

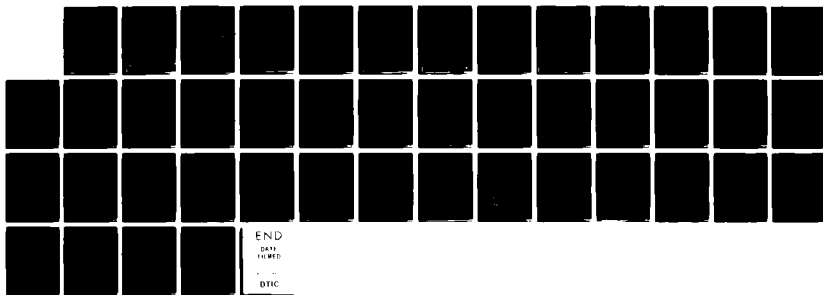
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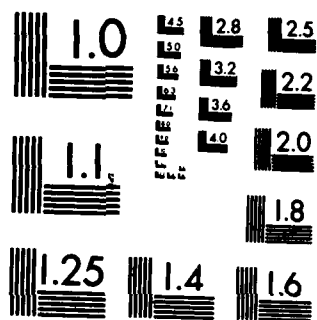
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AIR CONDITIONING (..(U) NAVAL CIVIL ENGINEERING LAB
PORT HUENEME CA R E KIRTS MAR 83 NCEL-TN-1658

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ENERGY CONSUMPTION IN SINGLE BUILDINGS

AUTHOR: R. E. Kirts

DATE: March 1983

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NAVAL CIVIL ENGINEERING LABORATORY
PORT HUENEME, CALIFORNIA 93043

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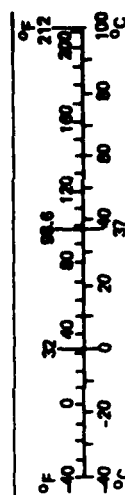
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
in ft yd mi	inches	<u>LENGTH</u> 2.5 30 0.9 1.6	centimeters	cm
	feet		centimeters	cm
	yards		meters	m
	miles		kilometers	km
in ² ft ² yd ² mi ²	square inches	<u>AREA</u> 6.5 0.09 0.8 2.6 0.4	square centimeters	cm ²
	square feet		square meters	m ²
	square yards		square meters	m ²
	square miles		square kilometers	km ²
oz lb	ounces	<u>MASS (weight)</u> 28 0.45 0.9	grams	g
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	(2,000 lb)			
tsp fl oz c pt qt gal ft ³ yd ³	teaspoons	<u>VOLUME</u> 5 15 30 0.24 0.47 0.95 3.8 0.03 0.76	milliliters	ml
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	fluid ounces		milliliters	ml
	cups		liters	l
	pints		liters	l
	quarts		liters	l
	gallons		liters	l
	cubic feet		cubic meters	m ³
°F	Fahrenheit temperature	<u>TEMPERATURE (exact)</u> 5/9 (after subtracting 32)	Celsius temperature	°C

Approximate Conversions from Metric Measures

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mm cm m km	millimeters	<u>LENGTH</u> 0.04 0.4 3.3 1.1 0.6	inches	in
	centimeters		inches	in
	meters		feet	ft
	kilometers		yards	yd
cm ² m ² km ² ha	square centimeters	<u>AREA</u> 0.16 1.2 0.4 2.5	square inches	in ²
	square meters		square yards	yd ²
	square kilometers		square miles	mi ²
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g kg t	grams	<u>MASS (weight)</u> 0.035 2.2 1.1	ounces	oz
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ml l l m ³ m ³	milliliters	<u>VOLUME</u> 0.03 2.1 1.06 0.26 36 1.3	fluid ounces	fl oz
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	liters		gallons	gal
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°C	Celsius temperature	<u>TEMPERATURE (exact)</u> 9/5 (then add 32)	Fahrenheit temperature	°F

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price \$2.25, SD Catalog No. C13.10-286.



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shedding, economizer and enthalpy cycles, scheduled temperature reset, chiller control and chilled water reset, boiler control and hot water temperature reset, and condenser water temperature reset. Recent developments in HVAC control system hardware, such as pneumatic systems, electropneumatic systems, digital-electronic systems, and microcomputer-based control systems, are also discussed. The strategies are described and compared to each other in terms of cost effectiveness. The BLAST computer program is used to evaluate the various control strategies. The results illustrate the energy-saving potential of simple strategies, such as night and weekend setback and scheduled start-stop, which are inexpensive to implement and should be installed in most buildings. The most complex strategy is not necessarily the most effective due to the interactions between the building, climate, and HVAC system.

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INTRODUCTION

This report presents a discussion of the most common control strategies and equipment used to reduce the amount of energy consumed by heating, ventilating, and air conditioning (HVAC) systems. Emphasis is placed on comparatively small HVAC systems, such as those found in many office buildings and shops, and on stand-alone implementation. The control systems described herein can be implemented independently or as part of a phased program to reduce energy consumption. None of the control strategies requires connection with an energy monitoring and control system (EMCS) or other computer-based system; most of the strategies can be implemented through simple modifications to the HVAC control system using inexpensive, readily available components.

Because of the variety of HVAC systems in use at Naval facilities and differences in climate and building design and use, not all the strategies presented can be employed at every facility. Also, the marginal improvement in energy savings will decrease as additional energy-saving strategies are employed. Consequently, site-specific analyses will be required to determine which strategies will be the most cost effective and the order in which the strategies should be implemented.

BACKGROUND

The performance of heating, ventilating, and air conditioning systems at Naval facilities has been a subject of increasing concern to facility managers in recent years. Heating and air conditioning systems often account for a large portion of facility energy costs and sometimes fail to provide the level of personal comfort desired. In addition, maintenance and repair of HVAC systems has been a persistent problem at some facilities. The condition of many Navy HVAC systems and the increasing cost of energy resulted in the establishment, in 1979, of the Air Conditioning Tune-Up (ACT-UP) Program by the Naval Facilities Engineering Command. The goal of the ACT-UP program, administered by the Naval Energy and Environmental Support Activity, is to improve the operation and maintenance of major air conditioning systems. The program has produced guidance (Ref 1) for evaluating, rehabilitating, and improving the efficiency of large HVAC installations. Using the process presented in the "Procedure Manual for Air-Conditioning Tune-Up (ACT-UP) Program," many Naval facilities have been evaluated for possible improvements to their HVAC systems. Analysis of the ACT-UP studies completed to date indicates that, although some system problems were caused by faulty design and installation, the majority of problems can be attributed to improper or inadequate maintenance and repair of the control system. Problems encountered during ACT-UP surveys include: inoperable or disconnected sensors, control elements, and actuators; control elements out of calibration; excessive leakage in pneumatic control lines; and inoperative control air dryers and filters.

