Pneumatic Controls for HVAC

FUNDAMENTALS OF PNEUMATIC CONTROLS

Schneider Electric
1354 Clifford Avenue (Zip 61111)
P.O. Box 2940
Loves Park, IL 61132-2940
United Stated of America

LITHO IN U.S.A. 10/85, REV . 6/89

F-24820-2

OUTPUT PRESSURES

FOR
PNEUMATIC TEMPERATURE TRANSMITTERS

ACTUAL TEMPERATURE

TEMPERATURE RANGE (F)					OUTPUT
-25 — +125 F	0 — 100 F	40 — 140 F	-40 — +160 F	40 — 240 F	PSIG
-25	0	40	-40	40	3.0
-22	2	42	-36	44	3.24
-19	4	44	-32	48	3.48
-16	6	46	-28	52	3.72
-13	8	48	-24	56	3.96
-10	10	50	-20	60	4.2
- 7	12	52	-16	64	4.44
- 4	14	54	-12	68	4.68
- 1	16	56	- 8	72	4.92
2	18	58	- 4	76	5.16
5	20	60	0	80	5.4
8	22	62	4	84	5.64
11	24	64	8	88	5.88
14	26	66	12	92	6.12
17	28	68	16	96	6.36
20	30	70	20	100	6.6
23	32	72	24	104	6.84
26	34	74	28	108	7.08
29	36	76	32	112	7.32
32	38	78	36	116	7.56
35	40	80	40	120	7.8
38	42	82	44	124	8.04
41	44	84	48	128	8.28
44	46	86	52	132	8.52
47	48	88	56	136	8.76
50	50	90	60	140	9.0
53	52	92	64	144	9.24
56	54	94	68	148	9.48
59	56	96	72	152	9.72
62	58	98	76	156	9.96
65	60	100	80	160	10.2
68	62	102	84	164	10.44
71	64	104	88	168	10.68
74	66	106	92	172	10.92
77	68	108	96	176	11.16
80	70	110	100	180	11.4
83	72	112	104	184	11.64
86	74	114	108	188	11.88
89	76	116	112	192	12.12
92	78	118	116	196	12.36
95	80	120	120	200	12.6
98	82	122	124	204	12.84
101	84	124	128	208	13.08
104	86	126	132	212	13.32
107	88	128	136	216	13.56
110	90	130	140	220	13.8
113	92	132	144	224	14.04
116	94	134	148	228	14.28
119	96	136	152	232	14.52
122	98	138	156	236	14.76
125	100	140	160	240	15.0

OTHER RANGES

40 to 100 F RANGE	
ACTUAL TEMPERATURE	OUTPUT PSIG
40	3.0
42	3.4
44	3.8
46	4.2
48	4.6
50	5.0
52	5.4
54	5.8
56	6.2
58	6.6
60	7.0
62	7.4
64	7.8
66	8.2
68	8.6
70	9.0
72	9.4
74	9.8
76	10.2
78	10.6
80	11.0
82	11.4
84	11.8
86	12.2
88	12.6
90	13.0
92	13.4
94	13.8
96	14.2
98	14.6
100	15.0

50 to 90 F RANGE	
ACTUAL TEMPERATURE	OUTPUT PSIG
50	3.0
52	3.6
54	4.2
56	4.8
58	5.4
60	6.0
62	6.6
64	7.2
66	7.8
68	8.4
70	9.0
72	9.6
74	10.2
76	10.8
78	11.4
80	12.0
82	12.6
84	13.2
86	13.8
88	14.4
90	15.0

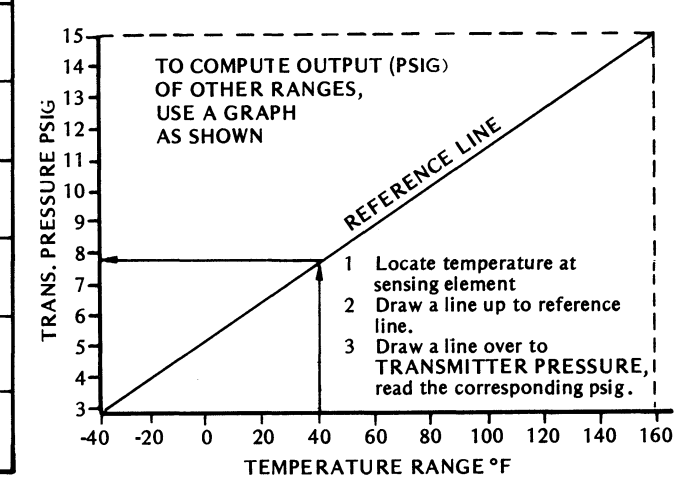
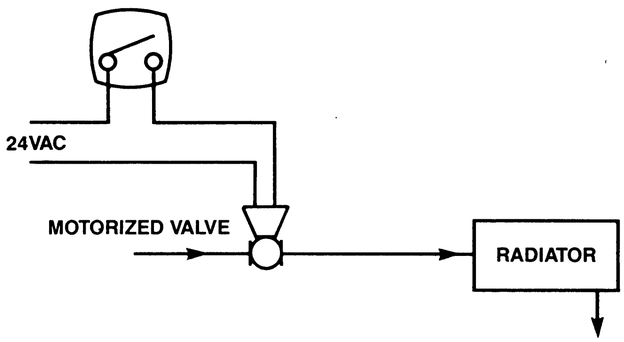


TABLE OF CONTENTS

TYPES OF CONTROL SYSTEMS 1

Control systems are classified according to the source of power used to operate the devices within the system. These devices can be electric/electronic, self-contained or pneumatic controls.

A. **Electric/Electronic** – 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.

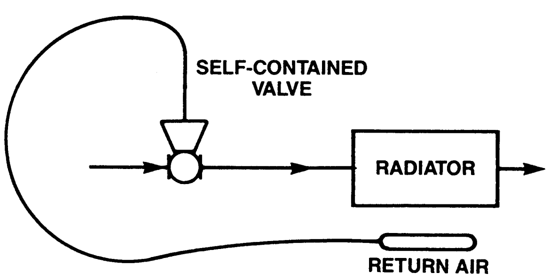


LOW VOLTAGE TWO-POSITION CONTROL
FIGURE 1-1

Figure 1-1 illustrates a simple electric control application. The electric thermostat operates the control valve motor to open the radiator valve when an increase in room temperature is required and closes the valve when the desired temperature is reached.

Electronic systems use very low voltages (typically 15V or less) for sensing and transmission. An electronic amplifier is used to increase the minute voltage variations from the sensing device to the larger values necessary to operate the controlled electric actuator.

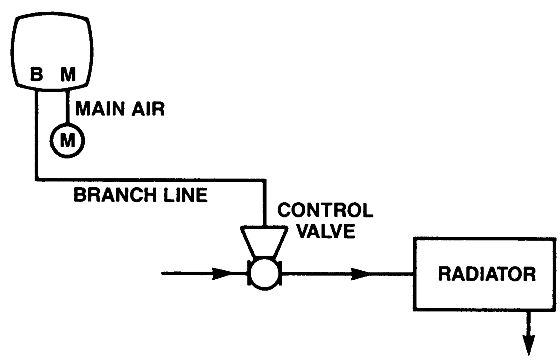
B. **Self-contained** – The source of power as well as the sensing element and the final control device (such as a valve) are contained within a single instrument. A self-contained control usually includes a bellows and a bulb connected by a length of tubing and filled with a vapor, gas or liquid. A change in the temperature of the medium surrounding the bulb causes the bellows to expand or contract with enough force to operate the valve.



SELF-CONTAINED CONTROL VALVE
FIGURE 1-2

Figure 1-2 illustrates a simple application of a self-contained control. The valve which controls the supply of steam 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 operating principle is essentially the same as that of a pneumatic system. The difference lies in the fact that the variations of pressure in the bellows are directly produced by changes in temperature at the bulb.

C. **Pneumatic** – In a pneumatic system, compressed air is the source of power for the controllers and controlled devices. The devices are connected by either copper or polyethylene tubing.



SIMPLE PNEUMATIC CONTROL
FIGURE 1-3

Figure 1-3 illustrates a simple application of a pneumatic control. An operating pressure of 15-20 psig is supplied to the controller through a main air line. The thermostat functions as a pressure regulator actuated by the air temperature. The line supplying the valve actuator (branch line) maintains a pressure of 15 psig or less, according to the need for heat in the room.

It should be noted that the control valve actuator does not consume air. Except when it is changing position, air does not flow into the actuator. The valve assumes a position determined by the branch air line pressure, the tension of the spring and the force exerted by the steam pressure on the valve. However, the spring tension varies with the position of the valve stem. Therefore, the thermostat, by regulating the air pressure, effectively determines the position of both the actuator and the valve.

In any heating, ventilating or 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 temperature regulation from a single thermostat for an entire building due to exposures and interior temperature 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. A control system of this type includes a thermostat in each room which controls some type of valve, damper actuator, unit ventilator, 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. However, where maximum flexibility and the most accurate control is desired, individual room control provides the most satisfactory results.

ADVANTAGES OF PNEUMATIC CONTROL SYSTEMS

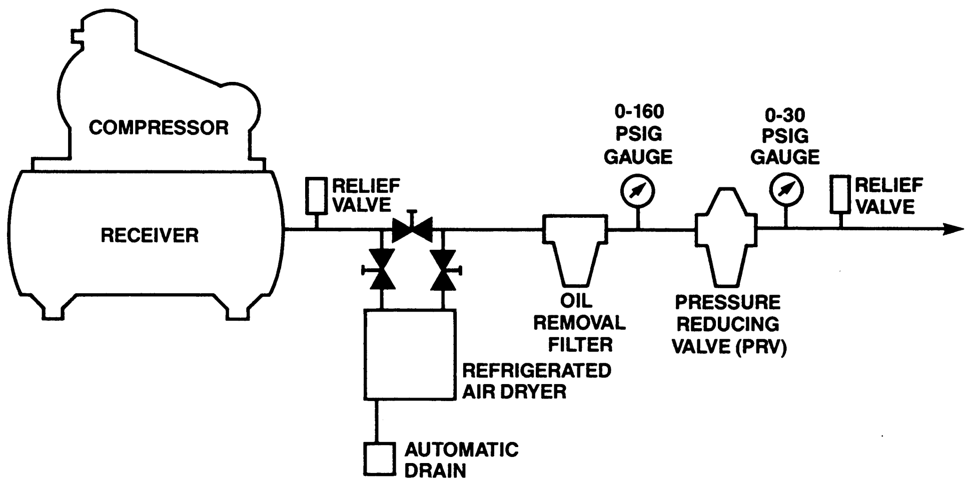
Pneumatic control systems are commonly used for controlling heating and air conditioning equipment in commercial applications. Pneumatic control offers a number of distinct advantages.

1) Pneumatic equipment is inherently adaptable to modulating operation, yet on/off operation 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) It is suitable where explosion hazards exist.
5) Cost is usually less in large installations, especially where codes require that low voltage electric wiring be run in conduit.

THE AIR STATION 2

Pneumatic control systems use compressed air to supply energy for the operation of valves, motors, relays and other pneumatic control equipment. Consequently, the circuits consist of air lines. Pneumatic control systems are made up of the following elements:

1) A source of compressed air which is stored in a receiver tank at a pressure capable of supplying all the pneumatic devices in the system with operating energy.
2) A refrigerated air dryer and oil removal filter to ensure dry, oil-free air downstream.
3) A pressure-reducing station which reduces receiver tank pressure to a normal operating pressure of 15-25 psig, depending on system requirements.
4) Air lines (which can either be copper or polyethylene tubing) connect the air supply to the controlling devices (thermostats and other controllers). These air lines are called **"mains."**
5) Controlling instruments such as thermostats, humidistats and pressure controllers are used to position the control devices.
6) Intermediate devices such as relays and switches.
7) Air lines leading from the controlling devices to the controlled devices. These air lines are called **"branch lines."**
8) Controlled devices such as valves or damper actuators.



SINGLE PRESSURE PNEUMATIC AIR STATION
FIGURE 2-1

The elements of a simple, single-pressure pneumatic control 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 25 HP. Currently, the most efficient air compressors in this size range are piston-style reciprocating units.

Reciprocating air compressors come in three basic designs: Oil lubricated, oil-free 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. The oil-free type contains a crosshead and oil seals to isolate the oil-filled crankcase from the compression chamber to prevent migration of virtually any oil. Oil-less compressors use sealed bearings and self-lubricating 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 refrigerated dryers and coalescent filters are necessary in a system to dry the air and to remove any oil vapors and dirt particles. Even the use of oil-free or 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 is a natural by-product of air compression. As the hot compressed air cools, the moisture content is released. By adding an automatic tank drain to the compressor storage tank, the tank drain will automatically remove water, oil, dirt and scale which has settled to the bottom of the storage tank.

Also, a refrigerated air dryer placed downstream from the storage tank will remove additional moisture carry-over which would normally enter the system. The refrigerated dryer should be equipped with an automatic drain trap as an added measure to remove any remaining condensation. A manual bypass valve for the refrigerated air dryer is available in order that the dryer may be routinely serviced without interrupting the system operation. A refrigerated dryer ensures that dry air is available to the system, but it cannot ensure that all oil aerosols or submicron dirt particles are removed. Therefore, an additional coalescent-type filter should be installed after the refrigerated dryer to remove these aerosols and to trap submicron dirt particles.

A pressure reducing valve (PRV) downstream of these devices maintains the operating pressure (15-25 psig) for the system. A pressure relief valve set at 30 psig is installed as the final air station device. The maximum safe operating pressure for most 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 most of these contaminates can be removed by the automatic drain trap.

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

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 compressor 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 low revolutions 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.

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

$$\text{SCFM} = \frac{\text{Total air consumption (SCIM)}}{\text{Desired compressor operation (\% running time)} \times 1728}$$

SCFM = Free cubic feet per minute of compressor capacity that will be required.

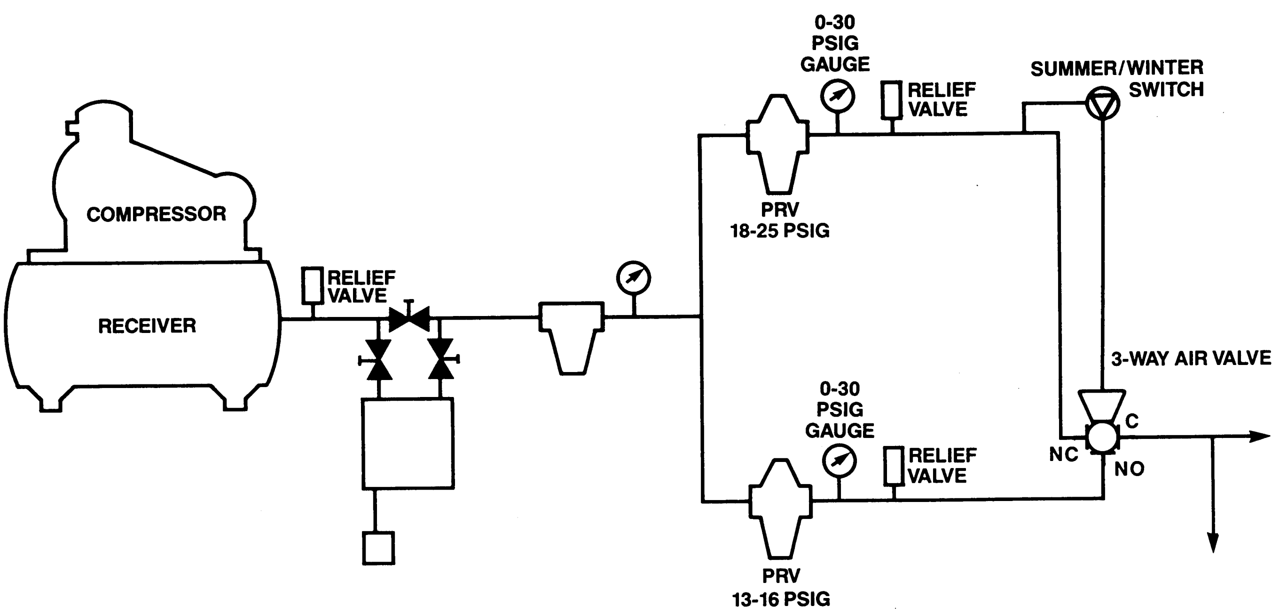
Total air consumption = sum of control devices used multiplied by the SCIM for each respective device.

Compressor operation = percentage of operating time required, such as 33-1/3% or 50% operation.

1728 = number of cubic inches per cubic foot of air.

DUAL PRESSURE SYSTEM

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. Since two distinct main air pressures are necessary, there are two separate pressure reducing valves, each set at the system operating pressure necessary for one mode of operation.



DUAL PRESSURE PNEUMATIC AIR STATION

FIGURE 2-2

In the typical dual pressure system shown in figure 2-2, one pressure regulator reduces tank pressure to between 13 and 16 psig and the other to between 18-25 psig (specific pressures depend on control manufacturer). The lower pressure is normally supplied to the controlling device 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 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 thermostat. 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 the higher operating pressure out the common port to the thermostat.

PNEUMATIC CONTROLLERS 3

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. Controllers can be either **direct** or **reverse acting.** These devices include thermostats, temperature controllers, humidistats and pressure controllers.

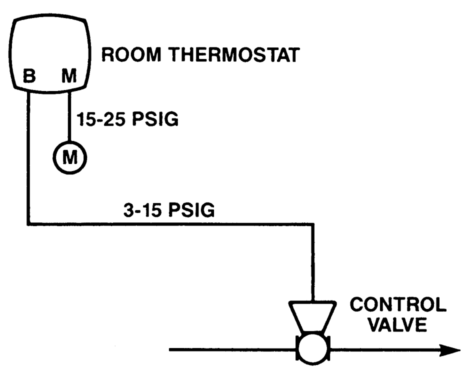
A **direct acting** controller increases its branch line pressure as the value of the condition it is sensing increases.

A **reverse acting** controller decreases its branch line pressure as the value of the condition it is sensing increases.

The term **sensitivity** is used to define the psig output change from the controller per unit of sensed variable change in the condition being controlled. For example, if a temperature controller changes its output 3 psig when the space it is controlling changes 1 degree F, the thermostat has a sensitivity of 3 psig per degree F.

The operating output range of most pneumatic controllers is 3-15 psig.

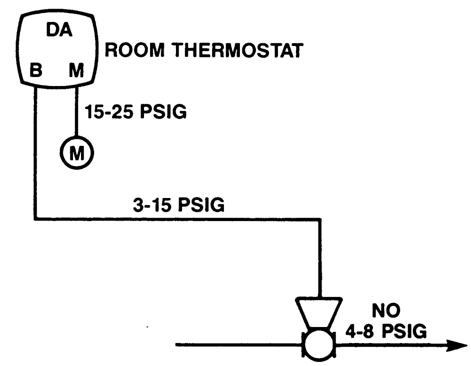
The change in value of the sensed medium which must occur to create that 3-15 psig output change is called **throttling range** or **proportional band**. For example, a thermostat has a temperature range of 55-85°F. Within that span a throttling range can be selected between 2 and 12°F. Assuming that a 4°F throttling range is selected the controller's branch line output will vary from 3 to 15 psig over a 4° change in room temperature.



TEMPERATURE CONTROLLER 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** and the valve could be either **normally open** or **normally closed.**

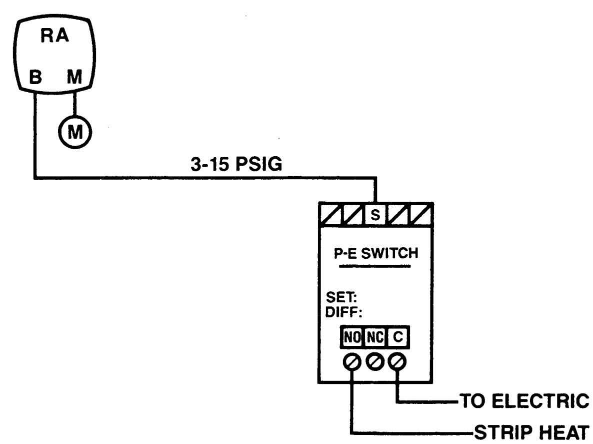


TYPICAL HEATING APPLICATION

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 4-8 psig. This combination is commonly used for controlling the flow of a heating medium (hot water or steam). With respect to the spring range, the valve will begin to close at a pressure from the controller of 4 psig and will be completely closed when the output (branch) signal is 8 psig. In the event of a system failure (loss of 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.

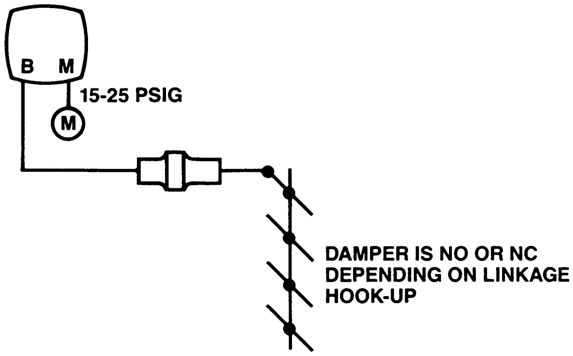
If the thermostat in figure 3-2 were to be reverse acting and the valve a normally closed type, the application could be used for controlling the flow of chilled water for cooling. As the temperature sensed by the reverse acting thermostat increases, its output pressure decreases causing the valve to open.



APPLICATION FOR ELECTRIC STRIP HEAT

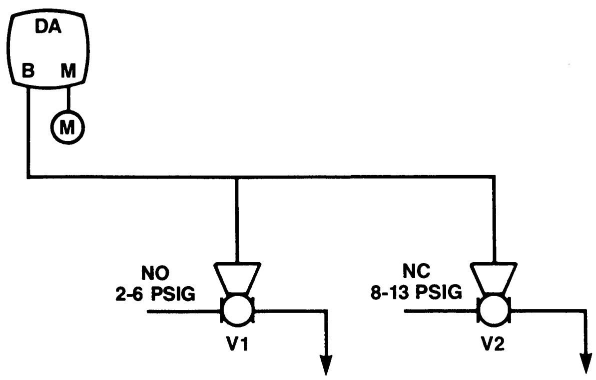
FIGURE 3-3

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 relay (P-E) which, in turn, activates the electric strip heater element when an increase in temperature is required.



BASIC DAMPER CONTROL
FIGURE 3-4

Figure 3-4 illustrates a combination of a controller and a damper actuator for pneumatic control. In this case, the controller sends an air signal to the damper actuator which positions the damper to control air flow across a coil, through a duct or into the 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 pressure loss, the damper will close.



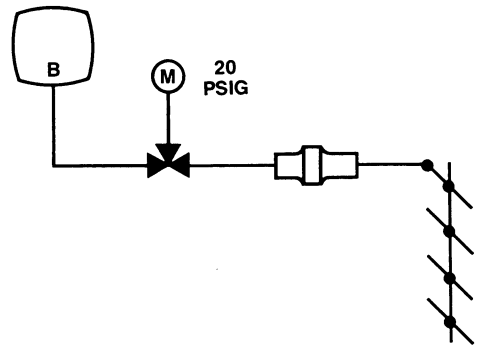
VALVES IN SEQUENCE TO
PROVIDE HEATING AND COOLING
FIGURE 3-5

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 is designated as direct acting, valve V1 is normally open and valve V2 is normally closed.

The **spring ranges** of the valves are also extremely important so as not to have both valves open at the same time. The spring ranges have been split to eliminate this possibility. Valve V1 is normally open with a spring range of 2-6 psig. Valve V2 is normally closed and has a spring range of 8-13 psig. With this arrangement, when the direct acting thermostat's output pressure is between 2 and 6 psig, valve V1 is closing and valve V2 remains fully closed. When the thermostat output pressure reaches 8 psig, valve V2 is starting to open. Above 13 psig, valve V1 is fully closed and valve V2 is fully open. The dead band between 6 and 8 psig eliminates the possibility of simultaneous heating and cooling.

SINGLE PRESSURE THERMOSTATS

Single pressure thermostats are either **one-pipe** (bleed-type) or **two-pipe** (relay-type) devices.



ONE-PIPE THERMOSTAT PIPING
FIGURE 3-6

A one-pipe thermostat (as the name implies) utilizes one connection to the thermostat. As figure 3-6 illustrates, the main air supply is fed through a restrictor tee into the branch line between the thermostat and the controlled device.

The restrictor tee allows only a fixed amount of air through its orifice. The thermostat opens and closes its leak port in response to changes in space temperature.

A two-pipe thermostat receives main air directly. Through relay action it will feed air to the branch line and/or exhaust it through its leak port in response to temperature changes. It provides a greater volume of air to the branch line than the one-pipe thermostat. This provides a faster response to a change in the room temperature.

The appearance of one-pipe and two-pipe thermostats is the same. Both types of devices utilize the same calibration procedures.

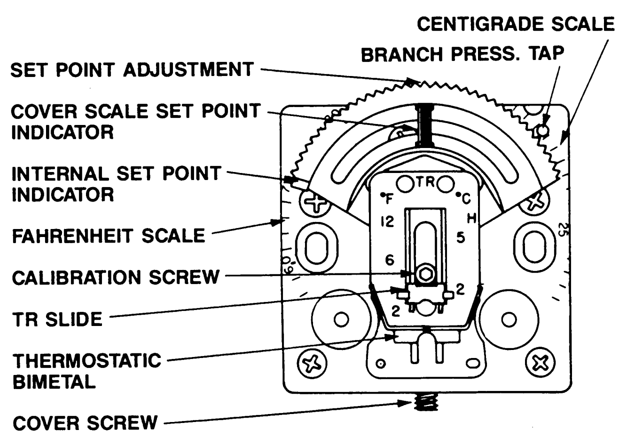


FIGURE 3-7

The branch pressure tap shown in figure 3-7 is provided in order to read the thermostat's output pressure. Methods of sampling branch pressure vary depending on the manufacturer.

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

1) Make sure that the main air line is connected to the thermostat and the branch line is connected to the actuator.
2) Main air pressure must be between 15 and 25 psig.
3) Attach the proper gauge tap fitting to the gauge port on the thermostat.

4) Using an accurate thermometer, measure the ambient temperature at the thermostat.
5) Change the set point of the thermostat to match the ambient temperature.
6) Adjust the calibration screw until the branch line pressure reads 9 psig (or the midpoint of the spring range for the actuator being controlled).

The thermostat is now in calibration. Set the thermostat to the desired control point.

DEAD BAND THERMOSTATS

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

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

The dead band thermostat operates in the same manner as a single pressure, single temperature thermostat. The bimetal assembly controls a single leak port which prohibits the two individual set points from overriding one another, resulting in simultaneous heating and cooling.

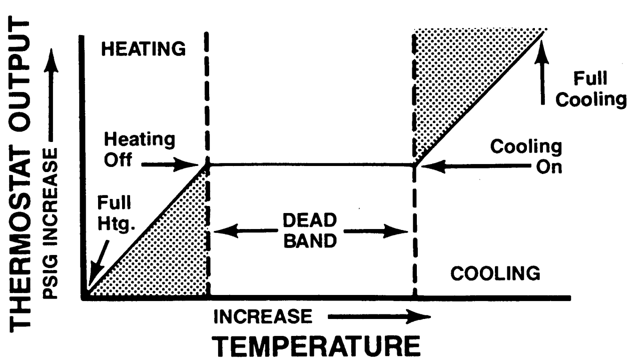


FIGURE 3-8

Figure 3-8 shows that the output of a direct acting thermostat rises to the dead band pressure. The dead band pressure is then maintained until the cooling set point is reached. At this point, the branch pressure will rise to main air pressure should the room temperature continue to rise.

Calibration procedures for Robertshaw dead band thermostats are described in the appendix.

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 both direct and reverse acting. This function in the thermostat is accomplished by changing the main air pressure depending on whether heating or cooling is desired.

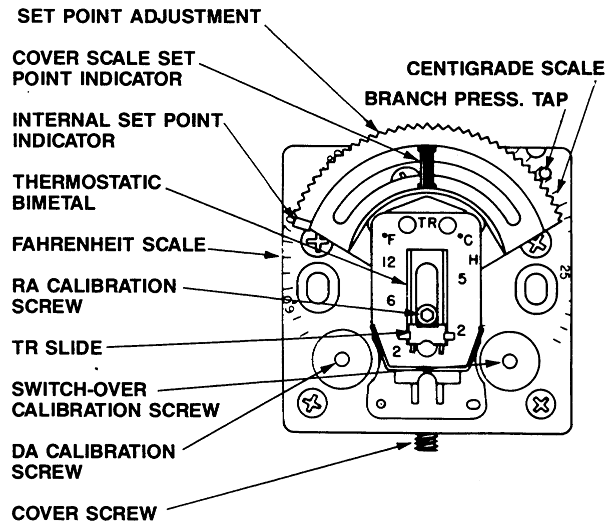


FIGURE 3-9

Figure 3-9 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 adjustment of a summer/winter switch, which positions a three-way air valve to change the main air pressure supplied to the thermostat, is the only manual adjustment.

Refer to the appendix for procedures used to calibrate and adjust Robertshaw summer/winter thermostats.

Day/Night

The day/night system is designed for applications requiring separate control points due to varying occupancy or seasonal loads. Day/night control is particularly adapted to buildings such as schools and office buildings. The purpose of day/night application is to control temperature at different set points during the day and night. A day/night thermostat is essentially the same as the summer/winter thermostat. The main difference is that there are two bimetals in the day/night thermostat and both are either direct or reverse acting. However, the two bimetals have separate set point adjustments to provide the set-back or set-up function.

When the main air pressure is 15-17 psig (depending on manufacturer) the day bimetal controls the thermostat's output pressure. With 18-25 psig main air pressure (depending on manufacturer) the night bimetal controls the output pressure. For example, a school may desire a 70°F space temperature in the daytime, but at night, when the building is unoccupied, it needs only to maintain control at 60°F. The main air pressure is changed in a similar manner to that of a summer/winter system. This brings the bimetal in control which is set for a desired control point of 60°F.

Some models of the day/night thermostats have a "local indexing" 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 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 until the main air pressure returns to the night pressure on the next cycle. At that time, it will reset automatically.

See appendix for calibration instructions.

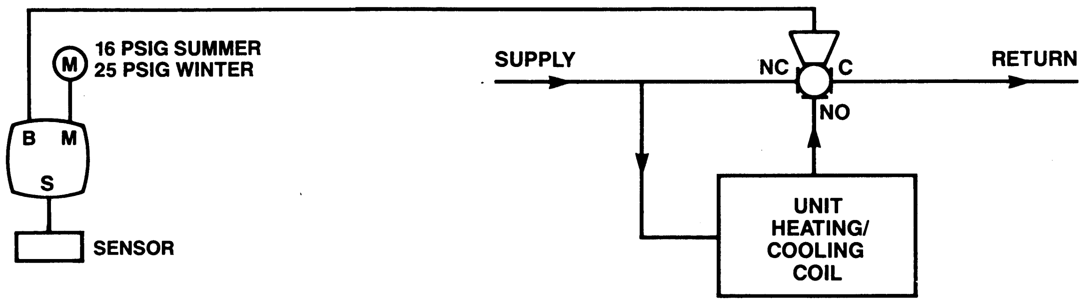
PNEUMATIC CONTROLLERS

UNIT TEMPERATURE CONTROLLERS

Unit temperature controllers are used primarily as return air controllers in induction units, fan coil units and unit ventilators.

The physical appearance of these devices is similar to other thermostats except the bimetal for temperature sensing is omitted. The bimetal temperature sensor for these devices is generally positioned in the system's return air flow. The sensed temperature is fed back to the controller which, in turn, provides the branch line signal to the controlled device.

There are different types of unit temperature devices available to be used for single or dual pressure applications.



DUAL PRESSURE UNIT TEMPERATURE CONTROLLER
FIGURE 3-10

Figure 3-10 illustrates a system utilizing a dual pressure unit temperature controller.

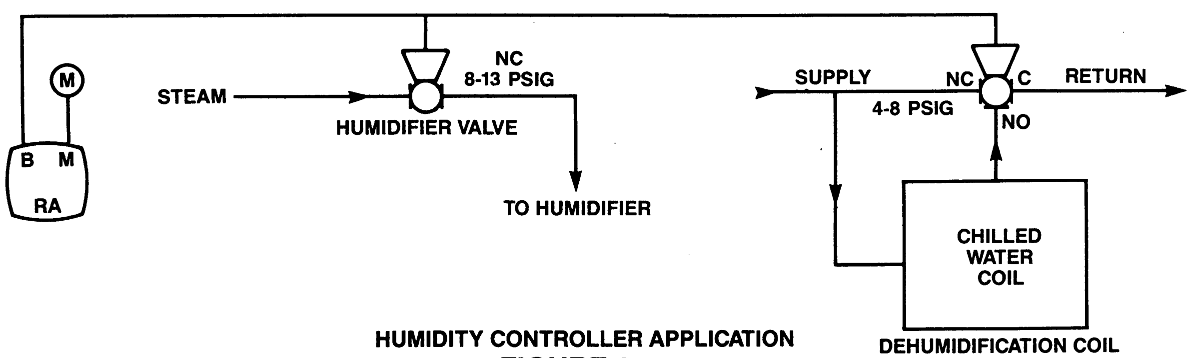
The controller can be either reverse or direct acting, depending on the main air pressure supplied to it and it is used to sense return air temperature. When the heating medium is supplied to the coil, the controller would normally be in the winter cycle, or direct acting. When the return air temperature drops below the set point of the controller, the unit valve would open to permit the heating medium to flow through the coil and increase return air temperature. When a cooling medium is supplied to the unit coil, the controller would normally be in the summer cycle or reverse acting. When return air temperature exceeds the set point of the controller, the branch line pressure will drop allowing the unit valve will open, permitting the cooling medium to flow through the coil and lower the return air temperature.

A significant point about unit controllers which should be noted is that when the controller is being used to control flow through one coil used for heating and cooling, then 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) then either a direct acting only, or reverse acting only unit controller can be used. In this case, the action of the controller chosen will depend on how the unit valves are piped to the coils.

HUMIDISTATS

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

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 most commonly used in humidistats because, since it is manufactured, its uniformity can be controlled.



