunction caused by piping a two-way valve backwards. Improper piping causes loss of control, vibration, excessive wear and noise.



THREE-WAY DIVERTING

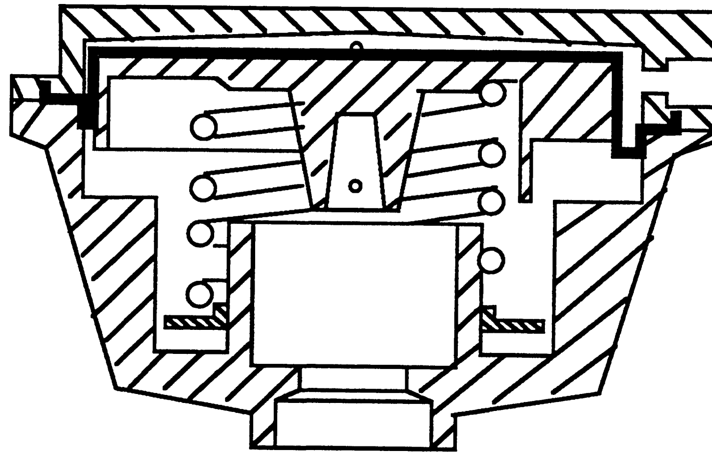
S tem Up (B) or C to (AB) or L
Stem Down (B) or C to (A) or U
Normally Open (N.O.) (B) or C to (AB) or L

FIGURE 5-8

A three-way diverting valve is shown in Figure 5-8. On some models the ports are labeled A, B and AB; and on other models the ports are labeled C, U and L. Three-way diverting valves have one inlet and two outlets. Media entering the inlet port can be diverted to either of the two outlet ports in any proportion desired by the movement of the valve stem.

Valve Actuators

Valve actuators are similar to damper actuators described earlier. Valve actuators have a shorter stroke length than damper actuators due to lesser distance the valve stem is required to move as compared to the movement necessary to open and close a damper. A valve actuator is shown in Figure 5-9.



VALVE ACTUATOR

FIGURE 5-9

The unbalanced forces on either side of the seat described earlier have an important influence on the selection of the spring range of the actuator. The force on the seat that is developed by the heat transfer medium must be overcome by the actuator through the application of additional signal pressure above the top end of the signal range for normally open valves, or by reducing the signal pressure below the bottom end of the range for normally closed valves. Normally open valves used on heating applications usually have lower spring ranges, such as 3-8 psig. When 0-3 psig is applied to the actuator, the valve is fully open. At 3 psig the normally open valve begins to close. When the pressure reaches 8 psig, the valve is fully closed. Assume that the available pressure from a controller is 15 psig (main air pressure). This gives an additional 7 psig to close the valve against the system pressure and hold it closed. The span of the spring range is actually increased by the amount of pressure drop across the valve. If the pressure drop across the valve exceeds the close off rating, the valve may not close properly.

A higher spring range is commonly used in conjunction with normally closed valves. For example, an 8-13 psig spring range provides the additional pressure needed below 8 psig to insure proper valve operation. At 13 psig the normally closed valve begins to close. When the pressure has decreased to 8 psig, the valve is fully closed. The pressure from the controller may decrease all the way to 0 psig, if necessary, to allow the full force of the spring to close the valve against the system pressure and hold it closed.

There are spring ranges available other than the two ranges described above. The 5-10 spring range is generally used with three-way valves because the 5-10 spring provides nearly equal force in either direction and, therefore, is able to provide adequate force to provide close off at either port.

In sequencing applications, a 3-7 psig spring is used on the normally open heating valve, and the 8-13 spring is used on the normally closed cooling valve. The deadband between 7 and 8 psig is incorporated into the design to prevent both valves from opening at the same time, eliminating the possibility of simultaneous heating and cooling.

There is a start point adjustment on most valve actuators which enables the technician to change the spring range. Should the ranges be too close or overlap, a slight adjustment can be made to provide proper operation. Also a positive positioning relay can be mounted on the valve actuator. This makes it possible to have the valve operate over a different range than its spring would normally allow, and enables the valve to move exactly as far, in either direction, as a controller demands.

Steam Valves

The steam valve should offer more resistance to the flow of steam than any other element in the system in order to effectively control flow. Therefore, the selection of valve pressure drop should be the first step in selecting pressures to be used in the system.

Things to consider for selecting a steam valve pressure drop are, as follows:

- The valve pressure drop in a proportional system should be at least 80% of the difference between the supply and return main pressure. The following exceptions apply:

  a) An 80% drop exceeds 42% of the absolute inlet pressure. In this case, the 42% of the absolute inlet pressure should be used as the valve pressure drop. This 42% of the absolute inlet pressure is called the critical pressure drop.

  b) In special circumstances, an 80% drop results in too low of a steam pressure in the heat transfer device. Example: Pressures are being selected for a building to be equipped with standing radiation. The radiators are selected to produce the designed quantity of heat when filled with steam at 1 psig. Atmospheric returns are to be used. Because the valve should have 80% pressure drop to give effective control, a supply main pressure should be selected which allows the 1 psig, in the radiator, to become 20% of the difference between supply and return main pressure. 5 psig is selected as the supply main pressure. **80% x 5 psig = 4 psig pressure drop**. This allows an 80% drop through the valve and allows 1 psig of steam to be in the radiator when the valve is wide open.

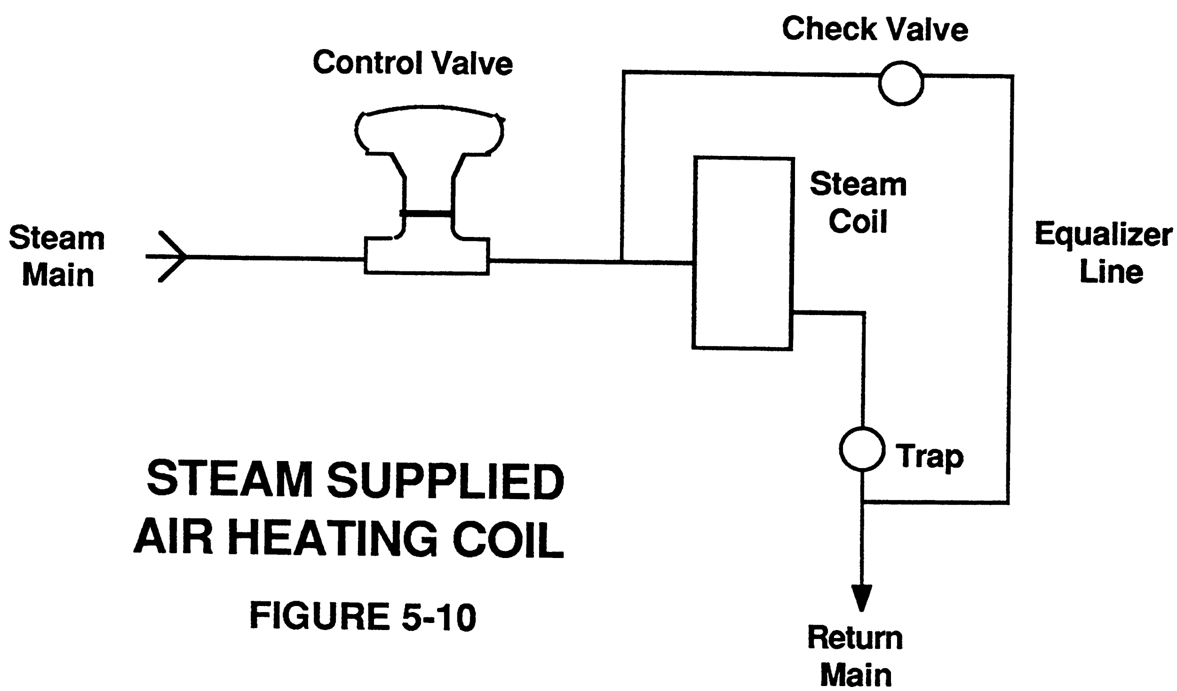
It should be noted that in most cases, the loss of heat output necessary to utilize an 80% pressure drop at the valve is quite small. For example: A radiator supplied from a 5 psig boiler through a valve having a 1 psig (20%) pressure drop produces 100,000 BTU/hour. The radiator is carrying a pressure of 4 psig of steam. The same radiator equipped with a smaller valve having a 4 psig (80%) pressure drop produces 92,000 BTU/hour which is a decrease in heat output of only 8%.

Because of this, it is possible to use fairly low supply main pressures and still utilize 80% of the supply to return main pressure difference for the control valve pressure drop without jeopardizing heat output of most steam supplied devices.

As previously stated, the supply main pressure should be sufficient to allow an 80% pressure drop through the control valve and still leave enough steam pressure downstream from the valve to produce the desired heat output. The supply main pressure should be held constant if possible. If the boiler pressure is not constant, a pressure regulator should be installed ahead of all steam supplied devices where output temperatures may vary rapidly with fluctuations of steam pressure.

Return main pressure should also remain constant, if possible, because a variation in return line pressure will cause fluctuation in steam flow through control valves, even though the valves do not change position. This means that from the standpoint of control, atmospheric returns with a condensate pump are superior to vacuum returns with a vacuum pump that can cycle over a range of several inches of vacuum.

<u>Steam Coils</u>



STEAM SUPPLIED AIR HEATING COIL

FIGURE 5-10

Figure 5-10 illustrates a steam supplied air heating coil. The illustration shows the most favorable design conditions and piping arrangements that lend itself to good control valve performance. General rules for this requirement are, as follows:

1. Main pressure should be held at design pressure ±1 psig.

2. Returns should be at atmospheric pressure unless lifts are required in the returns.

3. Traps should be sized to pass design condensate flow at 1 psig drop.

4. An equalizer line prevents formation of vacuum within the coil.

5 Control valve pressure drop should be 80% of the difference between supply and return main pressures.

The effect of fluctuating supply and return main pressures can be significant. For Example: assume a system has a boiler cycled that shuts off at 6 psig and cuts in at 2 psig. In the same system, assume that a vacuum pump is used which cuts in at 4 inches and cuts off at 8 inches of vacuum. The pressure difference between the supply and return mains can vary from a minimum of 4 psig to a maximum of 10 psig as the boiler and vacuum pump go through their cycles. This means that a 50% variation in capacity of the control valves in the building is possible as the pressure fluctuates. Control valves that have been sized correctly for the 4 psig pressure drop, will be 50% too large during periods when the 10 psig pressure drop existed across the supply and return mains.

An exception to this is the high vacuum systems. Their purpose is to lower the steam temperature and pressure as the heating load decreases. Vacuum systems are generally adaptable for the use of control valves. The usual practice is to maintain a control difference between supply and return main pressures, while varying supply main pressure with heating load.

A water line above the trap is important to insure clearing of condensate from the bottom rows of the coil. The equalizer line is important to both atmospheric and vacuum return systems. It is added protection so that a vacuum forming in the coil will not keep the water from draining into the trap.

Steam to Water Converters

Selection of a converter is very important and it is much better to oversize than undersize. Many converters are of the flash type. They have a small volume of water compared to the volume of steam where the leaving water temperature may be fairly close to the steam temperature. Therefore, the selection of supply pressures and valve pressure drop is very important.

Converters may be required to deliver hot water at varying degrees of temperature, such as water being used for space heating. This generally requires the water to be reset according to outdoor temperature. As the difference between the delivered water temperature and steam temperature becomes less, the size and cost of a converter usually becomes more. For example: A converter supplying water to a building at 120°F is only one-third the size of a converter supplying the same building with 200°F water. Unlike steam coils for air, the capacity of a converter varies drastically with changes in steam pressure delivered to the converter. For example: Consider a converter supplying 180°F water to a building. The converter is designed to operate on 10 psig saturated steam. If the pressure in the converter is reduced to 5 psig, a 17% reduction in capacity will result. If the steam pressure is reduced to 2 psig, the converter will be operating at 70% of design capacity. Considering this fact, the steam supply main to a hot water converter should be specified and selected. The control valves should be sized so that full design steam pressure can be obtained within the converter, yet allow sufficient valve pressure drop to give desired control. It has been stated that 80% is the best desired pressure drop, but a compromise of 42% pressure drop of the absolute inlet pressure may be used if available steam main pressure prevents an 80% pressure drop from being used.

For example: If a maximum of 10 psig of steam is available to a converter with an atmospheric return, an 80% valve pressure drop will leave 2 psig of steam in the converter. At this pressure, a larger converter is required. Under these circumstances, a compromise could be made by using a control valve with a 42% pressure drop of the absolute inlet pressure.

Steam Humidifiers

Humidifiers may be of the water spray, steam grid, or steam pan type. Of these three, the steam grid is the most efficient to control. Because steam is readily diffused without adding heat to the air, the air receives very little sensible heat.

Pan humidifiers normally require 5 psig steam because the water must boil in order to evaporate at a high enough rate. Therefore, the valve pressure drop must not be greater than steam main pressure, less 5 psig. Spray and steam grid humidifiers require only 0.5 psig to 1 psig respectively to force the steam through the jets or nozzles. Therefore, the valve pressure drop is determined by subtracting the humidifier inlet pressure from the supply main pressure.

Water Valves

Water valve selection and water system design must be coordinated to accomplish the best proportional control of a water system. In original design, all control valve locations must be considered so that the system will deliver design flow at full load and not generate uncontrollable conditions at minimum load. Valve selection based on pressure differentials at control valve locations depends on valve sizing at full load conditions and valve controllability and close off at minimum load condition.

Three factors affect control valve pressure differentials:

1. Flow variations.

2. Type and size of piping distribution system.

3. Pump characteristics and regulation of supply pressure differential.

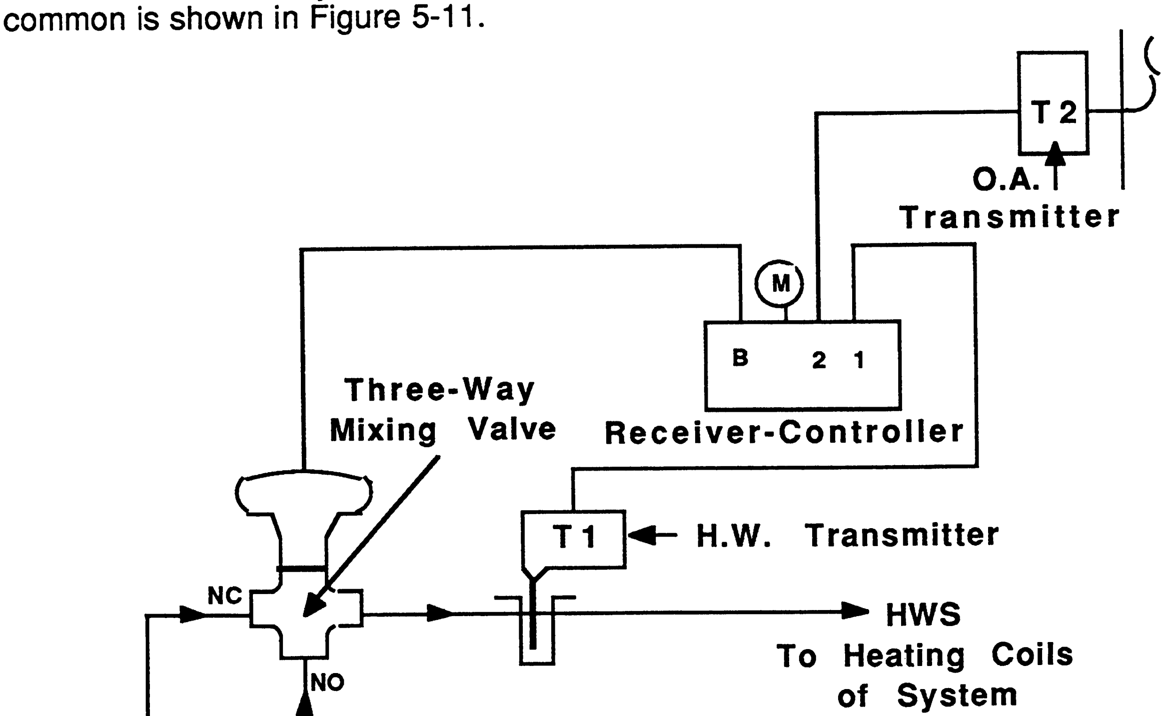
There are three methods of providing control of a water system:

1. Supply water temperature control is the most effective method of controlling BTU output of a water supply.

2. Flow control is suitable for control of individual terminal units, such as fan coils and induction units.

3. A combination of temperature and flow control of supply water is the best control method for heating systems.

There are several ways to vary supply water temperature to coils. One of the most common is shown in Figure 5-11.



RESET SCHEDULE

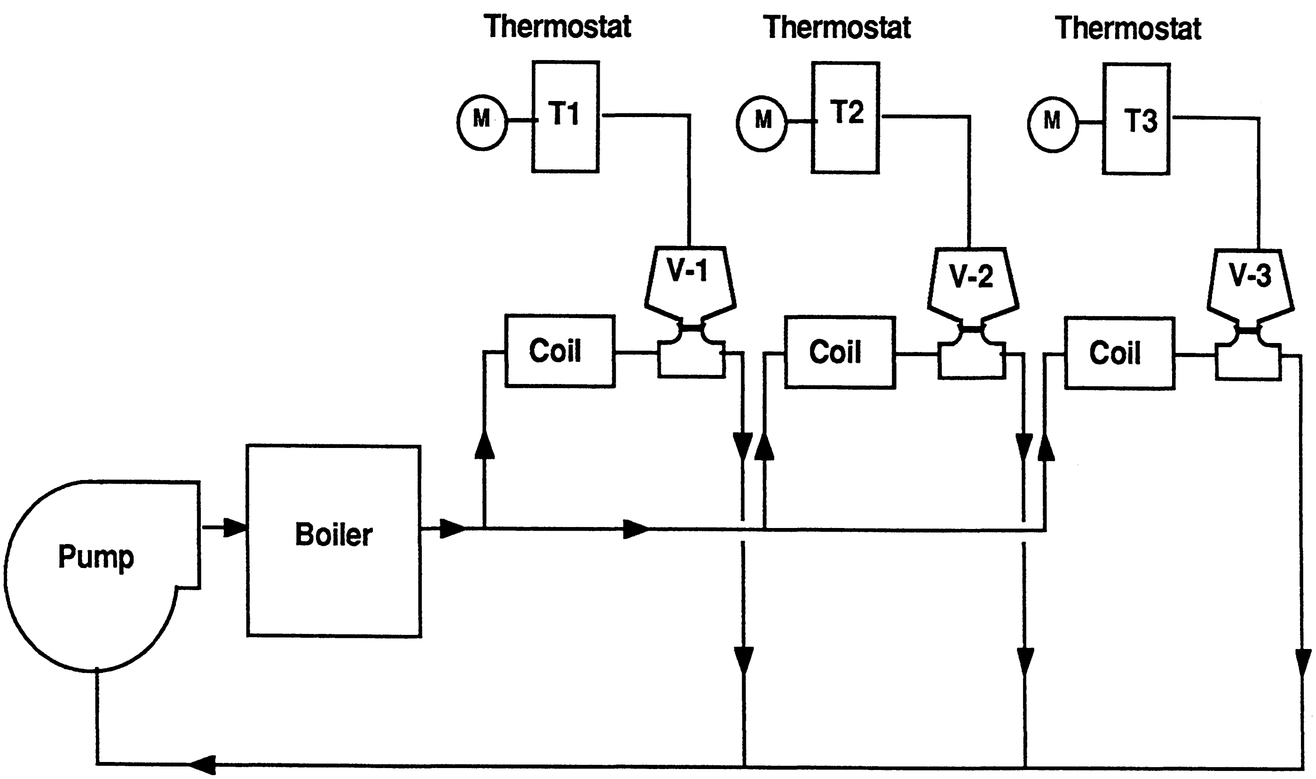
T1 - H.W.	T2 - O.A.
190 Degrees	-10 Degrees
100 Degrees	65 Degrees

SUPPLY WATER TEMPERATURE CONTROL

FIGURE 5-11

In this system, transmitter T2 measures outside air temperature. Transmitter T1 measures supply water temperature and maintains a preselected schedule of water temperature in accordance with the outside air temperature. Also shown is a reset schedule which is predetermined in accordance with design conditions and system requirements.

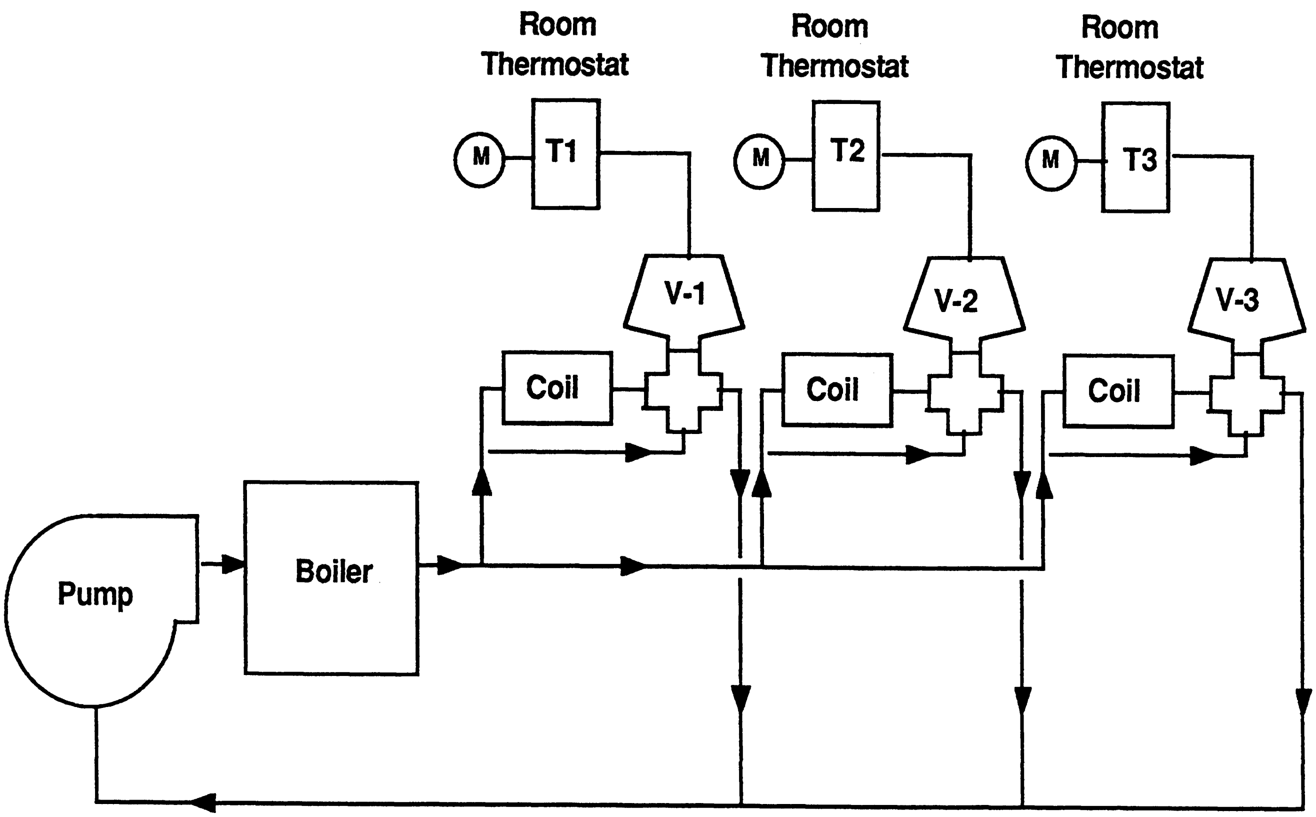
The second method of controlling a water system is flow control. Using this type of control, the BTU output of supply water does not vary directly in proportion to flow. Flow control is the simplest method to use for providing individual control of terminal units being fed from a common heat source. Control can be accomplished with either two-way valves or three-way valves.



FLOW CONTROL USING TWO-WAY VALVES

FIGURE 5-12

When two-way valves are used, as shown in Figure 5-12, for control of individual units, the flow through the supply system is varied, causing pressure variations dependent upon the type and size of the distribution system.

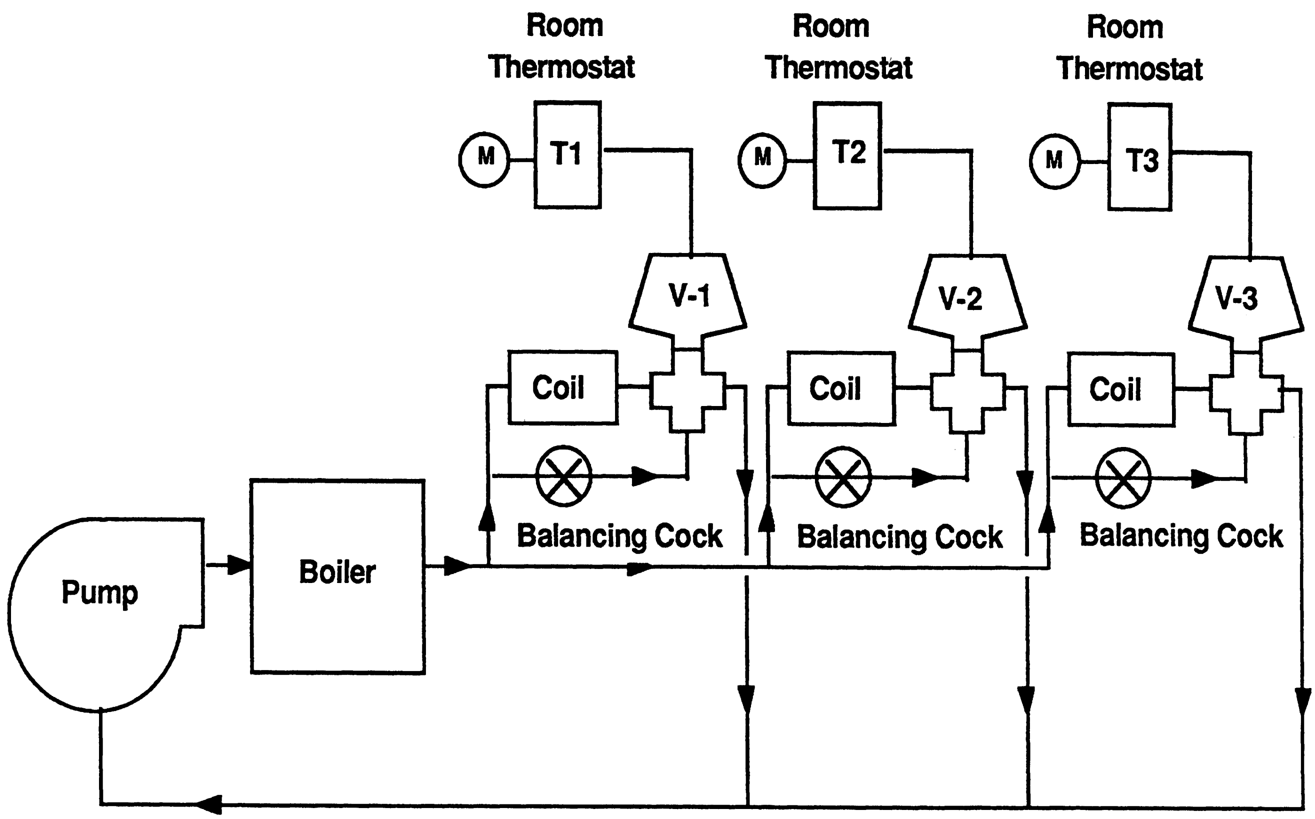


COIL BYPASS USING THREE-WAY VALVES

FIGURE 5-13

Three-way valves are often used to control flow to coils, particularly when it is desired to maintain a constant flow through the mains. Figure 5-13 illustrates this application. As the valve reduces the flow through the coil, it increases the flow through the bypass around the coil. Therefore, the total flow through the system remains relatively constant and the system friction loss remains relatively constant.