Many of the problems revealed by the ACT-UP program have also been found in the smaller HVAC systems located in single buildings. These small HVAC systems are characterized by packaged through-the-wall units, self-contained rooftop units, and small built-up systems. Packaged units are equipped with factory-installed control systems, while built-up HVAC systems utilize control systems designed for the specific application and assembled from standard hardware components.

Solutions to the problem of high energy consumption by HVAC systems can be found in two approaches to the problem: upgrading control system hardware and employing improved control strategies. These two methods of reducing the energy consumption in HVAC systems are described in detail in the following paragraphs.

DISCUSSION

Approach

The alternative methods of reducing energy consumption in small HVAC systems will be described and compared to each other in terms of economic performance. The measure of economic performance will be the cost effectiveness of the strategy or equipment, defined as the present value of the energy savings divided by the present value of the costs of implementing the strategy. Results of computer simulations of the various options will be presented to illustrate the magnitude of savings available. Note, however, that the energy savings presented are for specific HVAC systems, buildings, and climates. The applicability of a specific strategy or hardware device to a particular problem must be assessed on a case-by-case basis to obtain a realistic measure of the effectiveness of the proposed improvement.

Strategies for Reducing HVAC Energy Consumption

1. Bring the existing control system up to design specification while retaining the original control strategy.

The key to implementing every HVAC energy reduction strategy is correctly operating conventional control hardware. Thus, the necessary first step is to confirm that the existing HVAC control systems are operating as they should, and if they are not, bring them up to specification. On the simplest systems, this would involve no more than verifying that the temperature sensor is calibrated and that the set point and deadband of the temperature controller are correctly set. More complicated control systems, such as those found on large, packaged HVAC units and built-up HVAC systems, will probably require the services of an air conditioning service contractor to bring them up to specification.

2. Employ a new control strategy.

Many control strategies, and the devices required to implement them, have been introduced in the past few years for the purpose of reducing the energy consumption of HVAC systems. While all of these

strategies can be implemented with energy monitoring and control systems, many can be put into use with simple, low-cost devices. These strategies are described below.

a. Scheduled start-stop.

Scheduled start-stop is a strategy that implements the first law of energy conservation: if the equipment does not need to be in service, shut it off! Scheduled start-up uses programmable timers, either electro-mechanical pin programmable types or all electronic types, to start and stop the HVAC system at predetermined times of the day. Many of the timing devices have provisions for programming special schedules for weekends and holidays. A typical application of scheduled start-stop is to turn on an HVAC system at 6:00 a.m. and off at 6:00 p.m. every weekday and to leave the system off during weekends and holidays. Most applications of scheduled start-stop also incorporate high and low limit switches in the control system to prevent extremes in building temperature and humidity. For example, the timer function might be overridden if the building temperature rises above 78°F or falls below 55°F. Typical installations are illustrated in Figure 1. The electric-pneumatic relay is a two-position, three-way air valve used to convert the electrical signal from the timer into the appropriate pneumatic signal.

Although programmable timers are among the simplest and least expensive energy conservation devices to install, experience has shown that improper operation and maintenance of these devices can lead to numerous problems. Often, programmable timers are reset or otherwise tampered with by building occupants. To be effective, timers (and other control devices for that matter) should be secured from access by unauthorized personnel. Another common problem is the tendency to put timers on so many pieces of equipment that the job of maintaining all of them (e.g., resetting them after a power outage) becomes a significant task. Often, timers fail to receive regular maintenance and, as a consequence, the HVAC equipment does not operate as anticipated.

b. Day-night setback.

Closely related to the concept of scheduled start-stop is the strategy of day-night setback. Day-night setback is the strategy of reducing the heating space temperature setpoint or raising the cooling space temperature setpoint during periods when the space is not occupied or when a large change in setpoint is acceptable from comfort considerations. Day-night setback is thus a scheduled change in controller setpoint rather than a scheduled enable-disable of an essential controller function. Energy conservation is achieved through a reduction in heating or cooling load during the period of the control system setback. Guidelines for implementing temperature setback have been published as NCEL Techdata Sheet 78-42, "Energy Conservation Calculations for Night Setback Systems"; Techdata Sheet 78-43, "Temperature Setback Guide for Central Airhandling/Reheat System"; Techdata Sheet 78-44, "Temperature Setback Guide for Packaged Heating and Cooling Systems"; Techdata Sheet 78-45, "Temperature Setback Guide for Steam Radiation/Convection System";

Techdata Sheet 78-46, "Temperature Setback Guide for Hot Water Radiation/Convection System"; Techdata Sheet 78-47, "Temperature Setback Guide for Heating and Ventilating System"; Techdata Sheet 78-48, "Temperature Setback Guide for Central Air-Handling/Mixing System"; and Techdata Sheet 78-49, "Temperature Setback Guide for Forced Air Furnace Heating System".

c. Optimum start-stop.

Optimum start-stop is a strategy that starts and stops HVAC equipment based on a comparison of indoor and outdoor conditions, equipment capacity, building characteristics, and a schedule of equipment performance. The devices that implement optimum start-stop determine the latest time in the morning that the HVAC equipment can be turned on and still satisfy the space comfort requirements at the start of the building occupancy period, and they also determine the earliest time for stopping equipment at the end of the day. Starting time can be determined from the relationship,

$$\tau_{\text{Start Heating}} = \tau_{\text{Start Scheduled Occupancy}} - \frac{[A(T_i - T_o) - B]}{Q_H} \quad (1)$$

where

τ = time, hr

T_i = desired indoor temperature, °F

T_o = measured outdoor temperature, °F

A = a constant that characterizes the heat loss rate of the building envelope, Btu/hr/°F

B = a constant that characterizes internal heat sources, Btu/hr

Q_H = capacity of the heating system, Btu/hr

The optimum stopping time is often approximated using the relationship,

$$\tau_{\text{Stop Heating}} = \tau_{\text{Stop Scheduled Occupancy}} - \frac{T_{\text{Setback}} C}{[A(T_i - T_o) - B]} \quad (2)$$

where

T_{Setback} = desired nighttime temperature setback, °F

C = a constant that characterizes the thermal inertia of the building

Similar relationships apply to the cooling mode of system operation.