HUMIDITY CONTROLLER APPLICATION
FIGURE 3-11

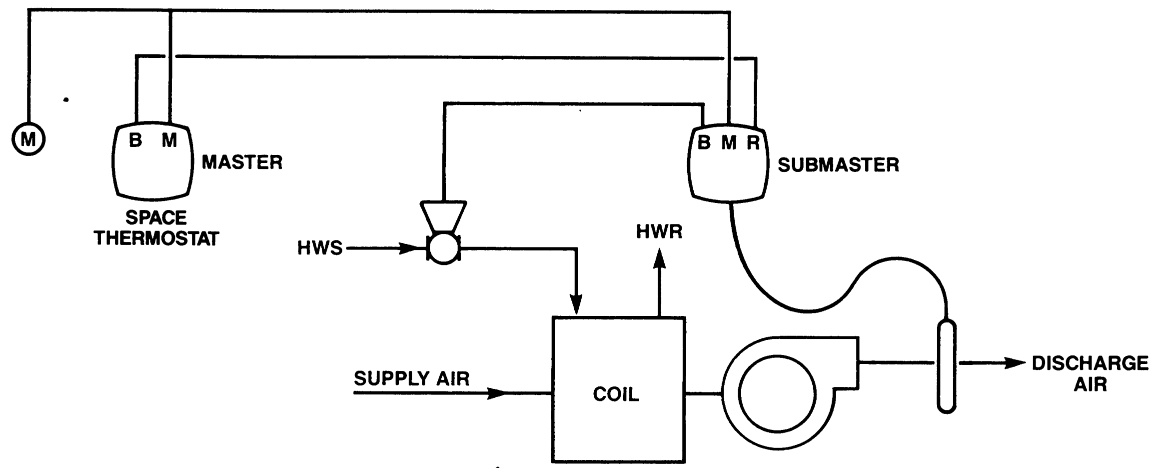
Figure 3-11 illustrates a humidistat application. The reverse acting humidistat is controlling a normally closed steam valve and 3-way chilled water valve. Hence, when the output pressure of the humidistat is between 8 and 13 psig, the normally closed steam valve is open and adding humidity to the system.

The chilled water mixing valve modulates open between 4 and 8 psig. Thus, as humidity in the space increases, the output signal decreases and allows the chilled water coil to provide dehumidification for the space.

Some humidistats may be set for direct or reverse action to correspond to a particular application. **Refer to the appendix for additional information.**

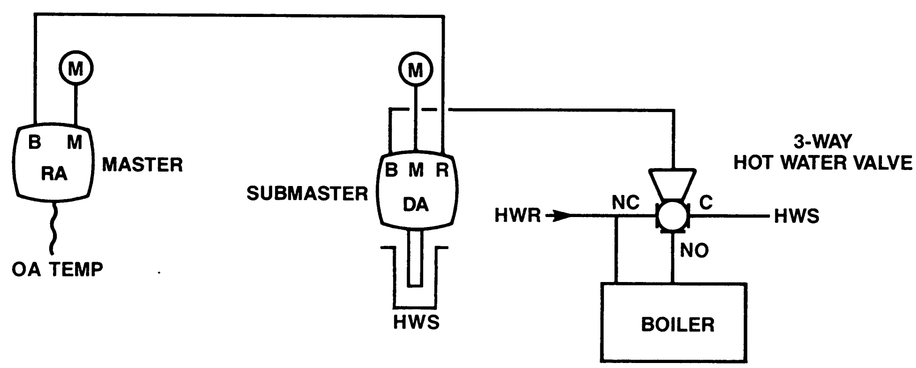
MASTER/SUBMASTER CONTROLLER

A master controller is a pneumatic controller which transmits its output signal to another controller. The second controller, or submaster, is similar to a standard type controller. The significant difference is that the submaster's set point will change as the signal from the master controller changes. The standard application for these devices is when some type of reset control is required.



RESET CONTROL OF UNIT DISCHARGE TEMPERATURE
FIGURE 3-12

Figure 3-12 shows a reset application for unit discharge temperature. The branch signal from the space thermostat is piped to the reset port of the submaster temperature controller. The output signal of the submaster is then piped to the final control device which controls the heating medium to a coil. The sensing element of the submaster is located in the discharge air stream. In this application, the discharge air temperature is varied as the space thermostat senses a change in temperature. When the space temperature changes, the signal from the thermostat varies to raise or lower the set point of the submaster controller. The submaster then senses discharge air temperature and varies its output signal to the heating valve to control the heating medium.



RESET CONTROL OF HOT WATER SUPPLY
FIGURE 3-13

Another application that utilizes master/submaster controls is shown in figure 3-13. In this example, the hot water supply temperature is reset based on a variation in the outside air temperature. The master controller is a remote bulb controller with a sensing element located to sense outside air temperature. The submaster controller has a rigid stemmed element mounted in the hot water supply line.

RESET SCHEDULE

HWS	OUTPUT	OA
200°	15 PSIG	10°
150°	9 PSIG	40°
100°	3 PSIG	70°

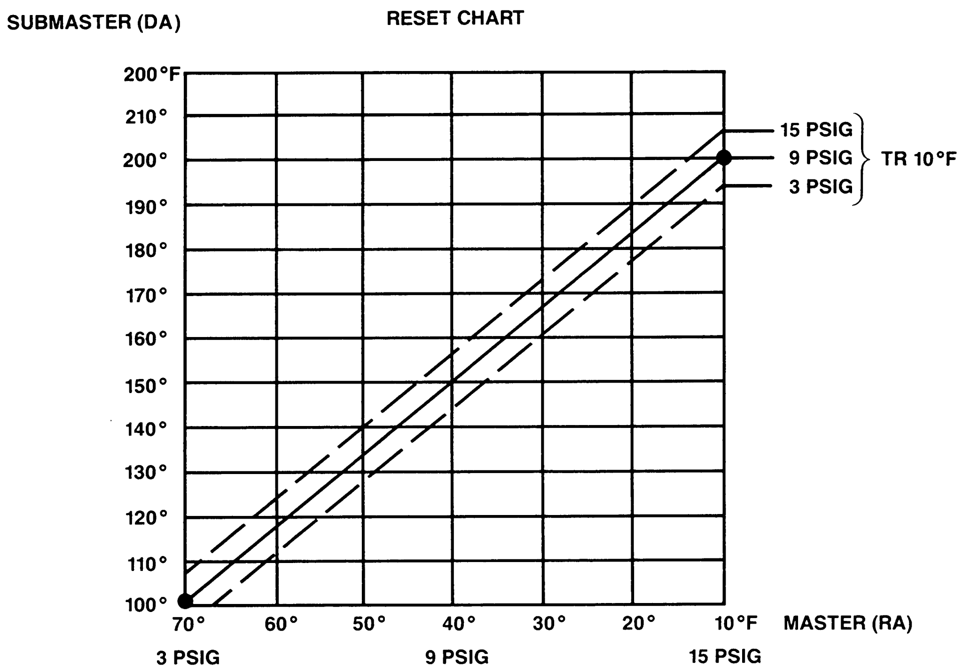


FIGURE 3-14

The schedule shown in figure 3-14 indicates at what temperature the water should be maintained as the outside air temperature varies. This information is important in order to properly calibrate the controllers. In the example, when the outside air temperature is 10°F, the system requires 200°F hot water. The branch line pressure from the master controller will be 15 psig when the outside air temperature is 10°F. Since the master controller is set for reverse acting, the branch pressure output of the device will increase as the outside air temperature decreases and vice versa.

Since the submaster is controlling a valve which is used for heating, and the valve is piped normally open to the boiler, the action of the submaster is direct acting. As the submaster sensing element senses a change in hot water supply temperature, it varies the branch pressure to the heating valve which, in turn, controls the flow of heating medium from the boiler.

The required reset range of the submaster controller determines the submaster's reset dial setting. In this example, the required reset range is 100 to 200°F. The reset dial should be set equal to the span of the reset range, or 100°F (200 minus 100°F). Since the lowest temperature of hot water desired is 100°F, this becomes the set point for the submaster controller. Once the set point adjustment and the reset adjustment have been set as required, the master controller will now reset the submaster from a minimum of 100°F up to 200°F and back down to 100°F. The master controller's output signal and the submaster in this application are combined to reset upward from the established set point, but not below it. Other facts about this example are also available from the reset chart in figure 2-14. The entire range (10 to 70°F) of the master controller represents a throttling range of 60°F. The minimum (3 psig) and the maximum (15 psig) output pressures of the master controller will occur within this 60°F throttling range.

A 10°F throttling range for the submaster controller has been selected. The submaster output pressure to the control valve is 3-15 psig. Regardless of the hot water temperature being supplied, the minimum (3 psig) and maximum (15 psig) output pressure of the submaster will occur within the 10°F throttling range. For example: when the outside air temperature is 40°F the system requires 150°F hot water. With a 10°F throttling range, the submaster's output pressure will be 3 psig at 145°F, 9 psig at 150°F, and 15 psig at 155°F. As the output pressure from the master controller varies, all three lines shown on the reset chart can be read up and down the chart to determine the hot water supply temperature in relation to the outside air temperature.

RECEIVER CONTROLLERS AND TRANSMITTERS

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 controlled condition (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 supply line to help maintain the proper volume of air between the transmitter and receiver controller. 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 0-100°F transmitter has a 100°F span. The output span of all transmitters is **always** 12 psig (15 minus 3).

Sensitivity is the psig change in the transmitter output signal per engineering unit (Example: Degree, % RH, etc.) change in the sensed medium. Transmitter sensitivity is calculated by the following method:

$$\text{Sensitivity} = \frac{\text{12 psig (output span)}}{\text{transmitter span}}$$

In the sensitivity formula on page 10, assume a 0-100°F range transmitter. The temperature span is 100°F (100 minus 0). By dividing the output span of 12 psig by the temperature span of 100°F, a sensitivity of .12 psig/°F is calculated. Thus, for each degree change in the sensed temperature, the output pressure from the transmitter will be varied .12 psig.

SENSITIVITY CHART

TRANSMITTER	RANGE	SPAN	SENSITIVITY
2232	30% - 80% RH	50%	0.24 psig / % RH
2323	0″ - 3″ WC	3″	4.0 psig / ″ W.C.
2323	0″ - 10″ WC	10″	1.2 psig / ″ W.C.
2220	50 to 90°F	40°	0.30 psig / °F
2252	40 to 140°F	100°	0.12 psig / °F
2252	0 to 100°F	100°	0.12 psig / °F
2252	40 to 240°F	200°	0.06 psig / °F
2252	−40 to 160°F	200	0.06 psig / °F
2252	−25 to 125°F	150°	0.08 psig / °F
2254	40 to 100°F	60°	0.20 psig / °F

FIGURE 3-15

The chart in figure 3-15 shows precalculated sensitivities for Robertshaw transmitters.

Throttling range refers to the number of degrees the temperature must change (%RH for humidity transmitter, etc.) in order for a receiver controller to change its branch pressure output between 3 and 15 psig.

Since the throttling range adjustment is designated as a percentage on the receiver controller (see figure 3-16), it is necessary to apply the following formula:

$$\text{TR in } \% = \frac{\text{TR}}{\text{span}}$$

For example, a 40-240°F temperature transmitter is selected and a 20 degree throttling range is desired. By using the previous formula, 20 divided by 200 equals 10%. Thus, the throttling range adjustment on the receiver controller would be set to 10%.

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

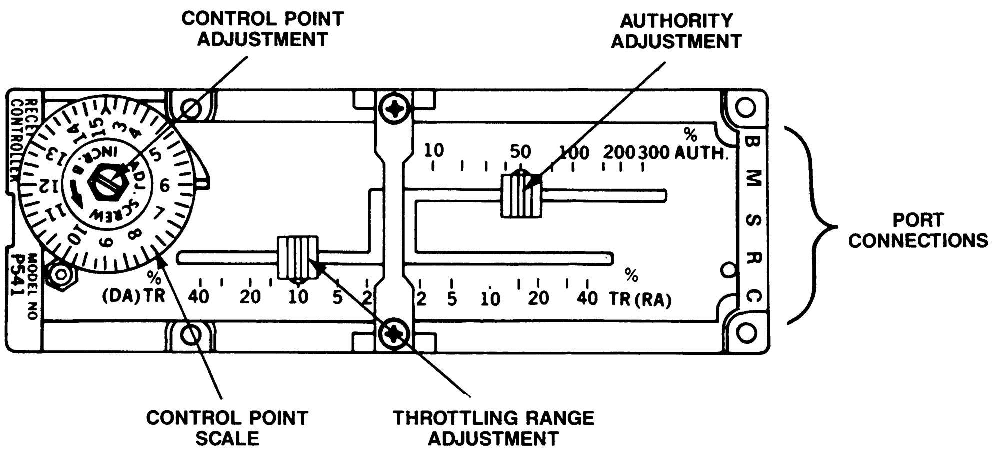


FIGURE 3-16

Figure 3-16 shows a receiver controller which can be used for single or dual input applications. There is a local control point adjustment and set point scale which may either be in psig, or match the range of the transmitter input. The "S" port connection is for the primary transmitter signal and the "R" port connection is for the secondary or reset transmitter input. The "C" port is a connection for a remote control point adjustment, should it be desired. Of course, the "M" port is for the main air connection and "B" is the branch output to the controlled device.

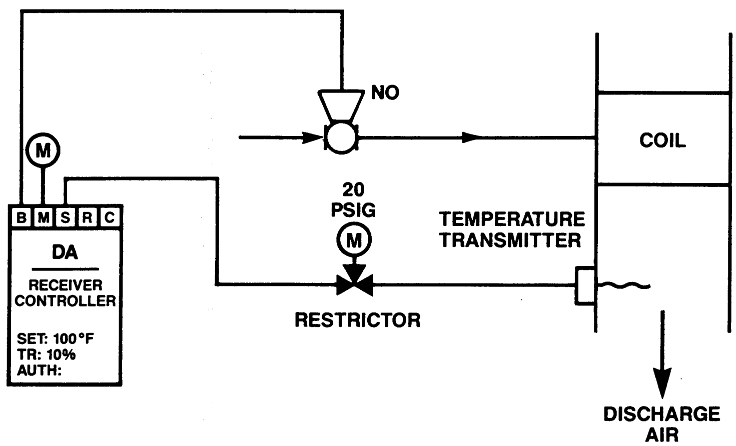
When using the receiver controller for a dual input application, another adjustment is provided which determines the effect of the secondary transmitter signal on the output signal of the controller. This adjustment is called **authority.** The typical application for using a secondary input and the authority adjustment is when reset control is required.

PNEUMATIC CONTROLLERS

A receiver controller can be used for either single or dual input applications as shown in figures 3-17 and 3-18. An input piped to the "C" port is used when a remote set point adjustment is required.

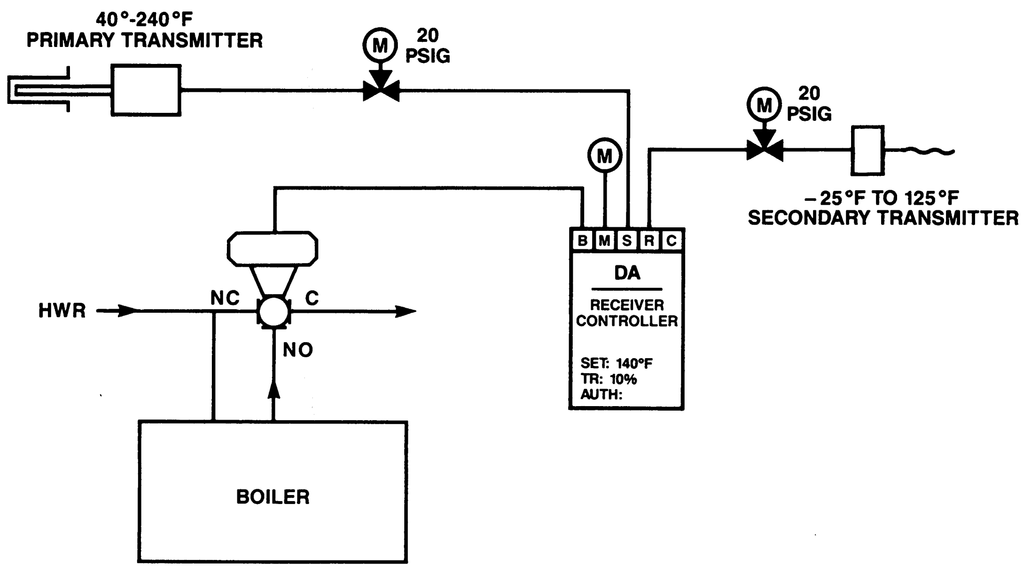
The remote set point adjustment uses a gradual switch to provide a means of manually raising or lowering the set point of the receiver controller without recalibrating the device to overcome temporary load changes.

Varying the input at port "C" from 3 to 15 psig will vary the set point 20% of the primary transmitter input span.



SINGLE INPUT RECEIVER CONTROLLER APPLICATION

FIGURE 3-17



DUAL INPUT RECEIVER CONTROLLER APPLICATION

FIGURE 3-18

In figure 3-17 a transmitter sensing discharge air temperature is used in conjunction with a direct acting receiver controller to position a normally open hot water valve in order to maintain desired discharge air temperature. The transmitter selected has a **coiled** element (or averaging bulb) in order to more accurately sense the duct temperature.

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

In this example, the primary transmitter senses the hot water supply temperature and its signal is transmitted (in psig) to the

"S" port of the controller. The secondary transmitter which is used to provide the reset effect is piped to the "R" port. Restrictors are placed in both transmission lines. Some manufacturers use restrictors within the receiver controller. External restrictors enable the transmitters to be located further away from the controller. Also, should a restrictor become clogged, replacement is easier when it is installed externally.

As the outside temperature drops, the set point of the receiver controller changes to increase the supply water temperature to offset the increased demand created on the building system. The opposite occurs on a rise in outside air temperature. The heating load decreases, thus the set point 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 authority (reset percentage) must be determined. For this example, an outside air transmitter with a −25 to 125°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 20°F was used. Based on a primary transmitter span of 200°F (240 minus 40°F), each degree change in the hot water temperature represents a .06 psig (12 psig÷200) change in the transmitter output pressure. 20 x .06 equals a throttling range setting of 1.2 psig or 10% (20 degrees÷200 degrees).

The third step is to calculate the authority (reset) percentage. In order to make this calculation it is necessary to refer to a reset schedule. The reset schedule (provided by the mechanical engineer) 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 100°F water when the outside air temperature is 60°F.

RESET SCHEDULE

Primary HW Temp	Secondary OA Temp
180°	10°
100°	60°

Throttling range: 10%
Authority: 150%

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 authority percentage the following formula can be applied:

$$\text{Authority} = \frac{\left(\begin{array}{c}\triangle T \text{ @ Primary} \times \\ \text{Primary Sensitivity}\end{array}\right) + \left(\begin{array}{c}\text{TR} \times \\ \text{Primary Sensitivity}\end{array}\right)}{\triangle T \text{ @ Secondary} \times \text{Secondary Sensitivity}} \times 100$$

The "Delta T" (△T) of the primary (difference between the two temperatures) is 80 degrees F. The "Delta T" of the secondary is 50 degrees F. Thus, the formula can be completed as follows:

$$\text{Authority} = \frac{(80°\text{F x .06}) + (20°\text{F x .06})}{50°\text{F x .08}}$$

$$\text{Authority} = \frac{4.8 + 1.2}{4.0} \text{ x } 100$$

$$\text{Authority} = \frac{6.0}{4.0} \text{ x } 100$$

Authority = 1.50 x 100

Authority = 150%

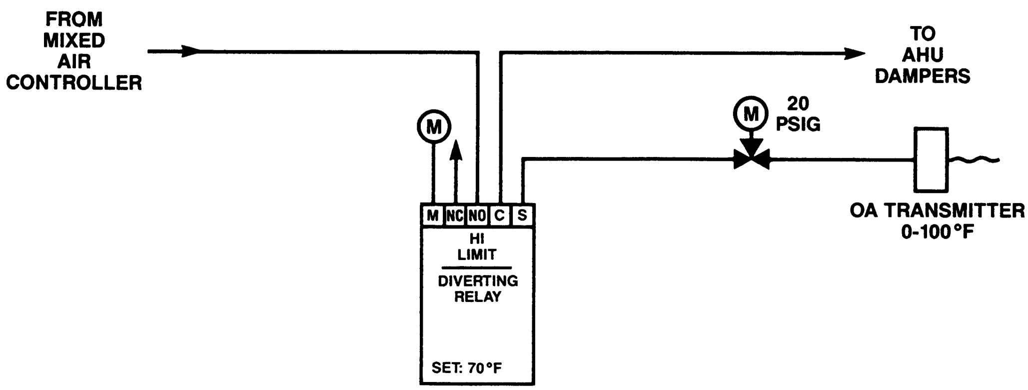
The authority adjustment on the receiver controller should be set to this percentage. The controller can now be calibrated. Set-up procedures vary according to manufacturer. **Calibration procedures for Robertshaw models are in the appendix of this manual.**

NOTES

PNEUMATIC RELAYS 4

Pneumatic relays are used to provide a multitude of switching functions in the pneumatic system. Applications for pneumatic relays are virtually unlimited. The most common are discussed here to provide an understanding of some of the more popular relays.

Diverting Relay: A diverting relay is a 3-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, into a signal for a final control device.

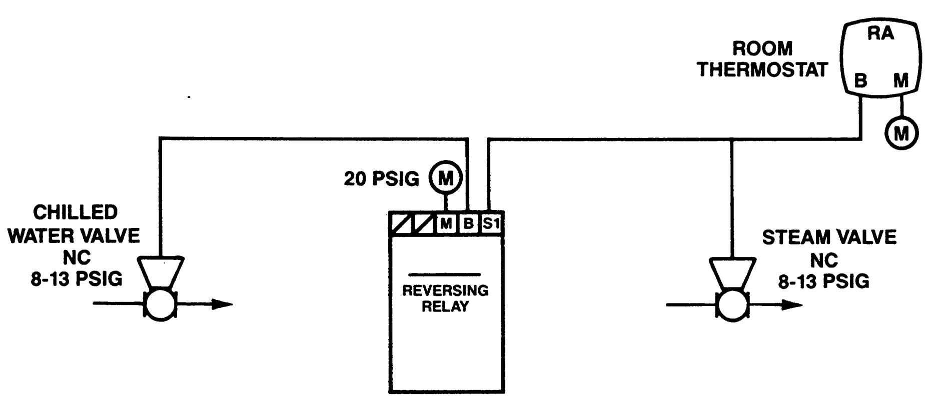


DIVERTING RELAY
FIGURE 4-1

In the illustration shown in figure 4-1, the diverting relay is being utilized as a high limit control to execute temperature changeover for an economizer application. The diverting relay allows the mixed air controller to control the air handling unit dampers up to an outside air temperature of 70°F. At this point (11.4 psig), the diverting relay switches and blocks the signal at "NO" and exhausts "NC" to allow the air handling unit dampers to close or go to minimum position. This relay can also be used as a low limit by piping the signal from the mixed air controller to "NC" instead of "NO" and exhausting out through "NO."

REVERSING RELAY

This device is designed for use in pneumatic control systems where the application requires reversing a signal from a controlling device. The relay's branch line pressure output decreases in direct proportion to an increase in input signal pressure.



REVERSING RELAY
FIGURE 4-2

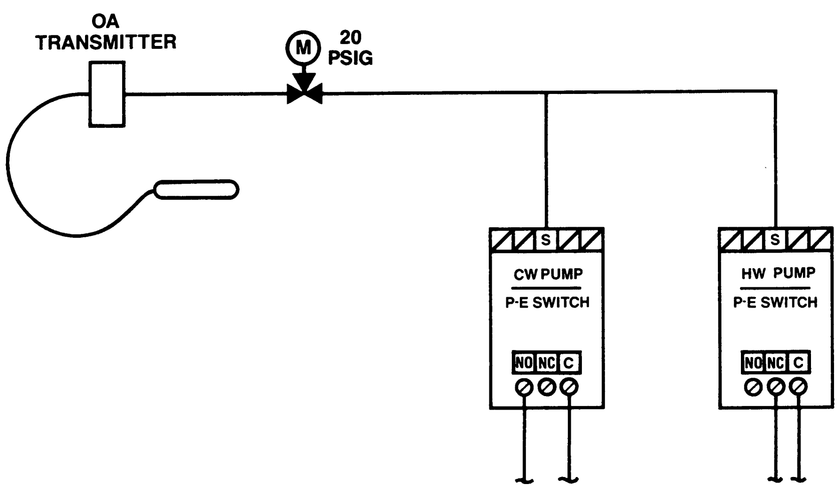
Figure 4-2 shows a typical way in which a reversing relay may be used. The reverse acting space thermostat is controlling both a heating and cooling valve. Each valve is normally closed. In this example the branch signal from the thermostat is piped to

PNEUMATIC RELAYS

the heating valve and then 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 increases, which causes the heating valve to open and permit the flow of hot water. The same signal goes to the reversing relay where it is reversed to a decrease in pressure to the cooling valve. This closes the cooling valve. As the thermostat senses an increase in temperature, its output pressure decreases, closing the heating valve. The signal to the reversing relay is reversed, causing its output pressure to increase and open the cooling valve.

PNEUMATIC-ELECTRIC RELAY

Pneumatic-electric relays, 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 ventilators, fan coil motors, unit heaters, air handling unit fans, chillers and hot water pumps. Several models are available; the main difference is in 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 available which will open either on a drop in pressure or on a rise in pressure. Special narrow differential models are also available.



PNEUMATIC-ELECTRIC RELAY
FIGURE 4-3

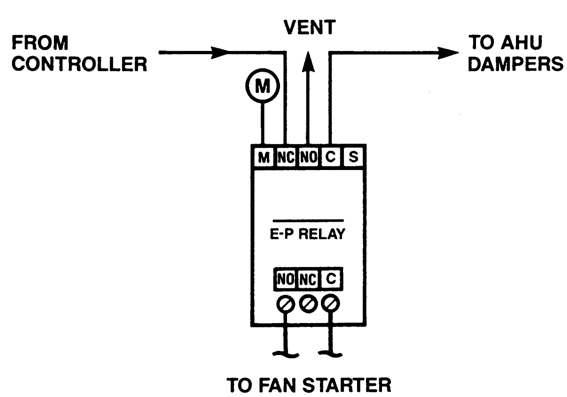
The application of a P-E switch is shown in figure 4-3. A change in the signal pressure from the outside air transmitter is used to open and close the contacts of the P-E switch which, in turn, energizes or de-energizes the pump starter. Separate P-E switches can be used to control the operation of the 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 transmitter. 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 would be wired to the normally open and common terminals of the P-E switch. An increase in the pressure signal from the outside air transmitter will cause 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 transmitter signal decreases below the set point of the P-E switch, the contacts will make and energize the hot water pump starter.

ELECTRIC-PNEUMATIC RELAYS

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 air connections.

Electric-pneumatic relays, or E-P's, are most often used as interlocking devices between an electrical circuit and a pneumatic circuit. Figure 4-4 utilizes an E-P to control a main air supply. It is wired electrically to the fan starter circuit so that when the fan motor is running, the proper ports are connected to allow 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 relay 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.

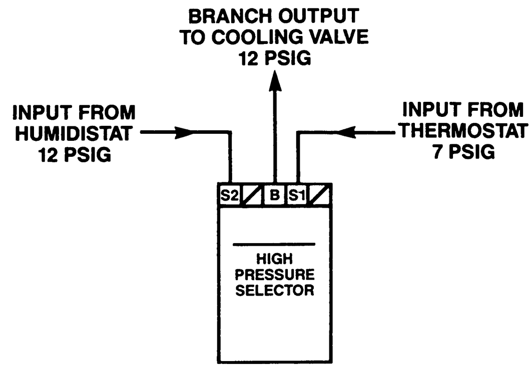


ELECTRIC-PNEUMATIC RELAY
FIGURE 4-4

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, high pressure selector or a combination of both. The low pressure selector receives input signals, compares, selects and transmits the lower of the two. The high pressure selector receives input signals, compares, selects, and transmits the higher signal.

The arrangement shown in figure 4-5 for a high pressure selector is often used for cooling and dehumidification, because either the thermostat or the humidistat can control the cooling valve. If both the thermostat and humidistat are calling for operation of the valve, the controller whose signal is the stronger will take preference.

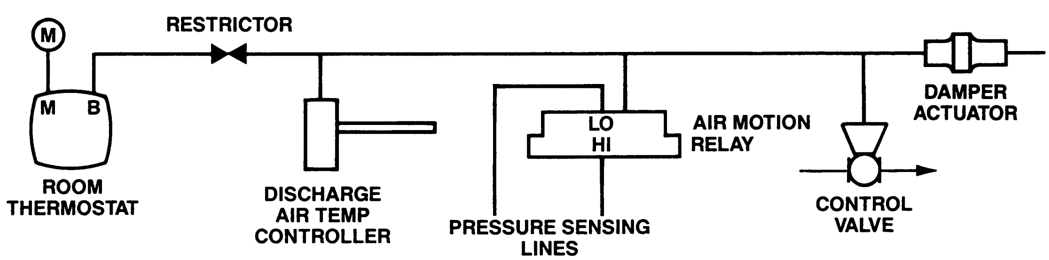


HIGH PRESSURE SELECTOR
FIGURE 4-5

AIR MOTION RELAY

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

The air motion relay can perform the same function as an E-P relay on the outside air damper on a standard unit ventilator cycle. This function is to allow the outside air damper to operate only when the unit fan is operating. When an E-P relay is used, the fan circuit could be energized and the outside air damper would operate even if the fan motor itself were to fail. Using an air motion relay, the unit fan must be functioning properly in order to create the pressure required to actuate the relay.



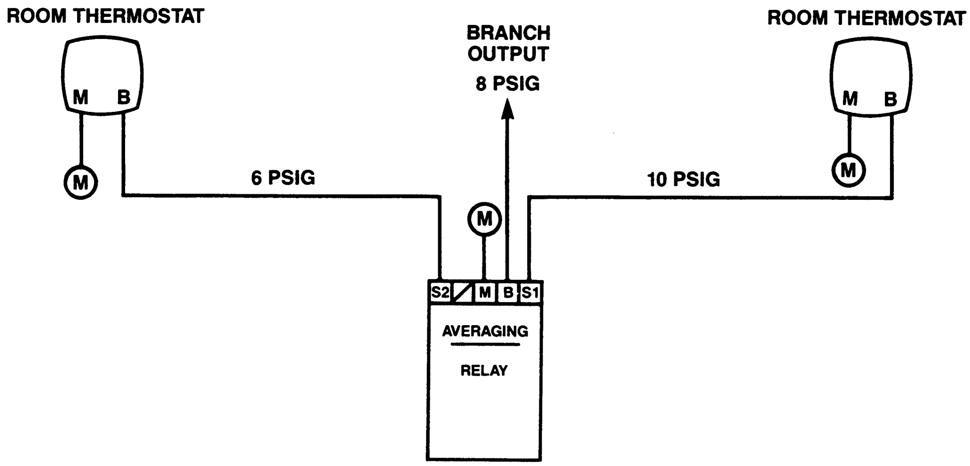
AIR MOTION RELAY
FIGURE 4-6

When using the air motion relay as shown in figure 4-6, there must be enough differential pressure to cause the relay to operate properly. In this example the low pressure sensing port is piped to the suction of the fan and the high pressure port is used as a reference port. It may be necessary in some cases to pipe the high pressure port to sense the fan discharge pressure if there is not enough differential pressure between the fan suction pressure and the reference pressure.

When the unit fan is operating, the relay senses the differential pressure across the fan, causing the signal port to close. This will allow the pressure downstream from the device to build up by placing the space thermostat in control of the pneumatic valve and damper actuator. When the unit fan is not operating, the result of equalized pressure across the diaphragm of the relay will allow the diaphragm to fall away from the signal port. Any pressure at the signal port will then exhaust the signal, causing the valve and damper actuators to return to their normal positions.

PNEUMATIC RELAYS

AVERAGING RELAY

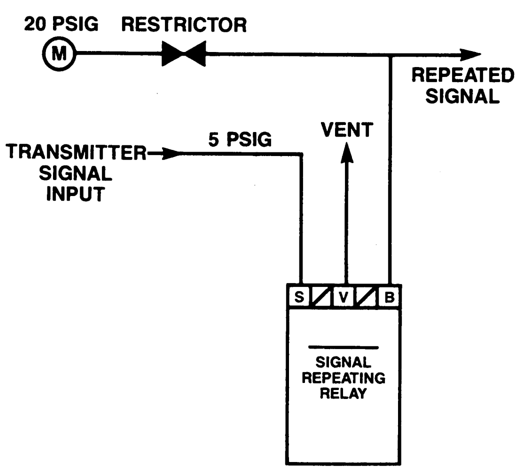


AVERAGING RELAY
FIGURE 4-7

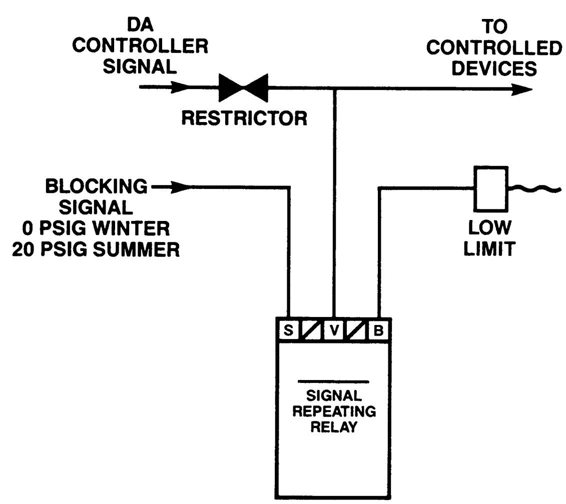
This type of relay is designed for use in pneumatic control systems where the application requires operation of a final control device by the average signal from two pneumatic controllers. In the example shown in figure 4-7 the two space thermostats send their respective branch signals to the averaging relay. The relay, in turn, transmits the average of the two signals to control the final control device.

SIGNAL REPEATING RELAY

This is a multipurpose device used primarily to repeat signals from transmitters. It is capable of accurately reproducing a low volume signal while increasing its volume.