Further examination shows that the total GPM circulated by the pump does not remain completely constant. At maximum demand, all water flows through the coil and there is a friction loss through the complete circuit. When the valve is in the other extreme position, all water flows through the bypass circuit. Because the bypass usually has less resistance than the circuit through the coil, there is an increase in flow. If the resistance through the bypass is one-half of that through the coil, the flow will increase. If this situation occurs at even a few coils throughout a system it can cause a seriously unbalanced circuit. If it occurs simultaneously at a large number of coils, the effect can be appreciable.



COIL BYPASS WITH BALANCING COCK

FIGURE 5-14

The bypass circuit should be sized for the same resistance as the circuit through the coil. If this is not done, then a balancing cock should be installed in the bypass circuit. This is shown in Figure 5-14.

Even with the circuits properly balanced for these extreme conditions, the intermediate positions of the valve can have a significant effect on the overall circuit resistance. When one-half the flow is through the coil and one-half through the bypass, the total friction loss through the circuit is considerably less than when all the water is going through one circuit. This has the same effect on the system when the valve is in one extreme position or the other.

Even with the use of three-way valves and a properly designed and balanced circuit, it is not possible to maintain complete system balance under all conditions of operation. The use of three-way valves does not cure all of the problems in the system. It is still necessary to carefully design the system and consider all variables that may affect its operation. The following items should be considered when using a three-way coil bypass valve:

1. As far as the coil output is concerned, a three-way valve controls output by varying flow the same as a two-way valve.

2. Piping costs can be higher for three-way than for two-way valves, especially where piping space is limited.

3. Most three-way valves are available only with linear flow characteristics. Equal percentage two-way valves are better suited to flow control of water supplied coils where close control is desired. However, this may be compensated to some degree by resetting the water temperature from outdoor temperature.

4. Constant flow in mains is possible in a system using three-way valves without the use of added bypass valves.

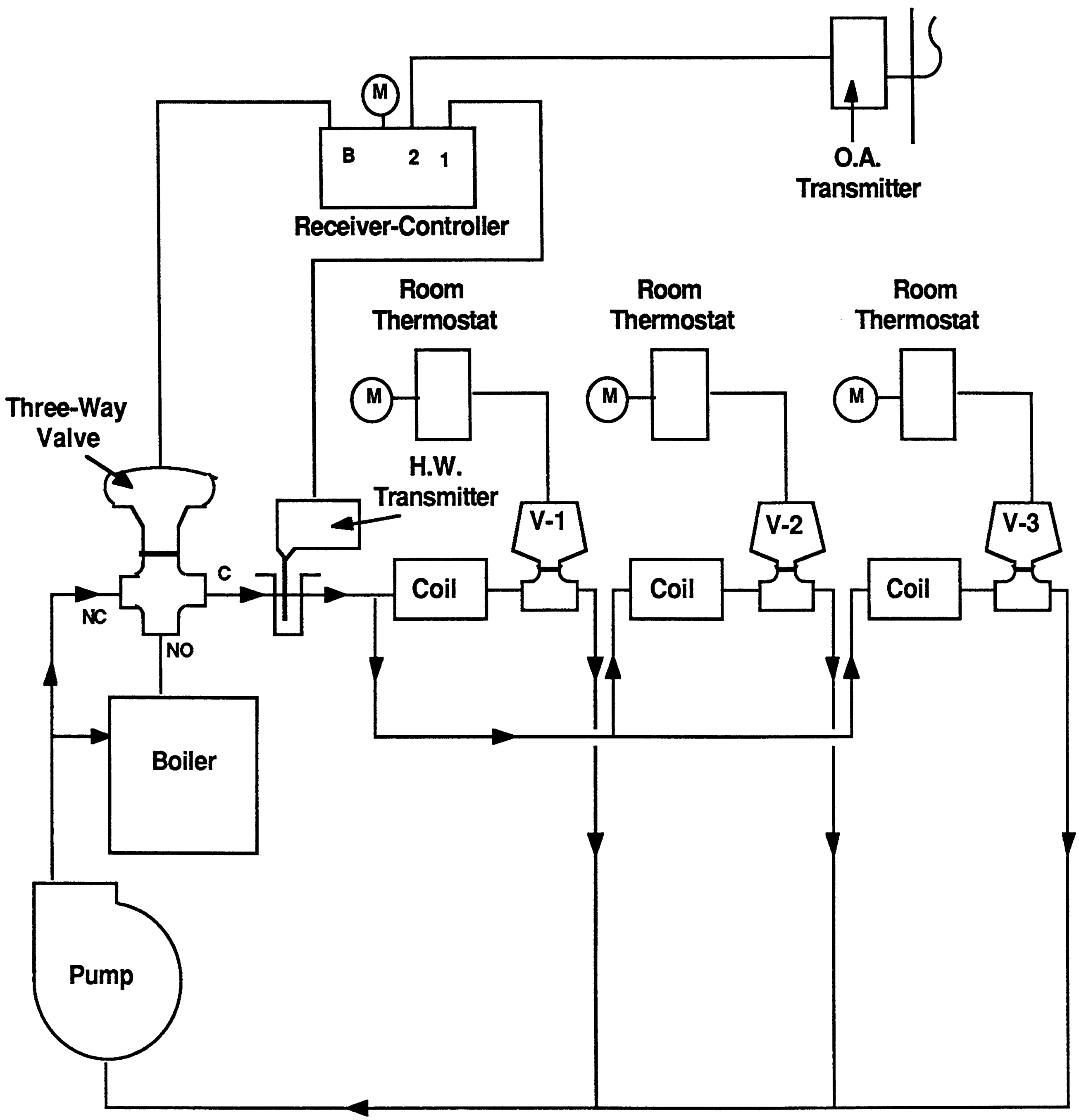
The third method of water system control is to control both temperature and flow of supply water. In large applications, having individual room control, it is not practical to individually control supply water temperature to each heating unit because of piping requirements. The best method is to control flow to each heating unit by using a valve controlled by a thermostat. By adding a supply water temperature control to the system, individual room control can be improved.

Some of the advantages to this method are, as follows:

1. During mild weather, supply water temperature is reduced causing individual heating limit valves to open wider than they would with a constant temperature of water. This keeps flow through the pump fairly uniform and reduces wear on the valves during this season.

2. During initial warm-up of a room or during heavy load changes, overshooting of temperature is not likely to occur because maximum heating unit output is regulated from the outdoor temperature.

3. With average conditions, heating unit valves will be working near midstroke rather than operating as two-position between closed and nearly closed positions. A typical piping arrangement for temperature and flow control of supply water is shown in Figure 5-15.

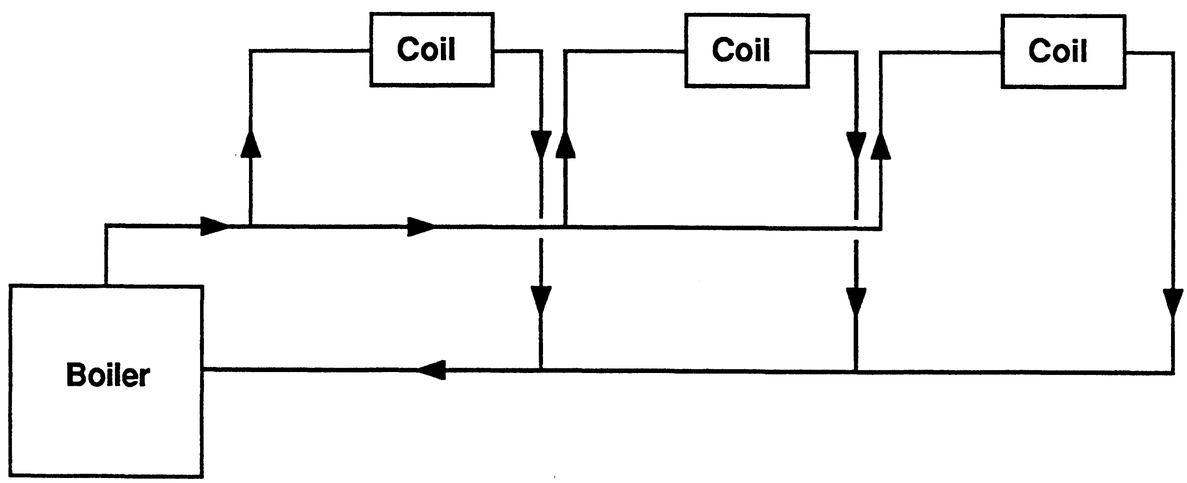


TEMPERATURE AND FLOW CONTROL OF SUPPLY WATER

FIGURE 5-15

Water Distribution Systems

The main function of a distribution system is to deliver the design flow and temperature of water to all parts of the system at full load without creating uncontrollable conditions at minimum load. The size of the distribution system, in terms of pressure drop through the entire system, determines the maximum pressure build-up at individual units under minimum load conditions. The type of distribution system will determine whether or not pressure differential can be regulated as loads change.



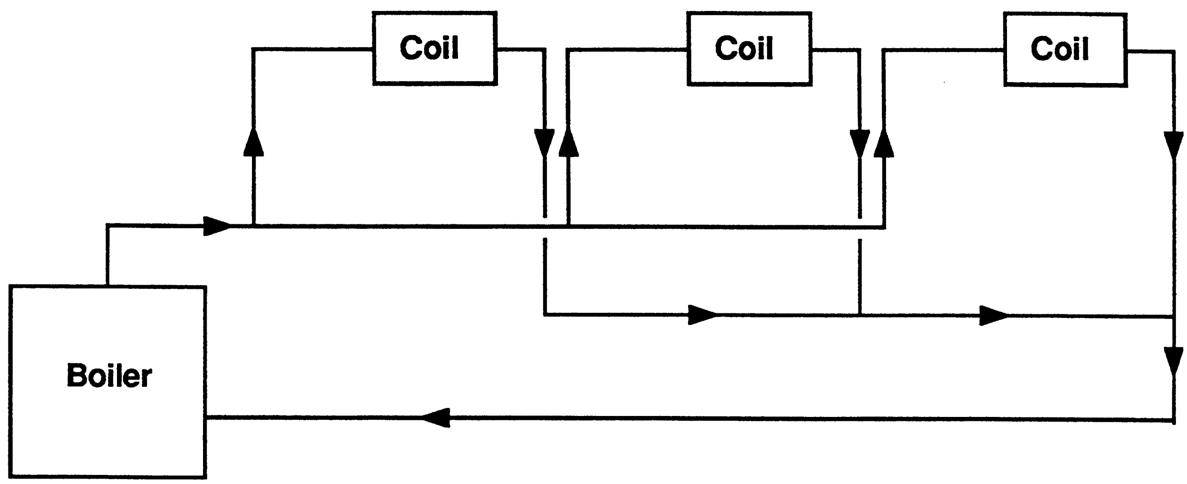
TWO-PIPE DIRECT RETURN SYSTEM

FIGURE 5-16

The two most common distribution systems are direct return and reverse return. Figure 5-16 is an example of a simple direct return system. As illustrated, the first heating load taken off the main is the first to be returned, and the last heating load taken off is the last returned. This means the length of the pipe circuits are unequal and are unbalanced.

In this type of system, the pump must be sized to overcome the friction loss of the longest water circuit. This means its pressure head will be too large for the first part of the system. Thus, an excessive flow of water through this part of the system will result, causing an uneven distribution of heat and possible noisy operation. Because the pump can deliver only a given flow against a given friction head pressure, the last part of the circuit will receive very little or no water and a corresponding amount of heat.

The direct return system can be balanced by placing balancing cocks into the circuit for each unit. However, the system remains in balance only as long as the flow remains constant. On a system with a large number of units, the adjustment of these balancing devices are time consuming and expensive. For this reason, the direct return distribution system is recommended only for small constant flow systems.



TWO-PIPE REVERSE RETURN SYSTEM

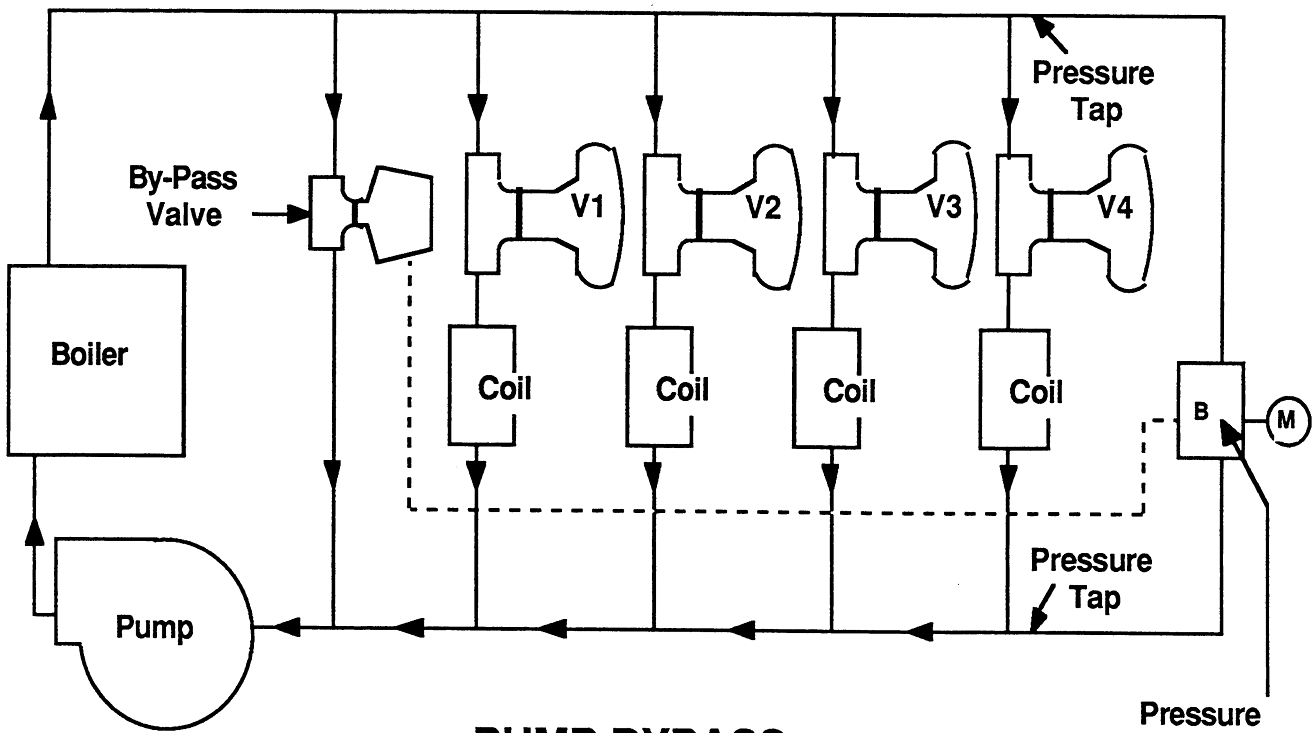
FIGURE 5-17

In a reverse return system shown in Figure 5-17, the first load taken off of the main is the last to be returned and all circuits are approximately the same length. This greatly simplifies the problem of establishing balance.

The reverse return system is the best arrangement for maintaining system balance and relatively constant pressure drop across the valves. However, even with this system, unless other provisions are incorporated, there can be a relatively large increase in the pressure drop across the control valves as the flow through the heating units is modulated from maximum to zero.

Control of Supply Pressure Differential

Decreases in flow, caused by the throttling of two-way valves, not only decrease piping pressure loss, but also increase the pressure differential generated by a typical pump. When valve pressure drop at minimum flow is more than three times the valve pressure drop at maximum flow, then a means of differential pressure control should be used.

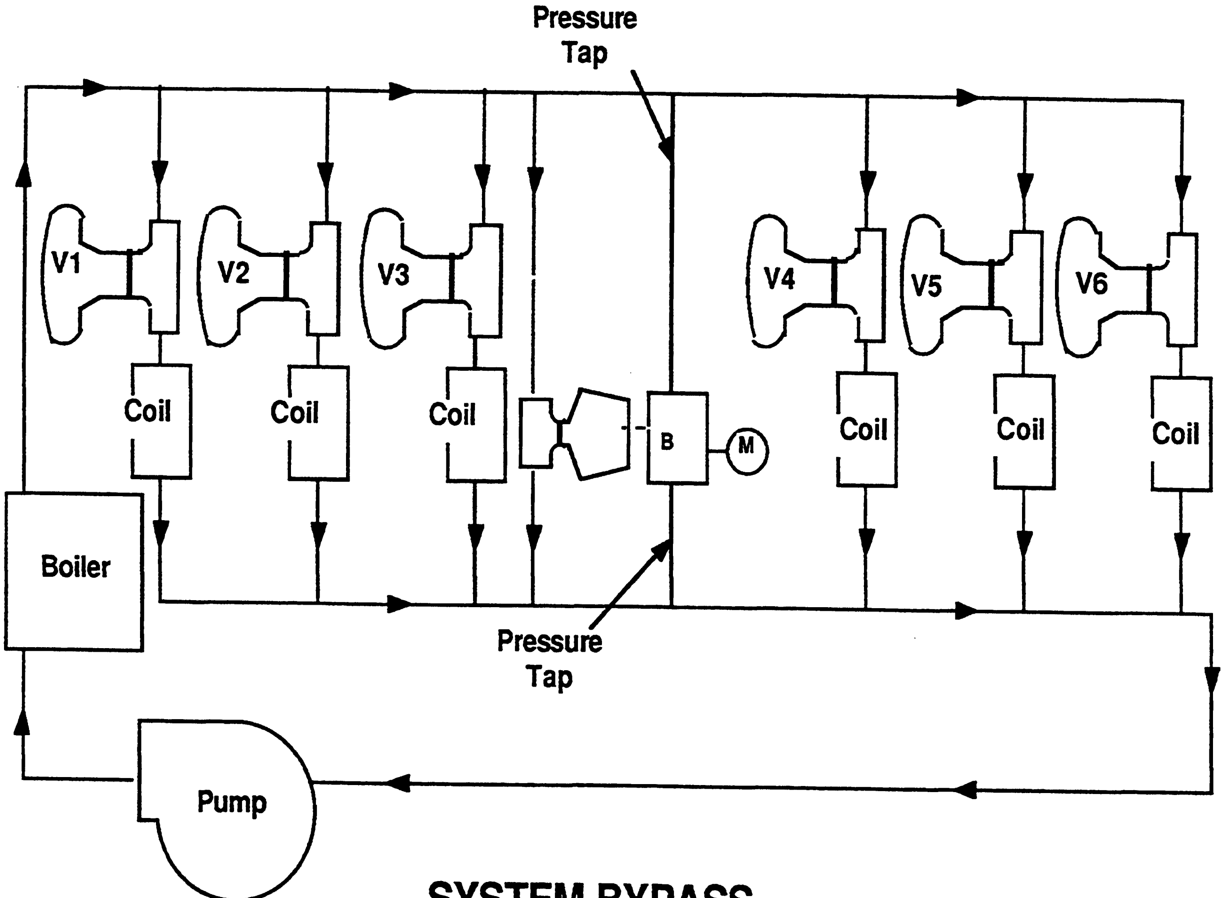


PUMP BYPASS

FIGURE 5-18

Many applications specify a pump bypass as shown in Figure 5-18 to maintain a constant pressure across the system. As the control valves on the individual heating units decrease the flow through the system during low heat demand, pressure across supply and return mains increase. This increase in pressure is too high to allow accurate control by the unit valves.

By adding a pump bypass valve and using a differential pressure controller to monitor pressure on either side of the pump and position the pump bypass valve, more accurate control can be obtained. This type of differential pressure control can offset any gain in pressure due to variations of pump head. However, it cannot compensate for pressure buildup because of design pipe loss. For this reason, the pump bypass is normally used where design piping drop is less than two times design unit valve drop.



SYSTEM BYPASS

FIGURE 5-19

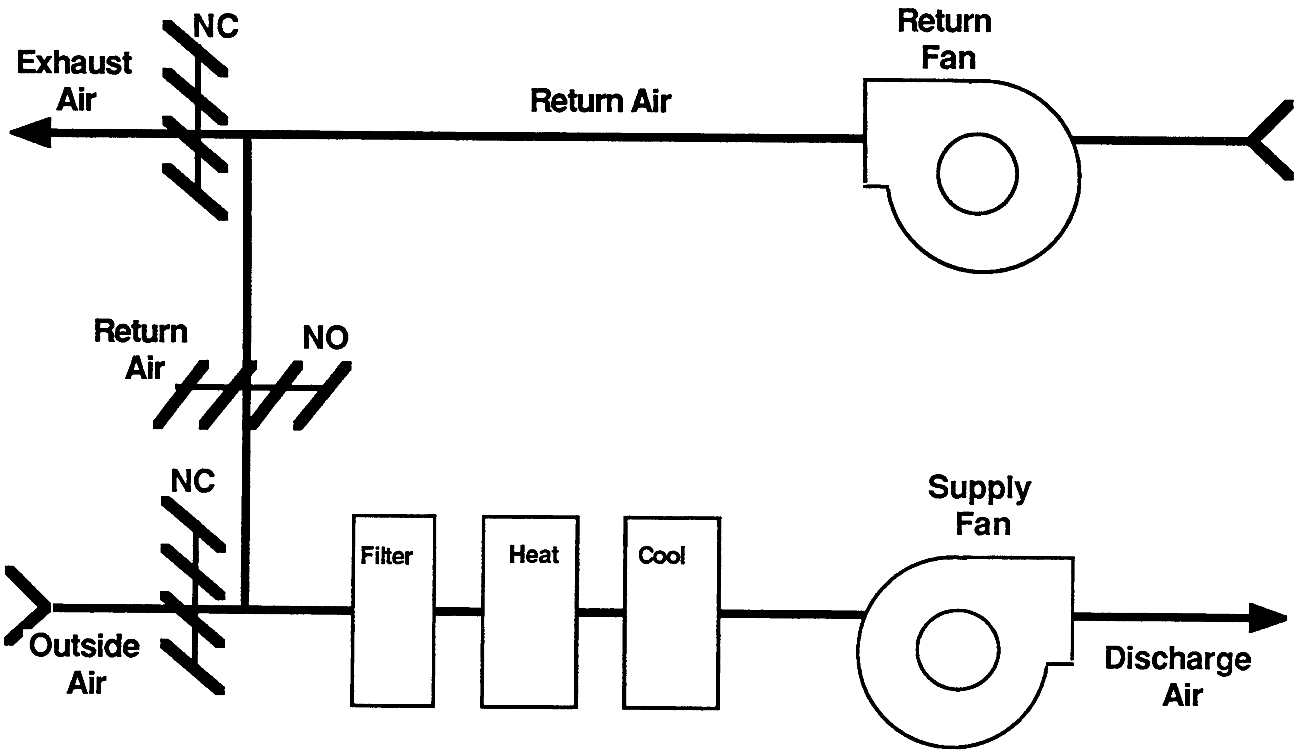
Another method of regulating system differential pressure is by using a system bypass valve as shown in Figure 5-19. By placing a bypass valve between the supply and return mains to keep water flowing through the mains at a constant rate, the frictional losses can be held relatively constant. The bypass valve should open when the system demand decreases and the unit control valves close decreasing the flow through the units. A differential pressure controller is used to monitor the head across supply and return mains. The controller modulates this system bypass valve to maintain a constant flow.

- CHAPTER 6 -

CHAPTER 6

TABLE OF CONTENTS

Basic Air Handling Unit



BASIC AIR HANDLING UNIT

FIGURE 6-1

A basic air handling unit consists of the various components shown in Figure 6-1. To follow the air pattern, start with the outside air intake to the building. Outside air is normally required, in some minimum quantity, to meet building codes during occupancy periods. The air passes through a normally closed outside air damper and the return air passes through a normally open damper to form the mixed air supply.