The constants A, B, C, Q_H , $T_{Setback}$ are input to an optimum start-stop controller at the time of installation. Values for A, B, and Q_H can be obtained from the building design calculations, while C will probably have to be determined experimentally.

New installations of optimum start-stop devices require an initial period of "tuning" to determine which set of constants best describe the characteristics of the building and HVAC system.

Optimum start-stop controllers can be obtained in either analog-electronic or digital-electronic models and are connected into the existing control system in the same manner as scheduled start-stop timers.

Simple, single-function optimum start controllers are often marketed under the generic term "smart thermostats." These devices utilize indoor and outdoor temperature sensors and solid state memory circuits to "learn" and then "remember" the amount of time it takes for the HVAC system to bring a building or zone to the design condition as a function of outdoor air temperature and then start the equipment at the appropriate time. For example, a smart thermostat in a single-family residence heating system would let the nighttime temperature drift down to a preset minimum value, say 55°F, then turn on the heating system at the appropriate time (based on the outdoor air temperature) so that the house will be heated to a preset temperature by a preset time, for example, heated to 70°F by 6:30 a.m.

Analysis (Ref 2) has indicated that the use of smart thermostats and other optimum start-stop devices can result in a cost savings beyond the savings attributable to energy conservation. It has been estimated that if an optimum start-stop control strategy is employed, the HVAC system need have a capacity of only 110% of the design load rather than the 150% overcapacity that is normally specified. Smaller equipment, of course, means a savings in capital cost and space. Also, most HVAC equipment operates at higher efficiency when run near design capacity. In the past, the capacity of HVAC equipment was often overspecified so that rapid warmup or cooldown could be achieved, but with an optimum start-stop controller the warmup or cooldown period is adjusted to meet the load.

d. Deadband control

The deadband control strategy saves energy by widening the range of environmental conditions that satisfy the control system (i.e., by widening the range over which neither heating nor cooling is provided). This is accomplished by reducing the heating and cooling load through changes in the control points. Unlike the previously described control strategies, however, deadband control may result in some building occupant dissatisfaction as a result of changes in the setpoints.

Deadband control is easily implemented on packaged heating and cooling systems equipped with conventional dual function (heating/cooling) thermostats by adjustment of the temperature setpoints (Figure 2a).

The full energy-saving capabilities of deadband control are realized only in the more complex types of HVAC systems - especially those systems that permit variation in the quantity or temperature of the air supplied

to the conditioned space (Figure 2b). Since implementation of the deadband control strategy in complex HVAC systems requires a detailed analysis of the characteristics of the individual system type and specific installation details, a representative example was chosen to illustrate implementation of the strategy in large systems. Figure 3 illustrates a typical mixed-air HVAC system (e.g., a multizone or dual-duct design) that provides a fixed quantity of outside air for ventilation purposes. With the conventional control system illustrated in Figure 3, the zone thermostats might be set at 73°F and the system throttling range might be 2°F. At 72°F, the mixing dampers allow the maximum amount of warm air to enter a zone, and at 74°F the dampers allow the maximum amount of cool air to enter. With the deadband control strategy the throttling range is increased to 10°F (setpoint remaining fixed at 73°F), and the system deadband is increased to 5°F. Zone-mixing dampers will then begin to supply warm air when the zone temperature drops to 70.5°F and will supply the maximum amount of warm air if the zone temperature drops to 68°F. Similarly, the cooling process is initiated when the zone temperature reaches 75.5°F and will operate at maximum capacity if the zone temperature reaches 78°F. Energy is conserved because no heating or cooling is used if the zone temperature is between 70.5°F and 75.5°F. Outside the deadband, the coils warm and chill the air supplies only enough to meet the load. Energy consumption is further reduced by resetting the hot and cold duct temperatures in response to the demands of the zones that require the most heating and cooling (this is discussed in more detail in section 2h).

Deadband control is implemented by modifying the existing control system as shown in Figure 4 (Ref 3). The zone thermostats must have a wide throttling range, but the other components are conventional pneumatic (or electronic) control system components. The thermostat output from a zone regulates the zone-mixing damper through a proportional relay. The thermostat output also goes to a high-low pressure selector, which relays the highest control pressure to the cold duct controller and the lowest control pressure to the hot duct controller. The duct air temperatures are thus regulated by the zones needing the most heating or cooling.

Deadband control can also be implemented in reheat-type HVAC systems. The control strategy for deadband control of reheat systems is illustrated in Figure 5 (Ref 3), and the necessary control configurations are presented in Figures 6 and 7. As with mixed-air systems, new or modified thermostats with a wide throttling range are required. Also shown in Figure 6b is an optional, low-pressure selector that samples the inputs from critical cooling zones (e.g., those in the interior of a building) and uses the output of the zone thermostat calling for the most cooling to regulate the cold plenum temperature (see section 2h).

Variable air volume (VAV) systems can also be equipped with deadband control, although the inherent energy efficiency of VAV systems may result in lower energy savings when compared to installations of deadband control on mixed-air and reheat systems. The control strategy is illustrated in Figure 8. Zone controls are similar to those for reheat systems (Figure 6) except that dampers, rather than valves, are the controlled devices. A simplified apparatus control system is presented in Figure 9 (plenum temperature reset, fan speed, and air-mixing damper

controls have been deleted for clarity). The winter/summer changeover switch reverses the action of the zone thermostats and changes the setpoint of the supply air thermostat. Changeover may be either manual or based on outdoor air temperature.

Variable air volume systems having terminal reheat may be modified for deadband control in a manner similar to that used for reheat systems (with modification for damper control).

e. Duty cycling.

Duty cycling as an HVAC control strategy means shutting down equipment for brief periods of time during the hours the building is occupied. A simple duty cycling strategy is implemented much like scheduled start-stop: an electromechanical or electronic timer sequences through a series of switch contacts to operate equipment in a fixed order and for fixed amounts of time. More complex duty cycle controllers vary the duration of the equipment periods in relation to sensed temperature conditions in the controlled zone and can be programmed to give certain equipment items priority over others with regard to startup and shutdown. They can also be programmed to prevent the simultaneous startup of equipment (to prevent electrical power demand peaks).