REPEATING APPLICATION
FIGURE 4-8



BLOCKING APPLICATION
FIGURE 4-9

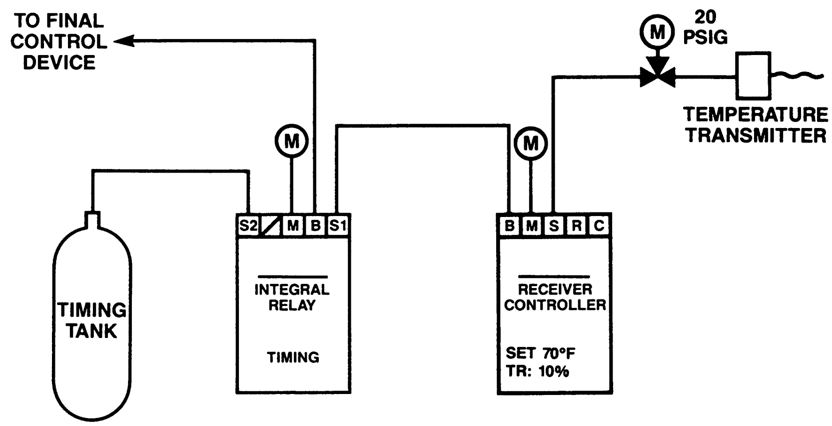
The application shown in figure 4-8 illustrates a 5 psig signal coming from a transmitter and piped to the S port on the relay. This 5 psig signal is repeated out of the B port at a greater volume enabling it to do more work, such as transmit the signal over a greater distance. Port V is left to vent to the atmosphere.

The signal repeating relay can also be used as a blocking relay. For example, as shown in figure 4-9, during winter operation with 0 psig pressure to port S, ports B and V are connected allowing for normal operation of the low limit. During the summer, with 20 psig fed to port S, ports B and V are blocked and the low limit is removed from the system.

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 and hunting. Control point **offset** may briefly be described as the deviation from a controller's set point that frequently occurs within the throttling range of the controller as significant load changes are effected.

The integral reset relay continuously increases or decreases its output as necessary to maintain its input pressure from a proportional controller's branch line at a constant value.



INTEGRAL RESET RELAY

FIGURE 4-10

In the example shown in figure 4-10, it is desired to maintain the supply air temperature at a constant 70°F. Because of the potentially large differential between the entering air temperature and leaving air temperature, a wide throttling range is required on the receiver controller for system stability (prevention of hunting or cycling). The offset that would result from a wide throttling range would be objectionable (up to 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 relay input (controller output) pressure at set point.

AMPLIFYING/RATIO RELAY

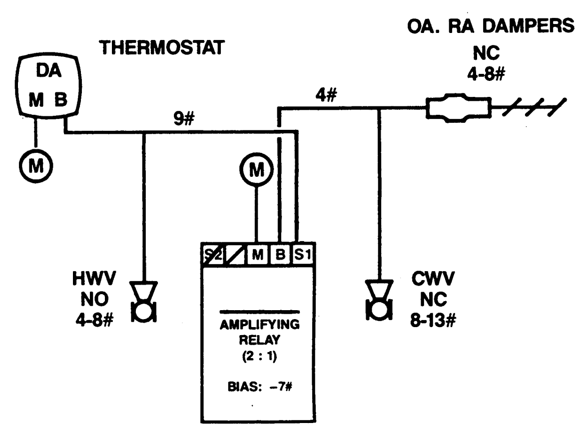


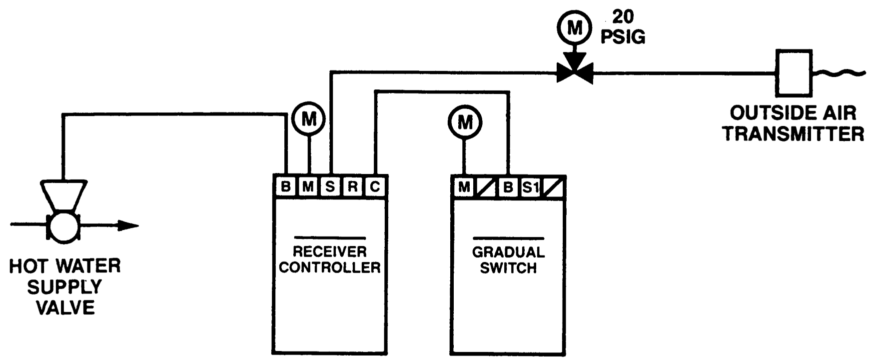
FIGURE 4-11

This relay is a proportioning device, used where amplification of a proportional input signal is needed. The relay branch line output pressure increases at a 2 : 1 ratio to the input signal pressure, and also amplifies the volume of air available to the final controlled device, minimizing system lag. A bias adjustment is provided which permits the effect of the input signal to be retarded up to −9 psig from the actual input pressure.

This device may be used in sequencing devices of the same spring range as in figure 4-11. Here we are using a thermostat to sequence a heating valve, a cooling valve, and outside and return air dampers. By utilizing the amplifying relay our direct acting thermostat can cycle the normally open hot water valve closed before opening the normally closed outside air dampers. Since the outside air dampers and the chilled water valve are both normally closed and are sequenced by their spring ranges, we can sequence this application by utilizing a **minus 7 psig** bias adjustment on the relay. **(OUTPUT = INPUT + BIAS × 2).** Presuming a 9 psig branch pressure input (closing the hot water valve) plus the −7 bias adjustment times 2 equals a 4 psig output from the relay, a continued increase in branch line pressure to the relay will then operate the outside air dampers and chilled water valve in sequence.

PNEUMATIC RELAYS

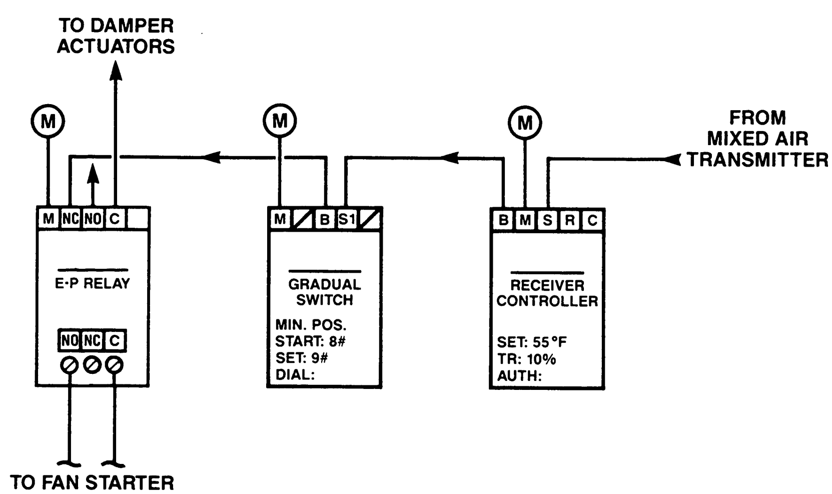
PNEUMATIC SWITCHES



GRADUAL SWITCH
FIGURE 4-12

Gradual Switch - This is a manually operated device designed to deliver a selected air pressure from 0-20 psig to its branch line. This switch is widely used in pneumatic control systems for remote positioning of final control devices and as a remote control point adjustment for receiver controllers. In this example (figure 4-12) the result of lowering or increasing the pressure signal from the gradual switch to the receiver controller will raise or lower the set point of the receiver controller to the control valve.

Minimum Positioning Switch - This device is a gradual switch with a built-in high pressure selector relay. It is used to maintain a minimum flow of outside air to meet code requirements for ventilation.



MINIMUM POSITION SWITCH
FIGURE 4-13

Figure 4-13 shows a minimum positioning switch used to position the outside air damper.

The controls are activated when the unit fan is on through the electric pneumatic relay. The transmitter and receiver controller are used to sense the mixed air temperature and position the dampers. The pressure required to maintain a minimum position is set. Whenever the controller output (branch) pressure is less than the minimum position switch set point, the switch will provide the minimum pressure to the 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 original position, closing the outside damper.

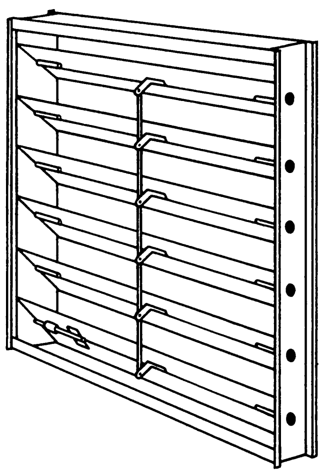
FINAL CONTROL DEVICES 5

DAMPERS

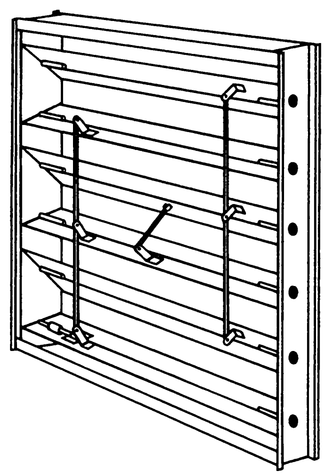
Airflow in a HVAC system is regulated by dampers which are installed in air-carrying ducts.

There are various types of dampers and blade designs including butterfly, flap, and louver. The louver-type damper is the most commonly used for controlling inlet or exhaust flow because it provides better control.

PARALLEL/OPPOSED BLADE DAMPERS



PARALLEL BLADE DAMPER
FIGURE 5-1



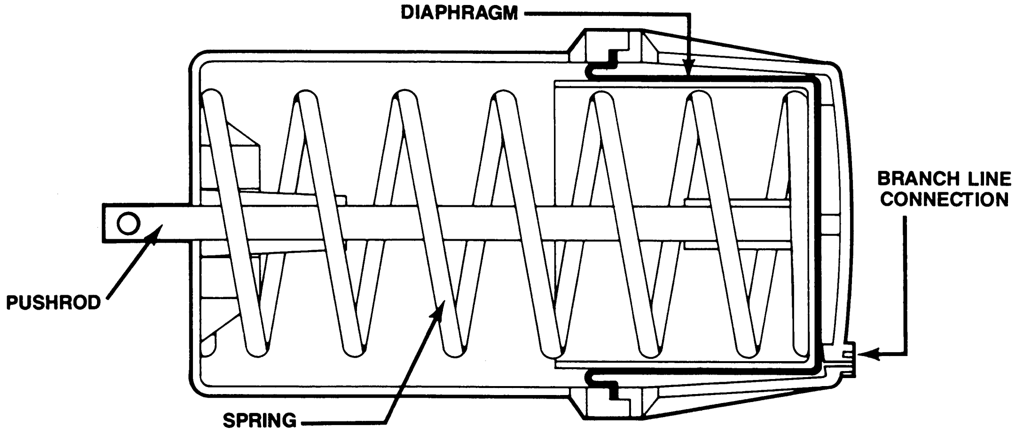
OPPOSED BLADE DAMPER
FIGURE 5-2

A parallel blade damper (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.

An opposed blade damper, as shown in figure 5-2, is constructed so that alternate blades rotate in opposite directions. They have an equal percentage airflow 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 airflow is required, such as variable air volume applications.

FINAL CONTROL DEVICES

DAMPER ACTUATORS



CUTAWAY OF DAMPER ACTUATOR

FIGURE 5-3

Pneumatic damper actuators position dampers according to the signal transmitted by a controller. The movement of the actuator's piston varies proportionally with air pressure applied to the diaphragm. This air pressure expands the diaphragm and forces the piston outward against the force of the spring. When air pressure is removed from the actuator, the spring returns the device to its normal position. See figure 5-3.

The **spring range** of the actuator restricts the movement of the piston to set limits. For example, an actuator with a 4-8 psig spring range is in its normal position when the air pressure applied to the device is 4 psig or less. Between 4 and 8 psig the stroke is proportional to the air pressure supplied. Above 8 psig the maximum stroke is achieved.

ROBERTSHAW DAMPER ACTUATOR SIZES

Stroke length (inches)	VS.	Damper Area (sq. ft.)
2		4
3		12
4		25
6		75

DAMPER STROKE VS. SQ. FT.

FIGURE 5-4

Damper actuators are rated by the damper area (in square feet) which they can operate. See figure 5-4. If the damper area is greater than the rating of the actuator, a larger, more powerful damper operator must be used. As an alternative, two or more smaller dampers with an actuator on each can be used, or two smaller actuators may be linked to the same damper.

CONTROL VALVES

The control valve is used as the means for regulating the flow of heating or cooling media. The flow being controlled may be steam, water or other heat transfer medium. The proper design and sizing of these control valves is essential to obtain the desired results from an air conditioning system.

A pneumatic actuated valve consists of the following components:

Actuator (operator) – The part of an automatic control valve which causes the valve stem to move.

Valve Body – The part of an automatic control valve through which the controlled medium flows.

Disc – The movable part of the valve that makes contact with the valve seat when the valve is closed and which varies the orifice controlling the flow. Discs are sometimes built so that the part of the disc that comes in contact with the seat may be replaced. This type of disc is known as a "renewable disc."

Guide – The part of the valve body that keeps the disc aligned with the valve seat. Top or bottom guides or both are used to accomplish this centering function.

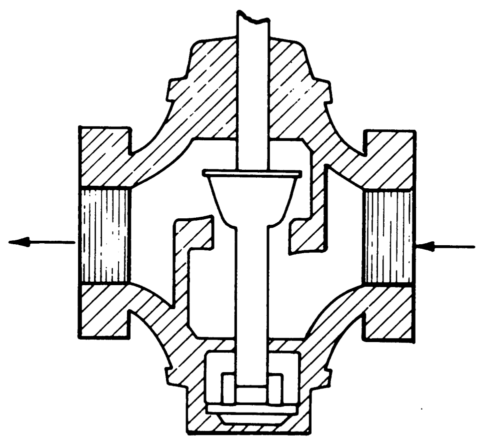
Port – This term refers to the flow controlling opening between the seat and the disc when the valve is wide open. It does not refer to body size or end connection size. Standard valve ports are the sizes normally used in the valve. Reduced port valves have flow controlling areas smaller than the valve port area of valves of the same body size.

Trim – Trim consists of all parts of a valve that are in contact with the flowing medium, but are not part of the valve shell or casting. Thus, seats, discs, throttling plugs, stems, packing rings, etc., are all trim components.

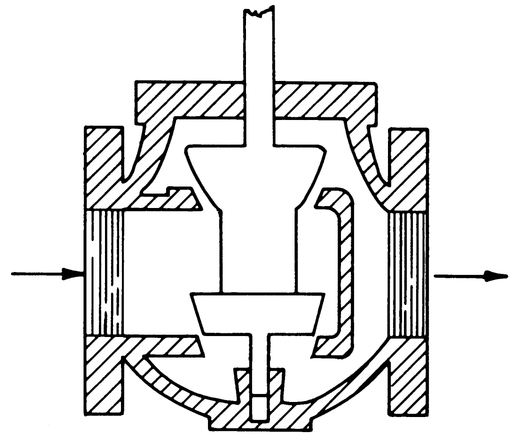
Control valves are classified according to the design of the valve body. They are known as 2-way valves, 3-way valves, single seated or double seated. When control valves are identified according to the control action, they are called normally open, normally closed, and 3-way mixing or diverting. Normally 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 3- way valve, they refer to the ports rather than the entire valve.

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 as valves in other cooling applications. Normally open and normally closed valves are often used in sequence or in combination for sequence operation.

CONTROL VALVES



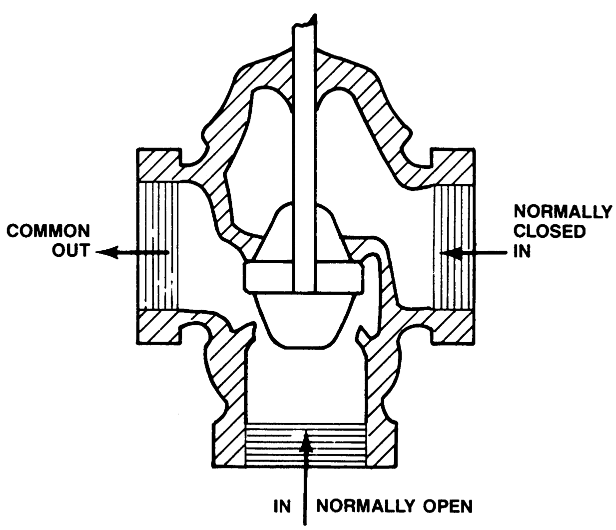
NORMALLY OPEN SINGLE-SEATED VALVE
FIGURE 5-5



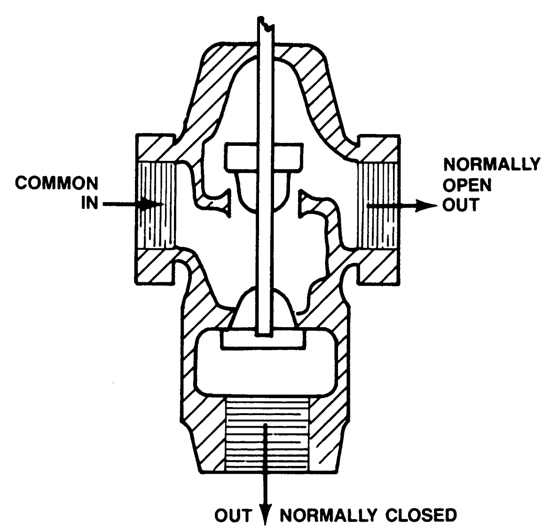
NORMALLY OPEN DOUBLE-SEATED VALVE
FIGURE 5-6

Single-seated valves have only one seat and one disc and are less expensive to construct than double seated valves. They are generally suitable for service where tight shutoff is required. Figure 5-5 shows a single seated valve. Since there is nothing in a single seated valve to balance the force exerted by the fluid pressure against the disc, the single seated valve will always require more force to close than a double seated valve of the same size.

Double-seated valves have two seats and two discs arranged in such a way that in the closed position there is very little fluid pressure forcing the media toward the open or closed position. For a valve of given size and port area, a double seated valve will always require less power to operate than a single seated valve. A double seated valve is shown in figure 5-6. An additional advantage of double seated valves is that they often have a larger port area for a given pipe size. A disadvantage is that they do not have tight shutoff since both discs are rigidly connected together and changes in temperature of the fluid being controlled will cause either the disc or the valve casting to expand, thus potentially allowing one disc to seat before the other. Impurities in the media may also affect the closing.



THREE-WAY MIXING VALVE



THREE-WAY DIVERTING VALVE

FIGURE 5-7

Three-way mixing valves have three openings. They always have 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 generally suitable for diverting service. This is because they have only one disc and two seats. If they are piped for diverting, the inlet pressure will slam the disc against the seat when it nears closing. This causes loss of control, vibration, excessive wear and noise.

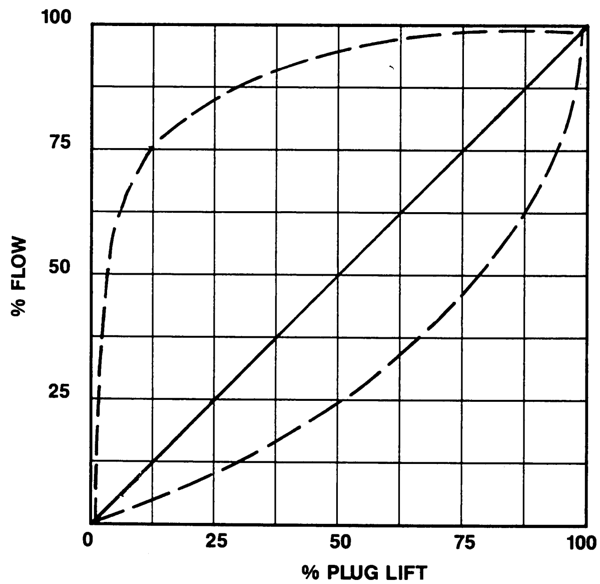
Three-way diverting valves always have one inlet and two outlets. Media entering the inlet port can be diverted to either of the two outlet ports in any proportion desired by the moving of the valve stem. Figure 5-7 illustrates a 3-way mixing and 3-way diverting valve.

FINAL CONTROL DEVICES

FLOW CHARACTERISTICS

Flow characteristic refers to the relationship between stem travel, expressed in percent of travel, and the flow of the medium through the valve expressed in percent of full flow. There are three common types of flow characteristics: quick opening, linear, and equal percentage.

1) **Quick Opening** – Maximum possible flow as soon as the stem starts to move upward. This type of valve usually has a flat seat with no throttling plug and is useful for 2-position control.
2) **Linear** – Regardless of percent of valve opening, the opening and flow are related in direct proportion. Double seated valves provide linear flow. They are used where high differential pressures are encountered or large flow capacities are required.
3) **Equal Percentage** – Each equal increment of opening increases the flow by an equal percentage. For example, suppose a valve stem lifts 30% of its total travel and the flow at this time is 4.0 GPM (gallons per minute). Next, suppose the valve stem moves an additional 10%. It would now be 40% open. The flow now is 6.4 GPM which is a 60% increase in flow. Next, assume the valve stem moves an additional 10% so that it is 50% open. The flow now is 10.2 gpm or another 60% increase in flow. Equal percentage discs are useful for control jobs where there will be wide variations in load from time to time. On water systems, utilizing either hot or chilled water, valves with equal percentage flow characteristics are almost always used. The equal percentage characteristic compensates for the nonlinear heat output characteristic of water coils.



VALVE RATINGS & TERMINOLOGY

1) **Capacity Index** – This is the quantity of water in gallons per minute at 60 degrees F that flows through a given valve with a pressure drop of 1 psi. It is sometimes called flow coefficient. The symbol for capacity index or flow coefficient is **Cv.** Once the Cv of the valve is determined, the flow of any fluid through the same valve can be calculated, provided the characteristics of the fluid and the pressure drop through the valve are known.
2) **Close Off** – The close off rating of the valve is the maximum allowable pressure drop to which the valve may be subjected while fully closed. This rating is a function of the power available from the valve actuator by holding the valve closed against pressure drop. The close off rating is independent of the actual body rating.
3) **Close Off Rating for 3-way Valves** – The close off rating for a 3-way valve is the maximum pressure difference between outlet and either of the two inlets for mixing valves or the pressure difference between the inlet and either of the two outlets for diverting valves.
4) **Maximum Fluid Pressure & Temperature** – These ratings are limitations placed upon a valve by the maximum pressure and temperature to which the valve may be subjected. The rating is derived from the type of packing, body material, disc material or actuator limitations.
5) **Pressure Drop** – Pressure drop is defined as the difference between the upstream pressure and the downstream pressure of the fluid passing through the valve.
6) **Critical Pressure Drop** – Fluid flow through a valve increases with increased pressure drop until a critical value is reached. Therefore, any drop in excess of the critical pressure drop should be avoided to insure noise-free operation and a minimum of valve wear.
7) **Body Rating** – This rating is defined as the correlation between safe permissible flowing medium pressure and flowing medium temperature. Each nominal valve body rating has definite corresponding permissible pressures at various temperatures. Thus the permissible pressure is usually dependent on the temperature.
8) **Nominal Body Rating** – This is the nominal pressure rating of the valve body expressed in psig. This rating is often cast on the valve body. The function of this rating is to provide a convenient method of classifying the valve by pressure class for identification.
9) **Rangeability** – Rangeability is defined as the ratio of the maximum controllable flow to the minimum controllable flow. For example, a valve with a rangeability of 50-1 and having a total flow capacity of 100 GPM, fully open, can control flow accurately down as low as 2 GPM. The valve may or may not have tight shut-off. However, the change in flow from 0 GPM to 2 GPM is accomplished with very little change in valve stem position.

CLOSE OFF RATINGS & SPRING RANGES

Pressure drop acting against the unbalanced area of a valve produces a thrust. This thrust must be overcome by the actuator through the application of additional signal pressure above the top end of the signal range for normally open control valves, or by reducing the signal pressure below the bottom end of the range for normally closed control valves. Normally open valves used on heating applications usually have lower spring ranges; example 4-8 psig. When 0-4 psig is applied to the actuator the valve is fully open. From 4-8 psig the normally open valve begins to close. It is fully closed at 8 psig. Consider that the available air pressure from a controller is 15 psig. This gives an additional 7 psig to hold the valve closed against flow. 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.

Spring ranges may vary slightly from one control manufacturer to another. For example, one manufacturer may supply an actuator with a 3-7 spring, another with a 4-8. This difference is very insignificant. An exception would be where valves are being sequenced for heating and cooling. The heating valve may be 3-7 psig, and the cooling valve 8-13 psig. The dead spot between 7 and 8 psig is incorporated into the design to prevent one valve from opening and the other from closing at the same time eliminating the possibility of simultaneous heating and cooling.

Some control valve manufacturers provide a means for making slight adjustments in spring ranges. 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 used to enable a valve to move exactly as far as a controller demands.

STEAM VALVES

The steam control valve should offer more resistance to the flow of steam than any other element in the system in order to effectively control flow. Thus, 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:

1) The valve pressure drop in a modulating system should be at least 80% of the difference between the supply and return main pressure. Exceptions are:
   a) The 80% drop exceeds 50% of the absolute upstream pressure. In this case, the 50% of the absolute upstream pressure should be used as the valve pressure drop.
   b) Due to special circumstances, an 80% drop would result in too low steam pressure in the heating device. Example – Pressures are being selected for a new building to be equipped with standing radiation. The radiators are selected to produce the designed quantity of heat when filled with steam at 1 psi. Atmospheric returns are to be used. Since the valve should have 80% pressure drop to give effective control, a supply main pressure should be selected which will allow the 1 psi, in the radiator, to become 20% of the difference between supply and return main pressure. 5# is selected as the supply main pressure. 80% × 5# = 4# valve pressure drop. This will allow an 80% drop through the valve and still allow 1# 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 out-put necessary to utilize an 80% valve drop is quite small. For example, a radiator supplied from a 5 psi boiler through a valve having a 1 psig (20%) drop will produce 100,000 BTU/hour. The radiator is carrying a pressure of 4 psi of steam. The same radiator equipped with a smaller valve having a 4# (80%) drop will produce 92,000 BTU/hour, a loss of 8%.

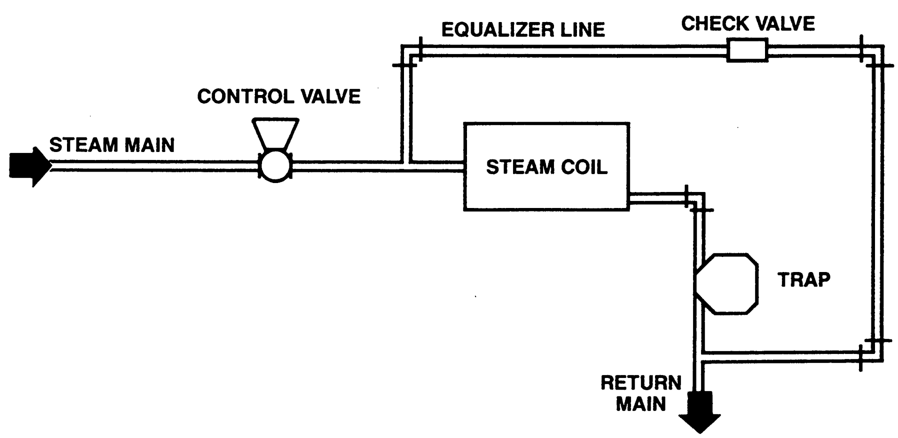
Because of this fact, it is possible to use fairly low supply main pressures and still utilize 80% of the supply to return main pressure difference for valve pressure drop without jeopardizing heat output of most steam-supplied devices.

SUPPLY & RETURN MAIN PRESSURES

As previously stated, the supply main pressure should be sufficient to allow an 80% 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-reducing valve 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, since 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.

CONSIDERATIONS FOR STEAM COILS



STEAM SUPPLIED AIR HEATING COIL

FIGURE 5-8

Figure 5-8 illustrates a steam-supplied air heating coil. The illustration shows the most favorable design conditions and piping arrangement that lends itself to good control valve performance. General rules for this requirement are as follows:

1) Main pressure should be held at design pressure ± 1 psi.
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 psi drop.
4) An equalizer line prevents formation of vacuum within the coil.
5) Control valve drop should be 80% of difference between supply and return main pressures.

The effect of fluctuating supply and return main pressures can be significant. For example, assume a system where the boiler cycles so that it shuts off at 6# and cuts in at 2#. In the same system, assume that a vacuum pump is used which cuts in at 4" and cuts off at 8" of vacuum. The pressure difference between the supply and return mains could vary from a minimum of 4 psi to a maximum of 10 psi as the boiler and vacuum pump go through their cycles. This means 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 originally for the 4 psi drop, would be 50% too large during periods when the 10 psi drop existed across the supply and return mains.

An exception to this would be a high vacuum system. Their purpose is to lower the steam temperature and pressure as the heating load decreases. Vacuum systems are generally adaptable for the use of automatic 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 HOT 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 exiting water temperature may be fairly close to steam temperature. Thus the selection of supply pressures and valve pressure drop is significant.

Converters may be required to deliver hot water at varying degrees of temperature. As the difference between the delivered water temperature and steam temperature goes down, the size and cost of a converter usually goes up. For example, a converter supplying water to a building at 120°F would only have to be 1/3 the size of a converter supplying the same building with 200°F water. Unlike steam coils for air, the capacity of a converter varies quite drastically with changes in steam pressure delivered to it. For example, consider a converter supplying 180°F water to a building. The converter is designed to operate on 10 psi saturated steam. If the pressure in the converter is reduced to 5#, a 17% reduction in capacity would result. If the steam pressure is reduced to 2#, the converter would be operating at only 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 50% drop may be used if available steam main pressure prevents an 80% drop from being used.

For example, if a maximum of 10# of steam is available to a converter with an atmospheric return, an 80% valve pressure drop will leave 2# of steam in the converter. At this pressure, a larger converter would be required. Under these circumstances, a compromise could be made by using a valve with a 50% pressure drop.

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. Since steam is readily diffused without the necessity of adding heat to the air, the air receives very little sensible heat.

Pan humidifiers normally require 5 psi steam since 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 psi. Spray and steam grid humidifiers require only 0.5 psi to 1 psi 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.

SELECTING STEAM VALVES

Once a steam distribution system has been designed, and steam pressures have been selected that will allow effective operation of the control valve, specific valves can be selected using the following steps:

1) Determine steam required (W) in pounds per hour.
2) Determine pressure drop across the valve.
3) Find capacity index required (Cv).
4) Select the valve having the required specifications.

The following formulas can be used to determine pounds per hour of steam for different types of equipment:

1) When BTU/hour (heat input) is known:

$$W = \frac{\text{BTU/hour}}{1000}$$

2) When EDR (equivalent direct radiation) is known:

$$W = EDR \times .24$$

3) When sizing coil valves for fan systems:

$$W = CFM \times 1.08 \times \frac{TD_a}{1000}$$

CFM = air quantity in cubic feet per minute.
TD_a = air temperature difference across the coil.

4) When sizing converter valves - steam to hot water:

$$W = GPM \times TD_w \times .49$$

GPM = water flow in gallons per minute.
TD_w = water temperature difference.

5) When sizing steam jet humidifier valves:

$$W = CFM \times 60 \times .075 \frac{(GR_1 - GR_2)}{7000}$$

CFM = air quantity in cubic feet per minute.
GR_1 = grains of moisture per pound of air leaving the humidifier.
GR_2 = grains of moisture per pound of air entering the humidifier.

6) When sizing pan type humidifier valves:

$$W = CFM \times 60 \times \frac{(GR_1 - GR_2)}{7000} \times 1.5$$

Determining pressure drop across the valve should be calculated as follows:

$$\text{Pressure Drop} = 80\% \ (P_1 - P_2)$$

P_1 = supply main pressure psi. P_2 = return main pressure psi. $P_1 - P_2$ should be the lowest pressure that will normally exist across supply and return mains nearest the control valve.