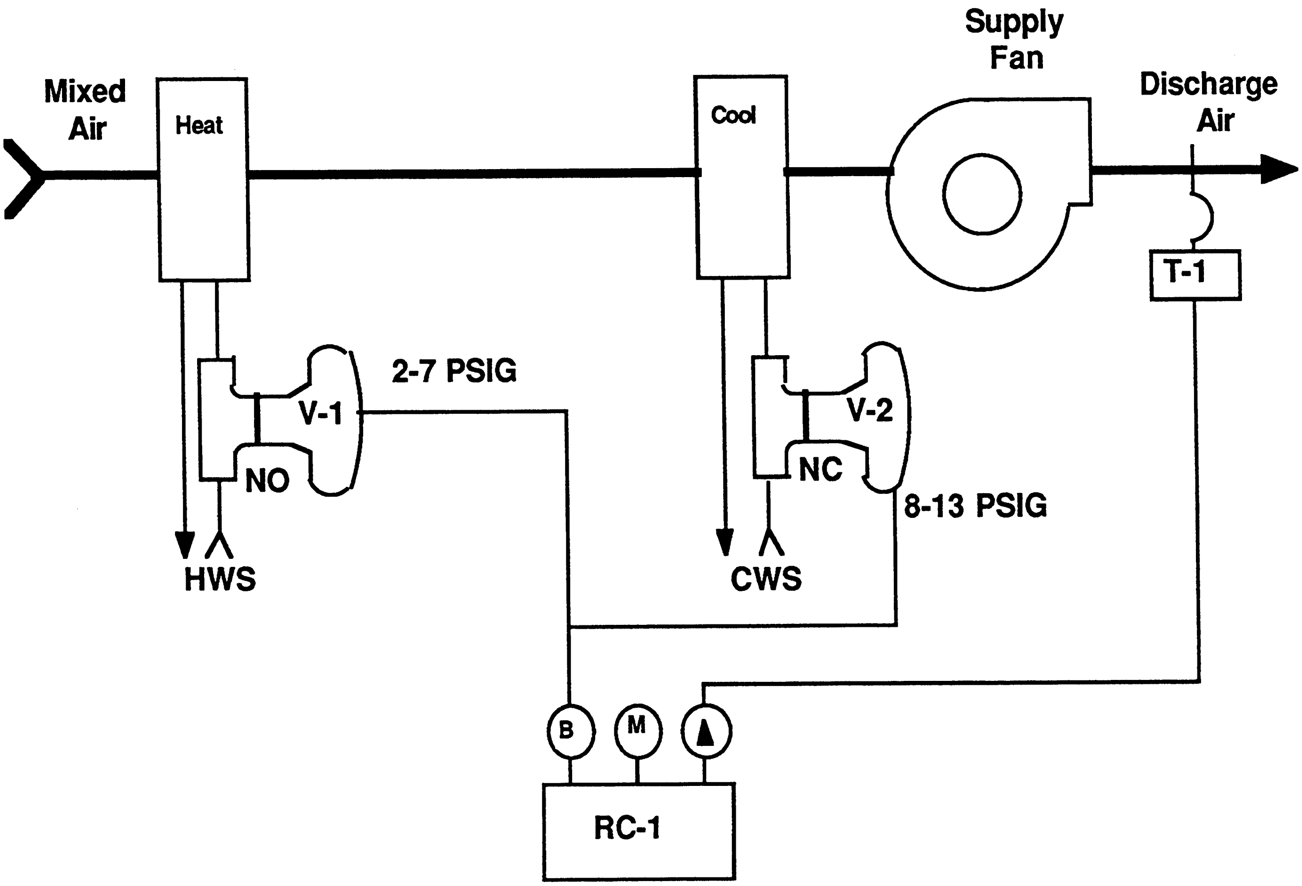
The mixed air is then conditioned through any required filter and passes through the heating and/or cooling coils. The discharge air from the supply fan feeds all parts of the system.

If the system includes a return air fan, air from the controlled spaces may be exhausted to the outside. In other instances, some, or all of the return air may be routed through the return air damper. The direction of return air flow is determined by the mixed air controls which position the outside, return and exhaust air dampers.

The amount of air exhausted from a building is usually less than the outside air drawn in because of the desire to maintain a slightly positive pressure within the building. Consideration is given to the normal exfiltration of the building through doors, windows and cracks.

A slightly positive pressure is desirable inside the building to help prevent drafts, heat gain or loss, and the infiltration of dirt, dust and, to a limited extent, humidity from outside.

Discharge Air Control



DISCHARGE AIR CONTROL

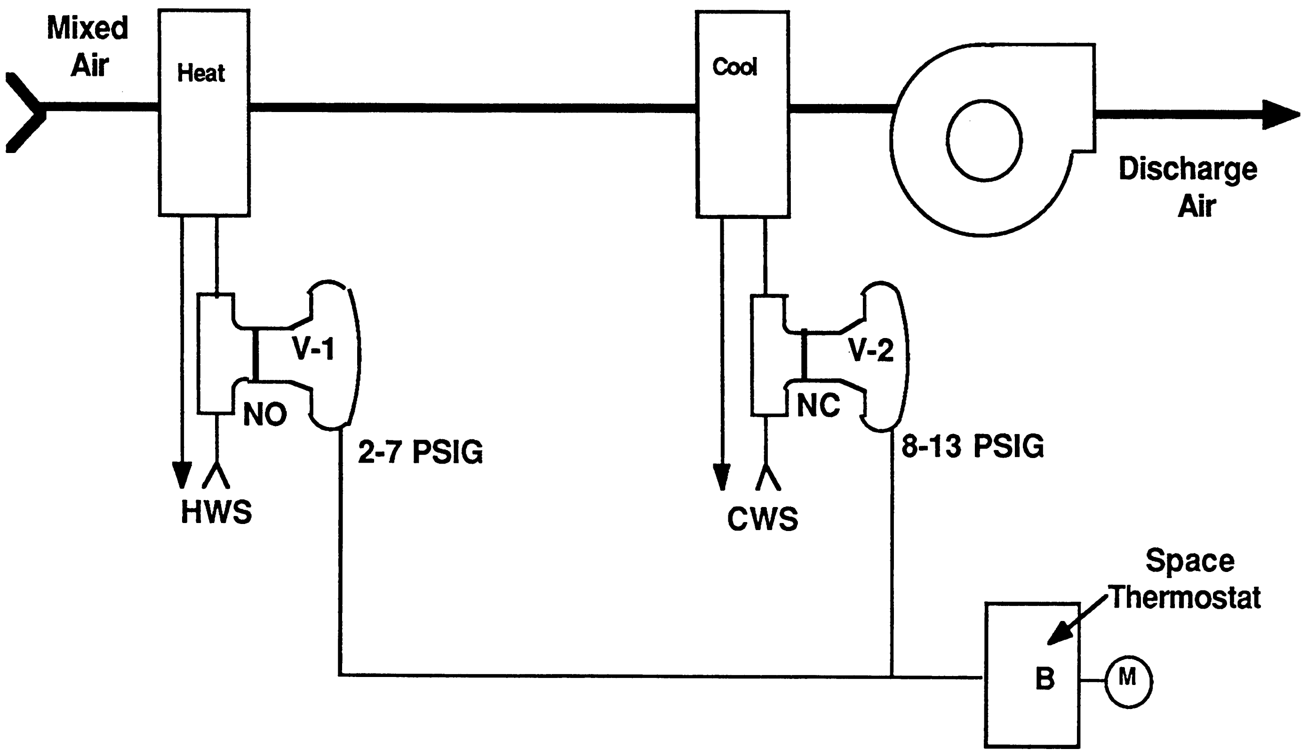
FIGURE 6-2

Figure 6-2 is an example of a discharge air controller with a temperature transmitter sensing the fan discharge temperature. The controller modulates the preheat and cooling coil valves in order to maintain the required discharge temperature.

In this example, a 0-100^{o}F range transmitter (T-1) is sensing discharge air temperature. A direct acting receiver controller (RC-1) positions the normally open hot water valve (V-1) and normally closed chilled water valve (V-2). As the discharge air temperature rises toward setpoint, the branch pressure of the receiver controller increases, which modulates the normally open heating valve toward the closed position. When the temperature rises above setpoint, the normally closed cooling valve is modulated toward the open position to provide more cooling capacity.

The proper selection of spring ranges and adjustment of start points is necessary to provide a dead band between the heating and cooling functions which prohibits simultaneous heating and cooling.

Single Zone System



SINGLE ZONE SPACE CONTROL

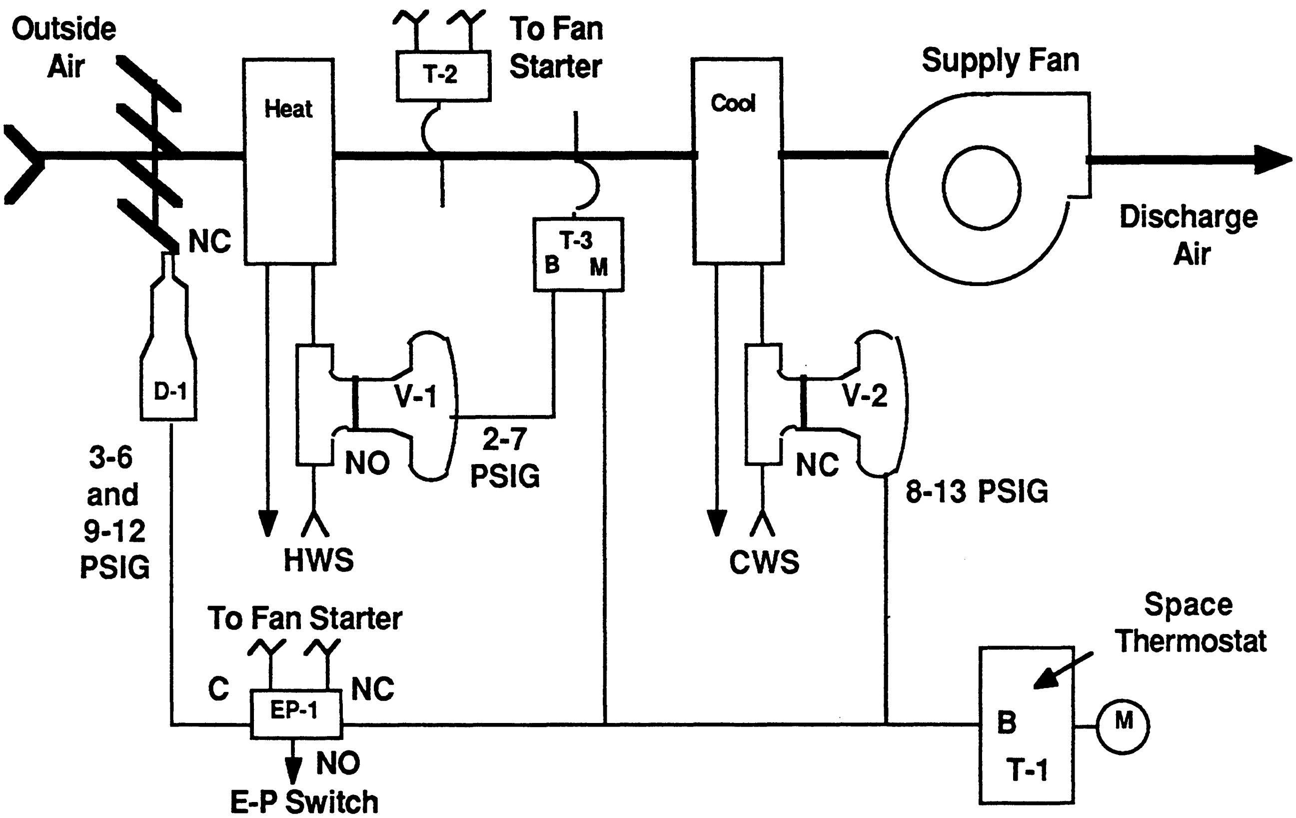
FIGURE 6-3

This system shown in Figure 6-3 provides simple control of space temperature. The temperature is controlled by the direct acting room thermostat which controls a normally open hot water supply valve (V-1) and a normally closed chilled water supply valve (V-2) in response to changes in room temperature.

In operation, the branch line from the thermostat provides a signal to the hot and chilled water valves. When the thermostat requires an increase in room temperature, the branch line pressure decreases and the hot water valve modulates toward the open position. As the temperature rises to the setpoint, the branch line pressure increases to close the hot water valve which has a spring range of 2-7 psig. Should the space temperature continue to rise, the direct acting thermostat will continue to increase the branch line pressure. If the temperature rises above the setpoint, the normally closed chilled water valve will begin to open. The chilled water valve has a spring range of 8-13 psig; therefore, it will throttle to the fully open position as the space temperature continues to increase.

Both valves will be closed between 7 and 8 psig branch pressure. This dead band between the two spring ranges ensures the heating valve is fully closed before the cooling valve opens and vice versa.

Single Zone System with Low-Limit and Outside Air Control



SINGLE ZONE SPACE CONTROL WITH LOW LIMIT AND OUTSIDE AIR CONTROL

FIGURE 6-4

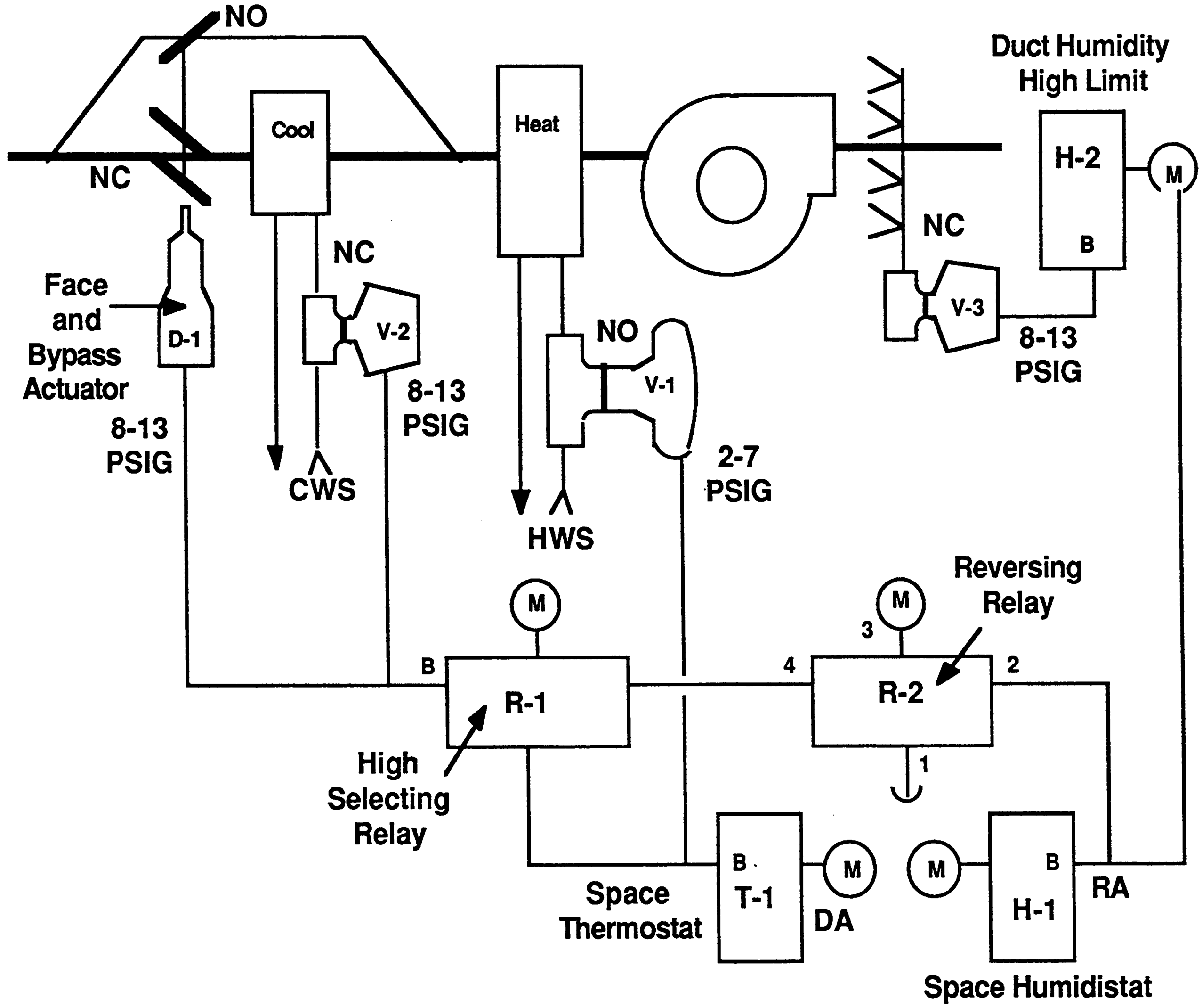
In this system shown in Figure 6-4, the same function is being performed by the space thermostat (T-1) as in the previously discussed single zone system. However, additional controls that are often found in conjunction with the thermostat and coil controls in a system l;ike this are added. This system also includes control of the outside air damper, the EP-1 switch that is wired into the fan starter circuit to provide shutdown at night, and low limit controls. In this sequence of operation, the fan starter is wired to the EP-1 switch controlling the branch line airflow to the outside air damper actuator (D-1). When the fan is off, the E-P switch exhausts the branch line to the outside air damper actuator allowing it to go to the fully closed position. Also wired into the control circuit to the fan starter is a freezestat (T-2) which is another electrical device. The other pneumatic device shown is the low limit thermostat (T-3).

T-3 is a bleed type device and is located directly downstream from the heating coil, and only becomes active when the mixed air temperature leaving the heating coil decreases below the setpoint, which would typically be 55°F. In the event that the low limit senses a heating coil discharge temperature below setpoint, the low-limit will bleed off the branch line pressure from the space thermostat allowing the hot water supply valve (V-1) to open further than the space thermostat would allow. If the low limit senses a heating coil discharge temperature at or above setpoint, it will pass the same pressure on to the heating valve that it receives from the space thermostat.

The low limit stat, mentioned above, will come into action if the low limit is not able to compensate for a drop in temperature by providing sufficient flow through the heating coil to enable it to recover. If the mixed air temperature leaving the heating coil continues to decrease, the freezestat would break the control circuit to the fan starter, shutting off the fan to prevent the possibility of coil freeze up. This is a safety control and generally has a manual reset. The setpoint of this device is typically 38°F and must be manually reset if it breaks the fan control circuit.

Other controls may be used in conjunction with this type of system, but this provides the basics. There could potentially be a minimum position control on the outside air damper in lieu of this type of outside air damper actuator. Also return air and exhaust air dampers could be included.

<u>Single Zone Control of Space Temperature and Humidity</u>



SINGLE ZONE SPACE TEMPERATURE AND HUMIDITY CONTROL

FIGURE 6-5

This system shown, in Figure 6-5, is similar to the previous single zone system which has been discussed, but with some additional control functions. This system provides humidity control in addition to the temperature control of the previous system. To facilitate humidity control, face and bypass dampers control flow through the cooling coil, and the heating coil provides reheat capability.

Basic temperature control in this system is accomplished by the direct acting space thermostat (T-1) which controls the hot water supply valve (V-1), the chilled water supply valve (V-2), and the face and bypass dampers (D-1).

As the temperature increases in the space, branch line pressure from the thermostat increases, causing the hot water supply valve to move toward the closed position. As the pressure continues to increase in response to an increasing space temperature, the signal passes through the high selecting relay (R-1) and starts to drive the chilled water supply valve toward the open position. At the same time the normally open bypass damper moves toward the closed position, and the normally closed face damper moves toward the open position.

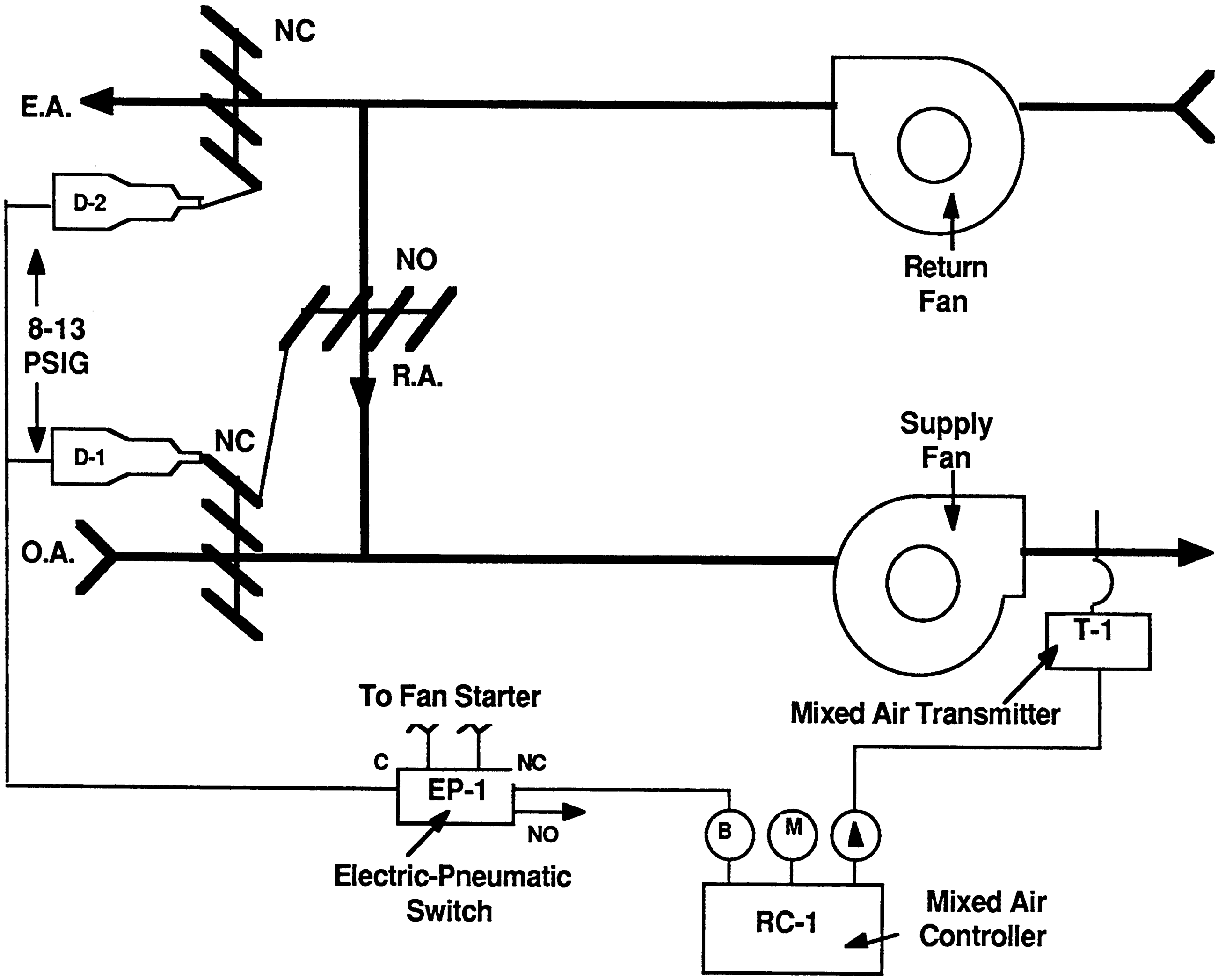
At the same time, the space humidistat, which is reverse acting, is sensing the humidity level in the space, and its branch line feeds to the reversing relay (R-2). The output from the reversing relay also feeds to the high selecting relay. If the humidity in the space increases above setpoint, and the branch line pressure of the reversing relay exceeds that of the branch line pressure of the thermostat, the pressure developed by the humidistat through the reversing relay will actuate the face and bypass damper and the chilled water supply valve to provide cooling to dehumidify the space.

At the same time, the branch line of the space humidistat (H-1) goes to the duct high limit controller (H-2) which is fed to the main air port of the duct high limit controller. Should the relative humidity in the duct increase above setpoint, typically 90%, it will not pass the signal from the space humidistat. The humidifier valve (V-3) is normally closed; if it receives no signal, the humidifier valve will be closed and no humidification will occur at this time. If the humidity is below setpoint in the space and below setpoint in the duct, the branch line pressure from the space humidistat is passed through the duct high limit controller to position the humidifier valve to some value to satisfy the space humidistat.

This type of system is applicable in an area with low humidity in the winter which could present a problem, as well as a need for dehumidification during the summer if the humidity is too high. As mentioned, it is possible to open the cooling coil valve and face damper to some value, thereby cooling the air because of a high humidity condition, even though the space thermostat might be calling for some heat. In this case, the air is reheated by the heating coil and supplied at a temperature necessary to satisfy the demands of the space.

Mixed Air Control Economizer Cycle

The purpose of this application is to provide the maximum amount of free cooling available from the outside air. Commercial systems are generally cooling oriented because of load conditions in the controlled space; as such, it is advantageous to maximize the use of outside air for cooling whenever possible. The type of economizer system will vary depending on geographical areas. In the past, some of the milder climate areas utilized no economizer at all, or outside air dampers where in a fixed position at all times. The conversion to economizer systems has been a popular energy savings strategy. There are a number of things that can be done to enhance the operation of an economizer system.