While duty cycling can save energy in some circumstances, it should be used with caution. Frequent on-off cycling can result in damage to some types of equipment, such as large electric motors, compressors, and boilers. Also, if the periods of shutdown are long enough to permit an appreciable change in the temperatures of the HVAC system components, the energy input required to bring the components back to operating temperature after startup can substantially reduce the anticipated energy savings.

f. Demand limiting and load shedding.

The demand limiting strategy for energy conservation limits the connected electrical load to prevent exceeding a predetermined value of peak electrical demand and thereby incurring demand charges from the electric utility company. The demand limiting strategy can be applied only where the facility is billed by electrical demand in addition to electrical power consumption. Demand limiting controllers are electronic devices that monitor the power demand of a building and predict near-term power demand. When the predicted value of power demand exceeds the preset upper limit value, the controller acts to reduce the demand by shutting off equipment in a scheduled order. For example, in the HVAC system illustrated in Figure 3, the chiller might be shut down first, followed by the circulation pumps, and finally the fans. Demand limiting controllers for large HVAC systems or for buildings having a multiplicity of HVAC system requirements are often programmed to shed loads according to a protocol-relating equipment priority and a revolving queue. This doctrine is intended to equitably distribute any discomfort attributable to equipment shutdown and prevent the same pieces of equipment from always being the first shed and the first to be reconnected.

A very simple demand limiting strategy is scheduled start-stop, with the equipment off-periods timed to coincide with daily periods of peak demand. These periods of peak electrical demand are available from the electric utility company.

Some utility companies also provide coded signals to customers over power or telephone lines that mark the beginning and end of intervals of high electrical demand. These signals can be interfaced with the more elaborate demand limiting devices to provide flexible and automatic control.

g. Economizer and enthalpy cycles.

The economizer cycle and the enthalpy cycle are HVAC control strategies that reduce energy consumption by reducing the load on the cooling equipment. The mechanical cooling load is reduced by employing a control device that selects, as the source of makeup air for the chilled air supply, the air stream that requires the least energy input to cool. The two sources of makeup air usually available are return air and outdoor air. An economizer controller will select between return air and outdoor air based on dry-bulb temperature: the air supply having the lower dry-bulb temperature is selected as the source of makeup air. An enthalpy controller selects between outdoor air and return air on the basis of enthalpy (or total energy content). The air supply having the lower enthalpy is selected as source of makeup air. Enthalpy is computed from measurements of dry-bulb temperature and relative humidity. An installation of an enthalpy controller is illustrated in Figure 10. The installation of an economizer controller is similar to that of an enthalpy controller, the differences being that humidity sensing is not used and, of course, an economizer controller is substituted for the enthalpy controller.

An enthalpy controller measures the enthalpy of the outdoor air and return air streams by summing signals from dry bulb temperature and relative humidity sensors, comparing the sums, then operating the air-mixing dampers to provide the correct proportions of outdoor and return air. The controller logic assumes that the outdoor air is cooler than the return air and, therefore, acts to increase the proportion of outdoor air as the mixed-air temperature rises. When the controller senses that the outdoor air enthalpy is greater than the return air enthalpy, it causes the mixed-air dampers to be positioned to provide only a required minimum amount of outdoor air.

In many areas of the country, an enthalpy controller should include a control circuit to close the outdoor air dampers to their minimum setting whenever the outdoor air dry-bulb temperature exceeds the return air temperature. Under this condition only sensible cooling takes place; thus, the air stream with the lower dry-bulb temperature is the most economical to cool. An exception to this circumstance is when moisture is added to the air stream (e.g., when an air washer or sprayed coil is used).

Enthalpy controllers cost more than economizer controllers and may be more difficult to maintain (primarily because of the humidity sensors), but the energy savings over economizer controllers can be significant. A site-specific analysis can determine which system is better suited to a particular application. In a warm, dry climate, where dry-bulb temperature effects are predominant, the economizer system has performed well and is usually the more cost effective (Ref 4).

An expansion of the economizer cycle, usually termed "air handler programming" or "ventilation-recirculation control," can often be beneficial in reducing the energy consumption in buildings having fairly uniform load requirements between zones - in single-zone applications. An air handler program operates to heat or cool the building using only outdoor air whenever feasible. During winter, when the air temperature is warmer than the space temperature, the fans and dampers are operated to warm the building with outdoor air. Conversely, during summer days when the outdoor air is cooler than the conditioned space, the fans and dampers are operated to provide cooling.

Outdoor air dampers should always be closed to the minimum allowable setting during building warmup or cooldown periods to reduce the load on the HVAC system.

h. Scheduled temperature reset.

Scheduled temperature reset is a control strategy that reduces energy consumption by maintaining the temperature of the conditioned fluid (air or water) at a value just sufficient to meet the demands of the zone having the largest load. The temperature of the conditioned fluid is adjusted (reset) based upon either outdoor conditions or conditions in the building. The temperature of the heated fluid, the chilled fluid, or both fluids can be reset.

For example, consider the mixed-air system illustrated in Figure 11. In the absence of reset control, the hot duct temperature will be maintained at a constant temperature of, for example, 100°F, and the cold duct temperature will be kept constant at, for example, 55°F. These two air streams are then mixed in varying proportions to produce the desired zone temperature.

With temperature reset control, the temperature of the air in the hot duct is reduced as the heating load decreases, and the temperature of the air in the cold duct is increased as the cooling load decreases. The outdoor air temperature is often chosen as the measure of the heating or cooling load (Figure 11), although the temperature in critical building zones can also be used as the measure of the loads (see Figure 4). The reset controller operates to control the duct temperature, between upper and lower limits, in a preset proportion to the sensed load. The actual modulation of the duct temperature is accomplished by varying the amount of heated and chilled water that bypass the coils or by changing the temperature of the water supplied to the coils. The latter method results in larger energy savings and is discussed further in sections 2j and 2k.

Figure 6 illustrates a control system to implement reset of the hot and cold duct temperatures based on outdoor air temperature. In pneumatic control systems, temperature reset is usually implemented using a dual-input proportional controller, a device that amplifies a small pressure change from a sensor into a proportionate pneumatic control pressure and changes its calibration point by means of a second sensor input. Simple reset controllers, which reset the temperature in discrete steps rather than continuously, can be assembled using combinations of master and submaster thermostats or using thermostats and pressure selector relays. A recommended source of information on modern HVAC controls is Reference 5.