SELECTING STEAM VALVES

VALVE SIZING TABLE

C_v	STEAM CAPACITY IN #/HR.							$W = 2.1\,C_v\,\sqrt{\Delta P}\,\sqrt{P_1 + P_2}$						WATER CAPACITY GPM $Q = C_v\sqrt{\frac{\Delta P}{G}}$			
	5# STEAM @ PRESS. DIFF.			10# STEAM @ PRESS. DIFF.						15# STEAM @ PRESS. DIFF.				POUNDS DIFF. PRESSURE			
	2	3	4*	2	3	4	5	7	8*	5	8	10	12*	1	2	3	5
.25	4.38	5.4	6.25	4.6	5.68	6.59	7.39	8.69	9.3	7.84	9.92	11.1	12.12	.25	.354	.434	.56
.50	8.75	10.8	12.5	9.2	11.35	13.15	14.75	17.35	18.6	15.65	19.80	22.1	24.25	.50	.706	.865	1.10
.60	10.5	12.9	15.0	11.05	13.62	15.8	17.70	20.8	22.3	18.8	23.8	26.65	29.1	.60	.850	1.04	1.34
.80	14.0	17.3	20.0	14.72	18.15	21.1	23.6	27.8	29.8	25.1	31.7	35.6	38.8	.80	1.13	1.39	1.79
.90	15.8	19.5	22.5	16.56	20.4	23.7	26.6	31.3	33.5	28.2	35.7	39.8	43.6	.90	1.27	1.56	2.01
1.0	17.5	21.6	25.0	18.4	22.7	26.3	29.5	34.7	37.2	31.3	39.6	44.2	48.5	1.0	1.42	1.73	2.24
1.1	19.3	23.8	27.5	20.3	24.9	28.9	32.4	38.2	40.9	34.2	43.6	48.9	53.4	1.1	1.56	1.91	2.46
1.2	21.0	25.9	30.0	22.1	27.2	31.8	35.4	41.6	44.6	37.6	47.6	53.4	58.2	1.2	1.70	2.08	2.68
1.6	28.0	34.6	40.0	29.4	36.3	42.1	47.2	55.6	59.5	50.1	63.4	71.0	77.6	1.6	2.26	2.78	3.58
1.8	31.6	38.9	45.0	33.1	40.8	47.4	53.2	62.5	67.0	56.4	71.4	79.9	87.4	1.8	2.55	3.12	4.03
2.0	35.0	43.2	50.0	36.8	45.4	52.6	59.0	69.4	74.4	62.6	79.2	88.4	97.0	2.0	2.84	3.46	4.48
2.2	38.5	47.6	55.0	40.4	49.9	57.8	64.9	76.4	81.6	68.9	87.2	97.6	106.9	2.2	3.12	3.82	4.93
2.4	42.0	51.9	60.0	44.2	54.4	63.2	70.9	83.4	89.4	75.2	95.4	106.5	116.4	2.4	3.40	4.16	5.36
2.5	43.8	54.0	62.5	46.0	56.8	65.9	73.9	86.9	93.0	78.4	99.2	111.0	121.2	2.5	3.54	4.34	5.6
3.0	52.5	64.8	75.0	55.2	68.1	78.9	88.5	104.1	111.6	93.9	118.8	132.6	145.5	3.0	4.26	5.19	6.72
4.0	70.0	86.4	100.0	73.6	90.8	105.2	118.0	138.8	148.8	125.2	158.4	176.8	194.0	4.0	5.68	6.92	8.96
4.5	78.9	97.1	112.5	82.9	101	118.5	132.9	156.1	167.5	141.0	178	200	218	4.5	6.36	7.80	10.1
4.6	80.6	99.5	115.0	84.6	104.2	121	135.6	159.9	172	144.5	182	204.5	223	4.6	6.51	7.96	10.3
5.0	87.5	108.0	125.0	92.0	113.5	131.5	147.5	173.5	186	156.5	198	221	242.5	5.0	7.1	8.65	11.2
5.3	93.8	114.4	132.5	97.5	120.2	139.5	156.2	184	197	166	210	235	257	5.3	7.50	9.20	11.89
5.6	98.4	121.0	140.0	103.0	127.0	147.5	165.1	194.5	206	175.4	222	249	272	5.6	7.94	9.72	12.5
5.7	99.9	123.0	142.5	105.0	129.2	150	168	198	212	178.5	226	253	277	5.7	8.06	9.89	12.75
6.3	110.0	136.0	157.9	115.9	143.0	165.9	186	219	234	197.5	249.9	280	306	6.3	8.93	10.90	14.1
6.5	114.9	140.5	162.5	119.5	147.5	171	192	226	242	203.9	257.9	289	315	6.5	9.20	11.28	14.5
9.0	157.5	194.4	225.0	165.6	204.0	237	266	312.3	335	282	356.4	398	436.5	9.0	12.78	15.57	20.16
9.3	163.9	201.0	232.0	171.2	211	245	274	323	346	291.2	368	414	452	9.3	13.15	16.10	20.8
9.5	166.2	205.5	238.0	174.9	218	250	281	350	353	297.9	378	423	462	9.5	13.41	16.49	21.2
11	193	238.0	275.0	203	249	289	324	382	409	342	436	489	534	11	15.6	19.1	24.6
12	210	259	300	221	272	318	354	416	446	376	476	534	582	12	17.0	20.8	26.8
15	263	324	375	276	341	394	443	521	558	468	594	666	726	15	21.2	26.0	33.5
16	280	346	400	294	363	421	472	556	595	501	634	710	776	16	22.6	27.8	35.8
17	298	368	425	313	386	446	501	592	634	533	673	756	824	17	24.0	29.5	38.0
18	316	389	450	331	408	474	532	625	670	564	714	799	874	18	25.5	31.2	40.3
21	368	454	525	386	476	554	620	729	782	656	831	935	1019	21	29.97	36.4	47.0
22	385	476	550	404	499	578	649	764	816	689	872	976	1069	22	31.2	38.2	49.3
23	404	497	575	424	522	604	678	799	856	722	912	1021	1115	23	32.5	39.8	51.2
25	438	541	625	460	568	659	739	869	930	784	992	1110	1212	25	35.4	43.4	56.0
30	525	648	750	552	681	789	885	1041	1116	939	1188	1326	1455	30	42.6	51.9	67.2
38	666	820	950	699	636	998	1121	1320	1415	1190	1505	1689	1842	38	53.8	65.9	84.0
40	700	864	1000	736	908	1052	1180	1388	1488	1252	1584	1768	1940	40	56.8	69.2	89.6
47	824	1015	1175	865	1068	1239	1386	1632	1749	1472	1860	2065	2280	47	66.5	81.5	105.0
48	842	1038	1200	884	1089	1264	1419	1669	1785	1502	1900	2180	2330	48	67.9	83.4	107.2
70	1228	1512	1750	1288	1589	1841	2055	2429	2604	2191	2772	3094	3395	70	99.4	121.1	156.8
72	1260	1555	1800	1325	1635	1899	2125	2500	2680	2259	2850	3199	3490	72	102.0	124.7	161.0
75	1312	1620	1875	1380	1700	1975	2215	2610	2790	2350	2970	3330	3640	75	106.0	130.0	167.5
93	1639	2010	2320	1712	2110	2450	2740	3230	3460	2912	3680	4140	4520	93	131.5	161.0	208.0
94	1649	2030	2350	1730	2140	2470	2779	3270	3498	2945	3720	4170	4560	94	133	163	210
100	1750	2160	2500	1840	2270	2630	2950	3470	3720	3130	3960	4420	4850	100	142	173	224
102	1782	2210	2550*	1865	2319	2680	3010	3540	3799	3199	4040	4530	4940	102	144	177	228
104	1820	2250	2600	1910	2360	2730	3070	3620	3870	3259	4120	4620	5040	104	147	180.5	232
164	2870	3540	4100	3020	3720	4320	4840	5700	6100	5140	6490	7270	7950	164	233.2	284	366
170	2980	3680	4250	3130	3860	4460	5010	5920	6340	5330	6730	7560	8740	170	240	295	380
174	Not Available for Steam			Not Available for Steam						Not Available for Steam				174	246	302	389
200	3500	4320	5000	3680	4540	5260	5900	6940	7440	6260	7920	8840	9700	200	284	346	448
250	Not Available for Steam			Not Available for Steam						Not Available for Steam				250	354	434	560
360	Not Available for Steam			Not Available for Steam						Not Available for Steam				360	509	624	805

*Recommended Pressure Drop (80% x P_1)

FIGURE 5-9

Once the facts regarding steam quantity and pressure drop have been determined, a capacity index (Cv) requirement can be determined. This can be determined by using the valve sizing table shown in figure 5-9.

Steam valve sizing examples:

1) It is desired to control steam flow to a steam to water heat exchanger. The following data is given: 10 psi steam pressure; water flow 8 GPM; pressure drop 4#. The inlet temperature is 180°F and the outlet temperature is 200°F.

   $W = GPM \times TD_W \times .49$

   $W = 8 \times 20 \times .49$

   W = 78.4# / hour

   Find the Cv by referring to the valve sizing table in figure 4-9. Under 10 psi steam and a 4# pressure drop, it can be determined that a valve with a Cv of 3.0 can pass 78.4# / hour.

2) Determine the size of a pneumatic valve for a steam coil with the following information: a fan system which develops 20,000 CFM with a coil entering air of 70°F and leaving air of 90°F.

   $$W = CFM \times 1.08 \times \frac{TD_a}{1000}$$

   $$W = 10{,}000 \times 1.08 \times \frac{20}{1000}$$

   W = 216# / hour

3) A radiation zone has 4,000 EDR of radiation installed. Determine the amount of flow (W) required to flow through a zone control valve.

   $W = EDR \times .24$

   $W = 4{,}000 \times .24$

   W = 960# / hour

Once all of the above information has been determined, a valve can be selected that will meet the desired specifications.

1) Tight shut-off requirements must utilize a single seated valve. If this is not required, a double seated valve can often be used to advantage, since it has a higher close off rating.
2) Consider limitations in pressure drop. These limitations are based on normal life of the seat and disc when steam is flowing through the valve at the maximum expected pressure drop. Since the greatest steam velocity and wear occur when the valve is almost closed, and since two-position valves are always opened or closed, the pressure drop limitations for two-position valves are considerably higher than for modulating valves. The modulating valve may assume a nearly closed position for long periods of time.
3) Temperature and fluid pressure limitations must be considered. Either the valve body or the packing may limit the steam pressure and temperature that can be used.
4) Consider flow characteristics, normally linear or equal percentage for modulating valves and quick opening for two-position valves.
5) The maximum ambient temperature rating for actuators is important and vary according to manufacturers since different materials are used.
6) Determine the body pattern needed, such as screwed or flanged, angle, straight through, three-way, etc.
7) Select size of valve to meet Cv requirement. Some valves are available having the same Cv capacity in several body sizes. However, normally the Cv rating increases as the body size increases.
8) It is important to select a valve with the proper close-off rating so that it will close off under the highest pressure differences that can occur.
9) A positive positioning relay may be needed, particularly if valves must be sequenced with other equipment. For example, a heating valve may operate from 2-6 psig and the cooling valve from 8-13 psig. Also, if very accurate control is desired, the positioner will allow the valve to move exactly as far as the controller signal demands, and valve position will not be affected by packing friction, fluid pressures, etc.

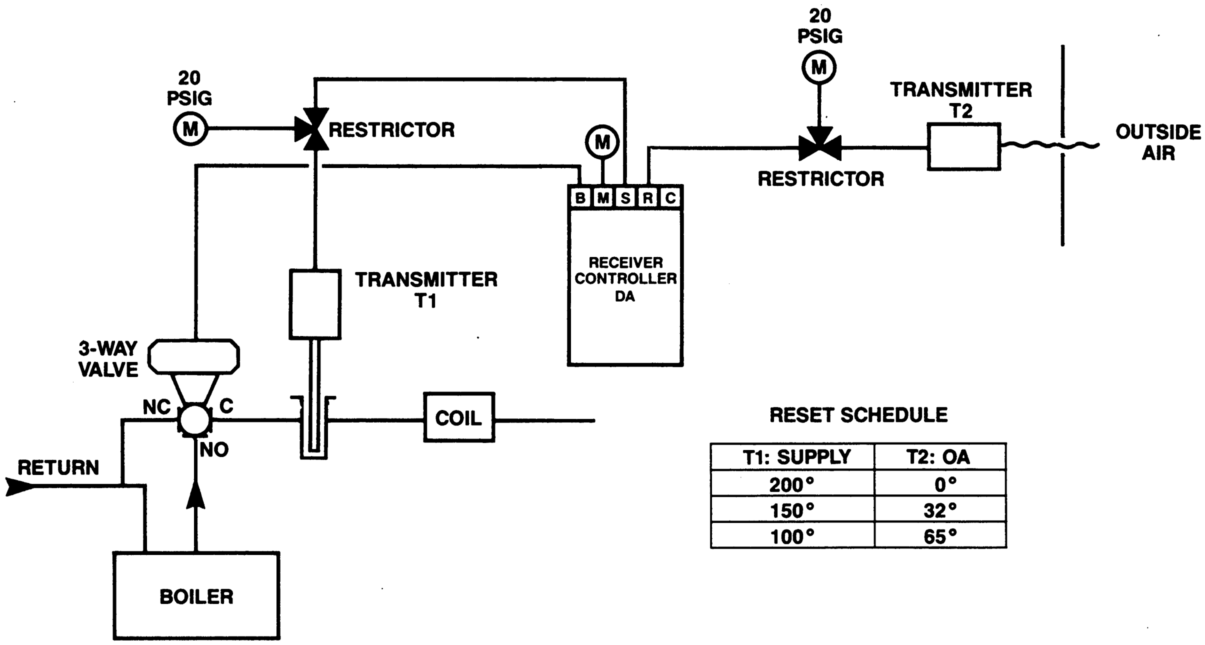
WATER VALVES

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. Supply water temperature control is the most effective method of controlling BTU output of a water supply. Flow control is suitable for control of individual terminal units, such as fan coils and induction units. However, a combination of temperature and flow control of supply water is the best control method for heating systems.



RESET SCHEDULE

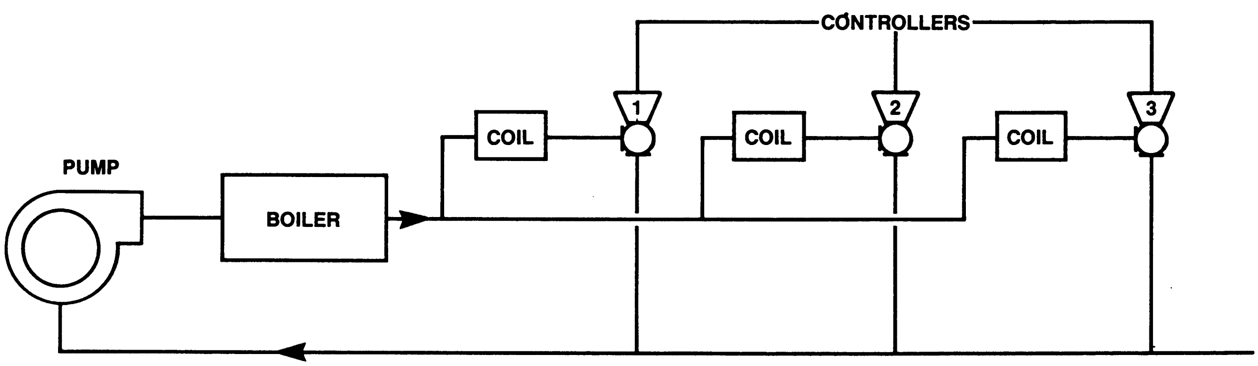
T1: SUPPLY	T2: OA
200°	0°
150°	32°
100°	65°

DISCHARGE WATER TEMPERATURE CONTROL
FIGURE 5-10

There are several ways to vary supply water temperature to coils. One of the most common is shown in figure 5-10.

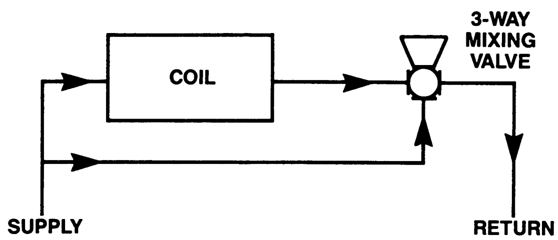
In this system, transmitter T2 measures outside air temperature. Transmitter T1 measures supply water temperature and maintains a preselected schedule of water temperature in accordance with the outside air temperature. Also shown is a reset schedule which is predetermined in accordance with design conditions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

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 2-way valves or 3-way valves.



FLOW CONTROL USING TWO-WAY VALVES
FIGURE 5-11

When 2-way valves are used as shown in figure 5-11 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. Selection and design of distribution systems is discussed under "Water Distribution Systems."

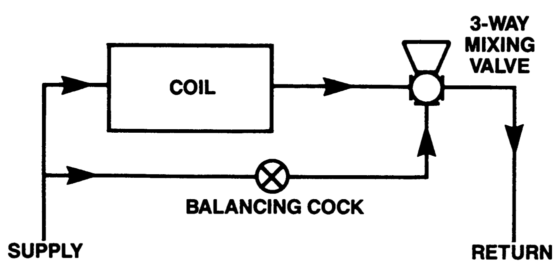


COIL BYPASS USING THREE-WAY VALVE

FIGURE 5-12

3-way valves are often used to control flow to coils, particularly when it is desired to maintain a constant flow through the mains. Figure 5-12 illustrates this application. As the valve reduces the flow through the coil, it increases the flow through the bypass around the coil. Thus the total flow through the system remains relatively constant and the system friction loss remains relatively constant.

Further examination will show 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. Since the bypass usually has less resistance than the circuit through the coil, there will be an increase in flow. If the resistance through the bypass is one half of that through the coil, the flow will increase. This situation occurring at even a few coils throughout a system can seriously unbalance the circuit. If it occurs simultaneously at a large number of coils, the effect can be appreciable.



COIL BYPASS WITH BALANCING COCK

FIGURE 5-13

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 5-13.

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 as when the valve is in one extreme position or the other.

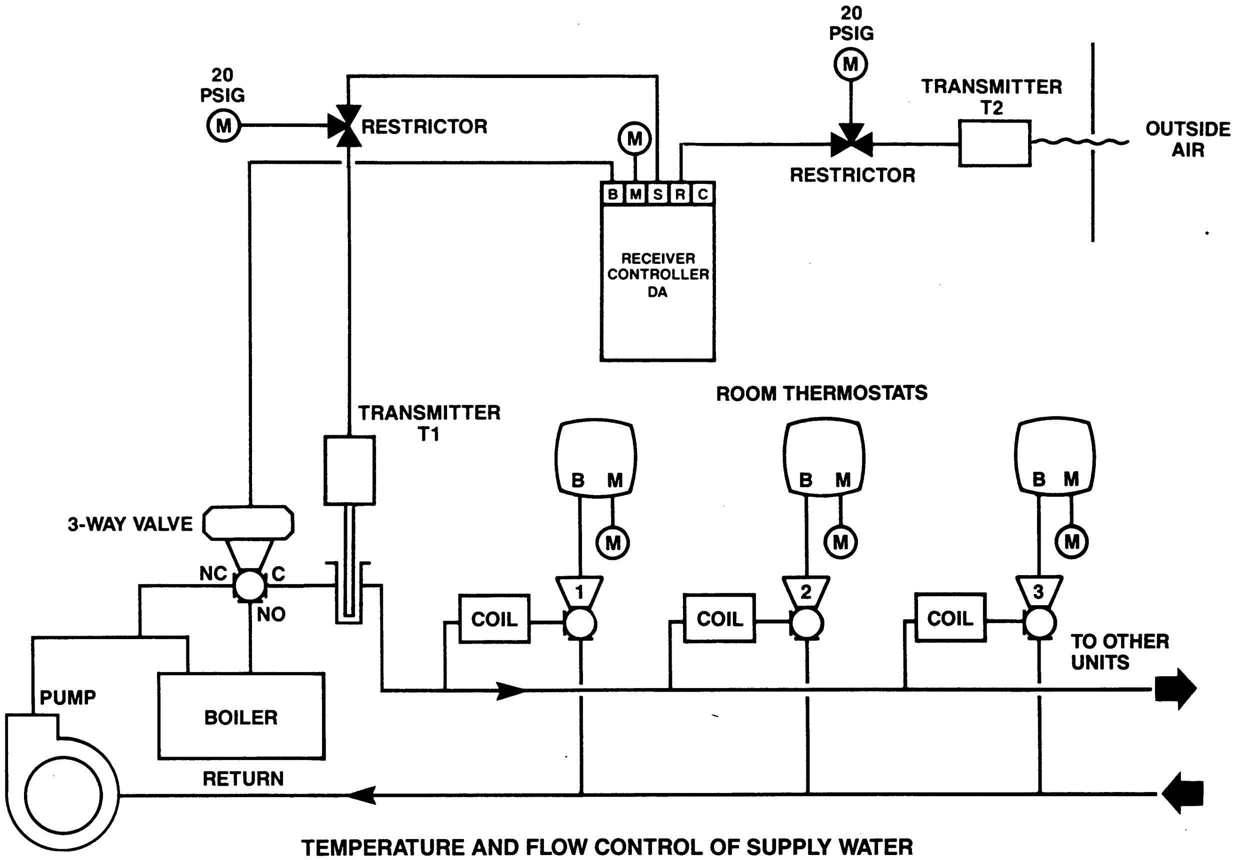
Even with the use of 3-way valves and a properly designed and balanced circuit, it is not possible to maintain complete system balance under all conditions of operation. The use of 3-way valves does not cure all of the system's problems. It is still necessary to carefully design the system and consider all variables that may affect its operation. The following things should be considered when using a 3-way coil bypass valve:

1) As far as the coil output is concerned, a 3-way valve controls output by varying flow the same as a 2-way valve.
2) Piping costs can be higher for 3-way than for 2-way valves, especially where piping space is limited.
3) Most 3-way valves are available only with linear flow characteristics. Equal percentage 2-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 use of scheduled water temperatures.
4) Constant flow in mains is possible in a system using 3-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 because of piping requirements to individually control supply water temperature to each heating unit. 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 unit valves to open wider than would be the case with a constant temperature of water. This keeps flow through the pump and boiler fairly uniform during this season.
2) During initial warm-up of a room or during heavy load changes, overshooting of temperature is not likely to occur since maximum heating unit output is regulated from the outdoor temperature.

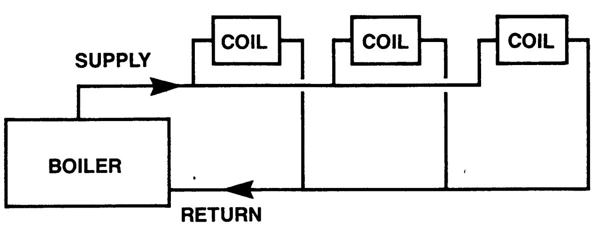


TEMPERATURE AND FLOW CONTROL OF SUPPLY WATER
FIGURE 5-14

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 5-14.

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.

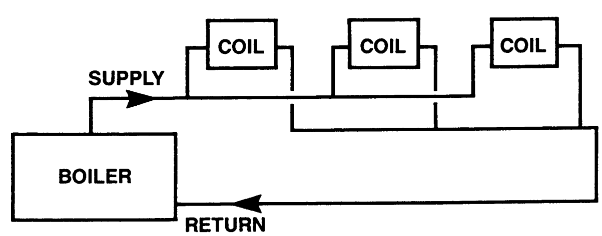


TWO-PIPE DIRECT RETURN SYSTEM
FIGURE 5-15

The two most common distribution systems are direct return and reverse return. Figure 5-15 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 would 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. Since 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 would be time-consuming and expensive. For this reason, the direct return distribution system is recommended only for constant flow small systems.



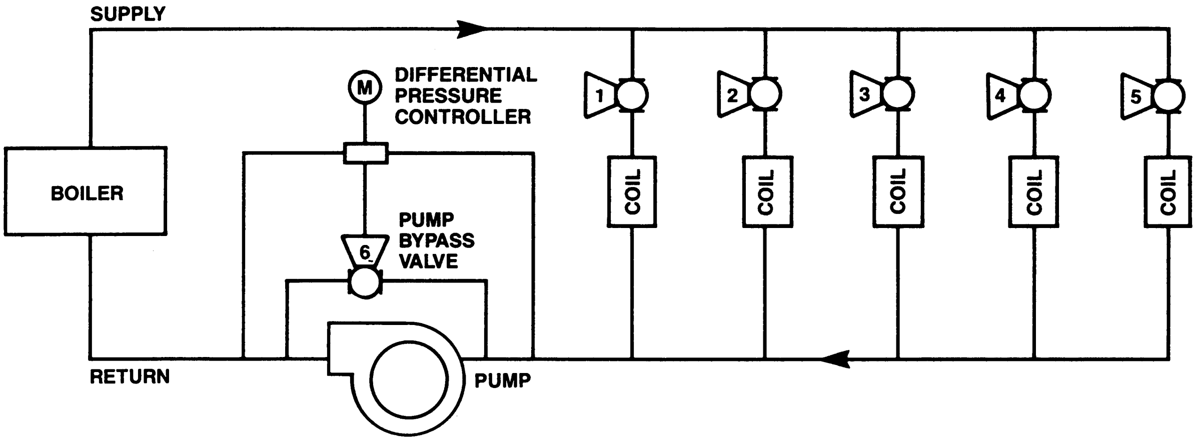
TWO-PIPE REVERSE RETURN SYSTEM
FIGURE 5-16

In a reverse return system shown in figure 5-16, 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 of all arrangements 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.

REGULATION OF SUPPLY PRESSURE DIFFERENTIAL

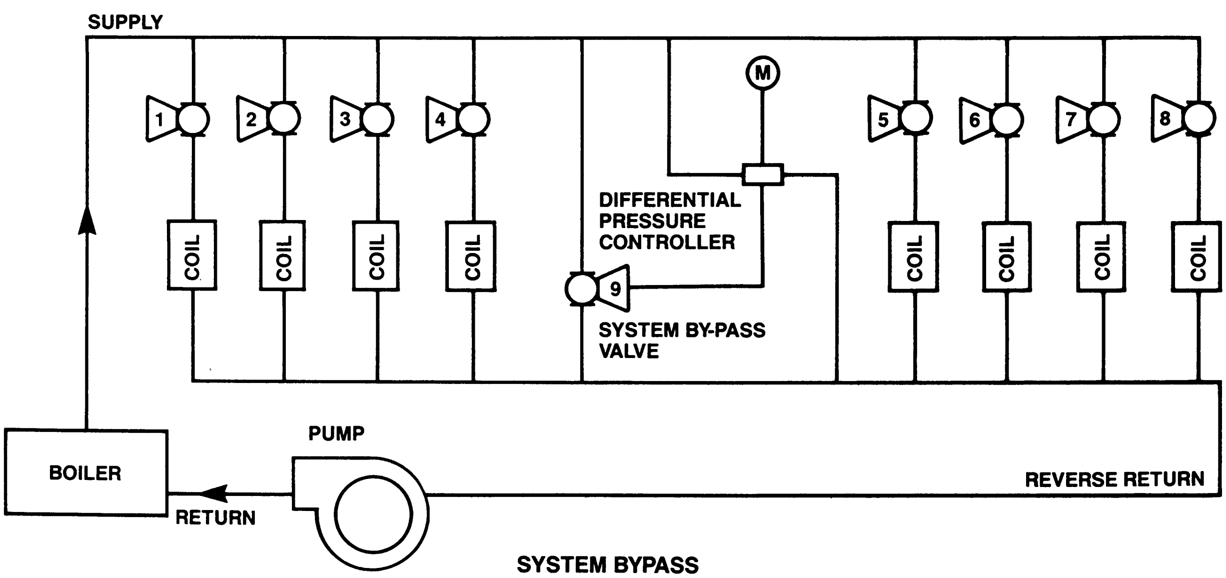
Decreases in flow not only decrease piping pressure loss, but also increase the pressure differential generated by a typical pump. When valve pressure drop at minimum flow is more than three times the valve pressure drop at maximum flow, then a means of differential pressure regulation should be used.



PUMP BYPASS
FIGURE 5-17

Many applications specify a pump bypass as shown in figure 5-17 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. The 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 build-up 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 5-18

Another method of regulating system differential pressure is by using a system bypass valve as shown in figure 5-18. 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 as the unit control valves close to decrease 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.

SELECTING WATER VALVES

After the water circulating system has been designed and system pressures have been determined, control valves can be selected.

1) Determine the needed gallons per minute (GPM).
2) Determine desired pressure drop across valve.
3) Find capacity index (Cv) needed.
4) Select a valve having proper requirements.

To find GPM it is necessary to know the BTU output and design water temperature drop of the heat-exchanging device. The formula below should be used to determine GPM:

$$GPM = \frac{BTU / Hour}{K \times TD_W}$$

TD_W = water temperature difference.

WATER TEMPERATURE °F	K	WATER TEMPERATURE °F	K
60	500	225	483
80	498	250	479
100	496	275	478
120	495	300	473
150	490	350	470
180	487	400	465
200	484		

K-FACTOR CHART
FIGURE 5-19

K = factor selected from the table shown in figure 5-19.

When equivalent direct radiation (EDR) is known, use the following method:

AVERAGE RADIATOR OR CONVECTOR TEMPERATURE °F	CAST IRON RADIATOR BTU / HR / EDR *	CONVECTOR BTU / HR / EDR †
215	240	240
200	209	205
190	187	183
180	167	162
170	148	140
160	129	120
150	111	102
140	93	85
130	76	69
120	60	53
110	45	39
100	31	27
90	18	16

* AT 70°F ROOM TEMPERATURE
† AT 65°F INLET TEMPERATURE

OUTPUT OF RADIATORS AND CONVECTORS
FIGURE 5-20

$$GPM = \frac{EDR \times (\text{Correct BTU/Hr/EDR value from table in figure 5-20})}{K \times TD_W}$$

An example for determining GPM when EDR is known could be as follows:

Determine the GPM required to pass through a control valve on a length of cast iron radiation which has an EDR rating of 180. The water temperature is 200°F entering and 180°F leaving:

$$GPM = \frac{EDR \times (\text{Correct BTU / Hr / EDR})}{K \times TD_W}$$

$$GPM = \frac{180 \times 209}{484 \times 20} \qquad GPM = \frac{37620}{9680} \qquad GPM = 3.88$$

Considering a design allowance for a 5 psi drop across the control valve, the valve sizing table in figure 4-9 indicates the valve with a Cv of 1.6 would do the job.

To calculate GPM when sizing hot water coil valves for fan systems, use the following formula:

$$GPM = \frac{CFM \times 1.08 \times TD_a}{K \times TD_W}$$

CFM = air quantity in cubic feet/minute.

TD_a = air temperature difference.

For sizing cold water coil valves for fan systems, the following formula should be used:

$$GPM = \frac{CFM \times \text{BTU / lbs. of dry air removed}}{113 \times TD_W}$$

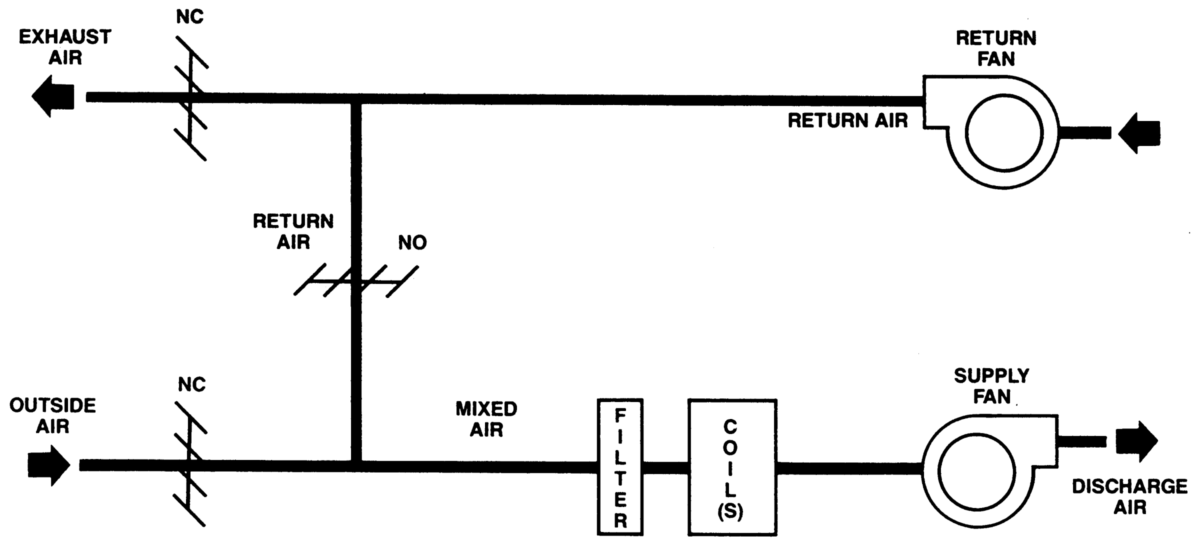
Dry air removed = total heat removed.

When utilizing this formula it is necessary to determine the pounds of dry air removed from an appropriate psychometric chart.

The pressure drop across a water valve should be at least 50-70% of the design pressure difference across the supply and return mains nearest the valve. This will provide the best modulating control. For example, if the drop across supply and return mains at design flow is 12 psi, the desired pressure drop across a valve should be at least 6-8.4 psi.

NOTES

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, or fresh air, is normally required in at least some minimum quantity to meet building codes during occupancy periods. The air passes through a normally closed outside air damper and mixes with the return air. This is the mixed air supply.

The air is then conditioned through any required filtering and then passes through the heating or cooling coil, or both. The discharge air from the supply fan then feeds all parts of the system.

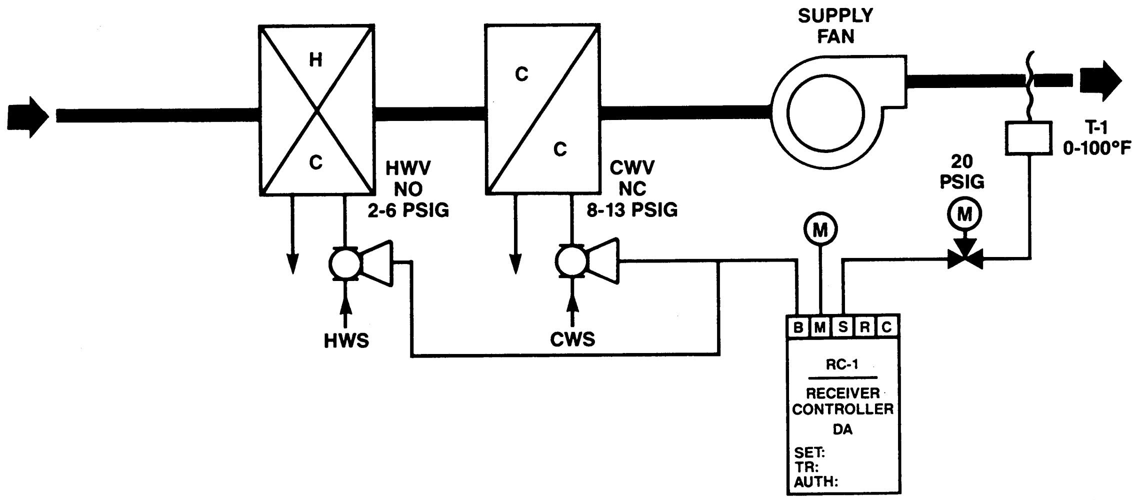
If the system includes a return air fan, air from the controlled space may be exhausted to the outside. In other instances, some or all of the return air may be routed through the return air damper. The direction of return air flow is determined by the mixed air controls which position the outside, return 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 building's normal exfiltration through doors, windows, cracks, etc.