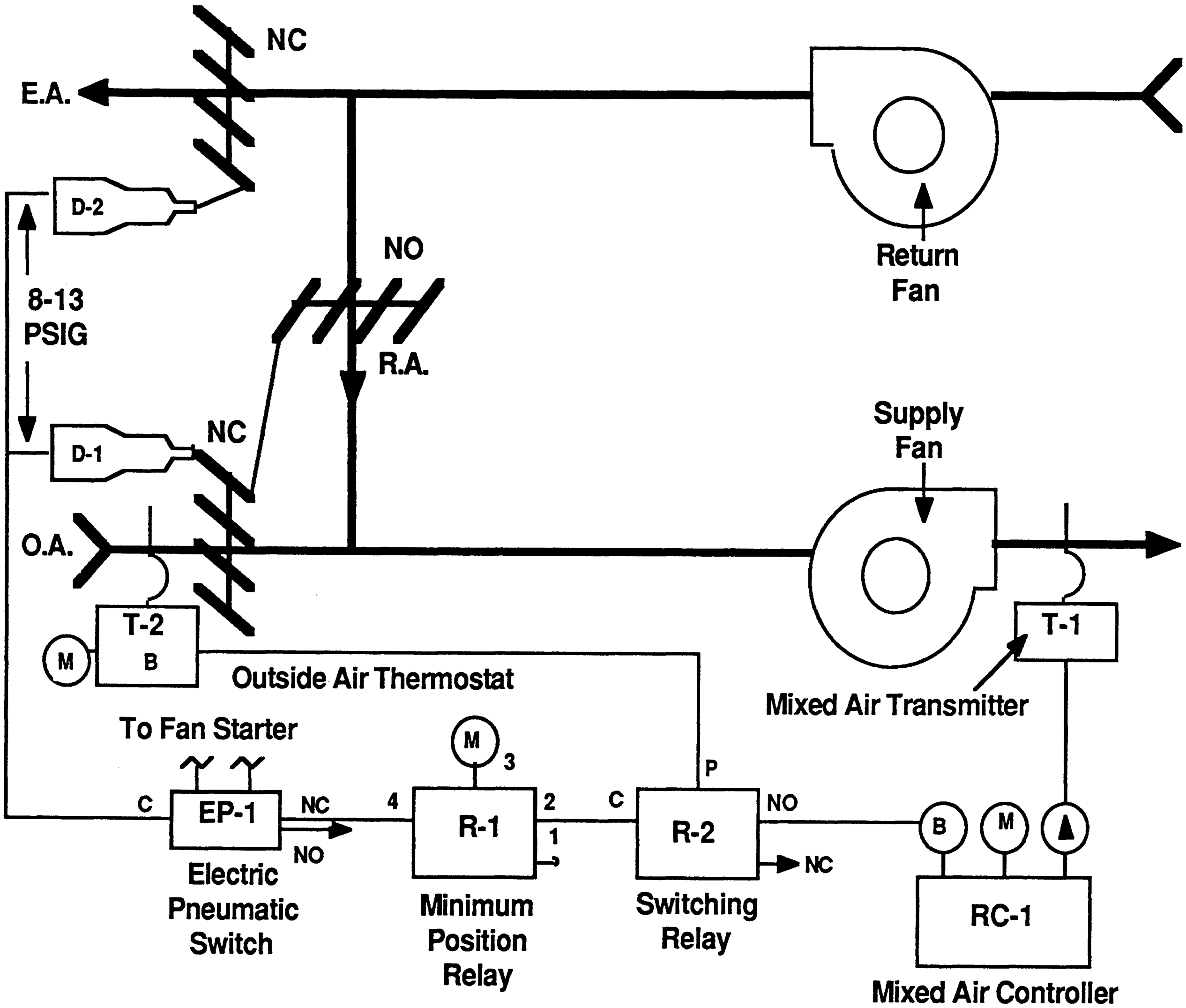


MIXED AIR CONTROL - ECONOMIZER CYCLE

FIGURE 6-6

The operation of the economizer cycle, shown in Figure 6-6, involves a mixed air transmitter (T-1) sending a signal to the direct acting mixed air receiver controller (RC-1). This branch line signal is sent to the normally closed port of the electric-pneumatic switch (EP-1), which is wired in series with the fan starter circuit. When the fan is energized, the EP switch is energized and the normally closed to common ports communicate, and the branch line pressure from the mixed air controller is sent to the outside and return (D-1) and the exhaust air damper actuators (D-2). When the fan is de-energized, the normally closed port of the EP switch is closed. The common port of the EP communicates with the normally open port, and the pressure of the actuators is exhausted to atmosphere, allowing the outside and exhaust air dampers to go to the closed position, and the return air damper to go to the open position. As long as the supply fan is energized, the dampers will modulate in response to changes in the mixed air temperature and will attempt to maintain a typical mixed air temperature of 55^{o}F.

Mixed Air Control, Economizer with High Limit and Minimum Position



MIXED AIR ECONOMIZER WITH HIGH LIMIT AND MINIMUM POSITION

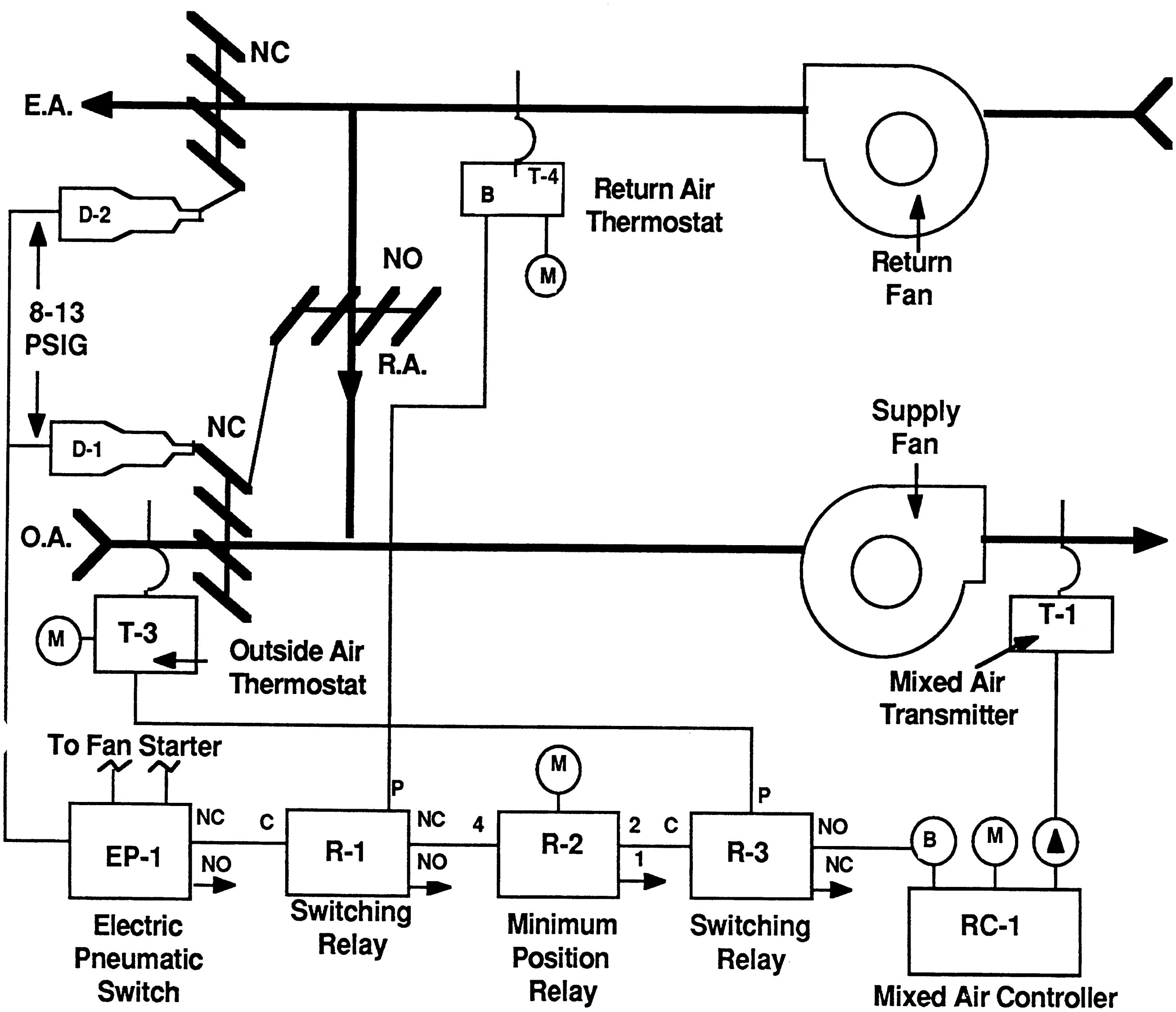
FIGURE 6-7

In this system shown in Figure 6-7, two functions are added to the economizer cycle. These are a high limit control to disable the mixed air controller (RC-1) if the temperature rises above the setpoint of the outside air thermostat (T-2) and a minimum position relay, so that the outside air damper (D-1) will be partially opened to a minimum position to satisfy the ventilation requirements in the building, regardless of other conditions such as low mixed air temperature during occupied hours.

The minimum position relay (R-1) is capable of providing a fixed signal which is adjustable or passing a higher signal which is provided at port 2 by the branch output of the mixed air controller.

The setpoint on the minimum position relay is typically 8.5 psig. As long as the branch output pressure of the mixed air controller is greater than 8.5 psig, this pressure will be transmitted out of port 4 of the minimum positioning relay to the normally closed port of the electric-pneumatic switch (EP-1). As long as the electric-pneumatic switch is energized, this pressure is passed on through the common port to the outside and return damper actuator and the exhaust air damper actuator (D-2) allowing them to modulate in response to the changing mixed air temperature.

As outside air temperature increases, the mixed air temperature will increase and the mixed air controller, which is direct acting, will increase its signal modulating the outside and exhaust dampers towards the open position and the return air towards closed position. This increased amount of warm outside air will not provide the necessary cooling. If the outside air thermostat (T-2) senses a temperature rising above its setpoint, typically 70^{o}F, it will trigger the switching relay (R-2), closing the normally open port receiving the output of the mixed air controller and exhausting the pressure out the normally closed port. This reduces the pressure at the minimum positioner below 8.5 psig. The minimum position relay is set to maintain a minimum 8.5 psig signal out of the branch line maintaining a minimum position on the outside and return and the exhaust damper actuators. In the example shown in this diagram, the start point of the damper actuators is shown as being 8 psig, and the setpoint of the minimum positioning relay is 8.5 psig. In this case, 8.5 psig is slightly above the start point and represents a position of the outside air damper that will allow enough outside air to enter the building while the fan is running to meet the minimum ventilation requirements.



MIXED AIR ECONOMIZER WITH HIGH LIMIT, MINIMUM POSITION AND WARM-UP CONTROL

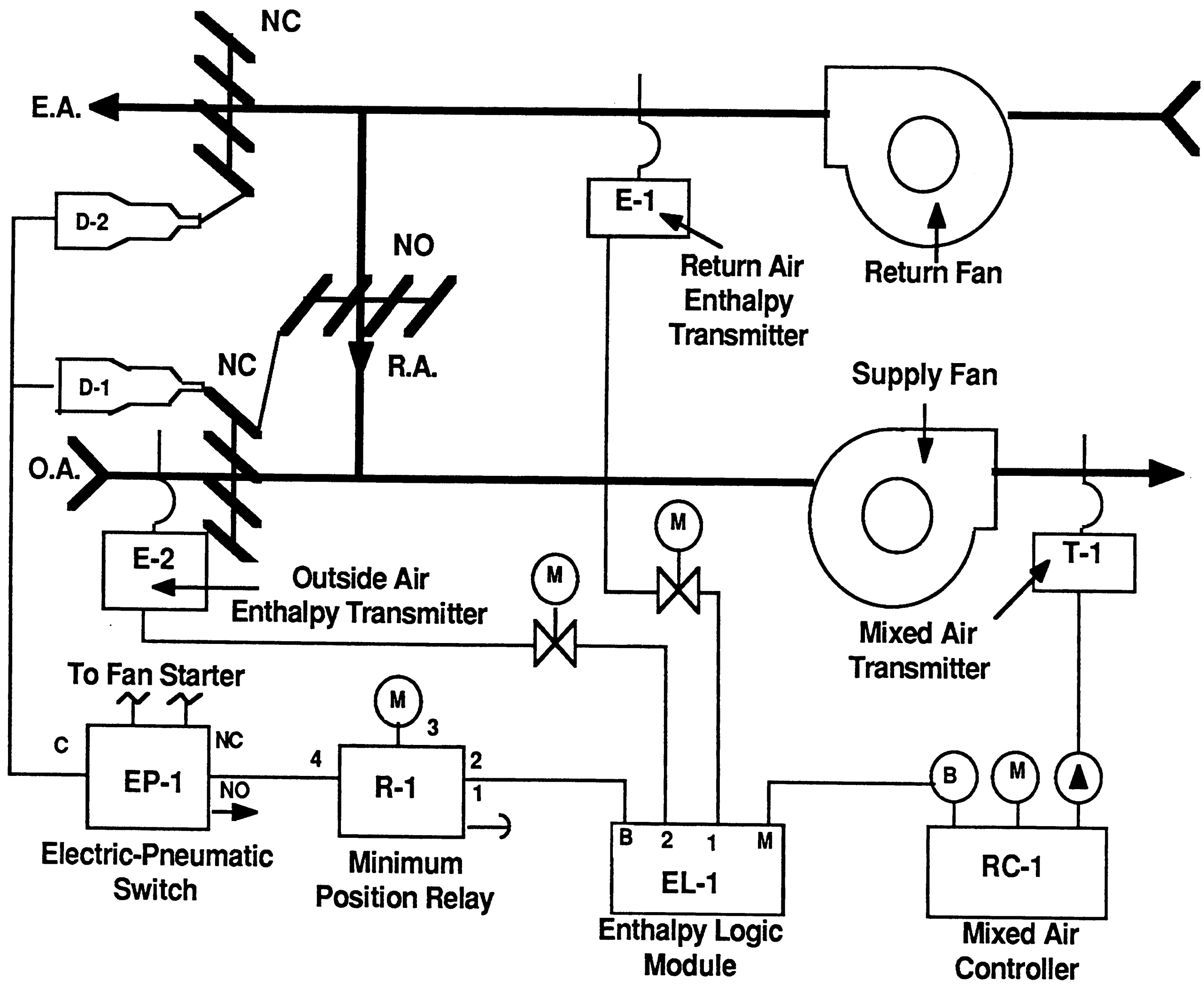
FIGURE 6-8

For this economizer application, morning warm-up control has been added and is shown in Figure 6-8. Warm-up control allows the building to rise to temperature on morning start-up before energizing the economizer dampers. Upon fan start-up, electric-pneumatic switch (EP-1) is energized as in the previous systems described; and the signal from the mixed air controller (RC-1) passes through the switching relay (R-3) and the minimum position relay (R-2) to the normally closed port of the switching relay (R-1). Until the return air temperature sensed by the return air thermostat (T-4) rises above the setpoint, typically 70^{o}F, the branch output of the mixed air controller is blocked at the switching relay (R-1).

Once the return air temperature rises above 70^{o}F, the relay switches and the normally closed and common ports are connected. This allows the branch signal to pass through to position the economizer dampers in response to either the signal from the mixed air controller or the minimum signal from the minimum position relay.

Mixed Air Control Economizer with Enthalpy Changeover and Minimum Position

In this application, an enthalpy controlled changeover point is used to monitor whether there is "free cooling" available from the outside air. This is in contrast to the previous systems where a fixed changeover point, or high limit to lock out outside air, was utilized. The amount of cooling energy needed is determined by the enthalpy content of the air which varies with both temperature and humidity. It is sometimes more economical to cool warmer outside air with a lower humidity content than it is to cool colder return air with a higher humidity content. The objective is to compare the enthalpy value of both the outside air and return air, and use whichever air has the lowest enthalpy value. Therefore, the air that has the lowest enthalpy value provides a lower load on the cooling coil. However, in some instances, the outside air will be capable, as determined by the enthalpy changeover, to provide enough cooling capacity without using the mechanical cooling system, therefore, providing "free cooling". This may occur on early or late hours of the day during the summer, longer periods of the day during spring and fall and all day long during the winter.

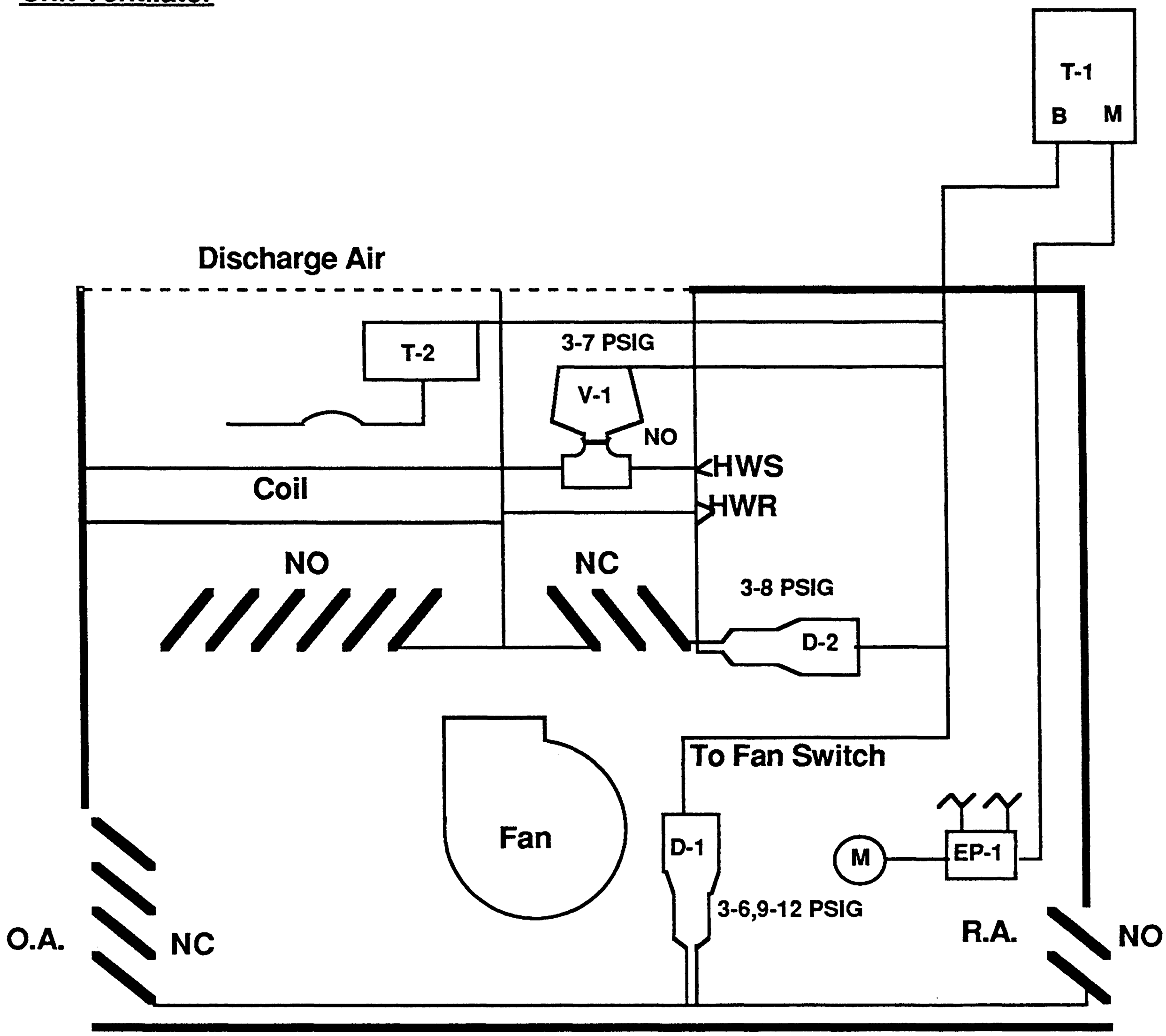


MIXED AIR CONTROL ECONOMIZER WITH ENTHALPY CHANGEOVER AND MINIMUM POSITION

FIGURE 6-9

In Figure 6-9, an enthalpy logic module (EL-1) which has two signal inputs, one from a return air enthalpy transmitter (E-1) and one from an outside air enthalpy transmitter (E-2), is used. As long as the signal from the outside air enthalpy transmitter is less than the signal from the return air enthalpy transmitter, the signal from the mixed air controller (RC-1) passes through the enthalpy logic module, providing a modulating signal to position the economizer dampers. When the outside air enthalpy rises above that of the return air enthalpy, the enthalpy logic module blocks the signal from the mixed air controller, which allows the outside, return and exhaust dampers to return to the minimum position as set on minimum position relay (R-1). The final control, of course, is through the electric-pneumatic switch (EP-1), which is in series with the fan starter, and when the supply fan is de-energized, the electric-pneumatic switch exhausts the control signal which positions the dampers to their normal position.

Unit Ventilator



UNIT VENTILATOR

FIGURE 6-10

Figure 6-10 depicts a unit ventilator. The unit ventilator is a self-contained unit with the exception of a hot water supply from a central boiler. It does not rely on any central air handling system. It is generally located beneath the windows on an outside wall where the normally closed outside air dampers would bring in air for ventilation or cooling from the outside, and the normally open return air dampers would be able to recirculate the return air. However, the majority of the time the outside and return air are mixed together in various proportions to satisfy space requirements.

When the fan is energized, the electric-pneumatic switch (EP-1) allows the thermostat signal to position the coil valve (V-1), face and bypass damper (D-2) and the outside and return damper (D-1). The fan draws the two air supplies through the heating coil to the space. The space thermostat (T-1) is direct acting and, as shown in the diagram, is piped to the normally open heating valve (V-1).

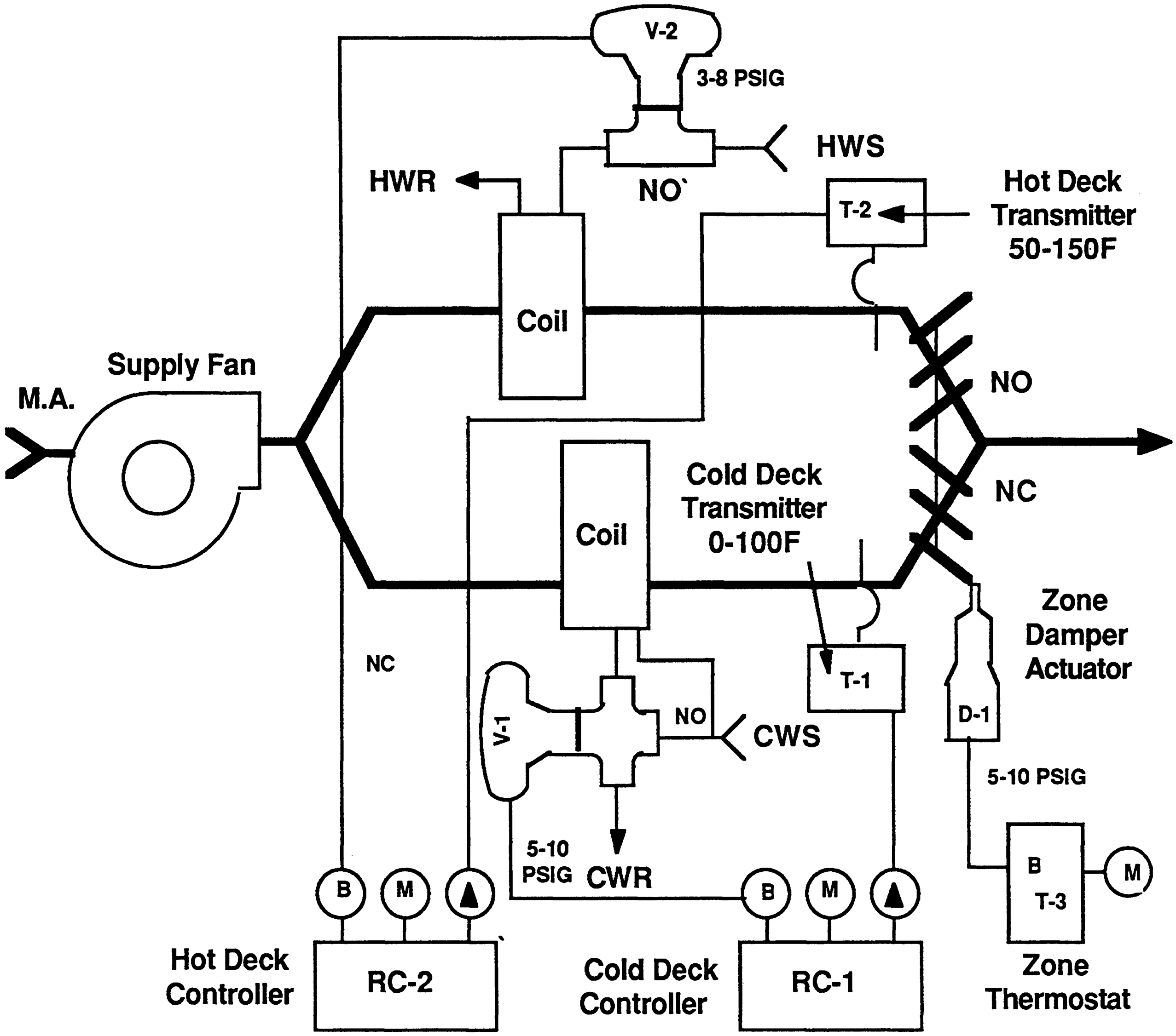
At the same time, whenever the fan is energized, the branch signal of the space thermostat (T-1) will also be fed to the face and bypass damper actuator (D-2) and the outside and return damper actuator (D-1). In this application the outside and return damper actuator (D-1) has a dual spring range of 3-6 psig and 9-12 psig.

When the temperature in the space is below setpoint, the space thermostat branch pressure will be low and the position of the outside air damper will be toward the closed position since it is normally closed, or at the minimum position if the branch pressure is between 6 and 9 psig. This allows the maximum recirculation of return air; and at the same time, the heating valve will be toward the open position, as will the face damper allowing the air to pass through the coil supplying additional heat to warm the space as the discharge air temperature is raised to achieve setpoint. As the space approaches setpoint, the branch pressure rises and modulates the normally open hot water supply valve toward the closed position. If the outside air temperature is quite cold and the discharge air temperature is lower than the setpoint of the low limit thermostat (T-2), typically 60^{o}F, the low limit thermostat will bleed the branch pressure down, allowing the heating valve to move toward the open position and the outside air damper to move toward the closed position, reducing the amount of outside air entering the unit. This could occur regardless of the space temperature being satisfied.

The dual spring range damper actuator (D-1) utilizes two separate springs internally. The first of these is actuated between 3 and 6 psig. This provides a minimum amount of damper movement upon system start-up. This allows the outside air damper to open to a minimum position to provide the required ventilation air if the space temperature is at or near setpoint. The space thermostat will have a branch line pressure of about 8 psig at this time.

Between 6 and 9 psig, there is no damper movement and the system operates on the return air and minimum outside air. This is to prevent the reheating of a large quantity of cold outside air, yet provide adequate ventilation as the hot water supply valve may be closed through this range. If the space temperature continues to increase, the hot water supply valve and face damper will be closed. The second spring of the outside and return air damper actuator will pass through its stroke (9-12 psig), closing off the return air and opening the outside air damper to obtain a maximum amount of outside air to provide cooling.