A word of caution: in some climates it is possible to lose control over building humidity by allowing the cooling coil temperature to be reset to too high a value. As a consequence, cold duct temperature reset may not be desirable in warm, humid climates, or the range of temperature reset may have to be more restricted.

Temperature reset control can also be used to change the cold plenum temperature in terminal reheat systems (see Figures 6b and 7).

In variable air volume systems, the heating or cooling load is met by regulating the quantity of air supplied to a zone rather than the temperature of the air supplied. Expressed as an equation,

$$Q = 1.1(\text{CFM})(T_S - T_R) \quad (3)$$

where

Q = load, Btu/hr

CFM = air flow rate, ft^3/min

T_S = temperature of supply air, $^{\circ}\text{F}$

T_R = temperature of return air, $^{\circ}\text{F}$

The return air temperature in a conventional VAV system might be set at, for example, 73°F . The supply air temperature might be 100°F in the heating mode and 55°F in the cooling mode of operation. Changes in zone loads are normally met by changing the CFM supplied to the zone and, since fan power consumption is directly proportional to the volumetric flow rate, substantial energy savings are achieved when compared to constant air volume systems. Examination of Equation 3 shows that any temperature reset control strategy will reduce the magnitude of the quantity $(T_S - T_R)$ so that the load must be met by increasing the air flow rate. Increasing the air flow rate results in an increase in electrical power consumption for fans, which offsets the reduction in thermal energy supplied to the coil. For this reason, temperature reset strategies are seldom employed in variable air volume systems.

i. Chiller control and chilled water reset.

Chiller controls are multipurpose, electronic control devices designed to control the temperature of the chilled water supply in response to input from a master controller or a sensor. Resetting the chilled water temperature to the highest temperature compatible with the requirements of the air handler system and building cooling load can result in substantial energy savings: typically a 1% to 2% savings in energy consumption is obtained for every 1°F that the chilled water temperature can be raised.

Automatic chilled water temperature reset controllers save energy by reducing the energy input needed to produce chilled water for cooling. For example, a chiller without temperature reset capability will supply chilled water at, say, 45°F regardless of the cooling load. With a chilled water temperature reset control, however, the exiting chilled

water temperature is increased as the cooling load decreases. Chilled water temperature can be reset based on outdoor air temperature or maximum zone cooling load.

Chiller controllers are available for centrifugal, reciprocating, and absorption chillers. The cooling capacity (and hence, chilled water temperature) of centrifugal machines can be modulated by varying the speed of the compressor motor, by varying the setting of the compressor inlet guide vanes, or by a combination of these two methods. Control of reciprocating chillers is achieved by varying motor speed and unloading (or disconnecting) cylinders. Absorption machines are modulated by throttling the steam flow to the generator. In addition to providing a means to implement temperature reset, many chiller controllers also provide circuitry to provide smooth starting, self-diagnostic capability, and interface to EMCS systems.

Since many types of centrifugal and reciprocating compressors rely on oil in the refrigerant for lubrication and on refrigerant for cooling, a chiller controller must not allow the chiller compressor to run at speeds too low to provide sufficient refrigerant circulation for adequate cooling and lubrication.

The cost effectiveness of chiller temperature reset controllers in any application can be determined by a site-specific analysis of the cooling load profile, chiller characteristics, and energy and equipment costs.

j. Boiler control and hot water temperature reset.

The automatic regulation of steam and hot water boilers is a complex field of study that has been presented in many other reports. Thus, control of the boiler firing process will not be addressed in this document. However, in many cases the output of a boiler can be more carefully managed to reduce the amount of fuel consumed by the boiler.

A hot water temperature reset controller is a device that controls the temperature of the water entering the heating coil in response to an input from a master controller or sensor. By resetting the hot water temperature to the lowest value compatible with the heating load, boiler firing can be minimized. In HVAC systems that use hot water boilers, the boiler setpoint can be remotely reset (Figure 12a). In systems that use steam boilers, it is usually most practical to reset the steam-to-hot-water converter (Figure 12b). Temperature reset of the heating coil is usually not practical in systems that use steam coils as the hot duct heating equipment since variation of the coil temperature can only be achieved through precise control of the boiler pressure, which is difficult to achieve in practice. Also, many HVAC systems that have steam coils as their heat source are served by a control steam plant, and the requirements of other steam users preclude changing steam conditions.

k. Condenser water temperature reset.

The energy consumption of a chiller is reduced by decreasing the temperature difference through which heat is pumped by the refrigeration cycle. This can be accomplished by raising the temperature at which the refrigerant boils, by lowering the temperature at which the refrigerant

condenses, or by employing both actions. Raising the boiling point of the refrigerant is affected by raising the chilled water supply temperature by means of a temperature reset controller. In some circumstances it is also feasible to lower the temperature at which the refrigerant condenses by lowering the condenser water temperature. The condenser temperature reset controller resets the temperature of the condenser from a nominal value, say 110°F, downward when conditions of outdoor dry-bulb temperature and relative humidity can produce lower condensing temperatures.

A conventional cooling tower control system is illustrated in Figure 13a. The condenser water supply thermostat modulates the diverting valve to maintain the water at a constant temperature. The cooling tower fan is started when the diverting valve is partially open through the action of a snap-action pneumatic relay. The fan is off when the valve is in the full bypass position.

Temperature reset of the condenser supply water can be implemented in several ways. The method illustrated in Figure 13b uses a two-input pneumatic controller to generate a control pressure proportional to the wet-bulb temperature or enthalpy of the outdoor air. This control signal is used to reset the setpoint of the condenser water supply thermostat. The condenser water supply thermostat controls the diverting valve and fan in the manner described above.

3. Comparison of economic performance of selected strategies.

Estimates of the economic performance of the various HVAC control strategies discussed in section 2 can be derived from estimates of the energy savings associated with each strategy, the costs of installing the control devices, and the cost of maintenance. The estimated energy savings can best be obtained through computer simulation of the HVAC system performance. It is difficult, however, to define a "typical" building and HVAC system such that the results of an energy consumption analysis can be directly applied to other buildings. Climate, building construction, HVAC system design, and space conditioning requirements combine to make the HVAC energy consumption pattern of each building unique. For purposes of illustration, however, a building can be described and the alternative energy conservation strategies evaluated.