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

CONTROL APPLICATIONS

DISCHARGE AIR CONTROL



DISCHARGE AIR CONTROL

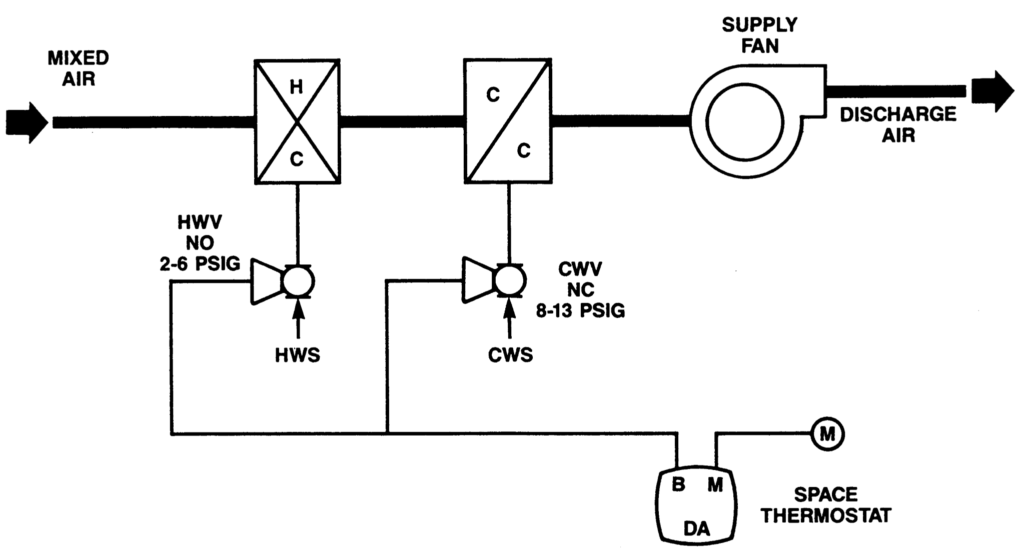
FIGURE 6-2

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°F range transmitter (T-1) is sensing discharge air temperature. A direct acting receiver controller (RC-1) positions the normally open hot water valve and normally closed chilled water valve. As the discharge air temperature rises, the branch pressure of the receiver controller increases, closing the normally open heating valve. When the temperature rises above set point, the normally closed cooling valve is modulated toward the open position to provide cooling.

The proper selection of spring ranges, to provide a dead band between the heating and cooling functions, prohibits simultaneous heating and cooling.

SINGLE-ZONE SYSTEM



SINGLE-ZONE SPACE CONTROL

FIGURE 6-3

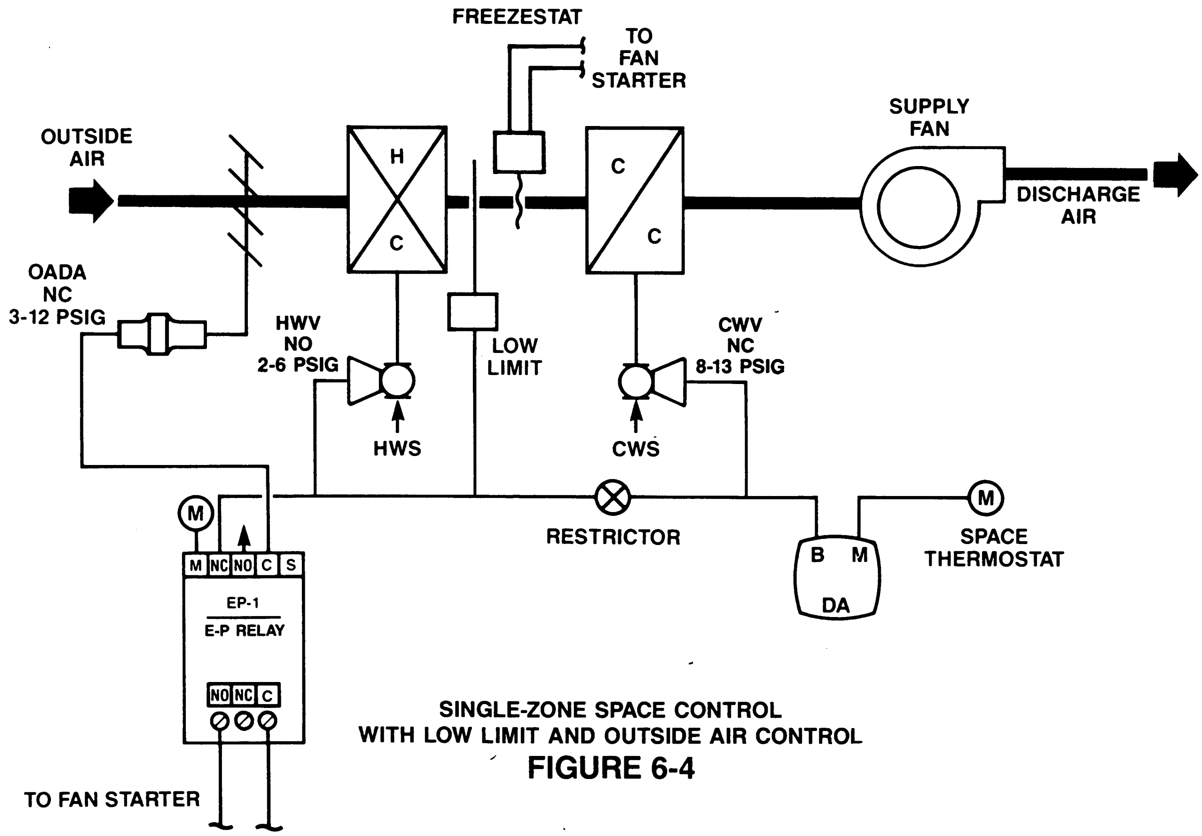
This system (figure 6-3) provides simple control of space temperature. The temperature is controlled by the direct acting room thermostat which controls a hot water supply valve and a chilled water supply valve in response to changes in room temperature.

In operation, the branch line from the thermostat provides a 3-15 psig signal to the hot and chilled water valves. When the thermostat requires an increase in room temperature, the hot water valve modulates open. As the temperature increases, the branch pressure increases. When the temperature rises to the set point, the branch pressure increases to close the hot water valve.

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 set point, the normally closed chilled water valve will begin to open. The chilled water valve has a spring range of 8-13 psig, thus it will throttle to the fully open position as temperature continues to increase.

Both valves will be closed between 6 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, FREEZESTAT, AND OUTSIDE AIR CONTROL



SINGLE-ZONE SPACE CONTROL
WITH LOW LIMIT AND OUTSIDE AIR CONTROL
FIGURE 6-4

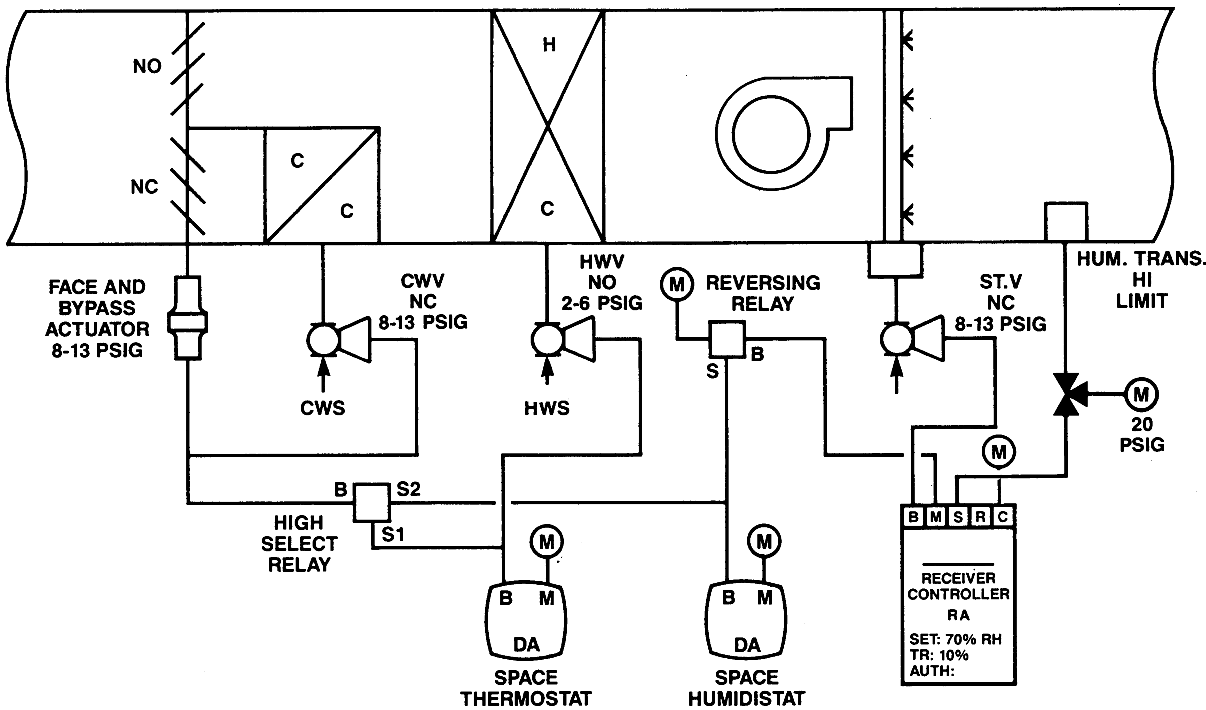
In this system (figure 6-4) the same function is being performed by the space thermostat as in the previously discussed single-zone system. However, additional controls that are often found in conjunction with the thermostat and coil controls in a system like this are added. This system also includes control of the outside air damper, the E-P relay that ties into the fan starter circuit to provide shutdown at night, and low-limit and freeze protection controls.

In this sequence of operation, the fan starter is tied to the E-P relay controlling the branch line airflow to the outside air damper actuator. When the fan is off, the E-P relay exhausts the branch line to the outside air damper allowing it to go to the fully closed position. Also tied into the control circuit for the fan starter is a freezestat which is another electrical device. The other pneumatic device shown is the low-limit control. This 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 drops below a nominal 50-55 degrees. The additional restrictor shown is an adjustable restrictor in the line ahead of the low-limit control and the hot water supply valve. This restrictor limits the amount of air flow to the devices downstream and, in the event that the low-limit control senses a decreased mixed air temperature, the low-limit will bleed off branch line air allowing the hot water supply valve to open further than the space thermostat may be allowing.

The freezestat mentioned above would come into action if the low-limit was not able to compensate for the 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 drop, 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 temperature range of this device is from 35 to 45°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 would potentially be a minimum position control on the outside air damper and control of return air and exhaust air dampers also.

SINGLE-ZONE CONTROL OF SPACE TEMPERATURE AND HUMIDITY



SINGLE-ZONE SPACE TEMPERATURE AND HUMIDITY CONTROL
FIGURE 6-5

This system (figure 6-5) is similar to the previous single-zone systems which have 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 done by the direct acting space thermostat which controls the hot water supply valve, the chilled water supply valve, and the face/bypass actuator sequence. Control of the hot water supply valve is direct acting. As temperature rises in the space, branch line pressure from the thermostat increases, closing off the hot water supply valve. As pressure continues to increase in response to rising space temperature, the signal passes through the high select relay and starts to drive open the chilled water supply valve. It then closes off the bypass damper and opens the normally closed face damper.

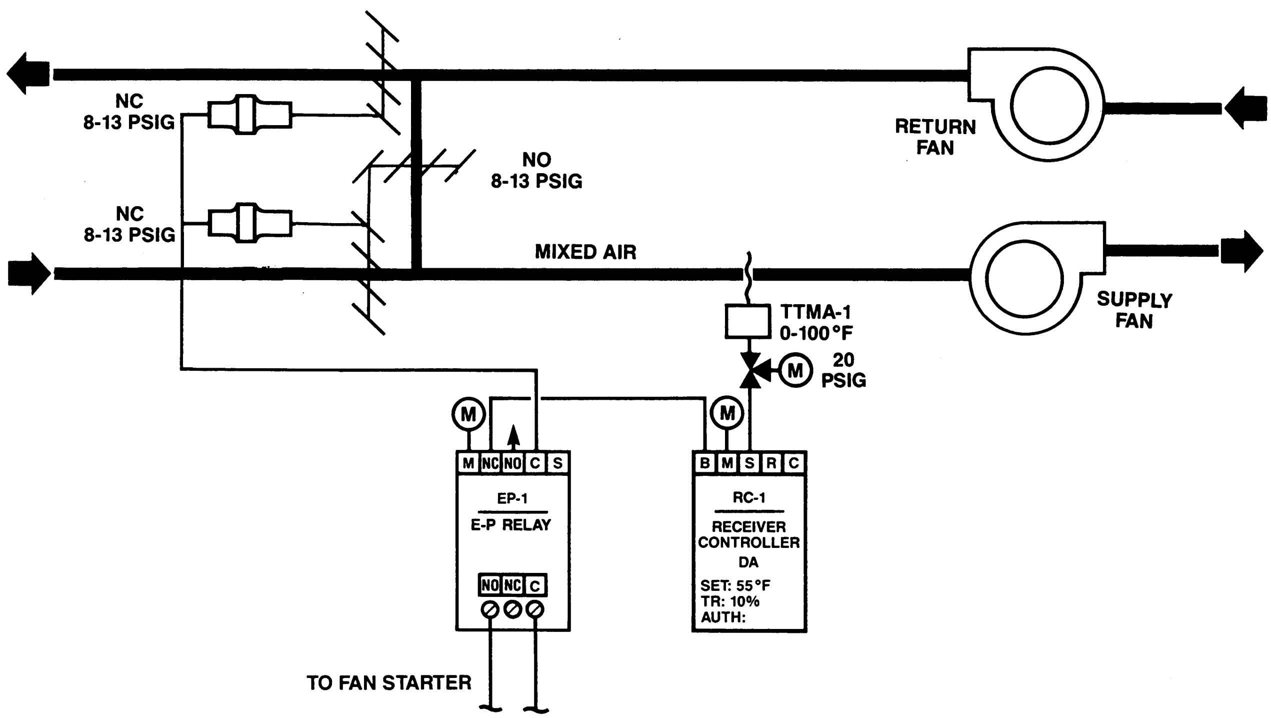
At the same time, the space humidistat which is also direct acting, is sensing the humidity level in the space and its branch line feeds the S2 port of the high select relay. If the humidity in the space rises above set point, and the branch line pressure of the humidistat exceeds that of the branch line pressure of the thermostat, the humidistat branch line pressure 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 humidistat goes to the reversing relay which, in turn, is fed to the main air port of the reverse acting receiver controller. The humidity transmitter located in the duct and the receiver controller, serve as a high limit. Should the humidifier output increase above 70 percent it will close the steam valve by reducing the branch line pressure to the normally closed steam valve.

This type of system would be used in an area where low humidity in the winter could present a problem, as well as a need for dehumidification during mild weather if the humidity were too high. As mentioned, it would be possible to open the cooling coil valve and face damper thereby cooling the air because of a high humidity condition (even though the wall thermostat might be calling for some heat). In this case, the air would be 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 generally are cooling oriented because of load conditions in the controlled space, and 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 were in a fixed position at all times. The conversion to economizer systems has been a popular energy savings tactic. Pneumatic systems have tended to be more sophisticated because of their being in larger buildings, whereas most electric or electronic control systems did not generally include economizers if the buildings were built in the 1950's and early 60's. There are a number of things that can be done to enhance the operation of an economizer system. Several of these options are covered elsewhere in this manual.

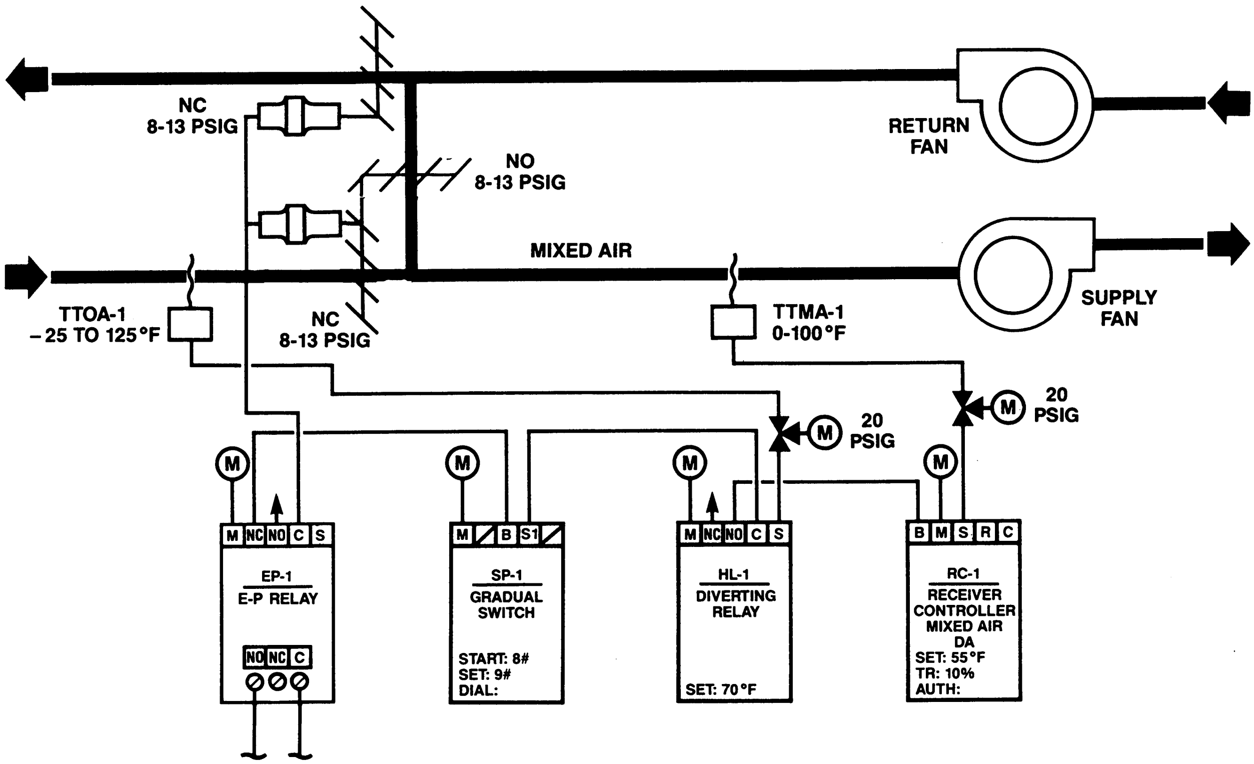


MIXED AIR CONTROL — ECONOMIZER CYCLE

FIGURE 6-6

The operation of the economizer cycle shown in figure 6-6 involves a mixed air transmitter sending a signal to the direct acting mixed air receiver controller, RC-1. This branch line signal is sent to the normally closed (NC) port of EP-1, which is connected in series with the fan starter circuit. When the fan is energized, EP-1 is energized and the NC to C ports are connected and the branch line signal from RC-1 is sent to the outside, return and exhaust damper actuators. When the fan is de-energized, the NC port of EP-1 is blocked and the C port is connected to the NO port and the signal is exhausted to atmosphere, allowing the outside air, return air and exhaust air dampers to go to the closed or open condition. As long as the supply fan is energized the dampers will modulate in response to changing mixed air temperatures and will attempt to maintain a mixed air temperature of 55°F.

MIXED AIR CONTROL, ECONOMIZER WITH HIGH LIMIT AND MINIMUM POSITION CONTROL



MIXED AIR ECONOMIZER WITH HIGH LIMIT AND MINIMUM POSITION

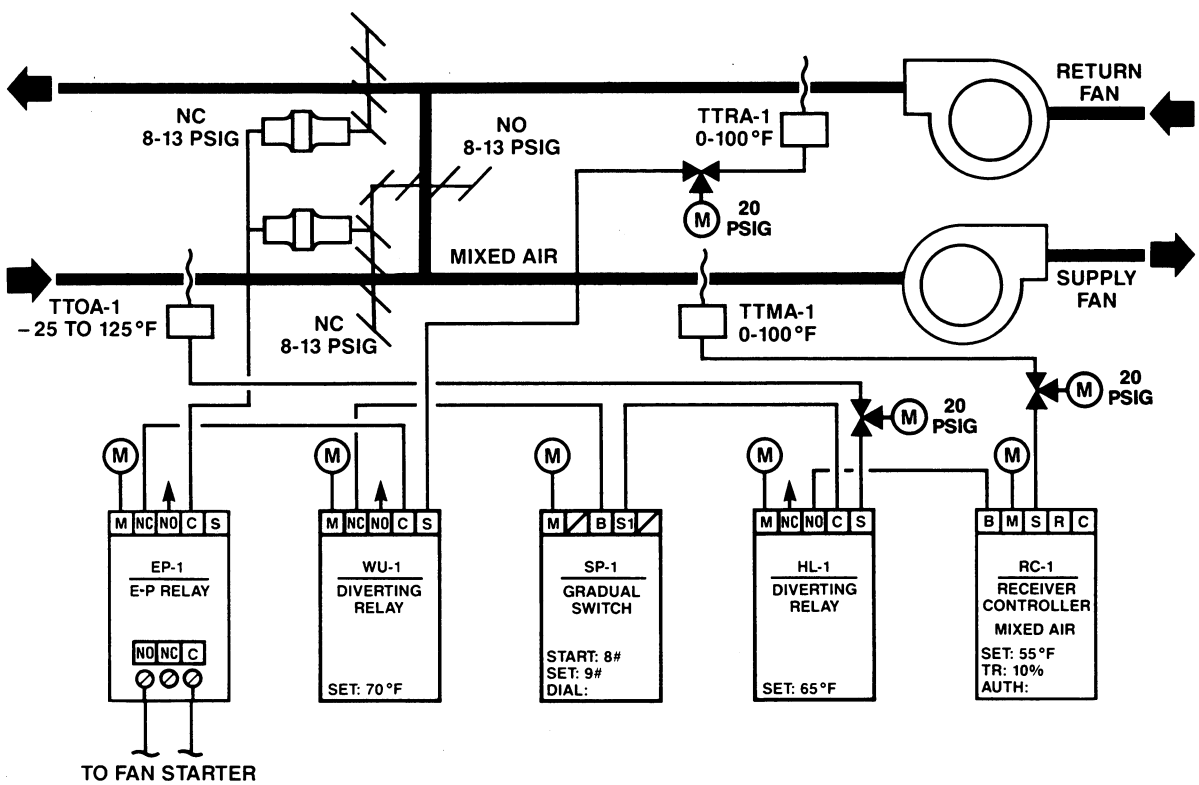
FIGURE 6-7

In this system (figure 6-7) two functions are added to the economizer cycle. These are a high limit control to deactivate the outside air intake if the temperature rises above a given set point, and a minimum position switch so that the outside air damper will remain open at a minimum position to satisfy ventilation requirements in the building, irrespective of other conditions (such as low or high mixed air temperature during occupied hours).

The minimum position switch is a gradual switch which is capable of providing a fixed signal which is adjustable or passing a higher signal which is provided at port S-1. The set point on SP-1 is 9 psig. As long as the pressure at port S-1 (branch output of RC-1) is greater than 9 psig, this pressure will be transmitted out of port B to the NC port of EP-1. So long as EP-1 is energized, this pressure is passed on through common to the outside, return and exhaust dampers, allowing them to modulate in response to the changing mixed air temperature.

As outside air temperature increases 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 to the closed position. If the outside air transmitter TTOA-1 senses a temperature rising above 70 degrees, it will trigger diverting relay HL-1, blocking the output of RC-1 and exhausting the common port out the NC port. This reduces the pressure at S-1 below 9 psig and the minimum position switch SP-1 will maintain a minimum 9 psig signal out the branch maintaining a minimum position on the outside, return and exhaust dampers. In the example shown on the drawing, the starting point is shown as being 8 psig (which corresponds to the range of the damper actuator) and the set point is 9 psig. In this case, 9 psig represents a movement of the outside air damper that will allow enough outside air to enter the building while the fan is running to meet minimum air requirements.

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



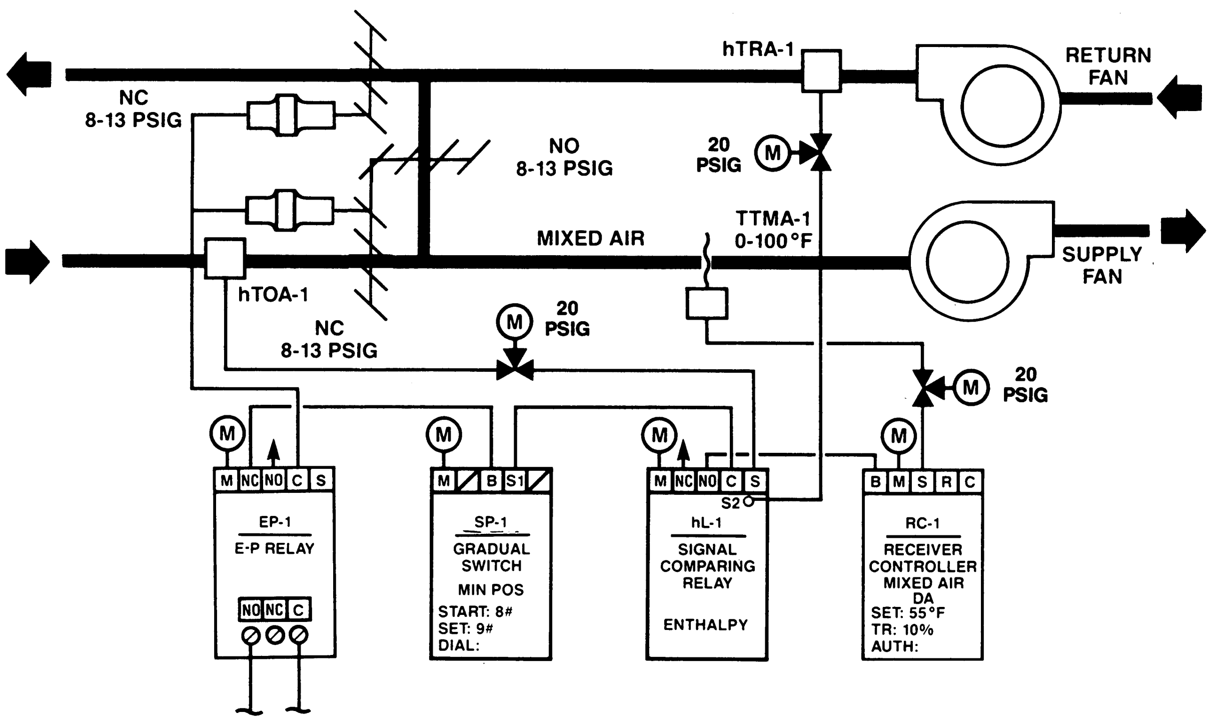
MIXED AIR ECONOMIZER WITH HIGH LIMIT, MINIMUM POSITION AND WARM-UP CONTROL

FIGURE 6-8

For this economizer application, warm-up control has been added (figure 6-8). Warm-up control allows the building to come up to temperature on morning start-up before energizing the economizer dampers. Upon fan start-up, EP-1 is energized as in the other systems and the signal from RC-1 (the mixed air controller) passes through the high limit diverting relay HL-1 and the minimum position switch SP-1 to the NC port of the warm-up relay WU-1. Until the return air temperature sensed at TTRA-1 rises above the set point in the warm-up relay which is 70°F, the branch output of the mixed air controller is blocked at the warm-up relay. Once the return air temperature rises above 70°F, the relay switches and NC and C 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, in the case of minimum position, in response to the minimum position switch SP-1.

MIXED AIR CONTROL – ECONOMIZER WITH ENTHALPY CHANGEOVER AND MINIMUM POSITION

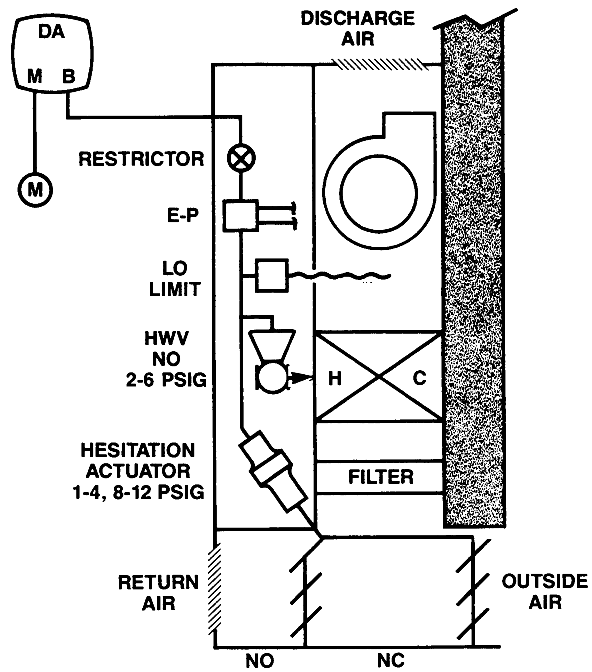
In this application, an enthalpy controlled changeover point is used to monitor whether there is "free cooling" available from the outside air. This is in contrast to the previous systems where a fixed changeover point or high limit to lock out outside air was utilized. The amount of cooling energy needed 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 enthalpy content than it is to cool return air with a higher enthalpy content.



MIXED AIR ECONOMIZER WITH
ENTHALPY CHANGEOVER AND MINIMUM POSITION
FIGURE 6-9

In figure 6-9, a signal comparing relay comparing signal inputs from an outside air enthalpy transmitter hTOA-1, and from a return air enthalpy transmitter hTRA-1 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 NO and C ports are connected and the mixed air receiver controller provides a modulating signal to position the economizer dampers. When the outside air enthalpy rises above that of the return air enthalpy, the diverting relay switches and blocks the NO port to exhaust the damper control signal back through C to NC, which allows the outside, return, and exhaust dampers to return to the minimum position as governed by SP-1. The final control, of course, is through EP-1, which is in series with the fan starter, and when the supply fan is de-energized EP-1 exhausts the control signal which positions the dampers to their normally closed or normally open position.

UNIT VENTILATOR



UNIT VENTILATOR
FIGURE 6-10

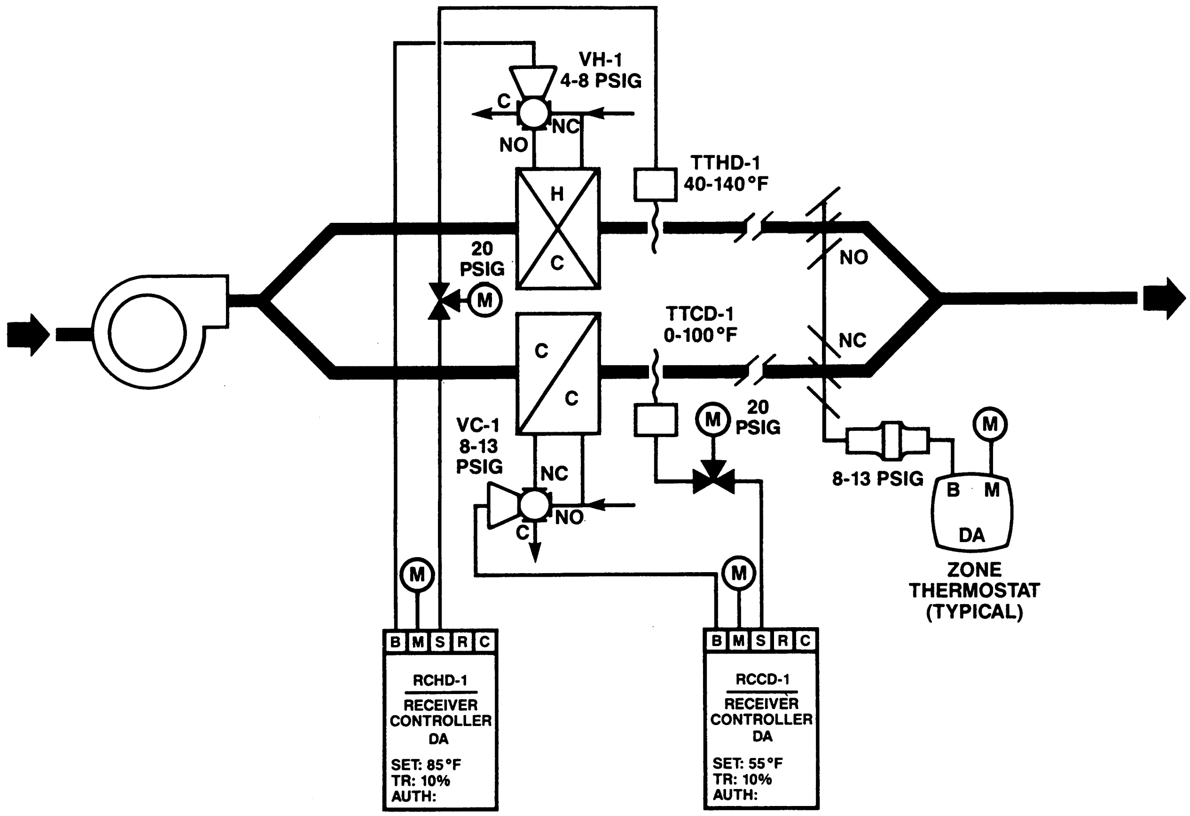
Figure 6-10 depicts a unit ventilator. The unit ventilator is a self-contained unit with the exception of a hot water supply from a central boiler. It does not rely on any central air handling system. It is generally located beneath the windows on an outside wall where the normally closed outside air dampers would bring in air for ventilation from the outside and the normally open return air dampers would be able to recirculate the return air after mixing it with the outside air.

When the fan is energized, the E-P relay allows the thermostat signal to position the coil valve and the damper actuator. The fan draws the two air supplies through the heating coil to the space. The thermostat is generally direct acting and, as shown in the diagram, is piped to position a normally open heating valve. At the same time, whenever the fan is energized, the branch signal of the thermostat will also be fed to the damper actuators. In this application the damper actuator has a dual spring range of 1 to 4 psig and 8 to 12 psig.

When the temperature in the space is below set point, the thermostat branch pressure will be low and the position of the dampers will be normally closed (or at a minimum position to the outside air and open to return air). This will allow the maximum recirculation of return air and at the same time the heating valve will be open, supplying additional heat to warm the space as the discharged air temperature is raised to achieve set point. As the room approaches set point, the branch pressure will rise and throttle the normally open hot water supply valve toward the closed position. If the outside air temperature is quite cold and the discharged air temperature is lower than 55 to 60 degrees, the low limit would bleed down the branch pressure, opening the heating valve and allowing the damper motor to close off outside air. This could occur regardless of the room temperature being satisfied.