Multizone and Hot and Cold Deck Control



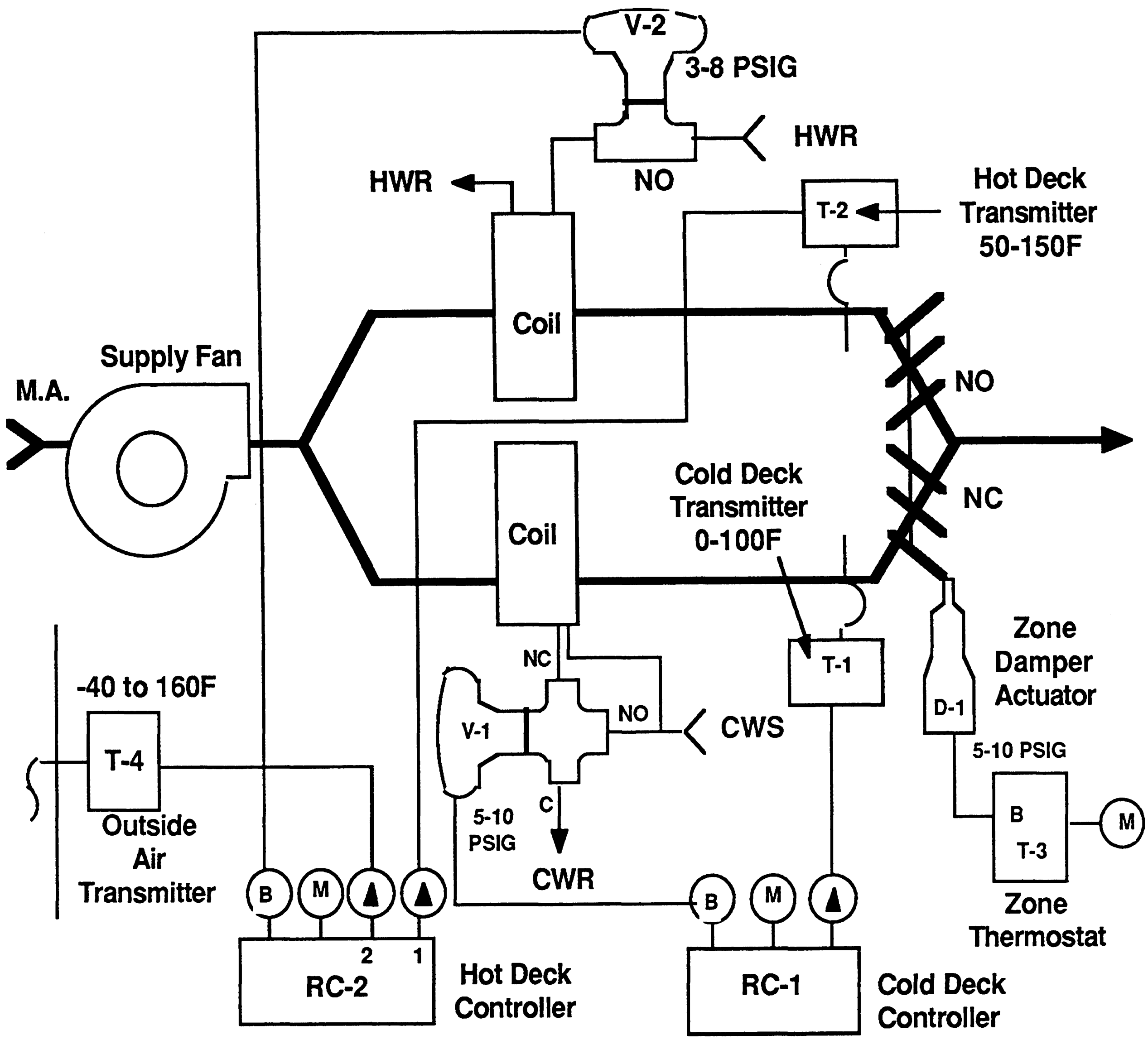
MULTIZONE HOT AND COLD DECK CONTROL
FIGURE 6-11

This is a relatively simple system shown in Figure 6-11 utilizing two transmitters, two receiver-controllers, a zone thermostat and damper actuator for each zone in the space. As the name multizone implies, there are many zones, typically 4 to 12; therefore, there are as many zone thermostats and zone damper actuators as there are zones. Mixed air is supplied to the fan in a manner described in Figures 6-6 through 6-9. During the heating season, this mixed air temperature will be about 55^{o}F. The air that reaches the hot deck will be heated to typically 90^{o}F. System requirements may require the hot deck temperature to be higher in some instances.

Transmitter (T-2) senses the temperature of the air leaving the heating coil and provides a signal to the hot deck controller (RC-2) to maintain a constant discharge temperature. The hot deck coil is controlled by the signal from the controller to the valve (V-2). The mixed air that passes through the cold deck provides the necessary cooling, if any of the zones require cooling. Also, this cool cold deck air will mix with the hot deck air if a zone should require an intermediate temperature as determined by the zone thermostat, such as T-3.

During the cooling season, chilled water will be available. A 55^{o}F discharge temperature of the cold deck is maintained by cold deck transmitter (T-1), providing a signal to the cold deck controller (RC-1) which controls the cold deck valve (V-1). This 55^{o}F air from the cold deck will provide the necessary cooling as determined by the respective zone thermostats. The mixed air during the summer months will be typically 85^{o}F. This warm air passes through the hot deck untreated, and will be available to temper the cold deck air for zones that do not require maximum cooling as determined by the respective zone thermostat.

Multizone and Hot and Cold Deck Control with Hot Deck Reset



RESET SCHEDULE

HD	OA
120	-10
55	65

MULTIZONE HOT AND COLD DECK CONTROL WITH HOT DECK RESET

FIGURE 6-12

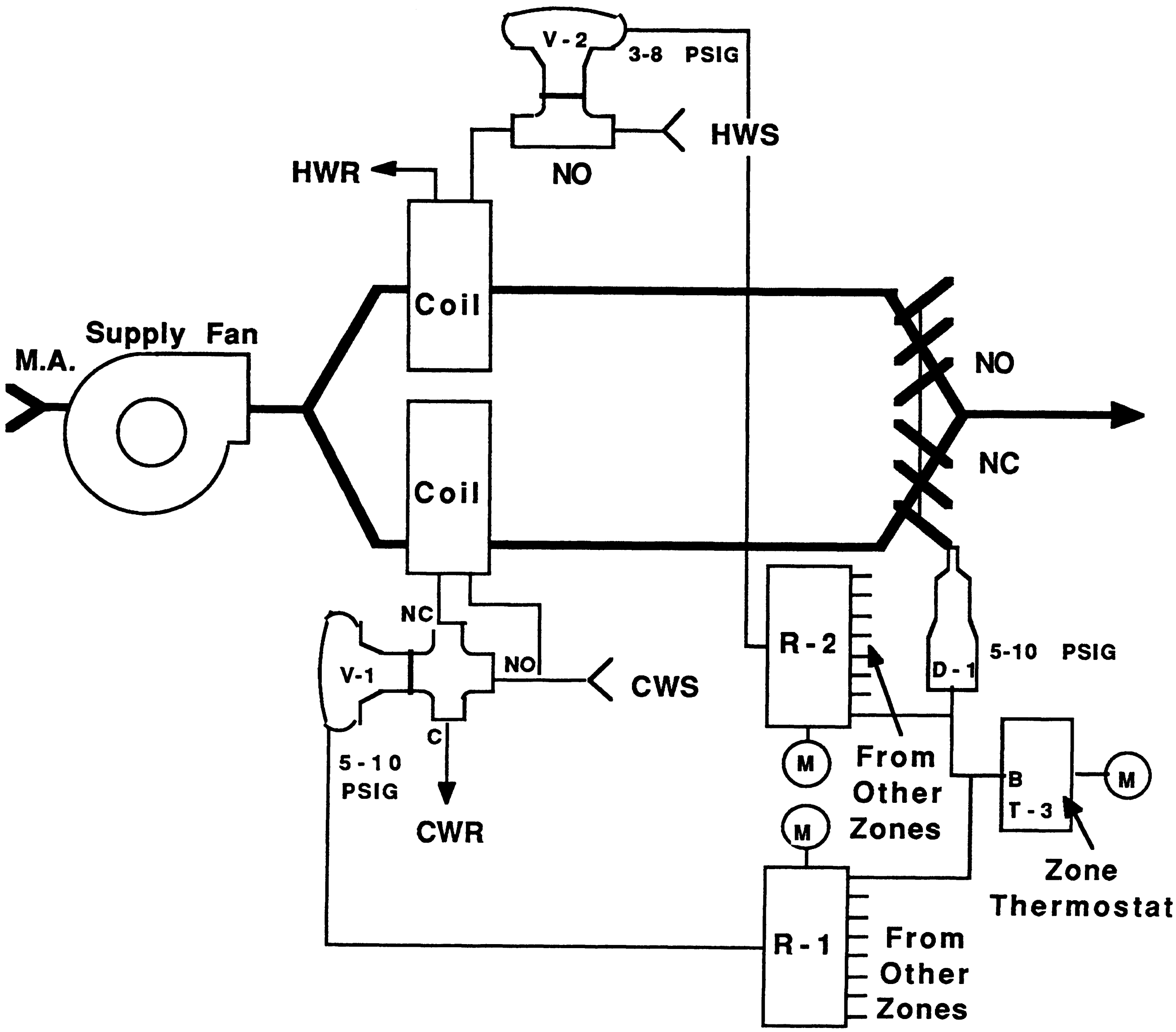
This system, shown in Figure 6-12, is essentially the same as the system discussed in Figure 6-11, with the addition of an outside air transmitter (T-4) which provides a signal to the hot deck controller (RC-2). The hot deck controller is a reset type controller. This makes it possible to reset the hot deck temperature according to outside temperature. The reset schedule shown represents the design temperature range of -10^{o}F to 65^{o}F outside air temperature.

Over this outside air temperature range, it is determined that a range of 55^{o}F to 120^{o}F hot deck temperature will be sufficient to offset the heat losses that these outside air temperature changes cause. This approach is typical for the more extreme, colder climates where several of the zones have an exterior exposure. When the outside air temperature reaches the warm conditions of 65^{o}F, the hot deck temperature will not be 55^{o}F, although the reset schedule indicates this. What actually occurs is that the hot deck temperature, at this time, will be the same as the mixed air temperature. By arranging the reset schedule in this manner, we can be assured that the hot deck valve (V-2) will be closed when the outside air temperature approaches 65^{o}F. The percent authority setting for this application works out to be 173%. Refer to Chapter 3 for a review of percent authority if this causes some confusion.

The cold deck operates in the same manner as shown in Figure 6-11. Chilled water will be available during the summer months to provide a cold deck temperature of 55^{o}F. The mixed air temperature will be approximately 85^{o}F which passes through the hot deck and will be available to temper the air supplied to any zone that does not require full cooling. During the winter months, the cold deck temperature will be 55^{o}F which is the mixed air temperature and will provide the cooling necessary for any zone that should call for such a condition.

The zone thermostats (T-3) provides a signal to their respective zone damper actuators (D-1) to modulate the dampers to supply the proportion of warm or cool air to satisfy the space requirements the year around.

Multizone Hot Deck and Cold Deck with Signal Selection



MULTIZONE HOT AND COLD DECK WITH SIGNAL SELECTION

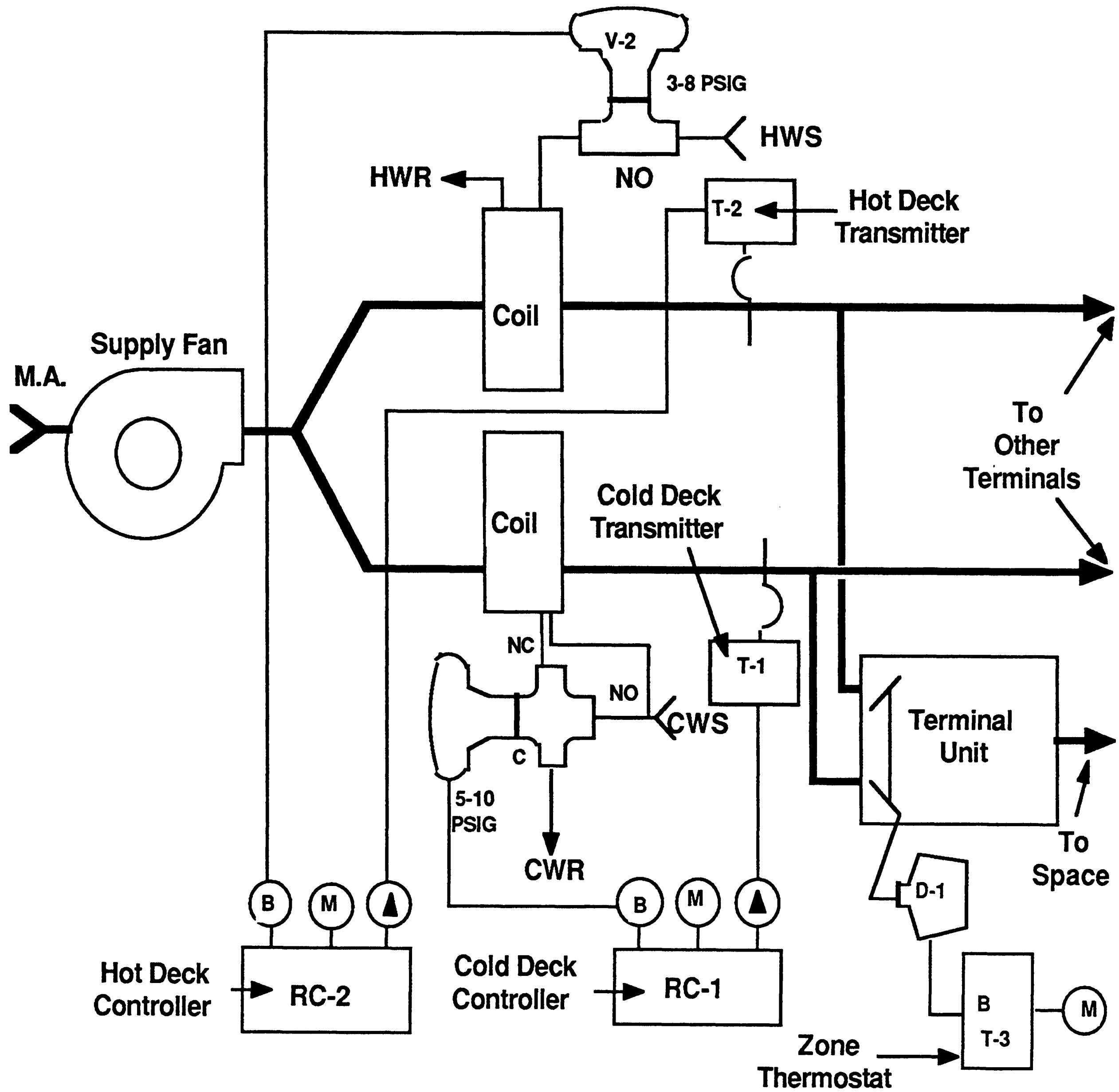
FIGURE 6-13

In the system shown in Figure 6-13, another approach has been taken which is signal selection. The cold deck temperature is maintained only at a level required by the zone requiring the greatest amount of cooling. A high signal selector relay (R-1) is used which accepts the branch input from up to 10 individual zones and passes on the highest input to control the chilled water valve (V-1). In this approach, less energy is required for the cooling process than in the previous systems where a constant cold deck temperature was maintained.

This system allows the temperature of the cold deck to be controlled upwards as demand decreases in the space.

For the hot deck, the same approach is applied. The zone of greatest demand will be the coolest and will be represented by the lowest signal from a zone thermostat from this cool zone. The signal is passed on to the hot deck valve (V-2) so that the system will effectively compensate for worst case demand. The advantage here is the same as described above for the cold deck in that the hot deck temperature is controlled downward to reduce the cost of heating, as space demand decreases.

Dual Duct System Control



DUAL DUCT CONTROL

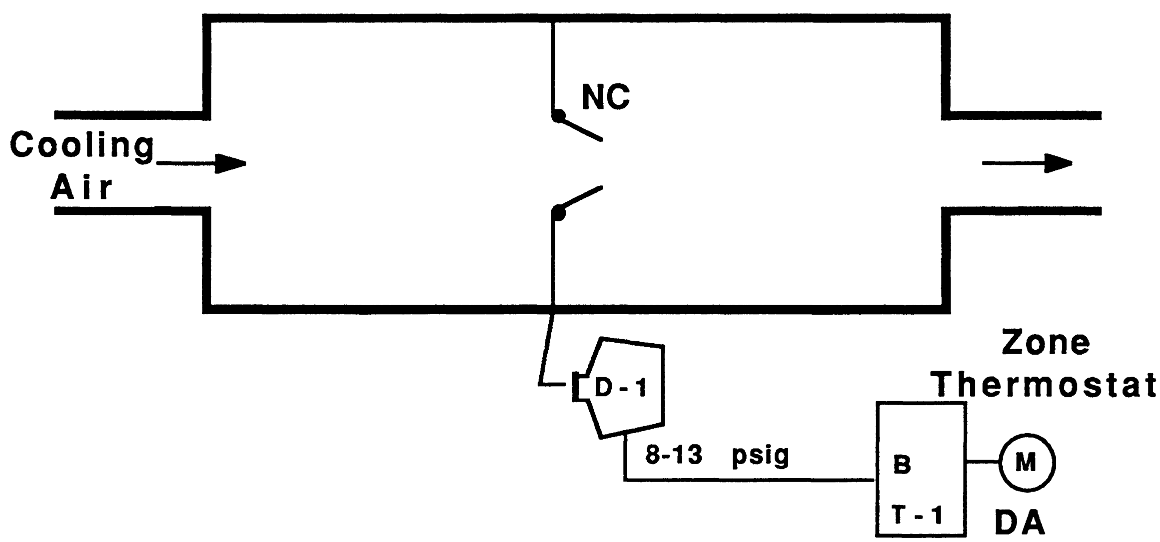
FIGURE 6-14

The system shown in Figure 6-14 is similar to the basic hot deck/cold deck for the multizone control that has already been described. The primary difference is in the delivery of the hot and cold air supply to the spaces. Dual duct systems are rare and are generally considered not to be energy efficient. They were originally designed to provide a continuous supply of hot deck air and cold deck air to the terminal unit of each space. The terminal units in each space mix these two air supplies as necessary to provide a discharge air temperature that will satisfy space conditions. These systems are rarely used today, and many older systems have been retrofitted to lock out the hot or cold deck during the inappropriate season.

In doing so, the necessity of reheating or recooling air is eliminated.

The variations shown in Figures 6-12 and 6-13 can be applied, just as effectively, to the dual duct system in an effort to reduce energy consumption. These functions are usually added on as an energy saving measure. As stated earlier, most of these systems predate the common application of both of these functions. The greatest opportunity for energy conservation is to eliminate the hot deck and continue the cold deck with the conversion of the space terminal units to variable volume devices. This requires the application of pressure controls on the system in order to throttle the fan as required in a variable volume system.

Terminal Unit, VAV



VAV TERMINAL UNIT

FIGURE 6-15

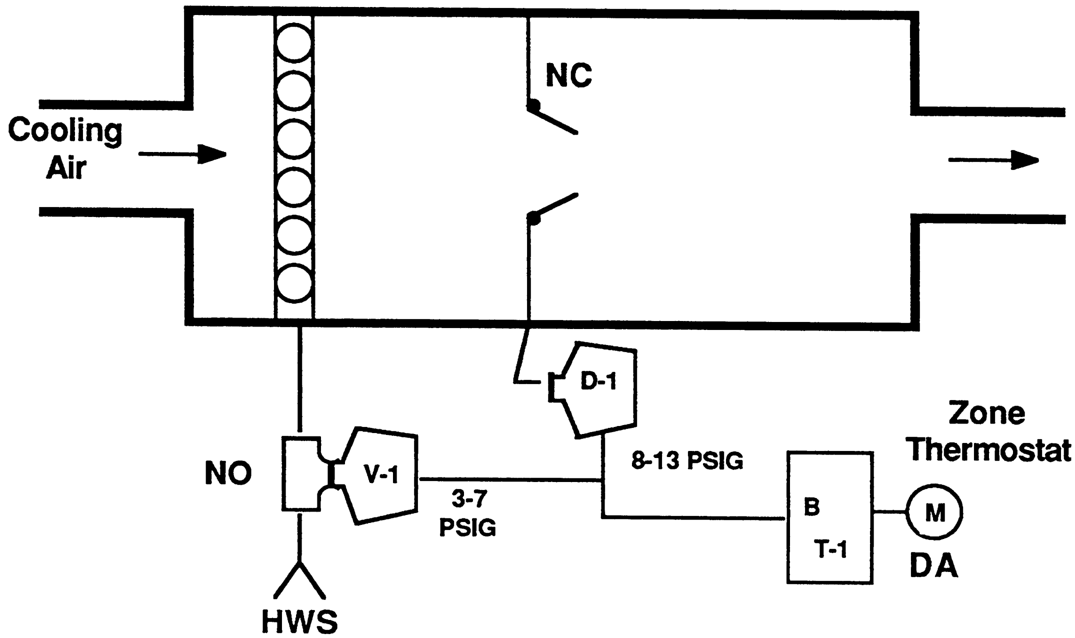
This terminal unit, shown in Figure 6-15, is representative of a typical VAV unit. In this example the terminal is supplied with cooling air, and the terminal dampers are normally closed. The thermostat (T-1) controlling the space temperature is direct acting and the damper (D-1) is normally closed. As the temperature in the space rises, the terminal dampers open further to allow greater flow into the space.

This terminal can also be supplied with warm air, and utilize a reverse acting thermostat which would close the dampers further on a rise in temperature.

Volume of air delivered to the space varies as the inlet duct static pressures varies although the thermostat has not called for any change.

Therefore, there is a need to compensate for this inlet duct static variance which will be discussed later.

Terminal Unit VAV, with Reheat

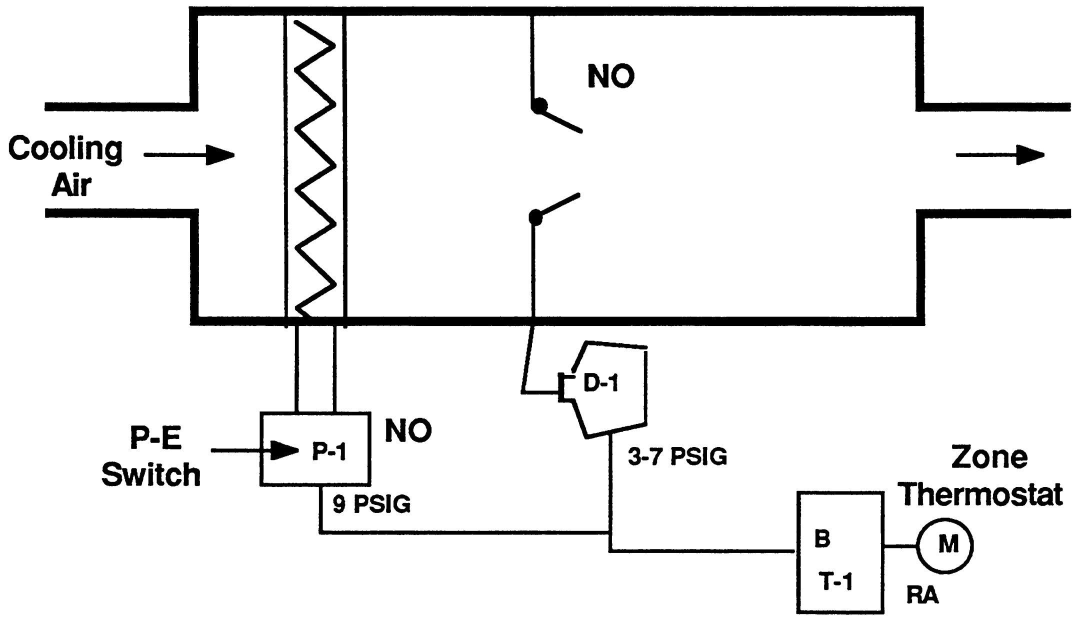


TERMINAL UNIT VAV WITH REHEAT

FIGURE 6-16

This is a variable volume terminal unit as previously discussed, however, a reheat coil and hot water valve are included to operate in sequence with the volume damper. In Figure 6-16 a direct acting thermostat (T-1) and a normally closed volume damper (D-1) are being used. As the temperature decreases, the volume damper moves toward the closed position by a decrease in branch line pressure. As the pressure continues to decrease in the branch line, the VAV unit goes to a minimum flow which is mechanically set. As the pressure decreases further, the normally open hot water valve starts to open. The hot water valve will throttle to full flow upon sufficient demand in the space, and, on a decrease in demand for heat, will throttle toward the closed position. The valve will be fully closed before the VAV unit starts to open beyond the minimum position to allow a greater flow of cooling air to the space.