The conservation strategies were evaluated through use of the BLAST computer simulation (Ref 6). The BLAST program calculates the heating and cooling loads for each zone of a building, then simulates the response of the HVAC system to the loads.

The building selected as an example is a three-story, light construction office building located in Washington, D.C. The building measures 100 feet on each side. The exterior wall is constructed of metal curtain, insulation, and gypsum board. Internal zone partitions are constructed of metal framing faced with gypsum board on both sides. Windows account for 30% of the wall area on each wall and are double glazed. The building is divided into 10 conditioned zones. Office occupancy schedules used in the simulations were taken from the BLAST library of schedules. Nominal office occupancy is 240 people. Lighting and other equipment contribute 3 W/ft² of the zone heating loads, but 30% of the heat gain due to lighting was directed to the return air

ducts. The lighting and equipment schedules were also from the BLAST library. Air infiltration was assumed to be 1/2 air change per hour for the exterior zones only. Since the building is pressurized by the fan system when the air handling system is operating, infiltration was modeled so as to occur only when the building was not occupied.

The building and central plant models were kept constant for all simulations. Energy conservation strategies were simulated through changes in temperature control and fan control schedules.

Two HVAC systems commonly used in small office buildings were simulated: a constant volume terminal reheat system and a variable air volume terminal reheat system.

The energy conservation strategies that were modeled are presented in Table 1.

The results of the BLAST computer program simulations are summarized in Table 2 and Figures 14 and 15. The BLAST simulations were performed by the National Bureau of Standards, Center for Building Technology, and are documented in detail in Reference 7. The results clearly illustrate the energy-saving potential of simple strategies, such as night and weekend setback. Note that the incremental energy savings tends to decrease as more strategies are added, since the baseline energy consumption decreases. Also, the most complex strategy need not be the most effective due to the interactions between the building, climate, and HVAC system. In neither of the two HVAC systems studied was the most complicated strategy (Strategy E) clearly better than the alternatives.

A sample calculation of the economic value of the energy savings is presented below using the following criteria:

Economic life	15 years
Annual operation and maintenance cost	10% of equipment cost
Electrically powered chiller operating cost (Ref 8)	5.4 ¢/kW-hr
Oil-fired, hot water boiler operating cost (Ref 8)	6.96 \$/MBtu
Discount rate	10%
Differential inflation rate for electricity .	6%
Differential inflation rate for fuel oil . .	8%
Chiller coefficient of performance	2

For the constant volume terminal reheat system, the value of the energy saved by implementing night-weekend setback (i.e., changing from Strategy A to Strategy B described in Tables 1 and 2) can be calculated as follows. The value of heating energy savings is:

$$(760-210) \times 10^3 \frac{\text{kW-hr}}{\text{yr}} \times 3,413 \frac{\text{Btu}}{\text{kW-hr}} \times \frac{\text{MBtu}}{10^6 \text{ Btu}} \times 6.96 \frac{\$}{\text{MBtu}}$$

$$\times \text{CPW}(15,10,8)$$

where CPW (15,10,8) is the compound present worth factor for a 15-year life, 10% discount rate, and 8% differential cost escalation rate. From standard tables, the value of CPW (15,10,8) = 13.112; therefore, the value of the heating energy savings is \$171,300.

In a similar manner, the value of the savings in cooling energy is equal to

$$(1,180 - 480) \times 10^3 \frac{\text{kW-hr}}{\text{yr}} \times \frac{1 \text{ kW-hr}_E}{2 \text{ kW-hr}_T} \times 0.054 \frac{\$}{\text{kW-hr}_E}$$

$$\times 11.508 = \$217,500$$

Therefore, the total anticipated life cycle savings due to implementation of night and weekend setback is approximately \$460,000.

Adding an enthalpy economizer system to night and weekend setback (Strategy C) increases life cycle savings by a comparatively small \$59,000.

A survey of the equipment available for implementing the setback strategy indicates that the necessary timeclock devices cost less than \$2,000. Consequently, night and weekend setback appears to be a very cost effective strategy to employ. If night and weekend setback saves 50% to 60% of baseline energy consumption, scheduled stop-start and optimum stop-start should save substantially more energy.

A simplified calculation procedure for estimating the energy savings of different control strategies is presented in Reference 9, and a thorough survey of the equipment available to implement the strategies is contained in Reference 10.

Recent Developments in HVAC Control System Hardware

For many years the most widely used type of control for large HVAC systems has been the pneumatic control. Although electric control systems have also been available, they have not been as widely used. Pneumatic systems are in widespread use because they offer several inherent advantages. First, pneumatic devices can be easily modulated since air pressure can be easily varied over a wide range. This permits wide rangeability for precise control. Second, most pneumatic devices are simple, reliable, and low in cost. Third, because pneumatic systems are operated by compressed air, pneumatic controls are safe to use where a fire or explosion hazard exists and can provide trouble-free operation in humid environments. Finally, the large variety of pneumatic sensors, controllers, switches, relays, and actuators available means that almost any control strategy can be implemented by assembling off-the-shelf components.

Pneumatic systems are not without weaknesses however. The nozzles and restrictors found in many pneumatic devices are susceptible to plugging from dirt; oil and water in the air lines can cause blockage and collect in bellows and diaphragms yielding incorrect response characteristics, and air lines can leak through joints and worn or broken components. Also, pneumatic sensors tend to have long response times (greater than 1 minute) and limited accuracy ($\pm 1\frac{1}{2}^{\circ}\text{F}$), and some advanced controller functions are difficult to implement with the pneumatic logic elements currently available.

In recent years, control system components that enable the HVAC designer to combine the advantages of pneumatic actuation with advantages of solid state electronic sensing and control circuitry have become available. Analog-electronic devices perform the sensing and control logic functions. The output of the controller is used to actuate pneumatic valves and damper motors by means of an electronic-to-pneumatic transducer, which varies the branch air line pressure in response to an electronic input. Manufacturers of electropneumatic components claim increased control accuracy and responsiveness, high reliability, and low cost as advantages over all-pneumatic systems.

The addition of modern, electric actuating devices permits design of an all-electronic HVAC control system. All-electronic control systems are widely used in Europe and, in recent years, have become available in the United States. A major advantage of electronic actuators over pneumatic actuators is increased rangeability and response time, which can result in increased control system bandwidth (i.e., the range of input conditions over which the system will respond satisfactorily).