The hesitation stroke damper actuator utilizes two separate springs internally. The first of these is actuated between 1 and 4 psig. This provides a minimum amount of damper movement upon system start-up and this allows the outside air damper to open to a minimum position to provide the required ventilation air. Between 4 and 8 psig nothing happens and the system operates on the return air. This is to prevent the reheating of cold outside air as the hot water supply valve is closed through this range. After the room temperature is satisfied and the hot water supply valve is closed, the damper actuator will fully stroke between 8 and 12 psig, closing off the return air and opening the outside air damper to get a maximum amount of outside air to provide ventilation.

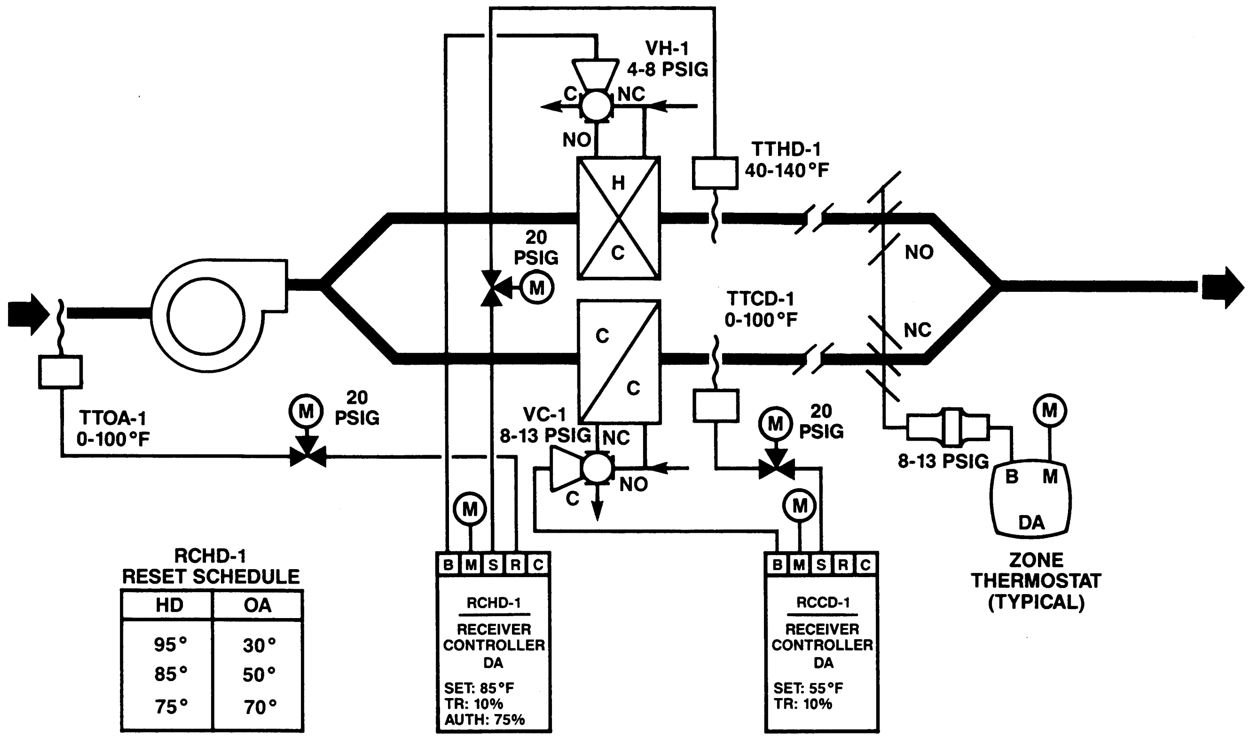
MULTIZONE AND HOT AND COLD DECK CONTROL



MULTIZONE HOT AND COLD DECK CONTROL
FIGURE 6-11

This is relatively simple system (figure 6-11) utilizing two transmitters, two receiver controllers, and a zone thermostat and damper actuator for each zone in the space. The receiver controllers control three-way mixing valves which control flow to the hot deck and cold deck coils. Both receiver controllers are direct acting. The flow through the heating coil in the hot deck is normally open to heating, and the flow through the cold deck is normally closed to cooling. Spring ranges for these valves are selected for close-off ratings that would correspond to their application.

MULTIZONE AND HOT AND COLD DECK CONTROL WITH HOT DECK RESET



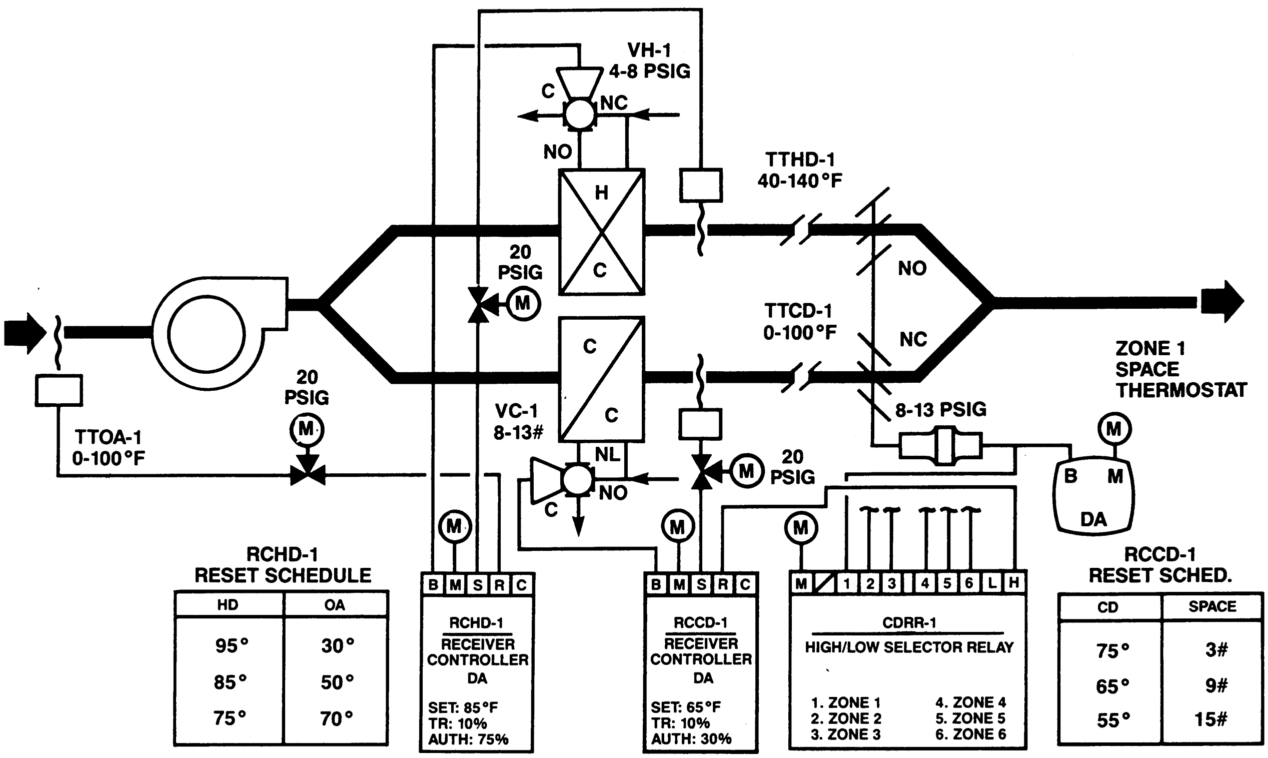
HD	OA
95°	30°
85°	50°
75°	70°

MULTIZONE HOT AND COLD DECK CONTROL WITH HOT DECK RESET

FIGURE 6-12

This system (figure 6-12) is essentially the same as the system in figure 6-11, with the addition of a temperature transmitter in the outside air which is piped to port R of the receiver controller RCHD-1. The reset schedule shown represents the design temperature ranges of 30 to 70 degrees outside air temperature. Over this outside air temperature range, it is determined that a range of from 75 to 90 degrees hot deck temperature will be sufficient to offset the heat losses that these temperature changes cause. The authority setting of this application works out to be 75%, utilizing a 10% throttling range with the transmitters as shown.

MULTIZONE AND HOT AND COLD DECK CONTROL WITH HOT AND COLD DECK RESET



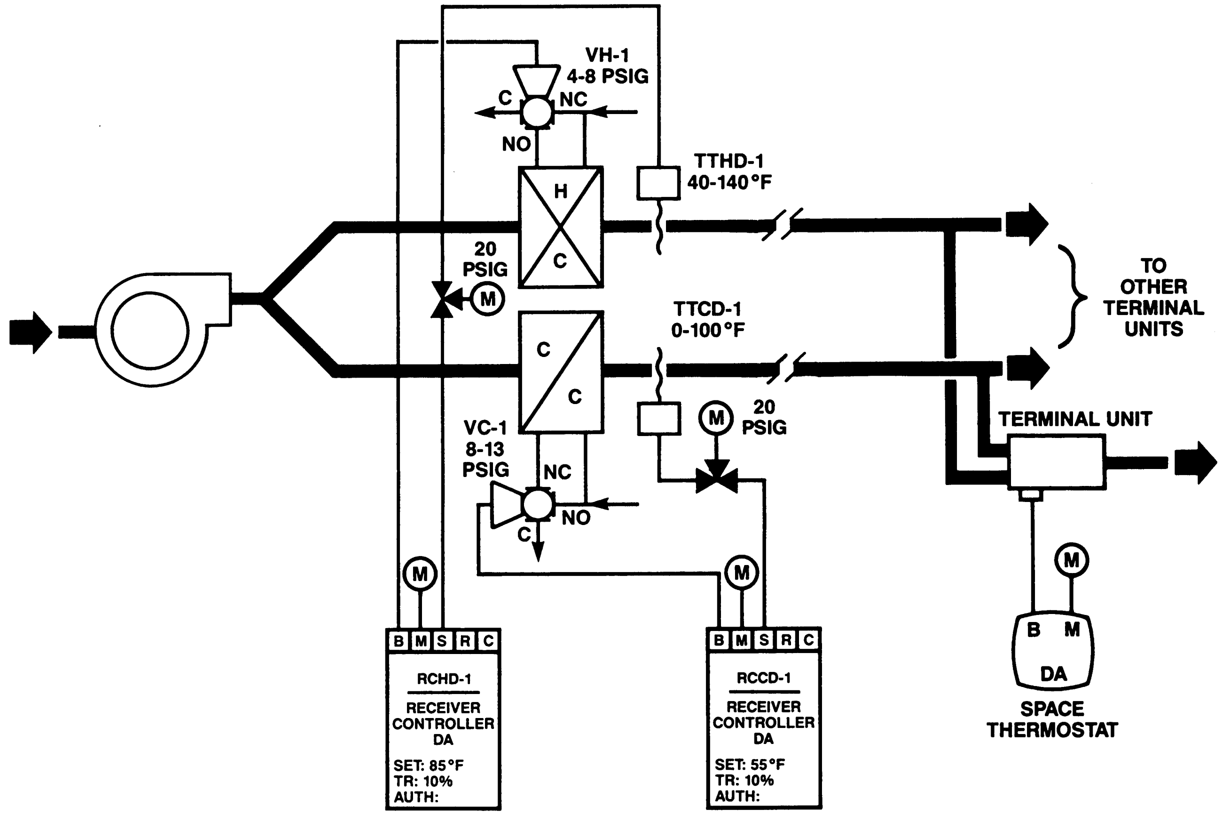
MULTIZONE HOT AND COLD DECK RESET

FIGURE 6-13

In this system (figure 6-13) the additional step of adding cold deck reset from space temperature has been taken so that cold deck temperatures are maintained only at a level required by demand in the various zones. A high/low multi-input selector relay is included which will accept the branch input from up to 6 individual zones and pass through the highest input to reset RCCD-1 per the cold deck reset schedule.

This type of system allows the temperature of the cold deck to be reset upwards as demand decreases in the space. The zone of highest demand is represented by the highest output passed through to reset the receiver controller so that the system will effectively compensate for worst case demand. The advantage here is that the cold deck supply temperature may be reset upwards to reduce cost of cooling, as space demand decreases.

HOT AND COLD DUCT OR DUAL DUCT SYSTEM CONTROL

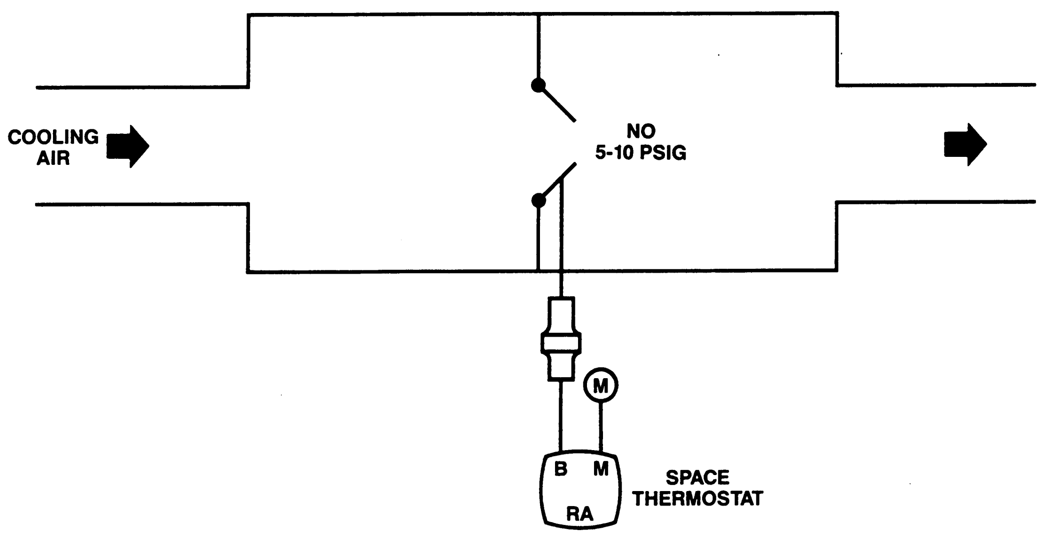


DUAL DUCT CONTROL
FIGURE 6-14

This system (figure 6-14) is very similar to the basic hot deck/cold deck or 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 relatively rare and are generally considered not to be energy efficient. They were originally designed to provide a continuous supply of hot deck temperature and cold deck temperature to each space. The terminal units for each space would mix these two air supplies as necessary to provide a discharge air temperature that would satisfy space conditions. These systems are very 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.

Some of these systems may also utilize hot deck reset (probably the most common) and, occasionally, cold deck reset. These functions would most probably have been added on as an energy-saving measure. As stated earlier, most of these systems predate the common application of both of these functions.

TERMINAL UNIT, VAV



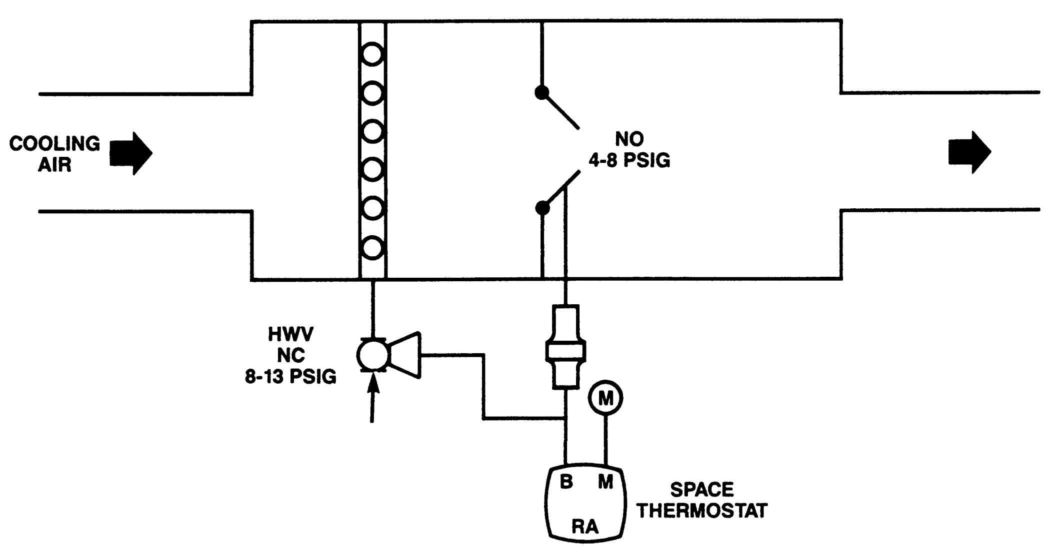
VAV TERMINAL UNIT
FIGURE 6-15

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

This terminal could also be supplied with warm air, and utilize a direct acting thermostat which would close the dampers to minimum on a rise in temperature.

Units of this type usually have a mechanical stop for minimum and/or maximum volume adjustment. Volume varies as the inlet duct static pressures vary.

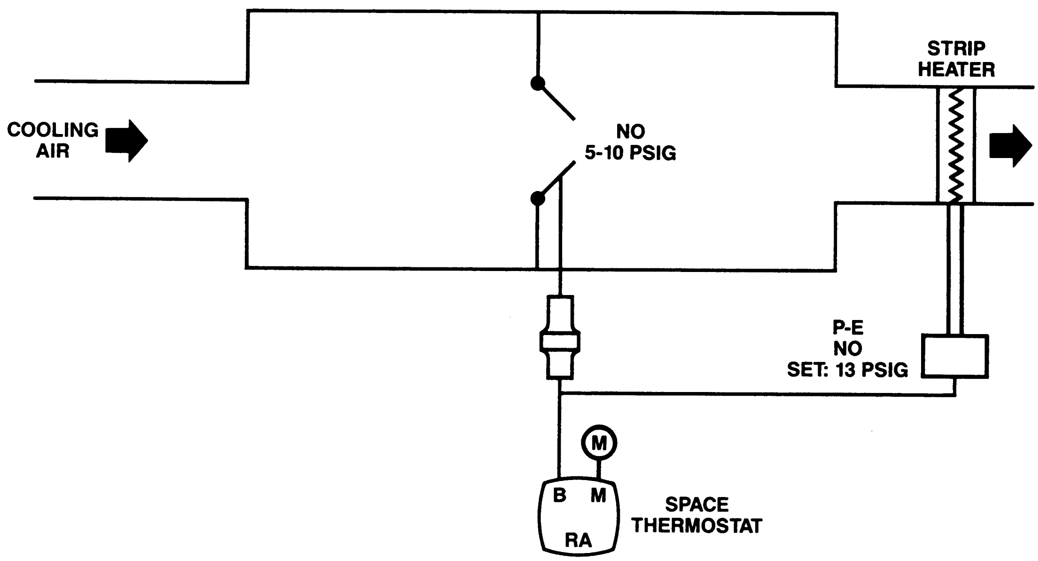
TERMINAL UNIT, VAV WITH REHEAT



VAV TERMINAL UNIT WITH REHEAT
FIGURE 6-16

This is a variable volume, throttling-type terminal as previously discussed, except a reheat coil and hot water valve are included to operate in sequence with the volume damper. In figure 6-16, a reverse acting thermostat and a normally open volume damper are being used. As the temperature decreases, the volume damper is closed by an increase in branch line pressure. As the pressure continues to rise in the branch line, the VAV unit goes to a minimum flow which is mechanically set. As the pressure continues to rise, the normally closed 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. It will be fully closed before the terminal unit starts to open to allow a greater flow of cooling air to the space.

TERMINAL UNIT, VAV WITH STRIP REHEAT

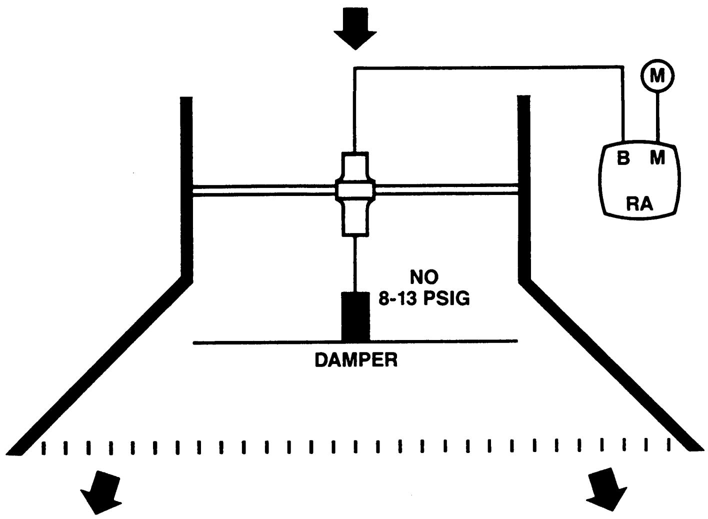


VAV TERMINAL UNIT WITH STRIP REHEAT
FIGURE 6-17

In figure 6-17, a strip heater in the duct is used and is controlled in sequence with the VAV damper. The reverse acting thermostat responds to an increase in space temperature by decreasing the branch line pressure, opening the VAV box to increase flow to meet demand in the space. As space temperature decreases, output of the controller increases driving the terminal dampers towards the minimum position. If this action is not sufficient to maintain space temperature at a comfortable level, the decreasing temperature will cause the room thermostat to increase the branch line pressure. This will close the normally open contacts in the P-E switch which is controlling the electric strip heater. This will provide reheat of the incoming air to the space until the space temperature reaches set point.

CONTROL APPLICATIONS

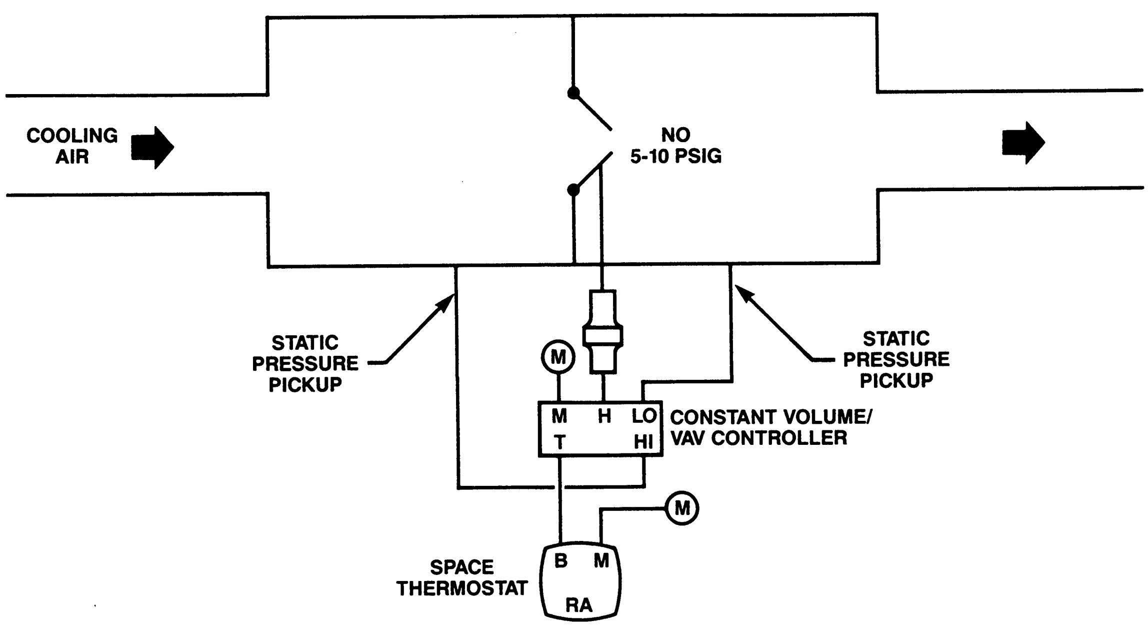
VARIABLE VOLUME DIFFUSER



VARIABLE AIR VOLUME DIFFUSER
FIGURE 6-18

This is a slightly different type of terminal unit (figure 6-18) with the terminal unit being part of the space diffuser. This unit would encompass mechanical limits for minimum and maximum volume and, as shown, would be normally open to cooling with a reverse acting thermostat.

TERMINAL UNIT, VAV WITH HIGH LIMIT



VAV TERMINAL UNIT WITH HIGH LIMIT
FIGURE 6-19

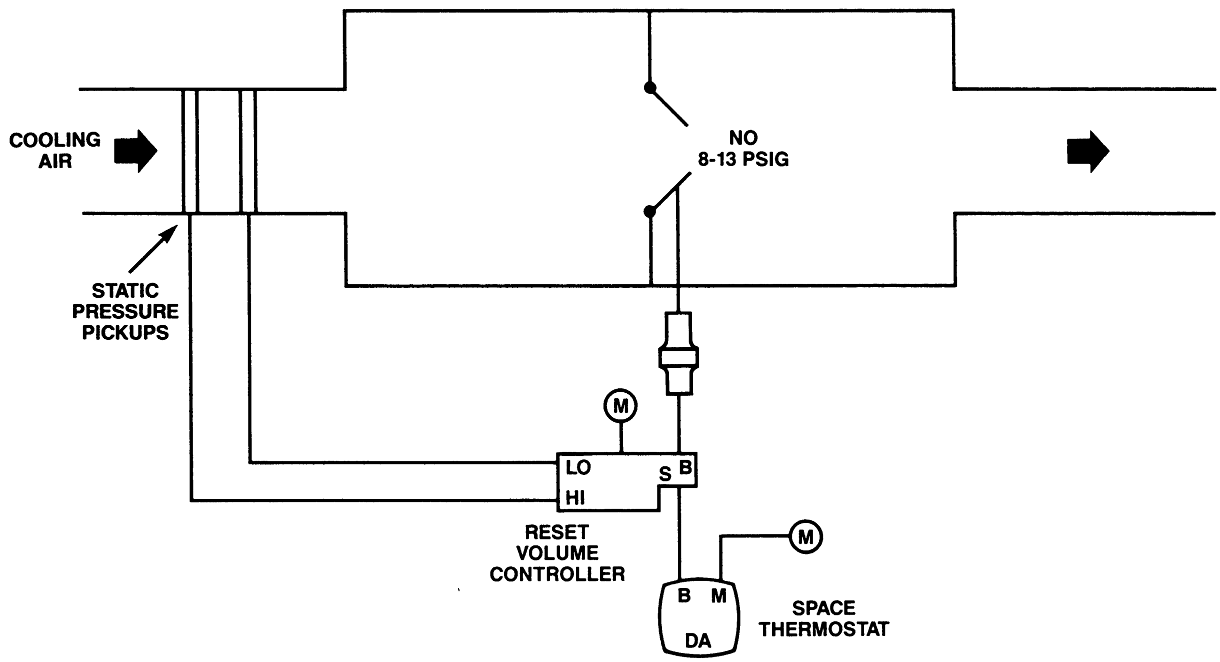
This terminal unit (figure 6-19) is controlled by a thermostat in the space and a constant volume/VAV controller sensing flow through the terminal unit. The controller contains a built-in high limit which limits flow through the terminal unit to a maximum setting that is established by the manufacturer (or upon application in the field by air balance specialists).

The terminal unit is supplied with cooling air and utilizes a normally open damper and a reverse acting thermostat. The space thermostat controls the action of the damper and, as the space temperature increases, the branch line pressure falls 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 controller will take over through the high select relay built into the controller. The constant volume/VAV controller will maintain flow at that fixed maximum setting.

This type of terminal unit control responds to changing 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 constant volume controller will respond by allowing the actuator to open 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, increases flow through this terminal unit. The static pressure sensors will pick up this increased flow and compensate for it by closing the terminal damper to maintain flow at the preset high limit.

TERMINAL UNIT, VAV PRESSURE INDEPENDENT

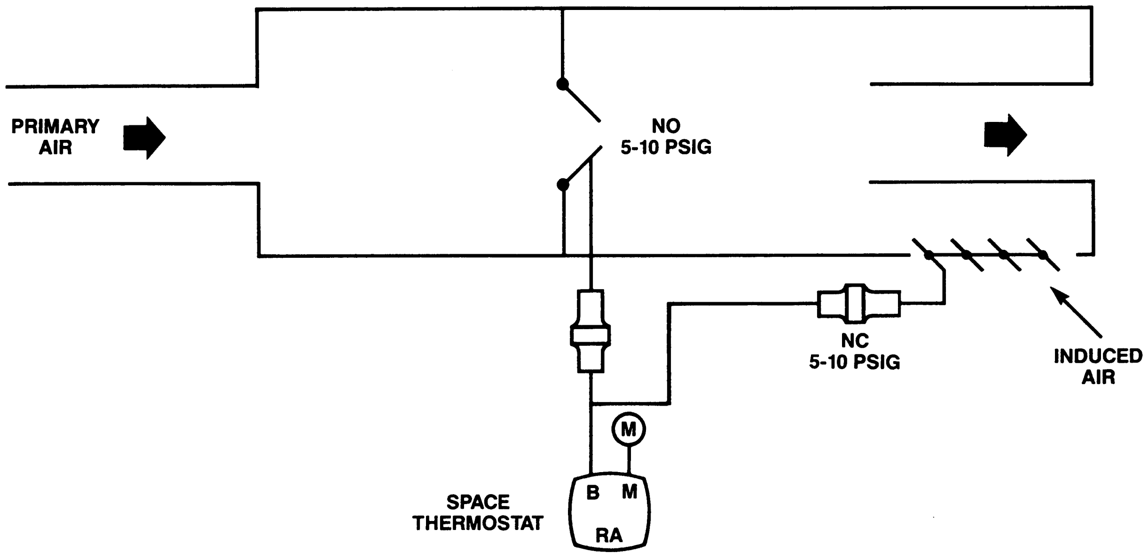


VAV PRESSURE INDEPENDENT TERMINAL UNIT
FIGURE 6-20

This type of terminal unit (figure 6-20) 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 it is controlling. The reset volume controller has minimum and maximum flow set points. The device in this particular application is a direct acting controller and is used in conjunction with a direct acting space thermostat. The space thermostat resets the set point of the controller up or down between 8 to 13 psig in response to changing space temperature. As the space temperature goes up, the branch line pressure from the thermostat goes up and resets the set point of the controller downward to allow a greater volume to flow through the terminal unit into the space. At any given control point the volume controller will respond to changes in inlet flow to either open or close the damper to maintain a flow rate that corresponds to the set point needed to satisfy space temperature.

This type of system is rarely used as a retrofit application, but has become quite common in new applications. The primary reason for it not being used in a retrofit situation is that the action of the thermostat and actuator must be changed in most applications to be used with this type of controller.

VAV INDUCTION BOX

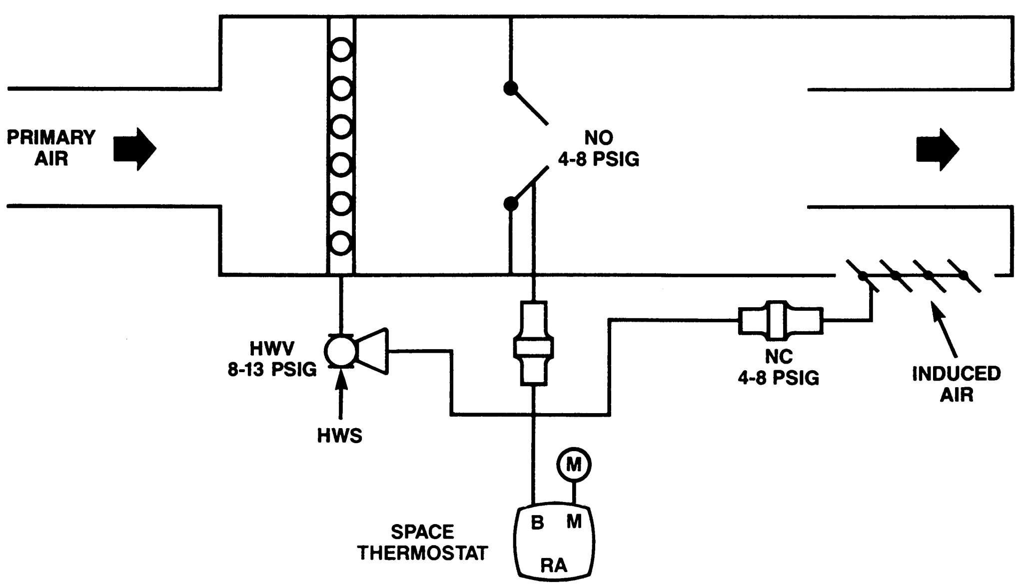


VAV INDUCTION BOX
FIGURE 6-21

Induction units as shown in figure 6-21 use primary system air discharged through the terminal unit and secondary air that is induced from the controlled space.

Shown is a typical unit with thermostatically controlled operators varying the primary and secondary dampers. On a demand for less cooling, the primary air dampers throttle, reducing primary air volume. The induced air dampers are simultaneously opened. Primary air discharges into a mixing section while inducing return air flow from the surrounding space. Mixed air is redirected into the supply duct and out of the diffuser.

VAV INDUCTION BOX WITH REHEAT

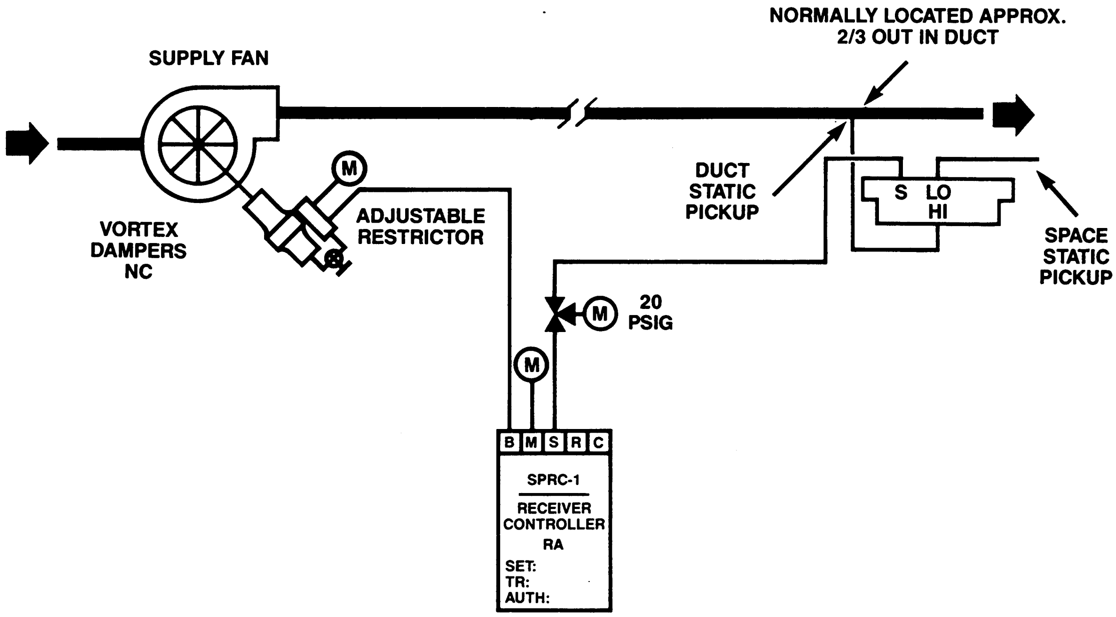


VAV INDUCTION BOX WITH REHEAT
FIGURE 6-22

Figure 6-22 is essentially the same as the induction box previously described, but with the important additional feature of a reheat coil in the primary air supply. This type of unit is used primarily in perimeter areas. It provides the capability of recovering in those areas that have a greater demand for heat, such as perimeter zones.