Terminal Unit VAV, with Electric Reheat

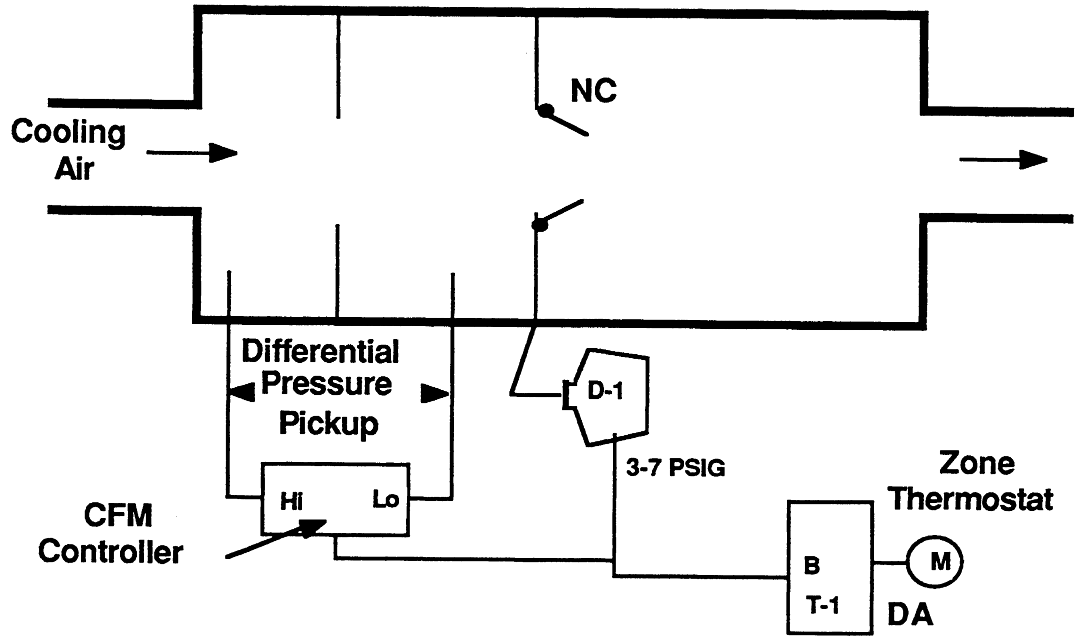


TERMINAL UNIT VAV WITH ELECTRIC REHEAT

FIGURE 6-17

In Figure 6-17 an electric heater in the duct is used and is controlled in sequence with the VAV damper. The reverse acting thermostat (T-1) responds to an increase in space temperature by decreasing the branch line pressure, opening the VAV damper (D-1) further to increase the flow of cooling air to meet the demand in the space. As space temperature decreases, output of the thermostat increases driving the VAV damper towards the minimum position. If this action is not sufficient to maintain space temperature at a comfortable level, the decreasing temperature causes the thermostat to further increase the branch line pressure. This will close the normally open contacts in the pressure--electric switch (P-1) which is controlling the electric heater. It provides reheat of the incoming air to the space until the space temperature reaches setpoint.

Terminal Unit VAV, with CFM Limiting



TERMINAL UNIT VAV WITH CFM LIMITING

FIGURE 6-18

The terminal unit, shown in Figure 6-18, is controlled by a thermostat in the space and a maximum CFM controller sensing flow through the terminal unit. The controller contains a high limit function which limits flow through the terminal unit to a maximum setting that is established by the manufacturer or by the air balance specialist in the field.

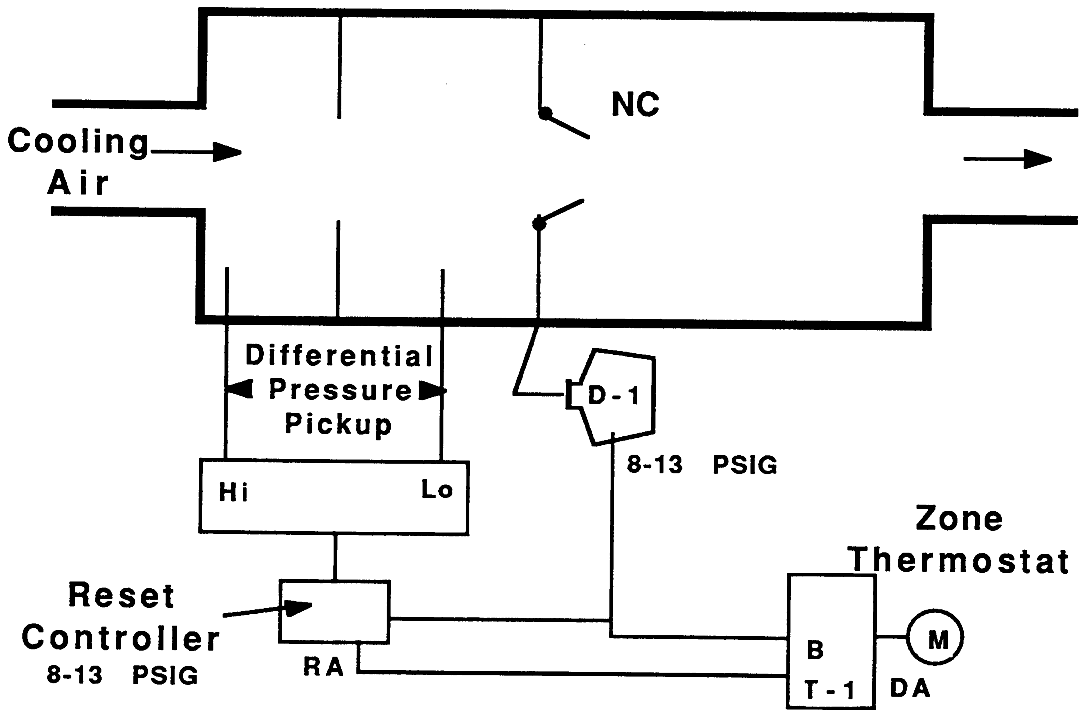
The terminal unit is supplied with cooling air and utilizes normally closed damper (D-1) and a direct acting zone thermostat (T-1). The zone thermostat controls the movement of the damper; and, as the space temperature increases, the branch line pressure increases allowing flow to increase through the terminal unit. As flow increases, the sensed pressure from the low and high static pressure pickups is sensed; and, at the predetermined high limit, the direct action of the CFM controller will take over through the high limit relay built into the CFM controller. This CFM controller maintains flow at that fixed maximum setting.

This type of terminal unit, with its maximum CFM limit control, responds to increased supply pressures. The duct supply pressure may change either upward or downward in response to demand in other zones within the building. If demand increases in other areas of the building, other terminals will open and decrease the total available static. The maximum CFM controller cannot respond to this situation. The temperature in the space may begin to increase at this time due to the lack of cooling air being delivered.

The space thermostat must respond to open the VAV damper further and add air volume into the space to meet the desired demand.

The other situation that may occur is an increase in static pressure due to terminal units closing down on a decrease in demand in other areas of the building which increases the static pressure, and, therefore, increase flow through this terminal unit. The static pressure sensors pick up this increased flow and compensate for it by closing the VAV damper to maintain flow at the preset high limit.

<u>Terminal Unit VAV, Pressure Independent</u>



TERMINAL UNIT VAV - PRESSURE INDEPENDENT

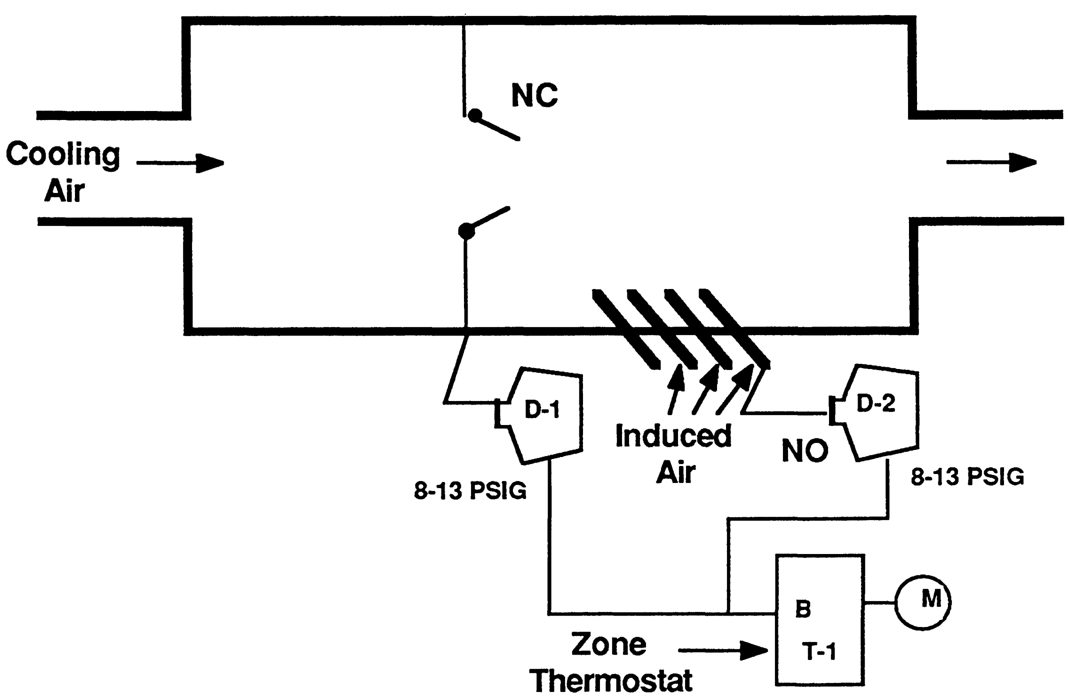
FIGURE 6-19

This type of VAV terminal unit, shown in Figure 6-19, utilizes a slightly different type of volume controller that maintains a constant volume of flow into the space in response to space demand. The controller used here is a reset volume controller which has a control output range of 8 to 13 psig. This matches the spring range of the actuator that is controlling the VAV damper (D-1). The reset volume controller has minimum and maximum flow setpoints. The device in this particular application is a reverse acting controller and is used in conjunction with a direct acting zone thermostat (T-1). This means that an increase in differential pressure will have a decrease in ouput to the normally closed dampers.

The zone thermostat resets the CFM setpoint of the reset controller to maximum or minimum between 8 to 13 psig in response to changing space temperature. As the space temperature increases, the branch line pressure from the zone thermostat increases and resets the CFM setpoint of the controller upward to allow a greater volume to flow through the terminal unit into the space. At any given control point, the terminal controller will respond to changes in inlet flow to move the damper to maintain the CFM rate that corresponds to the setpoint of the zone thermostat needed to satisfy load conditions.

This type of system is a good approach to VAV control. The obvious reason is that as pressure changes in the supply duct, the terminal unit can respond to the change and will continue to deliver the volume of air that the zone thermostat is requiring at the present load condition. This presents a much easier situation for the pressure controls on the central VAV fan to effectively control the system pressure which is discussed later.

Terminal Unit VAV - Induction



TERMINAL UNIT VAV - INDUCTION

FIGURE 6-20

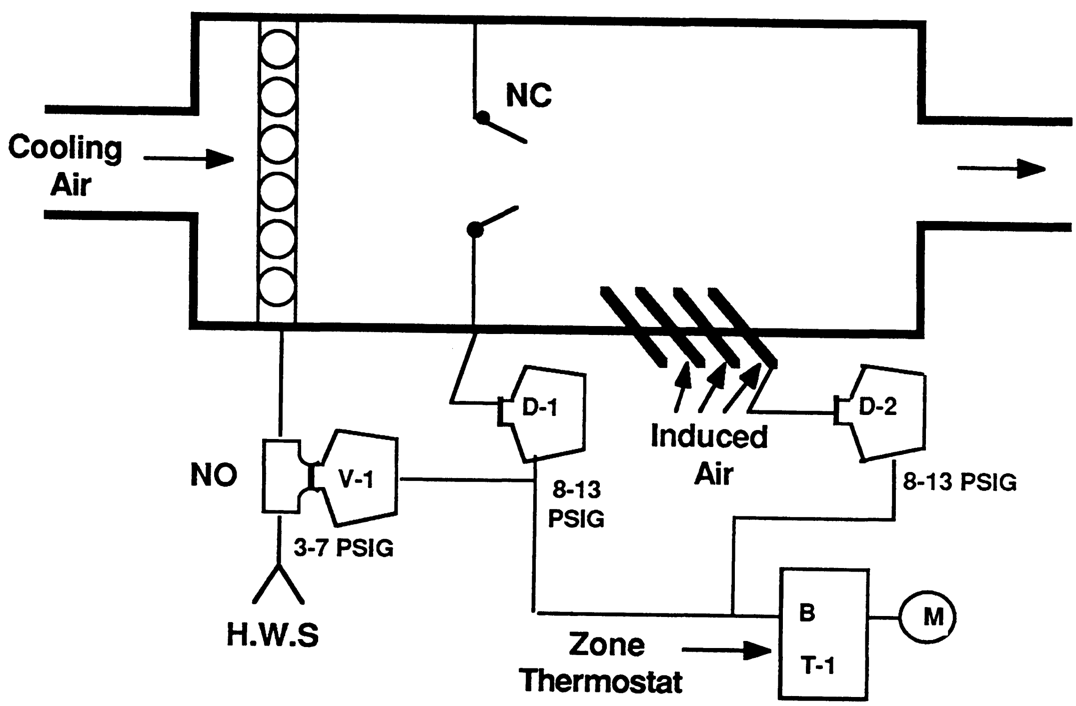
Induction units that are shown in Figure 6-20 use primary system air discharged through the terminal unit and secondary air that is induced from the controlled space. The units are located in the ceiling plenum the same as the units previously described. Return air from the space enters the plenum and provides the tempering effect to the primary air.

Shown, is a typical unit with thermostatically controlled actuators positioning the primary and induction dampers. On a demand for less cooling, the primary air damper (D-1) throttles toward the minimum position reducing the primary air volume. The induction air damper (D-2) simultaneously opens proportionally to the closing of the primary air damper.

Primary air discharges into a mixing section while inducing return air from the ceiling plenum. Mixed air is redirected into the supply duct and out of the diffuser to the space.

Reduction of the primary air mixed with the warm secondary air provides the reduction in cooling that may be required, yet provides the same total volume of air to the space. Unlike the terminal units previously described, there will not be a loss of air motion in the space during reduced cooling loads. Also, depending on the building design and occupancy, there may be no need for reheat, which is an increase in energy savings.

<u>Terminal Unit VAV - Induction with Reheat</u>



TERMINAL UNIT VAV - INDUCTION WITH REHEAT

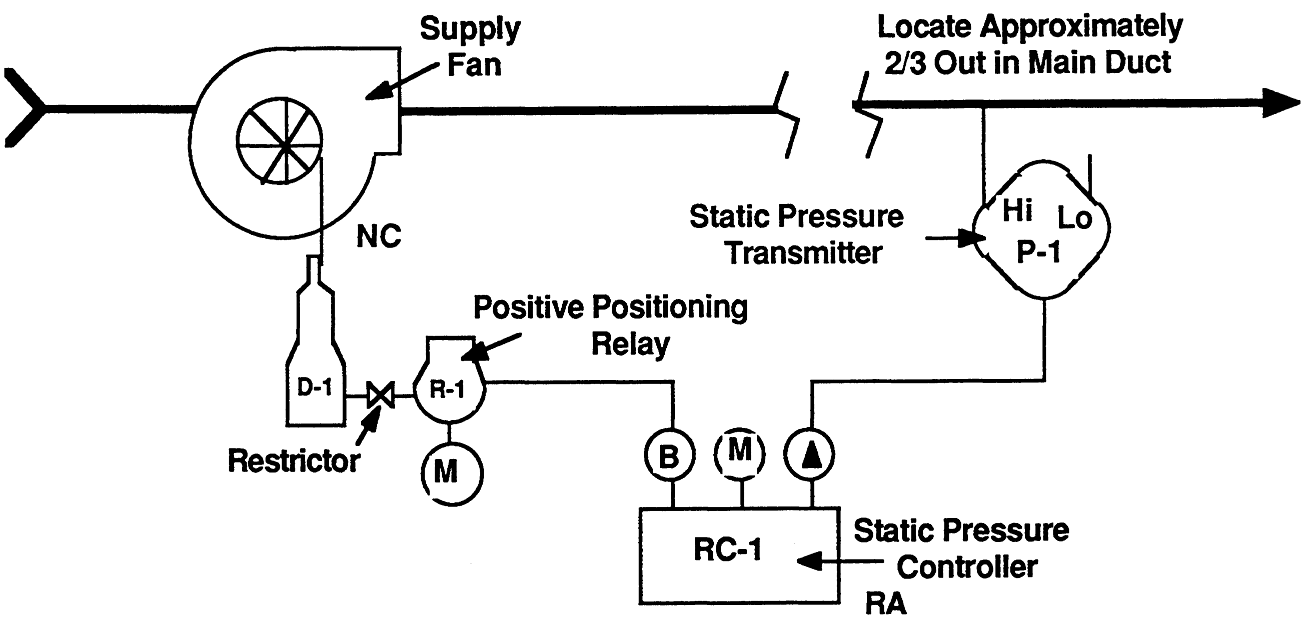
FIGURE 6-21

Figure 6-21 is essentially the same as the induction box previously described, but with the additional feature of a reheat coil in the primary air supply. This type of unit is generally used in perimeter areas.

If the temperature were to decrease in the space, the VAV terminal throttles toward the minimum position reducing the primary air volume, and the induction damper simultaneously opens proportionally bringing in warm return air to mix with the reduced volume of primary air. Upon a further decrease in temperature, the primary air is reduced to minimum, and the induction damper reaches the full open position. If this does not provide enough reduction in cooling, reheat is now in order. With a further decrease in space temperature, the hot water valve will begin to open which warms the minimum flow of primary air.

The zone thermostat (T-1) is direct acting and the normally closed VAV damper (D-1) and normally open induction damper (D-2) work in parallel. These dampers operate over an 8-13 psig range, and, therefore, operate in sequence with the normally open hot water valve (V-1) which has a spring range of 3-7 psig. This sequence makes certain that the primary air damper is at its minimum flow position, and the induction air damper is fully open before the hot water valve starts to open.

__Supply Fan Volume Control__



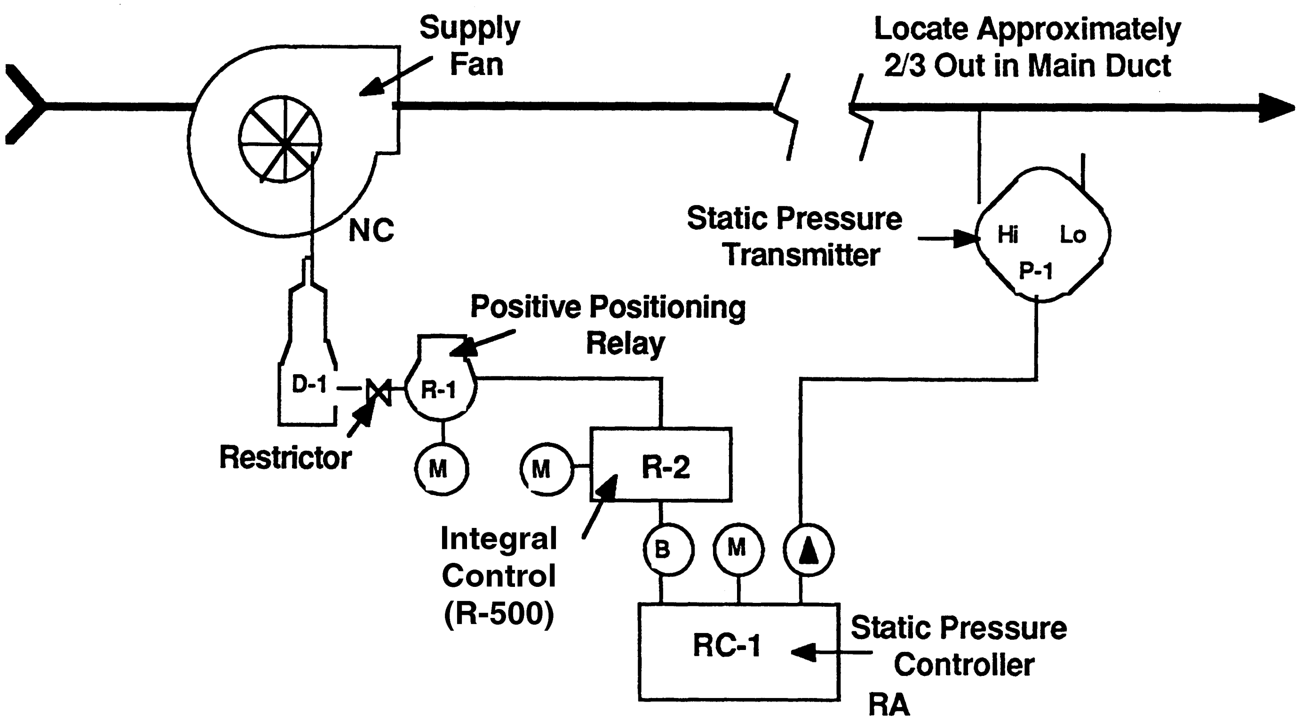
SUPPLY FAN VOLUME CONTROL
FIGURE 6-22

The purpose of fan volume control is to maintain a preset system static pressure. As shown in Figure 6-22, static pressure is sensed with a static pressure transmitter (P-1) by referencing atmospheric static pressure and a point where one-half of the system pressure loss has occurred. This will typically be two-thirds of the distance downstream from the fan in the main supply duct. The transmitter signal is sent to the reverse acting receiver-controller (RC-1) which modulates the normally closed supply fan vortex damper (D-1) to maintain the desired static pressure level.

A common problem is that the inlet vanes dampers may overshoot the control point, and some systems are so sensitive that hunting problems may occur even though the receiver-controller is set at its widest proportional band setting. It is recommended that a positive positioning relay be used on the actuator controlling the vortex damper. This provides accurate positioning in this high torque application and provides a span adjustment for the actuator. The actuator will typically have an 8-13 psig range. The adjustment of the positive positioning relay could widen this span to be more than 5 psig, and, therefore, widen the system throttling range and reduce the possibilities of hunting. To further insure against hunting, it may be necessary to install a restrictor to be used between the output of the positive positioning relay and the vortex damper actuator. This will slow down the actuator response for stable control.

Supply Fan Volume Control with Integral Control

A major consideration in controlling static pressure is hunting as the controller seeks its setpoint. To avoid this hunting problem, the throttling range is widened until the hunting ceases. If the throttling range is too wide, an objectionable offset to the setpoint could occur.



SUPPLY FAN VOLUME CONTROL WITH INTEGRAL CONTROL

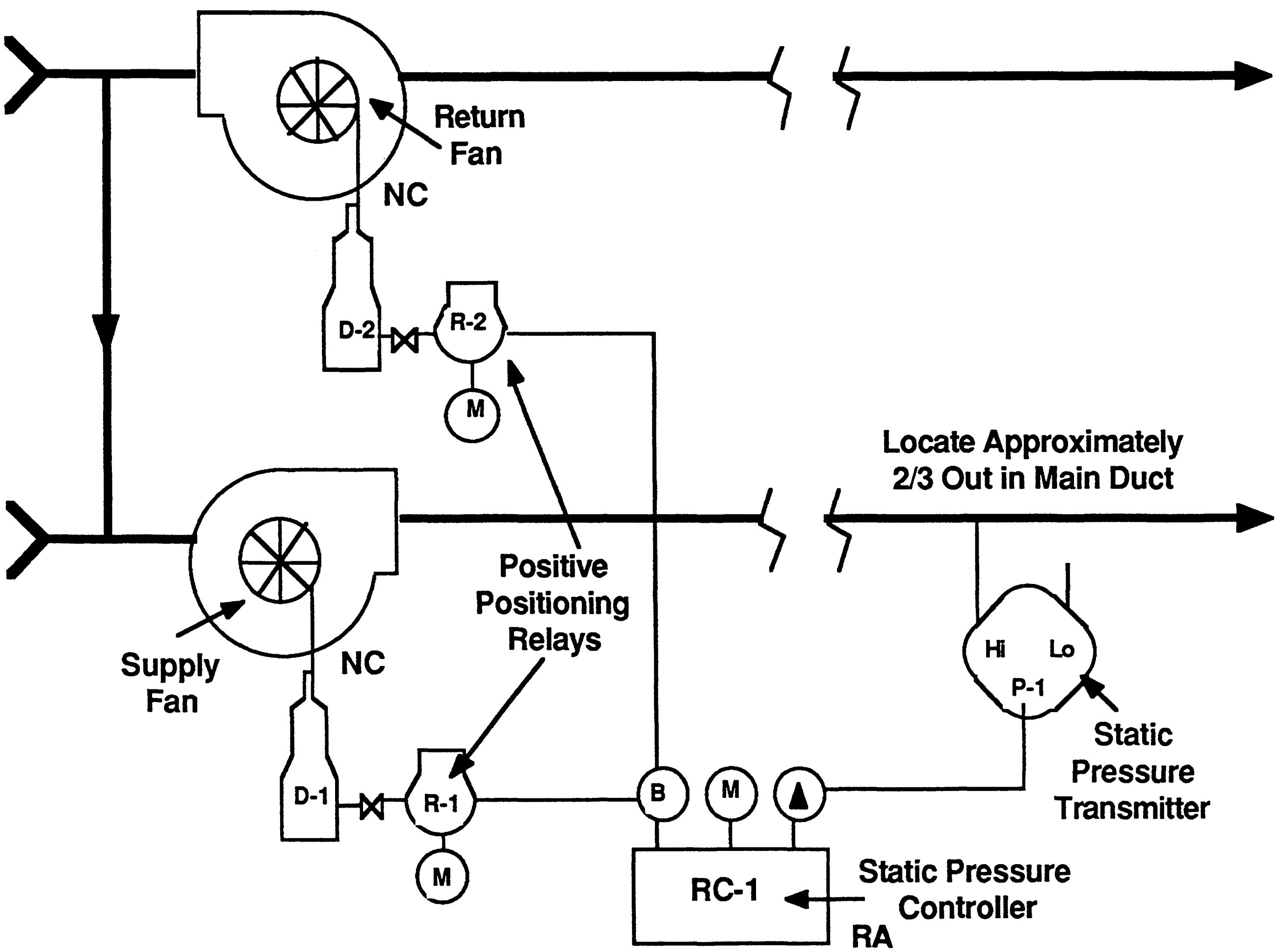
FIGURE 6-23

With the addition of an integral control relay, as shown in Figure 6-23, the offset problem can be substantially overcome. This relay (R-2) provides integral control to the proportional control of the receiver-controller. Although the offset may not be totally eliminated, it will be reduced substantially.

As explained in the previous example, the problem remains that the inlet vanes may overshoot their control point. By placing a restrictor between the branch output of the positive positioning relay and the actuator, the actuator will be slowed down which will allow for a more stable control to be achieved.

In order for the integral control relay to function properly, the system must be stable. If the system is hunting, the installation of an integral control relay will not solve the problem. The hunting problem must be solved by widening the throttling range and the use of the restrictor as described previously. Then the integral control relay may be used to reduce the offset that will occur so that precise control will be established.

<u>Open Loop Supply and Return Fan Capacity Control</u>



OPEN LOOP SUPPLY AND RETURN FAN CAPACITY CONTROL

FIGURE 6-24

The control system shown in Figure 6-24 is similar to the basic supply fan capacity control system, except that the control signal modulates the supply fan and return fan vortex dampers. The objectives are to maintain the supply static pressure at a preset level as system demand varies, and to maintain a fixed volume differential between the supply and return fans to maintain building static pressure at a fixed level. This method is often used because of its simplicity.