Digital-electronic systems are the latest development in HVAC controls. Digital control systems are based on microcomputer devices and act in response to a set of instructions (a program or "software") supplied to the system. Although digital computers are used as supervisory control of completely pneumatic control systems (e.g., EMCS systems), only the direct digital control (DDC) of HVAC system valves, dampers, motors, and other components will be addressed in this report. Digital control systems have all the advantages of analog-electronic systems plus the added benefits of being able to support, through software, control strategies that would be difficult, and perhaps impossible, to implement by other means. Also, through changes in the software the control strategy can be altered to meet changing needs. In addition to providing control functions, a microcomputer-based system can be configured to interface with an EMCS system, check sensor calibration, and provide alarms for hazardous conditions, faulty equipment, or maintenance reminders.

Microcomputer-based controllers also make it possible to implement feed forward control and adaptive control algorithms at the equipment controller level. Feed forward control responds to anticipated disturbances and is often beneficially employed in control of systems having slow response times (such as most HVAC systems). Adaptive control algorithms continually modify the characteristics of the controller (e.g., proportional gain) to provide optimum control under varying conditions of system performance. Thus, the characteristics of the control system change to account for changes in the system being controlled, such as changes in HVAC equipment performance caused by deteriorating heat exchanger or pump performance or a faulty actuator.

The increasing demands of energy conservation together with the rapid decline in the cost of microcomputer components will result in wide use of digital control systems. Digital controls can be readily retrofit to many existing HVAC systems through the replacement of pneumatic sensors and controllers with solid state circuitry, while retaining the existing pneumatic actuators. Several methods exist for converting the digital output of the controller into a pneumatic signal (Ref 11). Of course, digital controllers can also operate electrically actuated valves and dampers.

Control system type (whether a system is all-pneumatic, analog-electronic, digital-electronic, or a combination of types) undoubtedly has an influence on the performance and energy efficiency of an HVAC system, but the nature and magnitude of the effects are difficult to determine at the present time. Research continues on this subject.

SUMMARY AND CONCLUSIONS

There are many strategies available for reducing the energy consumption of HVAC systems in single buildings. Some strategies, such as nighttime setback and scheduled start-stop, are inexpensive to implement and should be installed in most buildings. Estimates of the effectiveness of more complex strategies (such as economizer/enthalpy cycles or demand limiting) can only result from more detailed analyses involving building envelope, climate, energy demand patterns, and HVAC system design. Computer codes are useful in analyzing the cost effectiveness of more complex conservation strategies. Applicable strategies should be implemented in the order of decreasing cost effectiveness. Care must be used when implementing several strategies to avoid conflicts in requirements on equipment.

A list of suppliers of energy management devices for small HVAC systems is presented in the Appendix (from Ref 12). The list may not be complete, and inclusion on the list does not imply any product endorsement.

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Table 1. Energy Conservation Strategies

Strategy	Description
A	Baseline case: Single temperature setpoint, constant cold duct temperature, fixed percent outdoor air.
B	Same as A except night and weekend setback is employed (A + setback).
C	Same as B except enthalpy economizer cycle is added (A + setback + enthalpy economizer).
D	Same as B except cold duct temperature is reset by zone having largest cooling load (A + setback + zone reset).
E	Same as D except enthalpy economizer cycle is added (A + setback + zone reset + enthalpy economizer).

Table 2. Annual Energy Consumption for Several Control Strategies

[Energy consumption is in units of 10^3 kW-hr.]

Strategy	Constant Volume Terminal Reheat					Variable Volume Terminal Reheat				
	Heat	Cool	Fan	Total	% Change ^a	Heat	Cool	Fan	Total	% Change
A	760	1,180	270	2,210	0	200	395	85	680	0
B	210	480	150	840	-62	50	235	50	335	-51
C	230	270	150	650	-71	55	170	50	275	-60
D	270	430	115	815	-63	40	215	55	310	-54
E	275	320	115	710	-68	40	175	55	270	-60

^a% change from baseline consumption.

Appendix

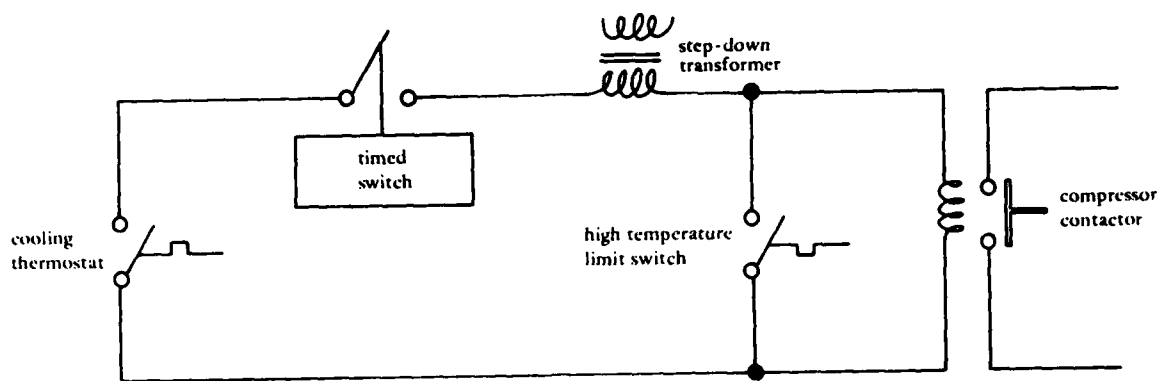
SUPPLIERS OF ENERGY MANAGEMENT DEVICES

Below is a list of suppliers of energy management devices for single buildings and other small energy users. The list was compiled by Energy Users News (Ref 12). Inclusion of a company on this list does not constitute a recommendation of the company's products by the Department of the Navy.