Note that the actuator spring ranges are slightly different here to accommodate the sequencing of the hot water supply valve. Instead of 5 to 10 psig actuators, they have a range of 4 to 8 psig so that the normally open primary volume damper is at its minimum flow position and the secondary air dampers are fully open before the normally closed water supply valve starts to open.

SUPPLY FAN VOLUME CONTROL



SUPPLY FAN VOLUME CONTROL

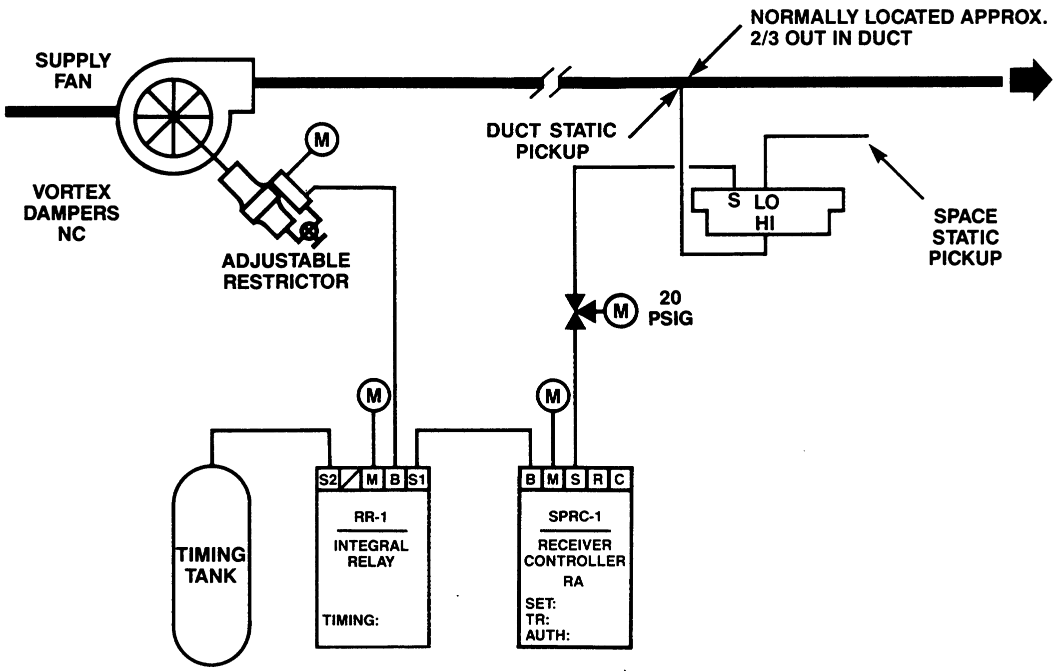
FIGURE 6-23

The purpose of fan volume control is to maintain a preset system static pressure. As shown in figure 6-23, static pressure is sensed with a differential static pressure transmitter by referencing space static pressure and a point two-thirds downstream from the fan in the supply duct. The transmitter signal is sent to the reverse acting receiver controller, which modulates the normally closed supply fan vortex damper to maintain the desired static pressure level.

A common problem is that the vortex vanes 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. In such cases it is recommended that a variable restrictor be used between the branch output of the receiver controller and the damper actuator. This will slow the actuator response until stable control is achieved.

SUPPLY FAN VOLUME CONTROL WITH RESET

A major consideration in controlling static pressure is hunting as the controller seeks its set point. On the other hand, if the throttling range is too wide, an objectionable offset to the control point could occur.

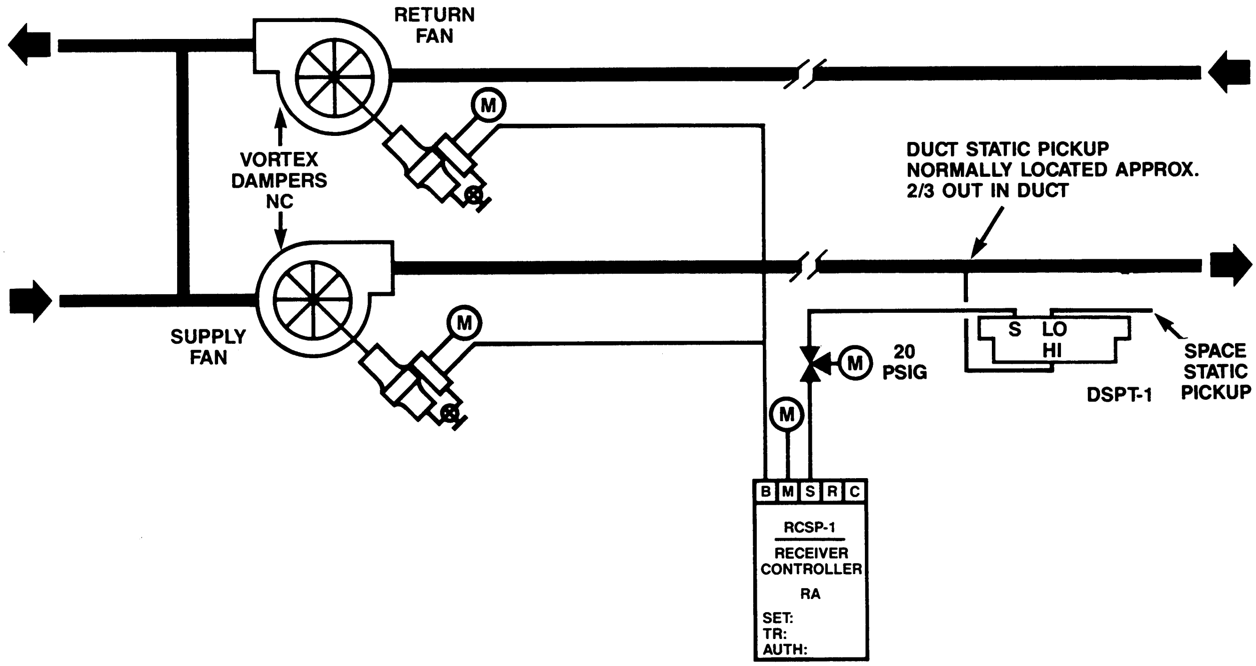


SUPPLY FAN VOLUME CONTROL WITH RESET
FIGURE 6-24

With the addition of an integral reset relay as shown in figure 6-24, the offset problem can be substantially overcome. This relay (RR-1) provides integral reset 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 vortex vanes may overshoot their control point. By placing a variable restrictor between the branch output of the integral relay and the actuator, the actuator movement will slow in order to achieve stable control.

OPEN LOOP SUPPLY AND RETURN FAN CAPACITY CONTROL



OPEN LOOP SUPPLY AND RETURN
FAN CAPACITY CONTROL

FIGURE 6-25

This control system (figure 6-25) 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 widely used because of its simplicity.

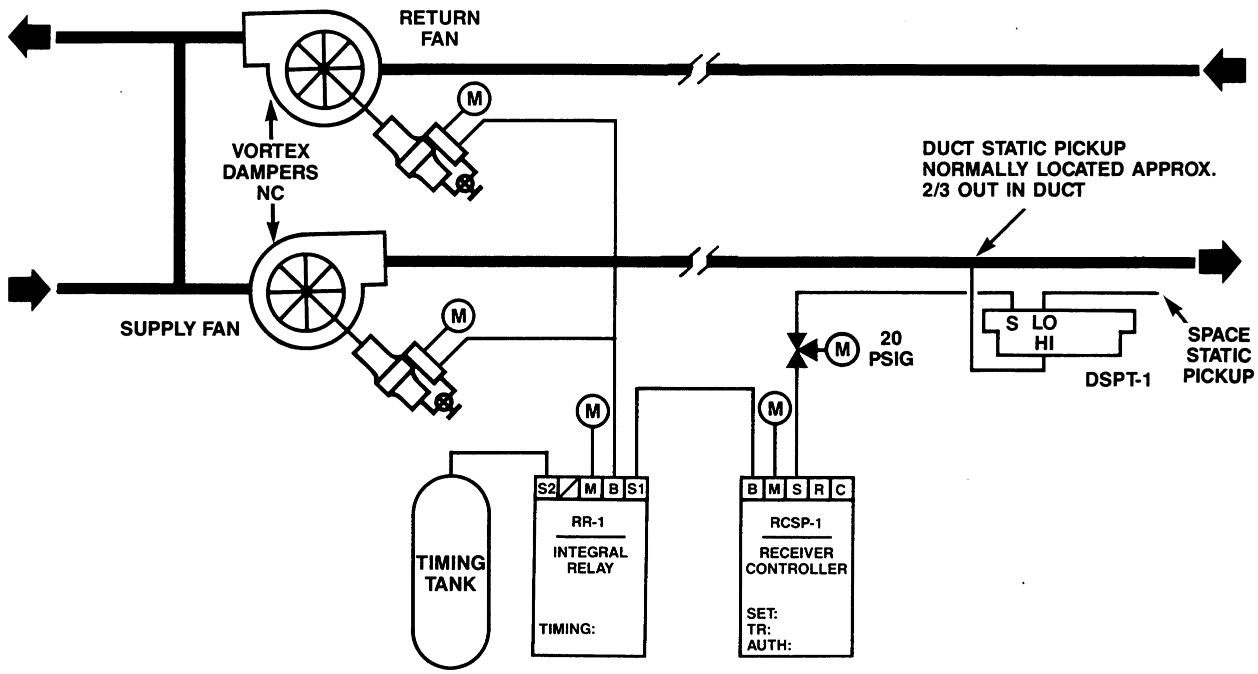
The differential static pressure transmitter signal is sent to a reverse acting receiver controller which modulates the supply fan vortex damper to maintain static pressure at a preset level as system demand varies. The receiver controller also modulates the return fan vortex damper to maintain a relatively constant CFM differential between the supply and return fans. This is accomplished by using a damper actuator with a positive positioner on it to adjust the start point and effective spring range 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 variable restrictors be installed between the positioner and the damper actuators to slow down response and aid in balancing the system.

OPEN LOOP SUPPLY AND RETURN FAN CAPACITY CONTROL RESET



OPEN LOOP SUPPLY AND RETURN FAN CAPACITY CONTROL RESET
FIGURE 6-26

Figure 6-26 is a duplication of the previous example with the addition of reset control. Again, this is accomplished by adding the integral reset relay RR-1.

As in the previous capacity control sequences, it is imperative that system stability be obtained before adding reset function.

APPENDIX 7

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 preventative maintenance program is followed. This section provides a suggested schedule for maintaining the air station and devices within the system.

MAINTENANCE SCHEDUL⁻

FREQUENCY	SERVICE REQUIRED
Once A Week	Drain compressor, tank, filter bowl, and any air lines that have drain cocks. Check compressor crankcase oil level. Check compressor safety-relief valve.
Once A Month	Inspect discharge air filter. Check pressure-reducing valve setting.
Once Every 3 Months	Change crankcase oil. Oil the compressor motors. Check compressor pressure switches.
Once Every 6 Months	Check for moisture, oil and dirt in air lines. Clean the intake air filter, felt and screen types. Check the compressor belt. Check the pressure relief valves. Check calibration, operation, nozzles, and restrictors of transmitters, temperature controllers, pressure controllers, thermostats and humidistats. Check piping of pressure transmitters and controllers. Clean elements and humidistats. Lubricate dampers, check damper actuators and close-off.
Once A Year	Replace cartridge-type intake air filters. Check calibration of receiver controllers. Check throttling ranges of humidistats, thermostats, temperature and pressure controllers. Lubricate packing, adjust packing or repack valves. Check valve for tight close-off. Check E-P and P-E relay operation. Check diverting, averaging, high/low, volume booster relay operation. Check diverting switch operation. Check gradual switch operation.

TEMPERATURE CONTROLLERS, RECEIVER CONTROLLERS AND TRANSMITTERS

After a control system is installed the control devices should not require maintenance unless the air lines become contaminated, the devices were abused, or the application requirements have changed.

In the event that recalibration is required, this section explains the procedures to be followed for selected Robertshaw devices.

CALIBRATION INSTRUCTIONS FOR ROBERTSHAW CONTROLLERS & TRANSMITTERS

ROOM THERMOSTATS

General Instructions

1. A thermostat should be mounted where it will be affected only by the average room temperature. Free circulation of air must exist at the selected location. Avoid locations that are affected by drafts or by radiant heat from the sun, water pipes, air ducts, etc.
2. Installation on outside walls should be avoided. If such a location is necessary, the thermostat should be mounted on an insulated backplate (accessory item).
3. Thermostats should be mounted **AFTER WALL SURFACES HAVE BEEN FINISHED.**

2211 & 2212 SERIES (except dead band models)

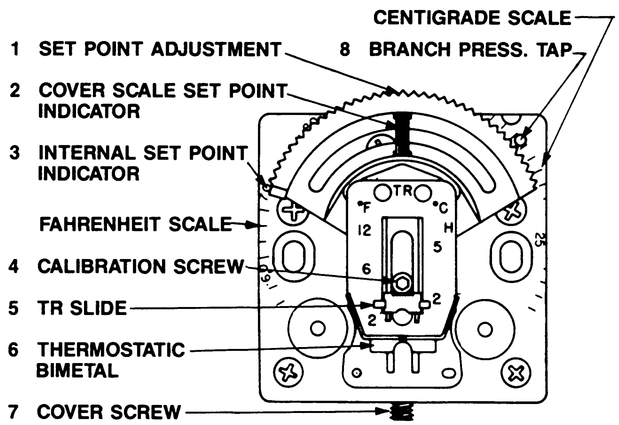


FIGURE 7-1

The 2212 Series (figure 7-1) thermostats are factory calibrated and are shipped with the throttling range set at 3°F. They should not require calibration upon installation.

If it is necessary to change the calibration or change throttling range setting, remove the thermostat cover and install a 22-138 branch tap gauge adaptor into the branch pressure tap hole (8) and measure the ambient temperature with an accurate thermometer.

This temperature **MUST BE WITHIN THE THERMOSTAT RANGE.** Move the Set Point Adjustment (1) to the measured ambient temperature, using the internal Set Point Indicator (3). Taking care not to breathe on or hold hand near the bimetal (6), use a 1/16" hex wrench (20-881 thermostat wrench) to turn the Calibration Screw (4) until the branch line pressure indicates 9 psig or the midpoint of the controlled device. Clockwise rotation increases the branch line pressure. Counterclockwise rotation will lower branch line pressure. Reinstall the thermostat cover.

DEAD BAND THERMOSTATS

If it is necessary to check calibration or to change calibration, install a branch tap (22-138 and suitable gauge) in the branch tap hole. The dead band pressure is factory set at 7 psig. If it is necessary to adjust this pressure to meet application requirements, the following procedure should be followed:

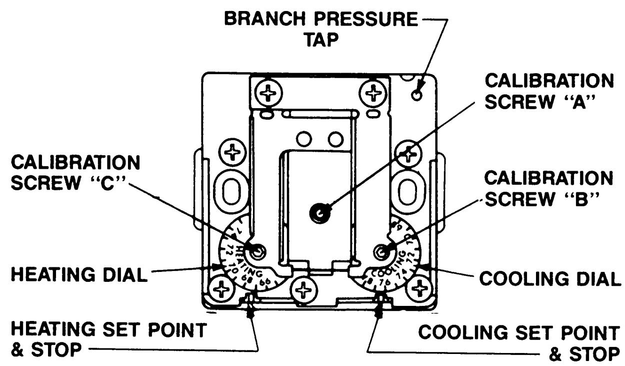
> Ambient temperature must be between 65 and 75°F. Turn the heating dial to 57°F and the cooling dial to 83°F. Adjust the dead band pressure by turning calibration screw A (see figures 6-2 or 6-3), until the desired dead band pressure is obtained.

Set Point Calibration

If it is necessary to check calibration or to change calibration, install a branch tap adaptor (22-138 and suitable gauge) in the branch tap hole and measure the ambient temperature with an accurate thermometer. The ambient temperature must be between 65 and 75°F.

Direct Acting Models

The direct acting models are calibrated to have a 4 psig branch pressure when the heating dial is set at ambient temperature, and a 10.5 psig branch pressure when the cooling dial is set at ambient temperature.



DIRECT ACTING

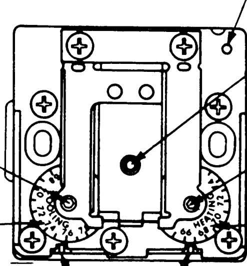
FIGURE 7-2

Referring to figure 7-2, use a 20-881 calibration wrench, and turn screw "C" to the 57°F setting. Turn screw "B" to position the cooling dial to the 83°F setting. Turn screw "C" to obtain the desired heating control pressure. If there is a difference in ambient temperature and dial reading, rotate dial to stop and continue rotating screw "C," slipping the screw inside the dial. Rotate the dial back to set point pressure. If dial does not match ambient temperature, a second try may be required. To calibrate the cooling dial, the heating dial must first be turned to the 57°F setting. This moves the heating bimetal away from the dead band lever. Turn screw "B" to obtain the desired cooling control pressure. If there is a difference in ambient temperature and dial reading, rotate dial to stop and continue rotating screw, slipping the screw inside the dial. Rotate dial back to set point pressure. If dial does not match ambient temperature, a second try may be required. Position heating and cooling dials to desired setting and replace cover.

DEAD BAND THERMOSTATS (Cont'd)

Reverse Acting Models

The reverse acting models are calibrated to have a 4 psig branch pressure when the cooling dial is positioned at ambient temperature, and 10.5 psig when the heating dial is positioned at ambient temperature.



REVERSE ACTING

FIGURE 7-3

To calibrate the reverse acting models, refer to figure 7-3 and proceed with calibration as outlined under the procedures for the direct acting models.

If, for some reason, a thermostat gets completely out of adjustment and does not respond to above calibration, first adjust screws "B" and "C" until, by observation, neither the heating or cooling bimetals are touching the dead band lever. Then proceed with calibration as outlined.

2214 & 2216 SERIES - (DAY/NIGHT

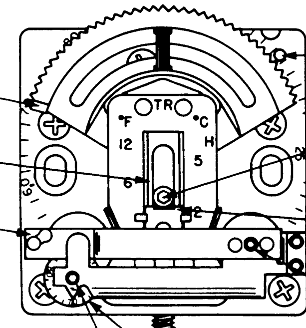


FIGURE 7-4

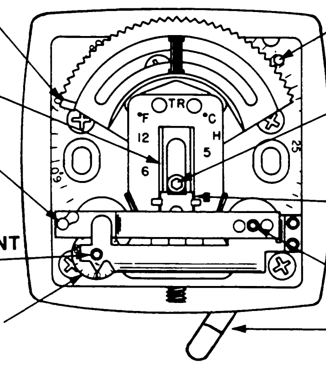


FIGURE 7-5

The 2214 (figure 7-4) and 2216 (figure 7-5) thermostats are factory calibrated with day sensor throttling range set at 3°F and normally should not need to be recalibrated. If the day throttling range adjustment (6) is changed, or if recalibration of the day or night set point becomes necessary, or if the switch point needs to be raised or lowered, install an adaptor (22-138) with a suitable gauge in the branch pressure tap hole (4). Measure ambient temperature with an accurate thermometer. This temperature must be within the range of the thermostat. NOTE: Do not breathe or hold hand near the bimetals (2 and 3) as this will not give an accurate reading.

Day Set Point Calibration

Position the day set point cam (1) to match the ambient temperature. Set main air pressure to 15 psig and adjust the day calibrating screw (5) using a 20-881 thermostat wrench until the branch tap gauge reads 9 psig (±1 psig). Clockwise rotation increases the branch pressure.

Night Set Point Calibration

Increase the main air pressure to 25 psig. Using the thermostat wrench in the night set point screw (8), position the night set point dial (7) to match the ambient temperature. Firmly hold the night set point dial with fingers and adjust the night calibrating screw (8) until the branch tap gauge reads 9 psig (±1 psig). Release the dial and allow it to rotate with the night calibrating screw (8) to the desired night control point.

Switching Adjustment

The 2214 Series devices are factory calibrated to switch from day to night action between 17 and 21 psig. This setting is to match existing Robertshaw systems, and no switching adjustment should be necessary. If the 2214 series thermostat is to be used with a competitive dual pressure system, it may be necessary to make a switching adjustment.

This switch-over adjustment should be done on a test bench where a variable main air supply is available. It is also necessary to read the branch line pressure while making the switching adjustment. The sequence of adjustment varies slightly depending upon the model thermostat used. Following is the sequence of adjustment for each model.

2214-122, 132, 522 (RA/RA)

Set the main air supply to the thermostat to the desired switch-over point (Example: system pressure is 13 psig day and 18 psig at night. Desired switch-over point would be between 15 and 16 psig.) Position the day set point (1) to 55°F setting and the night set point dial (7) to 80°F setting. The branch line pressure gauge or branch pressure tap should be reading approximately the main air pressure being fed to the thermostat. If not, recheck the day set point calibration. To lower the switch-over point, turn the switching screw (9) clockwise 1/8 turn at a time until the branch line pressure falls.

If it is desired to raise the switch-over point, the switching screw (9) should be turned counterclockwise. The calibration wrench (20-881) fits the switch over adjustment screw along with the calibration screws. The next step is to lower the main air pressure to the desired system day pressure and observe the branch line pressure. The branch line pressure should rise to approximately the main air pressure. Next, raise the main air pressure to the night setting and observe the action of the thermostat for proper function. As the main air pressure is raised past the switch-over point, the branch line pressure should drop off to zero on the way up and come back up as the main air pressure is lowered past the switch-over point on the way down. Calibration at both day and night settings should then be reached.

2214-121, 131, 521 (DA/DA)

Set the main air supply to the thermostat to the desired switch-over point (Example: system pressure is 13 psig day and 18 psig at night. Desired switch-over point would be between 15 and 16 psig.) Position the day set point (1) to the 85°F setting, and the night set point dial (7) to 50°F. The branch line pressure gauge or branch pressure tab should be reading approximately the same as the main air pressure being fed to the thermostat. If not, recheck the day set point calibration. To lower the switch-over point, turn the switching screw (9) clockwise 1/8 turn at a time until the branch line pressure falls.

If it is desired to raise the switch-over point, the switching screw (9) should be turned counterclockwise. The calibration wrench (20-881) fits the switch-over adjustment screw along with the calibration screws. The next step is to lower the main air pressure to the desired system day pressure and observe the branch line pressure. The branch line pressure should rise to approximately the main air pressure. Next, raise the main air pressure to the night setting and observe the action of the thermostat for proper function. As the main air pressure is raised past the switch-over point, the branch line pressure should drop off to zero on the way up and come back up as the main air pressure is lowered past the switch-over point on the way down. Calibration at both day and night settings should then be rechecked.

THREE-PIPE APPLICATIONS (2216 Only)

The 2216 Series thermostats may be used as either a two-pipe or three-pipe device. The third ("R" port) connection is longer than the other two ports, as the end of this port is cast solid. This means this is a two-pipe thermostat and may be installed without any alterations.

In the event that the thermostat is to be used in a three-pipe application, the installer should carefully cut approximately 1/16" off of the "R" port. When the 2216 is piped as a three-pipe thermostat, the pressure at the "R" port will be 0 psig during normal operation. That will mean that 0 psig will exist at this port at all times during day operation and main air pressure will be available at the "R" port during night operation. If 21 or greater psig main air pressure is available and the manual index lever is moved to the right, the pressure at the "R" port will drop to 2 psig or less.

2218 SERIES - SUMMER/WINTER

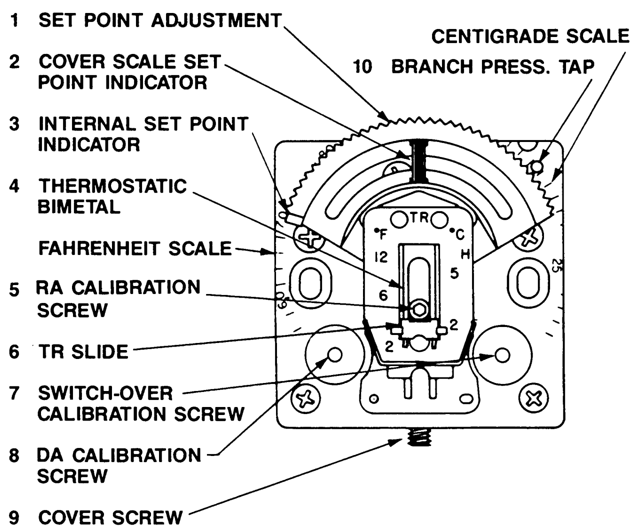


FIGURE 7-6

The 2218 (figure 7-6) is factory calibrated with the throttling range set at 3°F, and should not be recalibrated unless the throttling range is changed, or limited summer control action at 8 psig main pressure is required. If recalibration is required, proceed as follows:

1. Install branch pressure gauge adaptor 22-138, with a suitable gauge, into the Branch Pressure Tap Hole (10).
2. Measure the ambient temperature with an accurate thermometer. The temperature sensor must be within the temperature setting range of the thermostat.
3. Move the Set Point Adjustment (1) to the measured ambient temperature, using the Internal Set Point Indicator (3). Take care not to breathe on or hold hand near the bimetal (4).

STANDARD CALIBRATION

1. Set main air pressure to 16 psig.
2. Turn the RA Calibration Screw (5) using 20-881 Thermostat Wrench, until the test gauge indicates 12 psig. Clockwise rotation increases the branch pressure.

3. Raise the main air pressure to 25 psig.
4. Turn the DA Calibration Screw (8) until the test gauge indicates 12 psig. Counterclockwise rotation increases the branch pressure.

SPECIAL CALIBRATION – 8 PSIG Summer Control

1. Set main air pressure to 8 psig.
2. Turn RA Calibration Screw (5) until test gauge indicates 6 psig. Clockwise rotation increases branch pressure.
3. Raise the main air pressure to 25 psig.
4. Turn DA Calibration Screw (8) until test gauge indicates 6 psig. Counterclockwise rotation increases branch pressure.

ADJUSTING SWITCH-OVER POINT

This adjustment should be made, if possible, in the shop on a test stand, as it requires changing the main air pressure and this could affect other things in the system.

1. Set main air pressure to the switch-over point pressure you desire.
2. Move the Set Point Adjustment (1) to its furthest clockwise position.
3. Install a test gauge into the branch line and adjust the Switch-over Calibration Screw (7) until the branch pressure just drops to 0 psig. Switch-over point is now set to the pressure that was set in Step 1 above.
4. Recalibrate, following the instructions given for standard calibration, or if you are using the special summer calibration then follow the special calibration instructions.
5. Move the Set Point Adjustment (1) to the desired temperature.

ROOM HUMIDISTAT

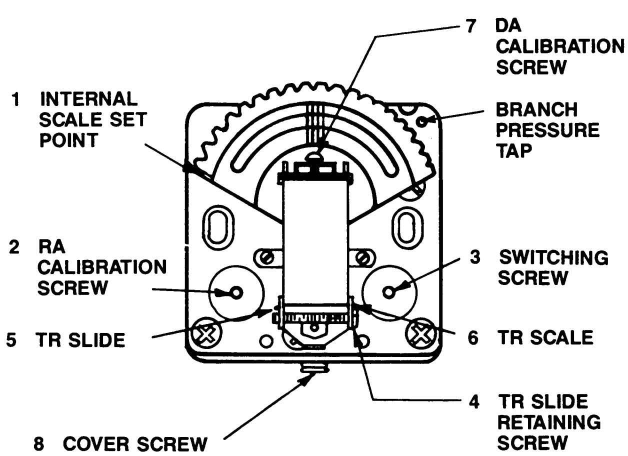


FIGURE 7-7

The 2230-018 Humidistat (figure 7-7) is factory set for a 10% throttling range, set in reverse action and calibrated for 9 psig when the ambient relative humidity equals the set point. It should not require calibration upon installation unless the throttling range is changed.

If adjustments are necessary,remove cover by turning the Allen screw (8) until bottom of cover can be moved away from the wall, and proceed as follows.

TO SET ACTION

Direct Action

Rotate the switching screw (3) ten complete turns counterclockwise. This change should not interfere with the factory calibration or TR setting, and no further adjustment should be necessary.

Reverse Action

Control is factory set in reverse acting mode. If control is set in direct acting mode, to restore to reverse action, rotate the switching screw (3) clockwise until it becomes snug. **DO NOT FORCE THE SCREW.**

TO SET THROTTLING RANGE

To change the throttling range, install a test gauge in the branch tap and rotate the cam by adjusting the set point until 8-10 psig branch pressure is obtained. (RH must be within the range of the humidistat.) Loosen the TR slide retaining screw (4) and slide the TR slide (5) to the desired throttling range setting on the TR scale (6), making sure to keep the TR slide (5) parallel with the TR slide retaining Screw (4). Tighten this screw to secure the slide (do not over tighten). Adjust the DA calibrating screw (7) to restore 8-10 psig branch pressure.

TO CALIBRATE AND SET CONTROL POINT

To check humidistat calibration, install a test gauge in the branch line, and use a sling psychrometer and a psychrometric chart to determine actual RH, which must be within range of the humidistat. Adjust the set point indicator (1) to the actual RH The branch pressure should be between 8-10 psig.

Reverse Acting

To change calibration in the reverse action mode, adjust set point as given for the calibration check procedure. Adjust the RA calibrating screw (2) to obtain 8-10 psig branch pressure. Clockwise rotation of screw (2) causes branch pressure to decrease; counterclockwise rotation causes it to increase. Do not force screw (2).

Direct Acting

To change calibration in the direct action mode, proceed as given for the calibration check. Then adjust the DA calibrating screw (7) to obtain 8-10 psig branch pressure.

To synchronize DA and RA calibrations, insert test gauge in branch line and adjust set point to actual humidity, as described above. Rotate switching screw (3) clockwise until it is snug. Then rotate ten turns counterclockwise and adjust DA calibrating screw (7) for 8-10 psig branch pressure. Then rotate switching screw clockwise until snug, and adjust the RA calibrating screw (2) for 8-10 psig branch pressure. The instrument is now synchronized from RA to DA and may be used in either mode without change in calibration.

Replace cover after making adjustments.

ROOM TEMPERATURE TRANSMITTER

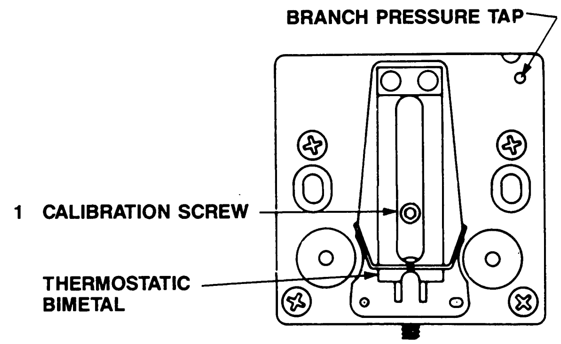


FIGURE 7-8

The 2220-053 (figure 7-8) is factory calibrated to provide a 3 to 15 psig signal over the range of 50 to 90°F. Additional field calibration should not be required. If minor field calibration is required, turning the calibration screw (1) clockwise increases the branch pressure; counterclockwise rotation decreases the pressure.

ROOM HUMIDITY TRANSMITTER

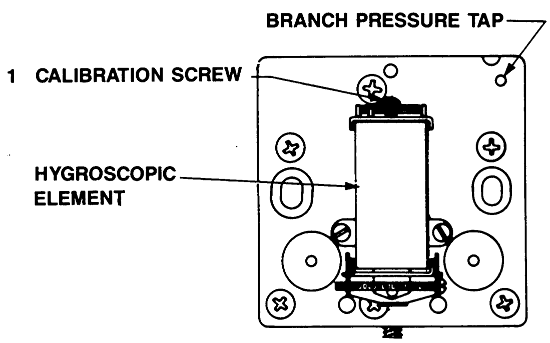


FIGURE 7-9

The 2232-053 room humidity transmitter (figure 7-9) can be calibrated by turning calibration screw (1) clockwise to decrease branch pressure and counterclockwise to increase branch pressure. The nylon hygroscopic element must be kept clean. Do not touch with fingers or foreign objects.

900-012 RECEIVER CONTROLLER/ TRANSMITTER CALIBRATION KIT

This kit was designed for the following uses:

1. The set-up and calibration of receiver controllers.
2. Checking transmitter operation and calibration.
3. Adjusting P-E relays and switching relays that require a field adjusted set point.

PNEUMATIC TEMPERATURE TRANSMITTER – 2252 SERIES

The 2252 Series temperature transmitter measures a system temperature and transmits a proportional pneumatic signal to a calibrated receiver gauge and/or receiver controller. It is a one-pipe, force-balance transmitter which utilizes an external restrictor in its supply line. It is not intended to be field calibrated. If the output pressure of the device does not correspond to the actual temperature, check the following:

1. The air supply to the restrictor must be 20 psig ± 0.5 psig and must be clean, dry and oil-free.
2. The restrictor and device filter must be free of obstructions.
3. Check piping to ensure that all connections are tight and leak-free.