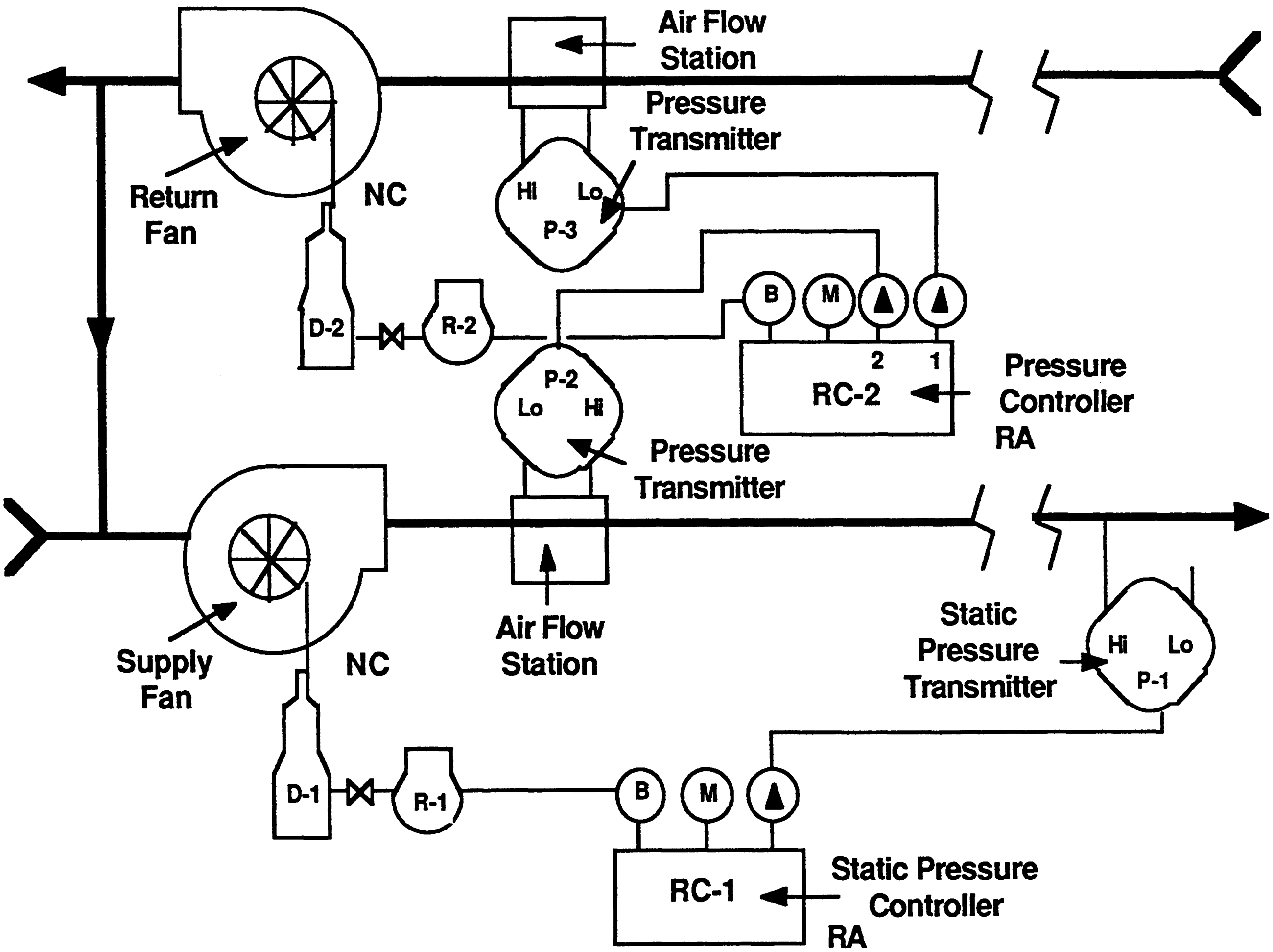
The static pressure transmitter (P-1) signal is sent to a reverse acting receiver-controller (RC-1) which modulates the supply fan vortex damper (D-1) to maintain static pressure at a preset level as system demand varies. The receiver-controller also modulates the return fan vortex damper (D-2) to maintain a relatively constant CFM differential between the supply and return fans. This is accomplished by using a damper actuator with a positive positioning relay on it to adjust the start point and operating span of the return fan vortex damper at minimum and maximum flow conditions. This adjustment ensures that the desired CFM differential is present at both ends of the demand curve, and assumes that the fan curves are matched closely enough to minimize errors as the flow modulates from maximum to minimum.

The accuracy of this control method in maintaining a fixed differential between supply and return fans depends on linear damper characteristics and matching fan curves. This may be difficult to obtain. However, this mismatch is not usually serious if the fans are of the same type. The fans are sized to operate at approximately the same percent of flow on their curves, and system flow reduction is limited to approximately 50%.

Should the system load vary significantly between major zones in the supply system, the return system resistance may not vary in direct proportion to supply system resistance. This control method does not sense the effect of resistance variations between supply and return systems. Therefore, the building pressure may vary when major load variations occur.

Again, on this system it is recommended that restrictors be installed between the positive positioning relays (R-1 and R-2) and the damper actuators to slow down the actuator movement.

Closed Loop Control System for Volumetric Synchronization of Supply and Return Fans



CLOSED LOOP CONTROL SYSTEM FOR VOLUMETRIC SYNCHRONIZATION OF SUPPLY AND RETURN FANS

FIGURE 6-25

The objective of this control scheme is to maintain a fixed volume differential (CFM) between the supply and return fans in order to establish the desired building pressurization level. Figure 6-25 control scheme illustrates the most precise means of accomplishing this objective.

System static pressure is controlled using the supply fan control method shown in Figures 6-22 and 6-23. This ensures that supply duct static pressure is maintained at a predetermined level sufficient for proper operation of the VAV terminal units under all system demand conditions.

Actual supply and return air volumes are used to control the return air fan, therefore, eliminating fan curve mismatch and system imbalance effects.

An airflow measuring station senses pressure in the return air duct. A pressure transmitter (P-3) converts this to an output signal that is sent to the pressure controller (RC-2) which modulates the return fan vortex damper (D-2) to maintain the return airflow rate at a predetermined CFM less than the supply airflow rate. A second airflow station with its pressure transmitter (P-2) measure the supply air pressure. This signal is used to reset the pressure controller (RC-2) modulating the return fan damper (D-2) so that the return fan CFM tracks the supply fan CFM as system demand varies. The percent authority setting of the pressure controller (RC-2) compensates for differing supply and return air duct areas and establishes the desired reset schedule. Differing supply and return air duct areas must be considered when comparing volume flow rates in two or more ducts.

A key to satisfactory fan capacity control is providing the precise control system response which is fast enough to prevent excess offset, but slow enough to avoid hunting. In practice, this may be difficult to achieve since system response time is difficult to predict. To match control response to system response, begin with a wide receiver-controller proportional band setting. Decrease this setting until hunting occurs, then widen setting until hunting ceases. This procedure often results in a proportional band setting so wide as to produce objectionable offset. If objectionable offset occurs, an automatic reset relay may be added as shown in Figure 6-23. The automatic reset relay allows the proportional band of the receiver-controller to be set as wide as necessary to prevent hunting. The automatic reset relay eliminates 90% of the offset.

Another problem mentioned previously is that inlet vanes dampers often overshoot their control point. Some systems are so sensitive that hunting still occurs at the widest proportional band setting of the receiver-controller. If so, a restrictor must be used, as shown, to slow down actuator response until stable control is achieved. Control stability is mandatory prior to applying automatic reset. When using the positive positioning relay, place the restrictor between the positive positioner output and the actuator.

- GLOSSARY -

GLOSSARY OF TERMS

TABLE OF CONTENTS

- GLOSSARY OF TERMS -

Absolute Pressure (PSIA): The sum of both atmospheric pressure (14.7) and gauge pressure (psig). **Example:** If a pneumatic gauge indicates 8 psig, the absolute pressure will be 22.7 psia (8 + 14.7).

Action: The direction of magnitude change of the output of a controller with respect to the change in the variable that is being sensed. **Example:**

Direct Action:	Variable increases, output increases.
Reverse Action:	Variable increases, output decreases.

Actuator: A device which is mechanically linked to a damper and positions the damper to regulate the flow of air; or is mounted on a valve and repositions the valve to regulate the flow of steam or water.

Ambient Temperature: Temperature of the medium (typically air) that surrounds an object.

Auxiliary Device: A control module which is generally placed between the controller and actuator that modifies the controller signal in some manner before the signal reaches the actuator. **Examples:** Relays and switches.

Averaging Element: A sensing device which is approximately 20 feet long that can extend across the entire duct and sense the average temperature of the gradient that may occur.

Averaging Relay: An auxiliary device that accepts the signals from two controllers and produces a signal that is the average of the two values received from the controllers.

Ball Flapper: An arrangement used in transmitters that varies the exhaust stream by repositioning a small sphere over an orifice shaped opening. This produces output linearity as the sensing element varies its pressure on the sphere.

- GLOSSARY OF TERMS -

Bias Start Relay: An auxiliary device that accepts a signal from a controller and delays the output until the signal from the controller reaches the setpoint of the relay.

Bleed Control: A pneumatic control arrangement in which the main air to the controller passes through a restrictor; and after passing this point, it becomes the branch line pressure.

Branch Lines: The tubing in a pneumatic control system which carries the output signal from controllers to auxiliary devices or actuators.

British Thermal Unit (BTU): The amount of heat required to change the temperature of one pound of water, one degree Fahrenheit.

Calibration Point: The output pressure of a controller when the set point and control point are equal. **Examples:** Normally this would be **[8]** psig (midpoint of controller output **[3 to 13]** psig). It may also be midpoint of actuator ranges: 3 - 8 = **[5.5]**, 5 - 10 = **[7.5]**, 8 - 13 = **[10.5]**, etc.

CFM: A unit of measure used to quantify air flow. The letters stand for Cubic Feet per Minute.

CIM: A unit of measure used to quantify air flow. The letters stand for Cubic Inches per Minute.

Close-Off: The maximum allowable pressure drop to which a valve may be subjected while fully closed.

Constant Volume Control: An air system in which the total quantity of air flow remains the same. The load is matched by changing the temperature of the air delivered.

Control Medium: The agent that is manipulated by the controlled device which, in turn, serves as the transport system of BTU's. **Examples:** Chilled water, hot water, steam, electricity.

Control Point The actual value of the controlled variable which the controller operates to maintain (under any fixed set of conditions).

- GLOSSARY OF TERMS -

Controlled Device: An apparatus that receives the signal from a controller and positions the damper or valve to match the capacity to the load. **Example:** Motorized damper or valve.

Controller: A device that receives a signal from a remote sensor or from its own integral sensor and then produces an output signal to a controlled device. **Examples:** Thermostats and Receiver-Controllers.

Critical Pressure Drop: The maximum pressure drop that should be allowed in order to minimize noise and is expressed in absolute pressure at the inlet port of a valve.

Day/Night Thermostat: A pneumatic thermostat which can be indexed by a change in main pressure to control at one setpoint during the day and another setpoint during the night. **Example:** 15 psig main during the day and 20 psig main during the night. (Synonymous to night setback or set-up).

Dew Point Temperature: The air temperature at which the condensation of water vapor begins as the air is cooled.

Differential: The change in the variable required to cause the contacts of a controller to move from one pole and contact to be made at the opposite pole.

Differential Pressure Control: A system in which two pressure sensors transmit their respective signals to a controller; the controller, in turn, produces an output to the controlled device that will vary in accordance with the difference of the two sensed pressures.

Direct Acting (DA): The action of a controller that produces an increase in output pressure as the variable increases.

Diverting Valve: A three-way valve which has one inlet, two outlets, and can direct full flow to either outlet or proportion the flow between the two outlets.

- GLOSSARY OF TERMS -

Dry Bulb Temperature: The temperature of air as indicated on a standard thermometer.

Dual Thermostat: A two-temperature thermostat, equivalent to two separate thermostats under one cover, which has two setpoints and two different outputs.

Electric-Pneumatic Switch (EP): An electrically operated air flow switch with normally closed and normally opened inputs which lead to a common output. Also known as solenoid air valve.

Equal Percentage Valve: A flow characteristic through a valve in which the seat-plug configuration is such that each equal increment of stroke movement will produce a change in flow equal to the continuously increasing increase over the linear increase.

Fail Safe: A characteristic of an actuator in which it returns to a known position upon a loss of control signal.

Feedback: A design feature of proportional control in which the controller receives a signal from the controlled device, thereby telling the controller what position the controlled device has assumed.

Finish Point: The pressure necessary to completely compress the spring of an actuator and cause the actuator to complete its stroke. **Example:** An actuator with a 5-10 psi spring range, the "10" is the finish point.

Firestat: A controller generally located in the return air, that will turn off the fan of air handling units when the temperature increases above setpoint.

Flow Coefficient (Cv): The flow rate in GPM of $60^{o}F$ water that will pass through a valve with a one pound pressure drop.

- GLOSSARY OF TERMS -

FPM: A unit of measure to quantify the velocity of air flow. The letters stand for Feet Per Minute.

Freezestat: A controller, located in the discharge of a water or steam coil, that will turn off the fan of air handling units when the temperature decreases below setpoint.

Gauge Pressure (PSIG): The amount of pressure above atmospheric pressure.

GPM: A unit of measure to quantify water flow. The letters stand for Gallons Per Minute.

High Limit: A controller used to prevent the variable it is sensing from increasing above a dangerous or undesirable condition.

Humidity Controller: A device which senses and controls the moisture content of air.

Hunting: The action of a pneumatic controller which causes the controlled device to continuously travel from one end of its stroke to the other.

Hydronics: The science dealing with the control of and use of water as a heat transfer medium in air conditioning systems.

Immersion Transmitter: A pneumatic sensor, with an extended element, which can be inserted into a well in order to sense the temperature in liquid lines.

Linear Valve: A flow characteristic through a valve in which the seat-plug configuration is such that each equal increment of stroke movement will produce a change in flow equal to the stroke movement.

Low Limit: A controller used to prevent the variable it is sensing from decreasing below a dangerous or undesirable condition.

Mains: The tubing in a pneumatic control system which carries the supply pressure from the air supply equipment to the controllers.

- GLOSSARY OF TERMS -

Manual Switch: An auxiliary device used to adjust the air pressure to a controlled device from zero to full supply pressure.

Master Controller: A controller which senses the condition of one variable and sends its signal to a second controller which changes the setpoint of the second controller.

Mixing Valve: A three-way valve which has two inlets, one outlet, and can direct full flow from either inlet or proportion the flow from the two inlets.

Motorized Damper: A damper which has a pneumatic actuator linked to it in such a manner as normally open or normally closed.

Motorized Valve: A valve body which has a pneumatic actuator mounted to it; the selection of the valve body determines whether the valve is normally open or normally closed.

Normally Closed (N.C.): A controlled device that reduces the flow of the controlled medium as the branch pressure is reduced, and will go to shut-off if the branch pressure goes to zero.

Normally Open (N.O.): A controlled device that increases the flow of the controlled medium as the branch pressure is reduced, and will go to full-flow if the branch pressure goes to zero.

Offset: The amount of difference between control point and setpoint in a proportional control system that is a result of the load being greater or less than 50% of capacity.

Paralleling: A control arrangement in which several actuators move through their stroke in unison as the signal from the controller changes, and this arrangement is derived by using actuators with the same spring ranges.

Percent Authority: The adjustment of a receiver-controller which determines the effect of the reset signal of the secondary transmitter as a percentage of the signal of the primary transmitter.

- GLOSSARY OF TERMS -

Pneumatic-Electric Switch (PE): An air pressure operated switch in which the contacts are made or broken in order to operate electrical devices in a pneumatic control system.

Positive Positioning: The characteristic of a controlled device in which it has the maximum force available at any point of the stroke.

Positive Positioning Relay: An auxiliary device that accepts a signal from a controller and sends a signal on to the actuator at a magnitude, up to main pressure, to extend the actuator to the point that the controller signal is calling for, and will decrease the pressure to the actuator to zero to retract the actuator to the point the controller signal is calling for.

Power Stroke: The extension of the actuator shaft as a result of the signal from the controller exceeding the opposing force of the spring.

Pressure Drop: The amount of pressure decrease between any two points in a system.

Proportional Control: A mode of control in which the controlled device may assume any position from fully closed to fully open, depending on the load at any given point in time; and the controlled device will be repositioned to a different position at the same ratio of change as the change in the variable; and a signal will be sent from the controlled device to the controller indicating what position the controlled device has assumed.

Quick Opening: A flow characteristic through a valve whereby the maximum flow is approached rapidly as the valve begins to open.

Range: (1) The minimum to maximum setpoint capability of a controller, (2) the minimum to maximum sensing capability of a transmitter, or (3) the start point to finish point of an actuator.
Examples:
Controller - 55-85°F
Transmitter - 40-240°F
Actuator - 5-10 psi.

- GLOSSARY OF TERMS -

Receiver-Controller: A device which receives the small signal changes from transmitters and amplifies these small change to a 3-13 psi output to the controlled device.

Relative Humidity: The ratio of the amount of moisture that is present in the air to the amount that can be in the air at that temperature.

Relay: An auxiliary device which receives a signal from the controller, conditions that signal in some manner, and then sends that signal on to a controlled device. **Examples:** Reversing relay, bias start relay or positive positioner.

Relay Control: A pneumatic control arrangement in which the main air to the controller passes through a valving mechanism to the controlled device. The valving mechanism is operated by the pressure variation of the pilot chamber.

Remote Bulb Thermostat: A thermostat in which the sensing element is filled with a fluid which is connected to a bellows by a capillary, and the expansion and contraction of the fluid is expressed on the bellows to regulate the rate of bleed at the bleed port.

Reset Control: The changing of the setpoint of a controller, from a signal, from a second controller or transmitter as a result of a change in the variable that the second controller is measuring.

Restrictor: A device which has a minute opening (such as .005" or .0075") which changes the velocity pressure of the air line to static pressure.

Return Stroke: The retraction of the actuator shaft as a result of the signal from the controller decreasing below the opposing force of the spring.

Reverse Acting (RA): The action of a controller that produces a decrease in output pressure as the variable increases.

- GLOSSARY OF TERMS -

Reversing Relay: An auxiliary device that produces a decrease in the output pressure to the controlled device as the input from the controller increases.

Selector Relay: An auxiliary device that chooses the higher or lower (depending on specific model) signal from several controllers and then passes that signal on to the controlled device.

Self-Actuated Control: A type of control system in which the source of energy is derived from the variable under control, rather than from an external source.

Sequencing: A control arrangement in which several actuators move through their stroke in succession as the signal from the controller changes. This arrangement is derived by using actuators with different spring ranges (such as 3-8 and 8-13) or by the use of a pneumatic relay.

Set Point: The position to which the control point setting mechanism is set.

Single-Seated Valve: A valve configuration in which there is one valve plug that comes to rest on one surface to obtain close-off.

Span: (1) The difference between the minimum and maximum setpoint capability of a controller, (2) the difference between the minimum to maximum sensing capabilitly of a transmitter, or (3) the difference between the start point and finish point of an actuator.

Examples:

Controller range - 55-85^{O}F, span is 30^{O}F;
Transmitter range - 40-240^{O}F, span is 200^{O}F;
Actuator range - 5-10 psi, span is 5 psi.

Start Point: The pressure necessary to begin compressing the spring of an actuator, therefore, causing the actuator to begin its stroke. **Example:** An actuator with a 5-10 psi spring range, the "5" is the start point.

- GLOSSARY OF TERMS -

Static Pressure: The force per unit area caused by the bursting force of the air against the walls of the duct as the air is moved through the duct.

Strap-On Thermostat: A controller designed for mounting on and sensing the temperature of a surface, such as that of a pipe.

Submaster Controller: A controller which receives a signal from one controller that changes the setpoint of this controller.

Summer/Winter: A pneumatic thermostat which can be indexed by a change in main pressure to control at one action during the summer and another action during the winter. This is necessary because it is controlling the same controlled device. **Example:**

15 psi during summer - reverse action;
20 psi during winter - direct action.

Switching Relay: An auxiliary device that chooses either of two signals to be the output to the controlled device, and the choice is made based on the signal received by the pilot chamber.

Throttling Range (Controller): The amount of change of the variable necessary for the controller to produce an output change of 3-13 psig.

Throttling Range (System): The amount of change of the variable necessary for the controller to drive the actuator through its complete stroke.

Transmitter: A device that measures the variable and sends a 3-15 psig signal to a receiver-controller and gauge, therefore, accomplishing control and indication. (A remote sensor)

Two-Position Control: A method of control in which the control device is either 100% open or closed; therefore, the controlled medium is flowing at these respective rates. Also known as On-Off control.

- GLOSSARY OF TERMS -

Vacuum Pressure: Pressure below atmospheric pressure, similar to the suction side of a centrifugal refrigeration machine, or the design of a steam heating system.

Variable: The temperature, humidity, or pressure that the control system operates to maintain.

Wet Bulb Temperature: The air temperature as indicated by a thermometer with a wetted wick, as the air is passed over the wick at a velocity of approximately 1,000 FPM.

Zone Control: An area being controlled by a controller with its respective controlled device and control medium.

- APPENDIX -

APPENDIX

TABLE OF CONTENTS

SECTION **PAGE**

APPENDIX

A source of clean, dry, oil-free air is essential to the proper operation of a pneumatic control system. The devices within the system should operate virtually trouble-free, provided a preventive maintenance program is followed. This section provides a suggested schedule for maintaining the air supply station and devices within the system.

MAINTENANCE SCHEDULE

Frequency	Service Required
Daily	Drain water from receiver tank of compressor, filter bowl and air lines that have drain cocks.
Weekly	Check compressor crankcase oil level.
Monthly	Check compressor safety relief valve. Check pressure-reducing valve setting. Clean intake air filter (felt or screen types). Check compressor belt tension. Check pump up time of compressor.
Once Each 6 Months	Clean fins of compressor drive wheel. Change compressor crankcase oil. Oil compressor motor. Check compressor pressure switch setting. Check calibration and operation of receiver-controllers, pressure controllers, thermostats and humidistats. Clean sensing elements of transmitters, thermostats and humidistats. Lubricate dampers and check for damper close-off. Check calibration and operation of relays. Check operation of EP, PE and manual switches.

MAINTENANCE SCHEDULE - Con't

Frequency	Service Required
Once Each Year	Lubricate packing and adjust packing or repack valves. Check valves for tight close-off. Replace cartridge of compressor intake.

GENERAL CALIBRATION PROCEDURE FOR THERMOSTATS

1. Measure the temperature of the variable with accurate thermometer.
2. Adjust setpoint to match temperature measured in Step No. 1.
3. Adjust calibration screw until branch line pressure measures 8 psig or mid-actuator range pressure.
4. Adjust setpoint to value desired.

GENERAL CALIBRATION PROCEDURE FOR SINGLE INPUT RECEIVER-CONTROLLERS

1. Measure the temperature of the variable that the transmitter is sensing with accurate thermometer.
2. Adjust proportional band setting to calculated valve.
3. Adjust calibration dial until branch line pressure measures 8 psig or mid-actuator range pressure.
4. Slide setpoint scale to match temperature measured in Step No. 1.
5. Adjust calibration dial until setpoint indicates value desired.

GENERAL CALIBRATION PROCEDURE FOR DUAL INPUT RECEIVER-CONTROLLERS

1. Measure the temperature of both variables with accurate thermometer.
2. Adjust proportional band setting to calculated value.
3. Adjust percent authority setting to calculated value.
4. Adjust calibration dial until branch line pressure measures 8 psig or mid-actuator range pressure.
5. Slide setpoint scale to match temperature measured at primary transmitter.
6. Adjust calibration dial until setpoint indicates desired value at primary transmitter under current conditions at secondary transmitter.

TABLE OF EQUIVALENT TEMPERATURES

Degree F	Degree C	Degree F	Degree C	Degree F	Degree C
-40	-40.0	- 1	-18.3	38	3.3
-39	-39.4	0	-17.7	39	3.8
-38	-38.8	1	-17.2	40	4.4
-37	-38.3	2	-16.6	41	5.0
-36	-37.7	3	-16.1	42	5.5
-35	-37.2	4	-15.5	43	6.1
-34	-36.6	5	-15.0	44	6.6
-33	-36.1	6	-14.4	45	7.2
-32	-35.5	7	-13.8	46	7.7
-31	-35.0	8	-13.3	47	8.3
-30	-34.4	9	-12.7	48	8.8
-29	-33.8	10	-12.2	49	9.4
-28	-33.3	11	-11.6	50	10.0
-27	-32.7	12	-11.1	51	10.5
-26	-32.2	13	-10.5	52	11.1
-25	-31.6	14	-10.0	53	11.6
-24	-31.1	15	- 9.4	54	12.2
-23	-30.5	16	- 8.8	55	12.7
-22	-30.0	17	- 8.3	56	13.3
-21	-29.4	18	- 7.7	57	13.8
-20	-28.8	19	- 7.2	58	14.4
-19	-28.3	20	- 6.6	59	15.0
-18	-27.7	21	- 6.1	60	15.5
-17	-27.2	22	- 5.5	61	16.1
-16	-26.6	23	- 5.0	62	16.6
-15	-26.1	24	- 4.4	63	17.2
-14	-25.5	25	- 3.8	64	17.7
-13	-25.0	26	- 3.3	65	18.3
-12	-24.4	27	- 2.7	66	18.8
-11	-23.8	28	- 2.2	67	19.4
-10	-23.3	29	- 1.6	68	20.0
- 9	-22.7	30	- 1.1	69	20.5
- 8	-22.2	31	- 0.5	70	21.1
- 7	-21.6	32	0.0	71	21.6
- 6	-21.1	33	0.5	72	22.2
- 5	-20.5	34	1.1	73	22.7
- 4	-20.0	35	1.6	74	23.3
- 3	-19.4	36	2.2	75	23.8
- 2	-18.8	37	2.7	76	24.4

TABLE OF EQUIVALENT TEMPERATURES - Con't

Degree F	Degree C	Degree F	Degree C	Degree F	Degree C
77	25.0	116	46.6	155	68.3
78	25.5	117	47.2	156	68.8
79	26.1	118	47.7	157	69.4
80	26.6	119	48.3	158	70.0
81	27.2	120	48.8	159	70.5
82	27.7	121	49.4	160	71.1
83	28.3	122	50.0	161	71.6
84	28.8	123	50.5	162	72.2
85	29.4	124	51.1	163	72.7
86	30.0	125	51.6	164	73.3
87	30.5	126	52.2	165	73.8
88	31.1	127	52.7	166	74.4
89	31.6	128	53.3	167	75.0
90	32.2	129	53.8	168	75.5
91	32.7	130	54.4	169	76.1
92	33.3	131	55.0	170	76.6
93	38.8	132	55.5	171	77.2
94	34.4	133	56.1	172	77.7
95	35.0	134	56.6	173	78.3
96	35.5	135	57.2	174	78.8
97	36.1	136	57.7	175	79.4
98	36.6	137	58.3	176	80.0
99	37.2	138	58.8	177	80.5
100	37.7	139	59.4	178	81.1
101	38.3	140	60.0	179	81.6
102	38.8	141	60.5	180	82.2
103	39.4	142	61.1	181	82.7
104	40.0	143	61.6	182	83.3
105	40.5	144	62.2	183	83.8
106	41.1	145	62.7	184	84.4
107	41.6	146	63.3	185	85.0
108	42.2	147	63.8	186	85.5
109	42.7	148	64.4	187	86.1
110	43.3	149	65.0	188	86.6
111	43.8	150	65.5	189	87.2
112	44.4	151	66.1	190	87.7
113	45.0	152	66.6	191	88.3
114	45.5	153	67.2	192	88.8
115	46.1	154	67.7	193	89.4

TABLE OF EQUIVALENT TEMPERATURES - Con't

Degree F	Degree C	Degree F	Degree C	Degree F	Degree C
194	90.0	210	98.8	226	107.7
195	90.5	211	99.4	227	108.3
196	91.1	212	100.0	228	108.8
197	91.6	213	100.5	229	109.4
198	92.2	214	101.1	230	110.0
199	92.7	215	101.6	231	110.5
200	93.3	216	102.2	232	111.1
201	93.8	217	102.7	233	111.6
202	94.4	218	103.3	234	112.2
203	95.0	219	103.8	235	112.7
204	95.5	220	104.4	236	113.3
205	96.1	221	105.0	237	113.8
206	96.6	222	105.5	238	114.4
207	97.2	223	106.1	239	115.0
208	97.7	224	106.6	240	115.5
209	98.3	225	107.2		

OUTPUT PRESSURE OF TEMPERATURE TRANSMITTERS
-40 TO 160°F

Temp. Degree F	Pressure	Temp. Degree F	Pressure	Temp. Degree F	Pressure
-40	3.00	- 4	5.16	32	7.32
-39	3.06	- 3	5.22	33	7.38
-38	3.12	- 2	5.28	34	7.44
-37	3.18	- 1	5.34	35	7.50
-36	3.24	0	5.40	36	7.56
-35	3.30	1	5.46	37	7.62
-34	3.36	2	5.52	38	7.68
-33	3.42	3	5.58	39	7.74
-32	3.48	4	5.64	40	7.80
-31	3.54	5	5.70	41	7.86
-30	3.60	6	5.76	42	7.92
-29	3.66	7	5.82	43	7.98
-28	3.72	8	5.88	44	8.04
-27	3.78	9	5.94	45	8.10
-26	3.84	10	6.00	46	8.16
-25	3.90	11	6.06	47	8.22
-24	3.96	12	6.12	48	8.28
-23	4.02	13	6.18	49	8.34
-22	4.08	14	6.24	50	8.40
-21	4.14	15	6.30	51	8.46
-20	4.20	16	6.36	52	8.52
-19	4.26	17	6.42	53	8.58
-18	4.32	18	6.48	54	8.64
-17	4.38	19	6.54	55	8.70
-16	4.44	20	6.60	56	8.76
-15	4.50	21	6.66	57	8.82
-14	4.56	22	6.72	58	8.88
-13	4.62	23	6.78	59	8.94
-12	4.68	24	6.84	60	9.00
-11	4.74	25	6.90	61	9.06
-10	4.80	26	6.96	62	9.12
- 9	4.86	27	7.02	63	9.18
- 8	4.92	28	7.08	64	9.24
- 7	4.98	29	7.14	65	9.30
- 6	5.04	30	7.20	66	9.36
- 5	5.10	31	7.26	67	9.42

OUTPUT PRESSURE OF TEMPERATURE TRANSMITTERS - Con't

-40 TO 160°F

Temp. Degree F	Pressure	Temp. Degree F	Pressure	Temp. Degree F	Pressure
68	9.48	99	11.34	130	13.20
69	9.54	100	11.40	131	13.26
70	9.60	101	11.46	132	13.32
71	9.66	102	11.52	133	13.38
72	9.72	103	11.58	134	13.44
73	9.78	104	11.64	135	13.50
74	9.84	105	11.70	136	13.56
75	9.90	106	11.76	137	13.62
76	9.96	107	11.82	138	13.68
77	10.02	108	11.88	139	13.74
78	10.08	109	11.94	140	13.80
79	10.14	110	12.00	141	13.86
80	10.20	111	12.06	142	13.92
81	10.26	112	12.12	143	13.98
82	10.32	113	12.18	144	14.04
83	10.38	114	12.24	145	14.10
84	10.44	115	12.30	146	14.16
85	10.50	116	12.36	147	14.22
86	10.56	117	12.42	148	14.28
87	10.62	118	12.48	149	14.34
88	10.68	119	12.54	150	14.40
89	10.74	120	12.60	151	14.46
90	10.80	121	12.66	152	14.52
91	10.86	122	12.72	153	14.58
92	10.92	123	12.78	154	14.64
93	10.98	124	12.84	155	14.70
94	11.04	125	12.90	156	14.76
95	11.10	126	12.96	157	14.82
96	11.16	127	13.02	158	14.88
97	11.22	128	13.08	159	14.94
98	11.28	129	13.14	160	15.00

OUTPUT PRESSURE OF TEMPERATURE TRANSMITTERS
40 TO 240^{o}F

Temp. Degree F	Pressure	Temp. Degree F	Pressure	Temp. Degree F	Pressure
40	3.00	77	5.22	114	7.44
41	3.06	78	5.28	115	7.50
42	3.12	79	5.34	116	7.56
43	3.18	80	5.40	117	7.62
44	3.24	81	5.46	118	7.68
45	3.30	82	5.52	119	7.74
46	3.36	83	5.58	120	7.80
47	3.42	84	5.64	121	7.86
48	3.48	85	5.70	122	7.92
49	3.54	86	5.76	123	7.98
50	3.60	87	5.82	124	8.04
51	3.66	88	5.88	125	8.10
52	3.72	89	5.94	126	8.16
53	3.78	90	6.00	127	8.22
54	3.84	91	6.06	128	8.28
55	3.90	92	6.12	129	8.34
56	3.96	93	6.18	130	8.40
57	4.02	94	6.24	131	8.46
58	4.08	95	6.30	132	8.52
59	4.14	96	6.36	133	8.58
60	4.20	97	6.42	134	8.64
61	4.26	98	6.48	135	8.70
62	4.32	99	6.54	136	8.76
63	4.38	100	6.60	137	8.82
64	4.44	101	6.66	138	8.88
65	4.50	102	6.72	139	8.94
66	4.56	103	6.78	140	9.00
67	4.62	104	6.84	141	9.06
68	4.68	105	6.90	142	9.12
69	4.74	106	6.96	143	9.18
70	4.80	107	7.02	144	9.24
71	4.86	108	7.08	145	9.30
72	4.92	109	7.14	146	9.36
73	4.98	110	7.20	147	9.42
74	5.04	111	7.26	148	9.48
75	5.10	112	7.32	149	9.54
76	5.16	113	7.38	150	9.60

OUTPUT PRESSURE OF TEMPERATURE TRANSMITTERS - Con't

40 TO 240°F

Temp. Degree F	Pressure	Temp. Degree F	Pressure	Temp. Degree F	Pressure
151	9.66	181	11.46	211	13.26
152	9.72	182	11.52	212	13.32
153	9.78	183	11.58	213	13.38
154	9.84	184	11.64	214	13.44
155	9.90	185	11.70	215	13.50
156	9.96	186	11.76	216	13.56
157	10.02	187	11.82	217	13.62
158	10.08	188	11.88	218	13.68
159	10.14	189	11.94	219	13.74
160	10.20	190	12.00	220	13.80
161	10.26	191	12.06	221	13.86
162	10.32	192	12.12	222	13.92
163	10.38	193	12.18	223	13.98
164	10.44	194	12.24	224	14.04
165	10.50	195	12.30	225	14.10
166	10.56	196	12.36	226	14.16
167	10.62	197	12.42	227	14.22
168	10.68	198	12.48	228	14.28
169	10.74	199	12.54	229	14.34
170	10.80	200	12.60	230	14.40
171	10.86	201	12.66	231	14.46
172	10.92	202	12.72	232	14.52
173	10.98	203	12.78	233	14.58
174	11.04	204	12.84	234	14.64
175	11.10	205	12.90	235	14.70
176	11.16	206	12.96	236	14.76
177	11.22	207	13.02	237	14.82
178	11.28	208	13.08	238	14.88
179	11.34	209	13.14	239	14.94
180	11.40	210	13.20	240	15.00

OUTPUT PRESSURE OF TEMPERATURE TRANSMITTERS
0 TO 100°F

Temp. Degree F	Pressure	Temp. Degree F	Pressure	Temp. Degree F	Pressure
0	3.00	35	7.20	69	11.28
1	3.12	36	7.32	70	11.40
2	3.24	37	7.44	71	11.52
3	3.36	38	7.56	72	11.64
4	3.48	39	7.68	73	11.76
5	3.60	40	7.80	74	11.88
6	3.72	41	7.92	75	12.00
7	3.84	42	8.04	76	12.12
8	3.96	43	8.16	77	12.24
9	4.08	44	8.28	78	12.36
10	4.20	45	8.40	79	12.48
11	4.32	46	8.52	80	12.60
12	4.44	47	8.64	81	12.72
13	4.56	48	8.76	82	12.84
14	4.68	49	8.88	83	12.96
15	4.80	50	9.00	84	13.08
16	4.92	51	9.12	85	13.20
17	5.04	52	9.24	86	13.32
18	5.16	53	9.36	87	13.44
19	5.28	54	9.48	88	13.56
20	5.40	55	9.60	89	13.68
21	5.52	56	9.72	90	13.80
22	5.64	57	9.84	91	13.92
23	5.76	58	9.96	92	14.04
24	5.88	59	10.08	93	14.16
25	6.00	60	10.20	94	14.28
26	6.12	61	10.32	95	14.40
27	6.24	62	10.44	96	14.52
28	6.36	63	10.56	97	14.64
29	6.48	64	10.68	98	14.76
30	6.60	65	10.80	99	14.88
31	6.72	66	10.92	100	15.00
32	6.84	67	11.04		
33	6.96	68	11.16		
34	7.08				

OUTPUT PRESSURE OF TEMPERATURE TRANSMITTERS
50 TO 150^{o}F

Temp. Degree F	Pressure	Temp. Degree F	Pressure	Temp. Degree F	Pressure
50	3.00	85	7.20	119	11.28
51	3.12	86	7.32	120	11.40
52	3.24	87	7.44	121	11.52
53	3.36	88	7.56	122	11.64
54	3.48	89	7.68	123	11.76
55	3.60	90	7.80	124	11.88
56	3.72	91	7.92	125	12.00
57	3.84	92	8.04	126	12.12
58	3.96	93	8.16	127	12.24
59	4.08	94	8.28	128	12.36
60	4.20	95	8.40	129	12.48
61	4.32	96	8.52	130	12.60
62	4.44	97	8.64	131	12.72
63	4.56	98	8.76	132	12.84
64	4.68	99	8.88	133	12.96
65	4.80	100	9.00	134	13.08
66	4.92	101	9.12	135	13.20
67	5.04	102	9.24	136	13.32
68	5.16	103	9.36	137	13.44
69	5.28	104	9.48	138	13.56
70	5.40	105	9.60	139	13.68
71	5.52	106	9.72	140	13.80
72	5.64	107	9.84	141	13.92
73	5.76	108	9.96	142	14.04
74	5.88	109	10.08	143	14.16
75	6.00	110	10.20	144	14.28
76	6.12	111	10.32	145	14.40
77	6.24	112	10.44	146	14.52
78	6.36	113	10.56	147	14.64
79	6.48	114	10.68	148	14.76
80	6.60	115	10.80	149	14.88
81	6.72	116	10.92	150	15.00
82	6.84	117	11.04		
83	6.96	118	11.16		
84	7.08				

OUTPUT PRESSURE OF TEMPERATURE TRANSMITTERS

50 TO 100°F

Temp. Degree F	Pressure	Temp. Degree F	Pressure	Temp. Degree F	Pressure
50	3.00	67	7.08	84	11.16
51	3.24	68	7.32	85	11.40
52	3.48	69	7.56	86	11.64
53	3.72	70	7.80	87	11.88
54	3.96	71	8.04	88	12.12
55	4.20	72	8.28	89	12.36
56	4.44	73	8.52	90	12.60
57	4.68	74	8.76	91	12.84
58	4.92	75	9.00	92	13.08
59	5.16	76	9.24	93	13.32
60	5.40	77	9.48	94	13.56
61	5.64	78	9.72	95	13.80
62	5.88	79	9.96	96	14.04
63	6.12	80	10.20	97	14.28
64	6.36	81	10.44	98	14.52
65	6.60	82	10.68	99	14.76
66	6.84	83	10.92	100	15.00

OUTPUT PRESSURE OF HUMIDITY TRANSMITTERS
10 TO 90% R. H.

% R. H.	Pressure	% R. H.	Pressure	% R. H.	Pressure
10	**3.00**	37	7.05	64	11.10
11	3.15	38	7.20	65	11.25
12	3.30	39	7.35	66	11.40
13	3.45	40	7.50	67	11.55
14	3.60	41	7.65	68	11.70
15	3.75	42	7.80	69	11.85
16	3.90	43	7.95	**70**	**12.00**
17	4.05	44	8.10	71	12.15
18	4.20	45	8.25	72	12.30
19	4.35	46	8.40	73	12.45
20	4.50	47	8.55	74	12.60
21	4.65	48	8.70	75	12.75
22	4.80	49	8.85	76	12.90
23	4.95	**50**	**9.00**	77	13.05
24	5.10	51	9.15	78	13.20
25	5.25	52	9.30	79	13.35
26	5.40	53	9.45	80	13.50
27	5.55	54	9.60	81	13.65
28	5.70	55	9.75	82	13.80
29	5.85	56	9.90	83	13.95
30	**6.00**	57	10.05	84	14.10
31	6.15	58	10.20	85	14.25
32	6.30	59	10.35	86	14.40
33	6.45	60	10.50	87	14.55
34	6.60	61	10.65	88	14.70
35	6.75	62	10.80	89	14.85
36	6.90	63	10.95	**90**	**15.00**

OUTPUT PRESSURE OF PRESSURE TRANSMITTERS
-.25 TO 6" H_20
* SETPOINT = .75" H_20

" H_20	Pressure	" H_20	Pressure	" H_20	Pressure
-.250	3.000	.439	7.169	1.101	11.171
-.223	3.167	.466	7.336	1.128	11.338
-.196	3.334	.493	7.503	1.155	11.505
-.169	3.501	.520	7.670	1.182	11.672
-.142	3.668	.547	7.837	1.209	11.839
-.115	3.835	.574	8.004	1.250	12.000
-.088	4.002	.601	8.171	1.277	12.167
-.061	4.169	.628	8.338	1.304	12.334
-.034	4.336	.655	8.505	1.331	12.501
-.007	4.503	.682	8.672	1.358	12.668
.020	4.670	.709	8.839	1.385	12.835
.047	4.837	.750	9.000	1.412	13.002
.074	5.004	.777	9.167	1.439	13.169
.101	5.171	.804	9.334	1.466	13.336
.128	5.338	.831	9.501	1.493	13.503
.155	5.505	.858	9.668	1.520	13.670
.182	5.672	.885	9.835	1.547	13.837
.209	5.839	.912	10.002	1.574	14.004
.250	6.000	.939	10.169	1.601	14.171
.277	6.167	.966	10.336	1.628	14.338
.304	6.334	.993	10.503	1.655	14.505
.331	6.501	1.020	10.670	1.682	14.672
.358	6.668	1.047	10.837	1.709	14.839
.385	6.835	1.074	11.004	1.750	15.000
4.12	7.002				

* Setpoint is adjustable between -.25" H_20 and 6" H_20. Only 4 setpoints are shown indicating inches of W.C. pressure inputs and the respective output pressure in psig.

OUTPUT PRESSURE OF PRESSURE TRANSMITTERS
-.25 TO 6" H_20
*** SETPOINT = 2" H_20**

" H_20	Pressure	" H_20	Pressure	" H_20	Pressure
1.000	3.000	1.689	7.169	2.351	11.171
1.027	3.167	1.716	7.336	2.378	11.338
1.054	3.334	1.743	7.503	2.405	11.505
1.081	3.501	1.770	7.670	2.432	11.672
1.108	3.668	1.797	7.837	2.459	11.839
1.135	3.835	1.824	8.004	2.500	12.000
1.162	4.002	1.851	8.171	2.527	12.167
1.189	4.169	1.878	8.338	2.554	12.334
1.216	4.336	1.905	8.505	2.581	12.501
1.243	4.503	1.932	8.672	2.608	12.668
1.270	4.670	1.959	8.839	2.635	12.835
1.297	4.837	2.000	9.000	2.662	13.002
1.324	5.004	2.027	9.167	2.689	13.169
1.351	5.171	2.054	9.334	2.716	13.336
1.378	5.338	2.081	9.501	2.743	13.503
1.405	5.505	2.108	9.668	2.770	13.670
1.432	5.672	2.135	9.835	2.797	13.837
1.459	5.839	2.162	10.002	2.824	14.004
1.500	6.000	2.189	10.169	2.851	14.171
1.527	6.167	2.216	10.336	2.878	14.338
1.554	6.334	2.243	10.503	2.905	14.505
1.581	6.501	2.270	10.670	2.932	14.672
1.608	6.668	2.297	10.837	2.959	14.839
1.635	6.835	2.324	11.004	3.000	15.000
1.662	7.002				

* Setpoint is adjustable between -.25" H_20 and 6" H_20. Only 4 setpoints are shown indicating inches of W.C. pressure inputs and the respective output pressure in psig.

OUTPUT PRESSURE OF PRESSURE TRANSMITTERS
-.25 TO 6" H_20
*** SETPOINT = 3.5" H_20**

" H_20	Pressure	" H_20	Pressure	" H_20	Pressure
2.500	3.000	3.189	7.169	3.851	11.171
2.527	3.167	3.216	7.336	3.878	11.338
2.554	3.334	3.243	7.503	3.905	11.505
2.581	3.501	3.270	7.670	3.932	11.672
2.608	3.668	3.297	7.837	3.959	11.839
2.635	3.835	3.324	8.004	4.000	12.000
2.662	4.002	3.351	8.171	4.027	12.167
2.689	4.169	3.378	8.338	4.054	12.334
2.716	4.336	3.405	8.505	4.081	12.501
2.743	4.503	3.432	8.672	4.108	12.668
2.770	4.670	3.459	8.839	4.135	12.835
2.797	4.837	3.500	9.000	4.162	13.002
2.824	5.004	3.527	9.167	4.189	13.169
2.851	5.171	3.554	9.334	4.216	13.336
2.878	5.338	3.581	9.501	4.243	13.503
2.905	5.505	3.608	9.668	4.270	13.670
2.932	5.672	3.635	9.835	4.297	13.837
2.959	5.839	3.662	10.002	4.324	14.004
3.000	6.000	3.689	10.169	4.351	14.171
3.027	6.167	3.716	10.336	4.378	14.338
3.054	6.334	3.743	10.503	4.405	14.505
3.081	6.501	3.770	10.670	4.432	14.672
3.108	6.668	3.797	10.837	4.459	14.839
3.135	6.835	3.824	11.004	4.500	15.000
3.162	7.002				

* Setpoint is adjustable between -.25" H_20 and 6" H_20. Only 4 setpoints are shown indicating inches of W.C. pressure inputs and the respective output pressure in psig.

OUTPUT PRESSURE OF PRESSURE TRANSMITTERS
-.25 TO 6" H_20
*** SETPOINT = 5" H_20**

" H_20	Pressure	" H_20	Pressure	" H_20	Pressure
4.000	**3.000**	4.689	7.169	5.351	11.171
4.027	3.167	4.716	7.336	5.378	11.338
4.054	3.334	4.743	7.503	5.405	11.505
4.081	3.501	4.770	7.670	5.432	11.672
4.108	3.668	4.797	7.837	5.459	11.839
4.135	3.835	4.824	8.004	**5.500**	**12.000**
4.162	4.002	4.851	8.171	5.527	12.167
4.189	4.169	4.878	8.338	5.554	12.334
4.216	4.336	4.905	8.505	5.581	12.501
4.243	4.503	4.932	8.672	5.608	12.668
4.270	4.670	4.959	8.839	5.635	12.835
4.297	4.837	**5.000**	**9.000**	5.662	13.002
4.324	5.004	5.027	9.167	5.689	13.169
4.351	5.171	5.054	9.334	5.716	13.336
4.378	5.338	5.081	9.501	5.743	13.503
4.405	5.505	5.108	9.668	5.770	13.670
4.432	5.672	5.135	9.835	5.797	13.837
4.459	5.839	5.162	10.002	5.824	14.004
4.500	**6.000**	5.189	10.169	5.851	14.171
4.527	6.167	5.216	10.336	5.878	14.338
4.554	6.334	5.243	10.503	5.905	14.505
4.581	6.501	5.270	10.670	5.932	14.672
4.608	6.668	5.297	10.837	5.959	14.839
4.635	6.835	5.324	11.004	**6.000**	**15.000**
4.662	7.002				

* Setpoint is adjustable between -.25" H_20 and 6" H_20. Only 4 setpoints are shown indicating inches of W.C. pressure inputs and the respective output pressure in psig.

OUTPUT PRESSURE OF TEMPERATURE TRANSMITTERS (CONDENSED)

Output (psig)	Temperature Range -40 to 160	40 to 240	0 to 100	50 to 150	50 to 100
3.00	-40	40	0	50	50
3.24	-36	44	2	52	51
3.48	-32	48	4	54	52
3.72	-28	52	6	56	53
3.96	-24	56	8	58	54
4.20	-20	64	12	62	56
4.44	-16	64	12	62	56
4.68	-12	68	14	64	57
4.92	-8	72	16	66	58
5.16	-4	76	18	68	59
5.40	0	80	20	70	60
5.64	4	84	22	72	61
5.88	8	88	24	74	62
6.12	12	92	26	76	63
6.36	16	96	28	78	64
6.60	20	100	30	80	65
6.84	24	104	32	82	66
7.08	28	108	34	84	67
7.32	32	112	36	86	68
7.56	36	116	38	88	69
7.80	40	120	40	90	70
8.04	44	124	42	92	71
8.28	48	128	44	94	72
8.52	52	132	46	96	73
8.76	56	136	48	98	74
9.00	60	140	50	100	75
9.24	64	144	52	102	76
9.48	68	148	54	104	77
9.72	72	152	56	106	78
9.96	76	156	58	108	79
10.20	80	160	60	110	80
10.44	84	164	62	112	81
10.68	88	168	84	114	82
10.92	92	172	66	116	83
11.16	96	176	68	118	84
11.40	100	180	70	120	85
11.64	104	184	72	122	86

OUTPUT PRESSURE OF TEMPERATURE TRANSMITTERS (CONDENSED)

Temperature Range

Output (psig)	-40 to 160	40 to 240	0 to 100	50 to 150	50 to 100
11.88	108	188	74	124	87
12.12	112	192	76	126	88
12.36	116	196	78	128	89
12.60	120	200	80	130	90
12.84	124	204	82	132	91
13.08	128	208	84	134	92
13.32	132	212	86	136	93
13.56	136	216	88	138	94
13.80	140	220	90	140	95
14.04	144	224	92	142	96
14.28	148	228	94	144	97
14.52	152	232	96	146	98
14.76	156	236	98	148	99
15.00	160	240	100	150	100

CONTROLLER OUTPUT PRESSURE CHANGE PER DEGREE F WITH VARIOUS THROTTLING RANGES

T. R.	PSIG CHANGE PER DEGREE F
2	5.00
3	3.33
4	2.50
5	2.00
6	1.66
7	1.42
8	1.25
9	1.11
10	1.00
11	.90
12	.83
13	.76
14	.71
15	.67

ABBREVIATIONS

ACU	=	Air Conditioning Unit
AHU	=	Air Handling Unit
C	=	Common
CR	=	Condensate Return
CWR	=	Chilled Water Return
CWS	=	Chilled Water Supply
DA	=	Direct Acting
DPC	=	Differential Pressure Controller
DPDT	=	Double Pole - Double Throw
EA	=	Exhaust Air
EP	=	Electric - Pneumatic Switch
FA	=	Fresh Air
HVU	=	Heating and Ventilating Unit
HVAC	=	Heating, Ventilating and Air Conditioning
HWR	=	Hot Water Return
HWS	=	Hot Water Supply
MA	=	Mixed Air
NC	=	Normally Closed
NO	=	Normally Open
OA	=	Outside Air
PA	=	Percent Authority
PB	=	Proportional Band
PE	=	Pneumatic Electric Switch
RA	=	Return Air
RA	=	Reverse Acting
RC	=	Receiver-Controller
SP	=	Setpoint
SPST	=	Single Pole - Single Throw
SS	=	Steam Supply
TR	=	Throttling Range
VAV	=	Variable Air Volume

TAC Training Center
1354 Clifford Avenue
P.O. BOX 2940
Loves Park, IL 61132-2940
www.tac.com

F-22763-3

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