If, after completing the above, the transmitter output varies from actual temperature, proceed with the following:

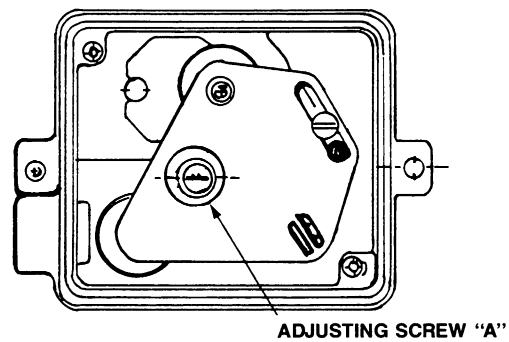


FIGURE 7-10

1. Refer to figure 7-10 and connect the transmitter to the output of the calibration kit (figure 7-11 on page 63).
2. Remove the cover and turn adjusting screw "A" to shift the output range (clockwise to increase).
3. If output correction is not obtained, no other adjustment should be attempted and the transmitter should be replaced.

RECEIVER CONTROLLER CALIBRATION

Single Input Calibration

1. Connect main air to 900-012 main air input. Close shutoff valve to output that is not being used.
2. Disconnect the transmitter from the receiver controller and connect the output from the 900-012 to the receiver controller.
3. Set desired throttling range on the receiver controller.
4. Determine the range of the transmitter, and choose the corresponding scale on the calibration gauge. Next, determine the desired set point.
5. Using the output adjustment of the 900-012, apply the pressure that corresponds to the desired set point to the receiver controller. This is done by using either the temperature scale corresponding to the transmitter range or the 3 to 15 psig scale on the calibration gauge.
6. Turn set point adjustment on the receiver controller until the branch line pressure is either 9 psig or is equal to the midpoint of the spring range of the device being controlled. (Example: 6 psig for a 4 to 8 psig range.)
7. Turn set point adjustment scale to match the calibrated set point and tighten or re-engage.
8. Throttling range may be checked by adjusting the input to the controller over the desired TR span. The branch line output of the controller should change from 3 to 15 psig over the span of the TR.

   Note: If further adjustment of the throttling range is necessary, steps 4 through 7 must be repeated to ensure proper calibration.
9. Reconnect transmitter to the primary port. Transmitter operation should also be checked whenever the receiver controller is calibrated.

RECEIVER CONTROLLER CALIBRATION (Cont'd)
Dual Input Calibration
(Reset or Compensation Application)

1. Connect main air to 900-012 main air input.
2. Determine the reset schedule for the application in question. For the purpose of these instructions, we will use the following example of a hot water temperature being reset by the outside air temperature. For an actual application in the field, the appropriate figures should be substituted.

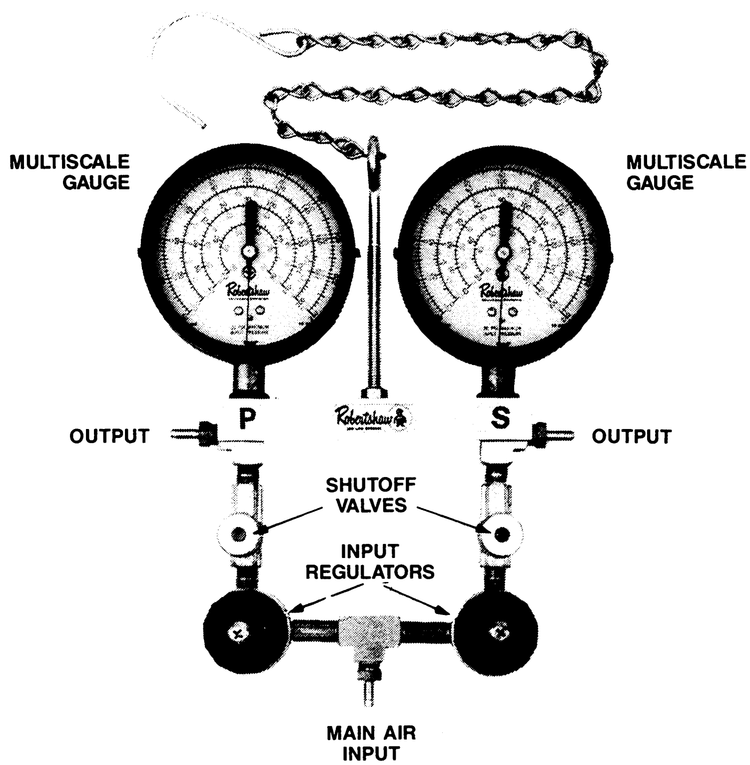
Temperature (Primary Transmitter)	Temperature (Secondary Transmitter)
200°F	0°F
150°F	35°F
100°F	70°F

Throttling Range: 10% (20°F)
Authority: 129%
OA Transmitter: −25 to 125°F range
HWS Transmitter: 40 to 240°F range

3. Disconnect the transmitters from the receiver controller. Connect the (P) output of the 900-012 to the primary transmitter input of the receiver controller. Connect the (S) output of the kit to the secondary or reset port of the receiver controller.
4. Set the throttling range and authority to the desired values.
5. Determine the ranges of the respective transmitters so that the appropriate gauge scales may be referred to.
6. Adjust the primary (P) input to the midpoint of the desired reset schedule (150°F in example schedule).

   Adjust the secondary (S) input to the midpoint of the desired reset schedule (35°F in example schedule).

   Note: If device being calibrated is being used with CPA (Control Point Adjustment or Remote Set Point Adjustment), this should be set to 9 psig.
7. Turn set point adjustment on the receiver controller until the branch line pressure is either 9 psig or is equal to the midpoint of the spring range of the device being controlled, (Example: 6 psig for a 4 to 8 psig spring range).
8. Turn set point scale to match the HWS temperature at set point (150°F in example), and tighten or re-engage.
9. To check calibration, raise OA (S) input to maximum of reset schedule (70°F in example) and lower HWS (P) input to minimum of reset schedule (100°F in example). Branch line pressure should be within ±1 psig of set point in Step 7. Next, lower OA (S) input to the minimum (0°F in example) and raise HWS (P) to maximum (200°F in example). Branch line pressure should again be within ±1 psig of set point in Step 7. Return inputs to midpoints as in Step 6. Branch line pressure should again be within ±1 psig of original set point in Step 7.
10. Throttling range may be checked by adjusting only the primary (P) input over the span of the throttling range. The branch line output should change from 3 to 15 psig over the span of the TR.
11. Reconnect transmitters to the proper ports. Transmitter operation should also be checked whenever the receiver controller is calibrated.



900-012 RECEIVER CONTROLLER/TRANSMITTER CALIBRATION KIT

FIGURE 7-11

APPENDIX

DEFINITIONS

Absolute Pressure: Gauge pressure plus atmospheric pressure (14.7 psig).

Actuator: A device which moves or stops the operation of the conditioning equipment in response to changes in branch line pressure of a controller. Examples: Damper actuator or valve actuator.

Air Control Damper: Used in central fan systems to control the mixture of air admitted to the system. Installed in the fresh air intake, return air and exhaust ducts.

Air Motion Relay: Used to sense suction and/or discharge pressures across a coil or fan.

Ambient Temperature: Temperature of air or fluid which surrounds object on all sides.

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

Auxiliary Device: Devices in automatic control systems which are not automatic controls. Such as, air compressors, transformers, thermometers, etc.

Averaging Element: Normally used for duct control when there is a large temperature gradient across the duct.

Averaging Relay: Used where the application requires operation of a final control device or setting a controller, by the average signal from two or more controllers.

Ball Valve Bleed: An arrangement which varies the exhaust stream by positioning a small ball in a shaped opening. This increases output pressure linearity and produces a significant amount of internal feedback.

Branch Lines: Air lines in a pneumatic ontrol system in which a varying air pressure is maintained by the action of controllers to position valves, dampers or similar devices.

British Thermal Unit (BTU): The amount of heat required to raise the temperature of 1 lb. of water one degree F.

Calibration Point: The set point at which control is calibrated. Normal calibration; 9 psig in branch line when ambient temperature equals set point.

Capacity Index (Cv Factor): The quantity of water in gallons per minute at 60°F that will flow through a given valve with a pressure drop of 1 psig.

CFM: Air quantity in Cubic Feet Per Minute. One cubic foot equals 1728 cubic inches.

CIM: Air quantity in Cubic Inches Per Minute.

Close-off: The close-off rating of a valve is the maximum allowable pressure drop to which the valve may be subjected while fully closed.

Compensated Control: A method of control in which the set point of one controller is automatically changed as the conditions at another control point change.

Constant Volume Control: Primarily for the control of total air flow from mixing units used in high pressure, high velocity dual duct air distribution systems.

Control Point: The condition actually being maintained by a controller.

Controllers: Devices which measure changes in temperature, pressure and moisture content and motivate actuators to make adjustments to counteract the change.

Critical Pressure Drop: Fluid flow through a valve increases with increased pressure drop until a critical value is reached. Any drop in excess of this value can cause noise and wear.

Cycling (or hunting): The periodic changing of the controlled variable above and below the set point.

Day/Night Thermostat: A pneumatic thermostat which can be indexed to control at one temperature during day operation and another temperature at night.

Deviation: Departure of the control point from the set point or the amount of that departure.

Differential: Applies to two-position controllers. The change in the controlled condition necessary to cause the controller to move from one extreme of its travel to the other.

Differential Pressure Control: A controller which measures and controls the difference between two separate pressures.

Direct Acting (DA): The action of a controller that increases its branch line pressure as the controlled variable increases.

Direct Acting Actuator Assembly: An actuator which extends with increasing pressure.

Direct Acting Valve Assembly: Valve assembly in which the valve plug is pushed down to close off the flow and raised to open.

Direct Bleed Leakport: An assembly within a pneumatic controller which incorporates a flat lever positioned in the exhaust stream of a flat-tipped nozzle. This produces a nonlinear output with negligible feedback.

Diverting Relay: Normal function is to divert air pressure from one supply line to either of two branch lines or either one of two supply lines to one branch line. Also some types can convert proportional pressure signals to positive pressure changes or feed and exhaust branch lines.

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

Double-seated valve: Fluid pressure is introduced between the two seats enabling the valve to close against high pressure. Should not be used when tight shut off is required.

Dry Bulb Temperature: Air temperature as indicated by an ordinary thermometer.

Dual Thermostat: A two-temperature thermostat. Equivalent to two separate thermostats in one case which have two different set points.

Electric-Pneumatic Relay (E-P): An electrically operated diverting valve designed for diverting air from one control point to another.

Equal Percentage: A flow characteristic through a valve whereby each equal increment of opening increases the flow by an equal percentage over the previous valve.

Feedback: A design feature of some pneumatic controls which provides a true proportional relationship between the movement of a sensing element and the output air pressure variation it produces.

Firestat: A control used in return air duct system to shut down air conditioning or ventilating fans when air temperature goes above a preset limit.

FPM: Air velocity in feet per minute.

Freezestat: A control used to protect against freeze-up of heating coils, cooling coils, or similar temperature applications.

GPM: Water flow in gallons per minute.

Gradual Switch: Pneumatic manual switches for adjusting the air pressure in a line to any valve from zero to full supply pressure.

High Limit: Prevents operation of equipment when it would cause dangerous or undesirably high temperature, pressure or relative humidity.

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

Insertion Thermostat: Controllers with extended elements which can be inserted into a duct or other enclosure in which temperature is to be maintained.

Limit Controller: Controller used in a control system to keep the temperature, pressure or relative humidity in a duct within some limit.

Linear: A flow characteristic through a valve whereby the opening and flow are related in direct proportion.

Mains: In pneumatic control systems mains are air lines carrying air at a constant supply pressure, usually 15 to 25 psig.

Master Controller: A controller which measures conditions at one point and resets the set point of another controller.

Mixing Valve: A three-way valve which has two inlets and one outlet. The valve is constructed with one disc between two seats. It mixes two fluids, in controlled proportions, and directs the mixture to the common outlet.

Motorized Damper: Consists of a damper to which a pneumatic actuator is connected. It is possible to mount the actuator so that the damper is either normally open or normally closed.

Motorized Valve: A pneumatic valve consisting of the actuator and the valve body.

Normally Closed (NC): A controlled device that moves toward the closed position as the branch line pressure decreases is normally closed.

Normally Open (NO): A controlled device that moves toward the open position as the branch line pressure decreases is normally open.

Pilot Bleed Relay: A relay system in some pneumatic controllers which translates the movement of the sensing element into a changing pressure to be fed to the actuator.

Pneumatic Actuator: A standard pneumatic actuator moves toward the advanced position as the branch line pressure increases and toward the retarded position as the branch line pressure decreases.

Pneumatic-Electric Relay (P-E): An air actuated device used to make or break electrical contacts in connection with the operation of the control system.

Positive Acting: In pneumatic control devices this is an abrupt change in the branch line pressure of a controller from zero to 15 psig. This causes either a full open or fully closed condition.

Positive Positioning Relay: An auxilliary device which can be fitted to a damper or valve actuator. It can position the actuator accurately with respect to signal pressure from the controlling instrument, regardless of the load.

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

Proportioning Control: A mode of control in which the controlled device may assume any position from fully closed to fully open.

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

Receiver Controller: This device controls final control devices such as valves or pneumatic actuators in response to air pressure signals. These signals may be supplied by pneumatic transmitters or gradual switches which supply a 3-15 psig signal over a given range.

Relative Humidity: The ratio of the amount of water vapor present in air to the greatest amount possible at the same temperature.

Remote Bulb Thermostat: A thermostat having an element in the form of a liquid or vapor – filled bulb connected by flexible capillary tubing to a bellows or diaphragm.

Reset Control: A device which can have its set point changed automatically by second controller because of changes in temperature, pressure or humidity.

Restrictor: In pneumatic control systems this device is used to maintain a constant supply pressure to transmitters.

Reverse Acting (RA): The action of a controller that decreases its branch line pressure as the controlled variable increases.

Reverse Acting Actuator Assembly: An actuator which retracts with increasing pressure.

Reverse Acting Valve Assembly: Valve assembly in which the valve plug is raised to close off the flow and pushed down to open.

Reversing Relay: This device is used to reverse a proportional signal from a controlling device.

Selector Relay: These relays are used where the application requires the comparison, selection and transmission of one of two proportional signals.

Self-contained Control: A control which has the source of power, sensing element and final control mechanism contained within a single instrument.

Sensitivity: The ratio of the rate of response of a controller to each unit of change of the controlled variable. Example: In a pneumatic thermostat, sensitivity is the PSIG change in control air pressure for each degree of temperature change felt by the sensing element.

Set Point: The degree of temperature, relative humidity or pressure which is desired to be maintained and at which the controller is set.

Single-seated Valve: Used for tight shut off but should not be used where high differential pressures exist.

Single Unit Control: An automatic conditioning system which is regulated by a single thermostat.

Span: Generally the difference between a temperature controller or transmitters lowest possible set point and the highest possible set point. Example: A 40°F to 240°F device has a 200° span.

Static Pressure: The outward push of the air against the walls of a duct.

Submaster Controller: A controller which is automatically set by a master controller as the condition changes at the master controller.

Summer/Winter: A term which designates a controller or control system for year-round heating-cooling control.

Surface Thermostat: Devices designed for mounting on and measuring the temperature of the surface, such as that of a pipe.

Throttling Range: The change in the controlled condition necessary for the controller output to change over a 3-15 psig range.

Transmission: Modern pneumatic control system designed around controllers which have had their functions of sensing and controlling split into separate devices, connected by a "transmission line" air piping.

Transmitter: A pneumatic transmitter measures air or fluid temperatures and transmits a 3-15 psig signal to a controlling and indicating device.

Two Position Control: A method of control in which the final control element is either on or off.

Vacuum Pressure: Pressure below atmospheric pressure.

Volume (Booster) Relay: A device used to amplify the volume of control air and minimize system transmission lag.

Wet Bulb Temperature: Air temperature indicated by a thermometer with a wet wick. As the moisture from the wick evaporates the air will be slightly cooler than the dry bulb reading in the same area.

Zone Control: An area being controlled which is divided into two or more zones and each has its own individual thermostat.

APPENDIX

ABBREVIATIONS

ACU	Air Conditioning Unit
AHU	Air Handling Unit
C	Common
CWR	Chilled Water Return
CWS	Chilled Water Supply
DA	Direct Acting
DPC	Differential Pressure Controller
DPDT	Double Pole-Double Throw
E-P	Electric-Pneumatic Relay
HVU	Heating and Ventilating Unit
HWR	Hot Water Return
HWS	Hot Water Supply
NC	Normally Closed
NO	Normally Open
OA	Outside Air
P-E	Pneumatic-Electric Relay
RA	Return Air or Reverse Acting
SP	Set Point
SPST	Single Pole-Single Throw
TR	Throttling Range
VAV	Variable Air Volume

Air Velocities & Velocity Pressures

Velocity Fpm	Velocity Pressure, In. w.g.	Velocity, Fpm	Velocity Pressure, In. w.g.	Velocity Fpm	Velocity Pressure, In. w.g.	Velocity, Fpm	Velocity Pressure, In. w.g.	Velocity, Fpm	Velocity Pressure, In. w.g.
400	0.01	2150	0.29	3050	0.58	4530	1.28	6200	2.40
565	.02	2190	.30	3100	.60	4600	1.32	6260	2.44
695	.03	2230	.31	3150	.62	4670	1.36	6310	2.48
800	.04	2260	.32	3200	.64	4730	1.40	6360	2.52
895	.05	2300	.33	3250	.66	4800	1.44	6410	2.56
980	0.06	2330	0.34	3300	0.68	4870	1.48	6460	2.60
1060	.07	2370	.35	3350	.70	4930	1.52	6510	2.64
1130	.08	2400	.36	3390	.72	5000	1.56	6560	2.68
1200	.09	2440	.37	3440	.74	5060	1.60	6610	2.72
1270	.10	2470	.38	3490	.76	5120	1.64	6650	2.76
1330	0.11	2500	0.39	3530	0.78	5190	1.68	6700	2.80
1390	.12	2530	.40	3580	.80	5250	1.72	6750	2.84
1440	.13	2560	.41	3620	.82	5310	1.76	6800	2.88
1500	.15	2590	.42	3670	.84	5370	1.80	6840	2.92
1550	.15	2620	.43	3710	.86	5430	1.84	6890	2.96
1600	0.16	2650	0.44	3750	0.88	5490	1.88	6940	3.00
1650	.17	2680	.45	3790	.90	5550	1.92	6980	3.04
1700	.18	2710	.46	3840	.92	5600	1.96	7030	3.08
1740	.19	2740	.47	3880	.94	5660	2.00	7070	3.12
1790	.20	2770	.48	3920	.96	5710	2.04	7120	3.16
1830	0.21	2800	0.49	3960	0.98	5770	2.08	7160	3.20
1880	.22	2830	.50	4000	1.00	5830	2.12	7210	3.24
1920	.23	2860	.51	4080	1.04	5880	2.16	7250	3.28
1960	.24	2880	.52	4160	1.08	5940	2.20	7300	3.32
2000	.25	2910	.53	4230	1.12	5990	2.24	7340	3.36
2040	0.26	2940	0.54	4310	1.16	6040	2.28	7380	3.40
2080	.27	2970	.55	4380	1.20	6100	2.32	7430	3.44
2120	.28	2990	.56	4460	1.24	6150	2.36	7470	3.48

Conversion Information

TABLE OF EQUIVALENT TEMPERATURES

°C.	°F.	°C.	°F.	°C.	°F.
−50	−58	75	167	200	392
−45	−49	80	176	205	401
−40	−40	85	185	210	410
−35	−31	90	194	215	419
−30	−22	95	203	220	428
−25	−13	100	212	225	437
−20	− 4	105	221	230	446
−15	+ 5	110	230	235	455
−10	14	115	239	240	464
− 5	23	120	248	245	473
0	32	125	257	250	482
5	41	130	266	255	491
10	50	135	275	260	500
15	59	140	284	265	509
20	68	145	293	270	518
25	77	150	302	275	527
30	86	155	311	280	536
35	95	160	320	285	545
40	104	165	329	290	554
45	113	170	338	295	563
50	122	175	347	300	572
55	131	180	356	305	581
60	140	185	365	310	590
65	149	190	374	315	599
70	158	195	383	320	608

TABLE OF VALUES FOR INTERPOLATION IN TABLE ABOVE

°F.	°C.	°F.	°C.	°F.	°C.
1	= 0.55	4	= 2.22	7	= 3.88
2	= 1.11	5	= 2.77	8	= 4.44
3	= 1.66	6	= 3.33	9	= 5.00

All decimals are repeating decimals.

°C.	°F.	°C.	°F.	°C.	°F.
1	= 1.8	4	= 7.2	7	=12.6
2	= 3.6	5	= 9.0	8	=14.4
3	= 5.4	6	= 10.8	9	=16.2

All decimals are exact.

TEMPERATURE

$$T_F = \frac{9}{5} T_C + 32$$

$$T_C = \frac{5}{9} (T_F - 32)$$

LENGTH

1 inch = 2.54000 cm.

MASS

1 pound = 453.5924 grams

1 kilogram = 1000 grams

STANDARD ATMOSPHERE

29,921 inches (76.0 cm.) mercury

29.921 inches (76.0 cm.) mercury column where mercury density is 13.5951 grams per cubic centimeter and gravity is 980.665 centimeters per second per second at sea level.

Correction factor for other levels from −1000 to +5000 ft.: approximately −0.5 psig per additional 1000 ft.

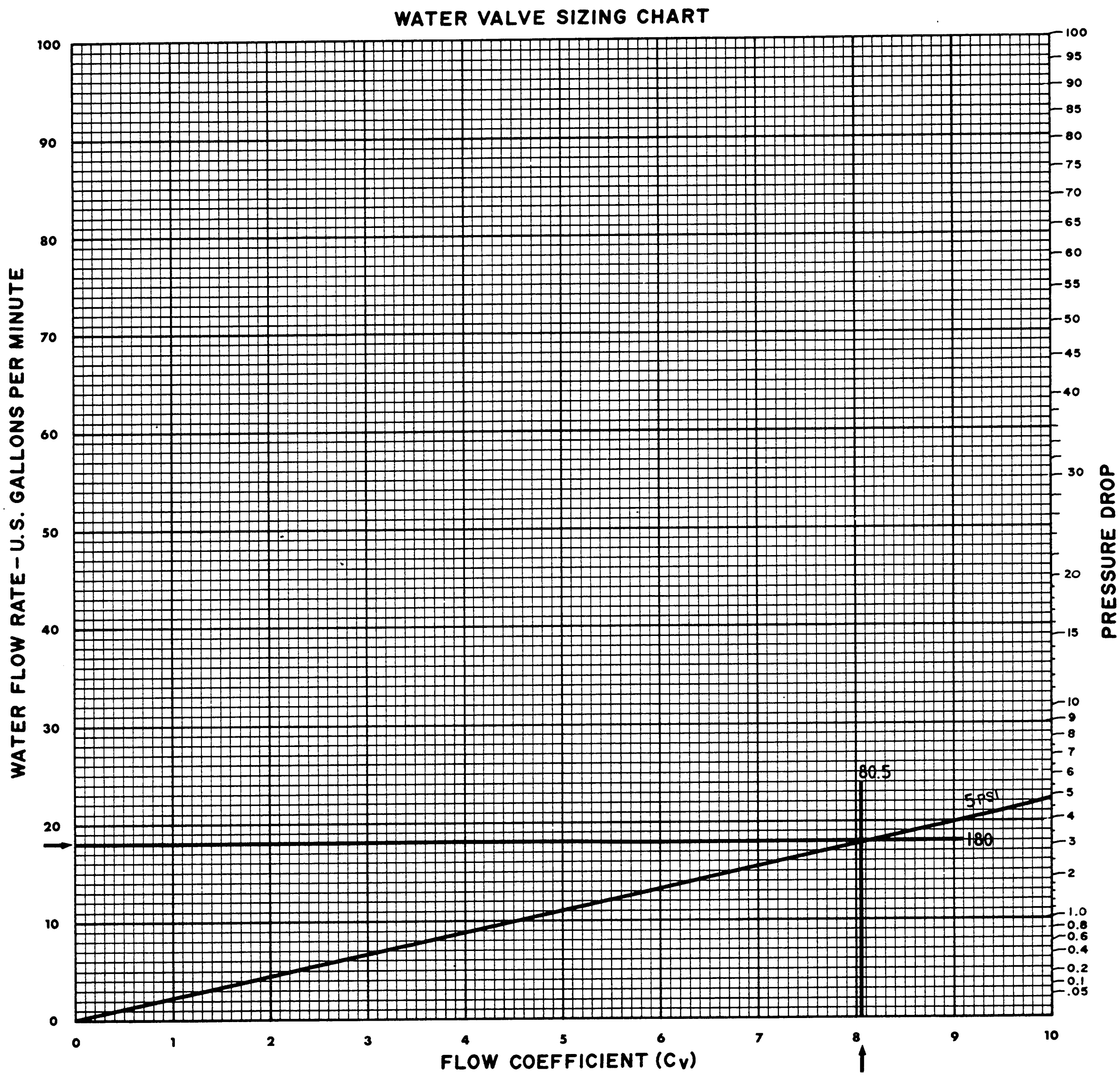
PRESSURE

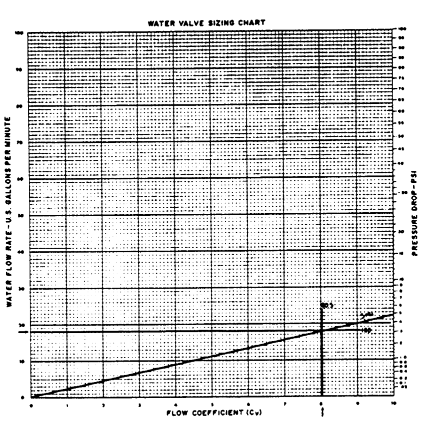
Units	lb./in.^{2}	kg./cm^2	bar	ft. H_2O (60°F)	Atm.	in. Hg. (32°F)	mm. Hg. (0°C)
1 Atmosphere	14.6959	1.033228	1.013250	33.934	1*	29.921	760*
1 kg./cm^2	14.2233	1*	0.980665*	32.843	0.967841	28.959	735.559
10 lb./in.^{2}	10*	0.70307	0.689476	23.091	0.68046	20.360	517.149
1 bar	14.5038	1.019716	1*	33.490	0.986923	29.530	750.062
1 meter Hg. (0^{2}C)	19.3368	1.35951	1.333224	44.65	1.31579	29.530	1000*
10 in. Hg. (32°F)	4.9115	0.34532	0.33864	11.341	0.33421	39.370	254*
100 ft. H_2O (60°F)	43.308	3.0448	2.9859	100*		88.175	2239.6

ENERGY

Units	BTU	Ft. lb.	Hp. hr.	Lb/in^2 x ft.^{3}	Amt. x cm.^{3}	Int. w. hr.	Int. joules	Kg.m.	IT. cal.	Abs. joules
10^4 Abs. joules	9.4770	7375.62	0.00372506	51.2196	98692.3	2.77694	9997	1019.72	2388.17	10000*
10^4 Kg. m.	92.938	72330	0.0365304	502.293	9678.41	27.2325	98037.1	10000*	23420	98066.5
10^4 Ft. lb.	12.8491	10000*	0.0050505	69.4444	133809	3.7650	13554.1	1382.55	3237.9	13558.2
10^4 Int. joules	9.4799	7377.8	0.0037262	51.235	98722	2.7778	10000*	1020.02	2388.9	10003
10^{-3} Int. Kw. hr.	3.41275	2656.0	0.0013414	18.4446	35540	1*	3600*	367.21	860*	3601.1
10^4 IT. cal.	39.683	30884	0.0155980	214.47	413255	11.6279	41860.5	4269.9	10000*	41873
10 Btu.	10*	7782.6	0.0039306	54.046	104138	2.93019	10548.7	1075.99	2519.96	10551.8
10^2 Lb./in^2 X ft.^{3}	18.5027	14400*	0.0037744	100*	192685	5.4216	19517.9	1990.87	4662.6	19523.8
10^5 Atm. X cm.^{3}	9.6026	7473.35	0.0037744	51.898	100000*	2.81374	10129.5	1033.23	2419.8	10132.5
10^{-3} Hp. hr.	2.5441	1980*	0.001*	13.750	26494	0.74548	2683.7	273.75	641.1	2684.5

Water Valve Sizing Chart





TO DETERMINE Cv: (1) Locate desired pressure drop on right side of chart.

(2) Draw locus line from origin to desired pressure drop.

(3) Draw horizontal line from desired flow rate at left of chart to locus line. *Water flow rate and Cv scales may be multiplied by any common factor.*

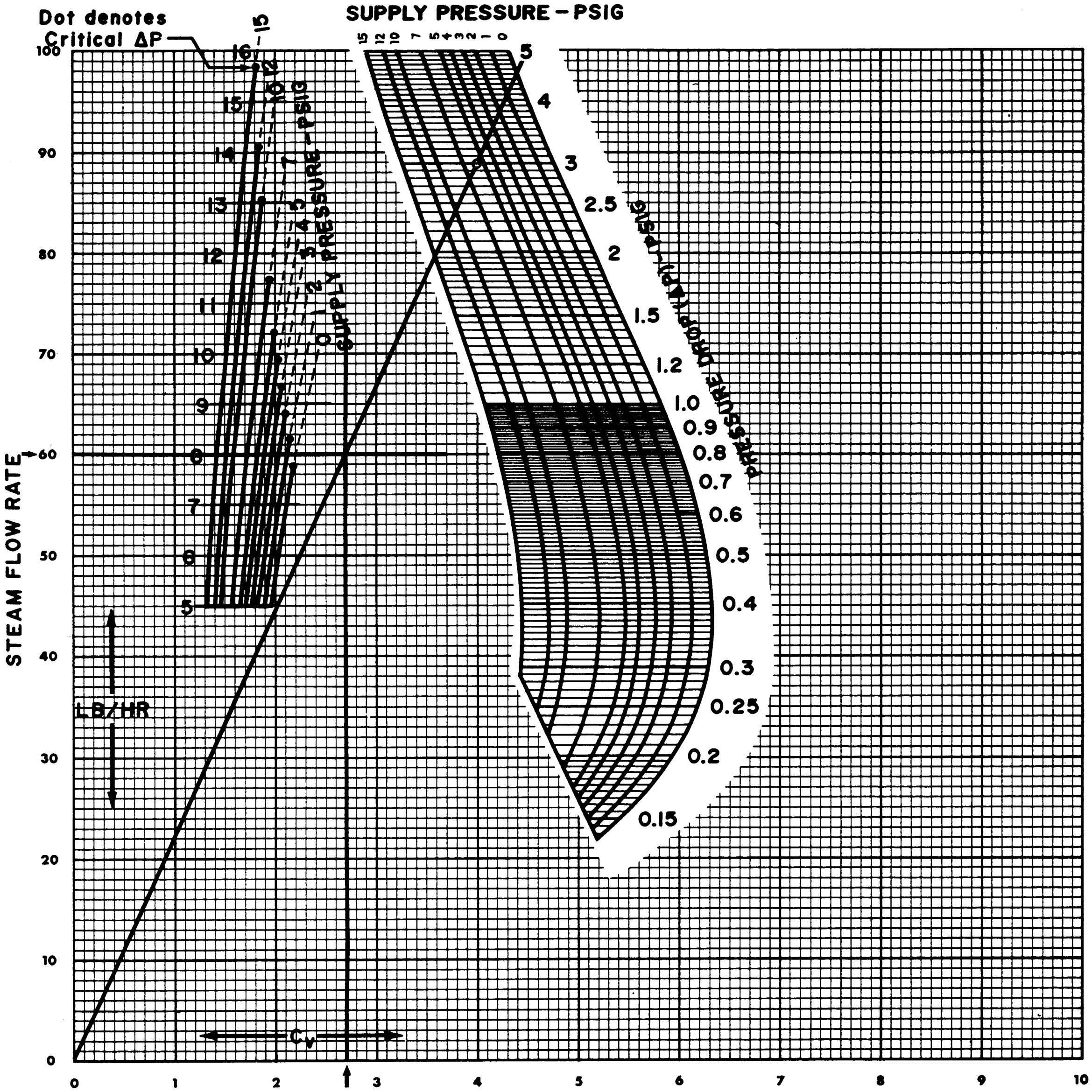
(4) Where horizontal line intersects locus line, draw vertical line down to bottom of chart and read Cv.

EXAMPLE: Given 180 GPM flow rate and 5 psi maximum allowable pressure drop, Cv of 80.5 is required.

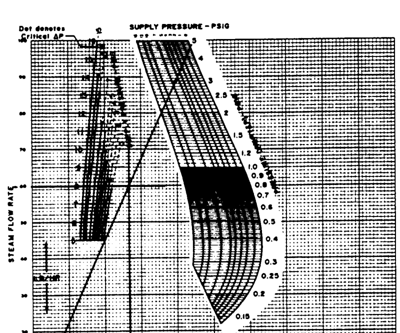
TO DETERMINE PRESSURE DROP: Mark intersection of flow rate (GPM) and selected Cv. Draw locus line from origin through intersection and extend to valve on pressure drop scale.

CONVERSION FACTORS: 1 ft. water column = 0.433 psi.
1 psi = 2.31 ft. water column

Low Pressure Steam Valve Sizing Chart



Based on FCI 62-1: $C_V = W/2.1\sqrt{\Delta P(P_1+P_2)}$ [W=lb/hr, P_1 = entering psia, P_2 = leaving psia, no superheat]



TO DETERMINE Cv: (1) Mark intersection of steam supply pressure (psig) and maximum allowable pressure drop (psi) within proper family of curves.

(2) Draw locus line from origin through intersection.

(3) Draw horizontal line for flow rate (lb/hr). *Steam Flow Rate and Cv scales may be multiplied by any common factor.*

(4) Draw vertical line down from intersection of flow rate and locus lines; read minimum Cv. (Steps 3 and 4 can be done by inspection.)

EXAMPLE: Given 5 psig steam supply, 3 psi maximum allowable pressure drop and 600 lb/hr flow rate. Minimum Cv of 27.3 is required.

TO DETERMINE PRESSURE DROP: Mark intersection of flow rate (lb/hr) and selected Cv. Draw locus line from origin through intersection. Where locus line crosses supply pressure curve, read horizontally to pressure drop scale.

NOTE: Chart based on sea level atmospheric pressure (14.7 psia). For higher elevations, reduce given supply pressure (psig) by 0.5 psi per 1000 feet before using chart.

NOTES