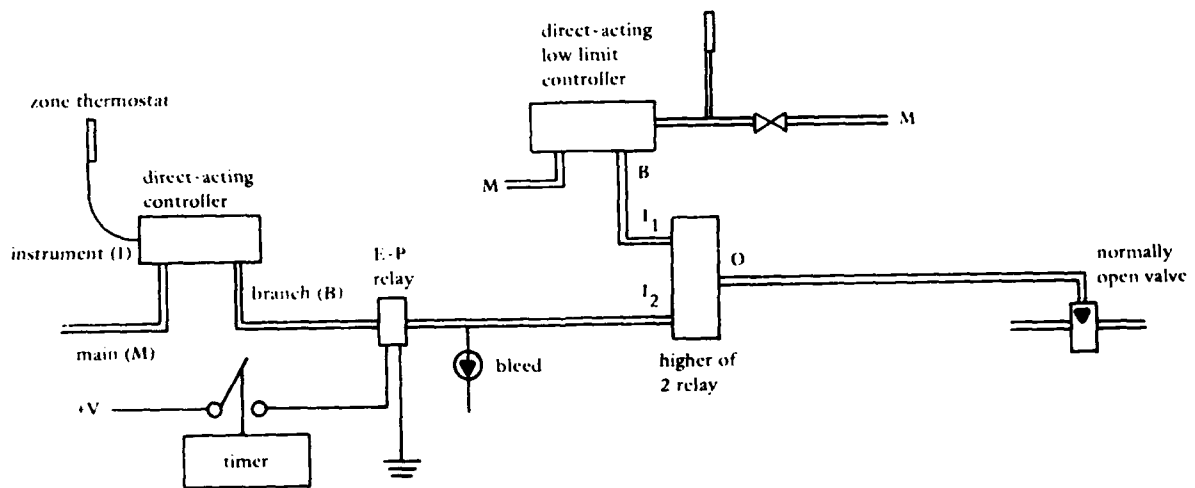
Advanced Electronic Controls Inc., Andrew Jackson Highway, Huntsville, Ala. 35801
Aegis Energy Systems Inc., 607 Airport Blvd., Doylestown, Pa. 18901
AMF Paragon Electric Co. Inc., 606 Parkway Blvd., P.O. Box 28, Two Rivers, Wis. 54241
American Multiplex Systems Inc., 1148 East Elm Ave., Fullerton, Calif. 92633
Andover Controls Corp., P.O. Box 34, Shawsheen Village Station, Andover, Mass. 01810
Atlantic Energy Technology, 73 Tremont St., Boston, Mass. 02108
Aviation Electronics, 2050-J Carroll Ave., Chamblee, Ga. 30341
Barber-Colman Co., 1300 Rock St., Rockford, Ill. 61101
Bud-Industries, 11370 Amalgam Way, Suite C, Rancho Cordova, Calif. 95670
Castellano Supply Co., 1652 S. Fulton Circle, Norcross, Ga. 30093
Cesco, 6505 218th St. S.W. 15, Mountlake Terrace, Wash. 98043
Com-Trol Inc., 285 River St., P.O. Box 724, Bucyrus, Ohio 44820
Conservation Concepts, 975 Arthur Godfrey Road, Suite 301, Miami Beach, Fla. 33140
CSL Industries, One Century Plaza, 2029 Century Park East, Los Angeles, Calif. 90067
Cutler Hammer Electrical/Electronic Control, Div. of Eaton Corp., 2620 Lance Drive, Kettering, Ohio 45409
Dencor Inc., 2750 S. Shoshone, Englewood, Colo. 80110
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Dupont Energy Management Corp., 625 South Good-Latimer, Dallas, Tex. 75226
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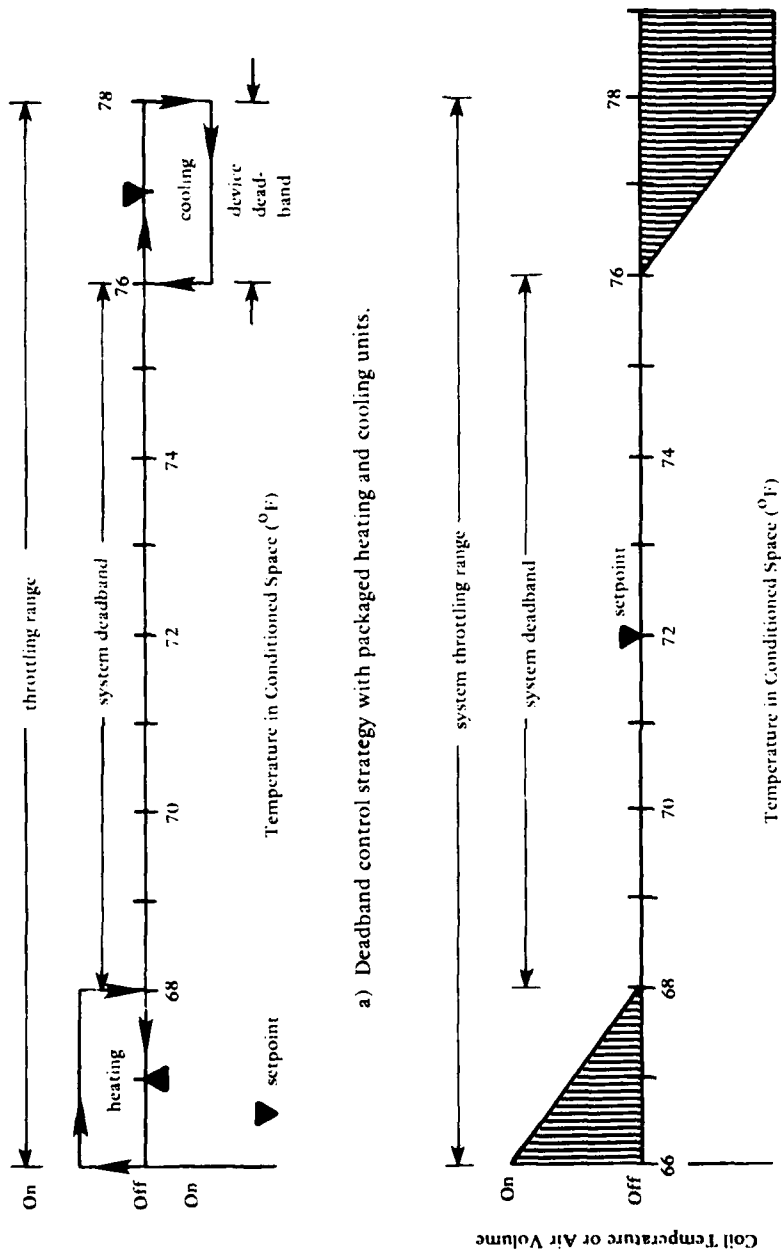


a) Electric control of cooling equipment.



b) Pneumatic control of heating equipment.

Figure 1. Applications of scheduled start-stop.



b) Deadband control strategy with mixed-air systems.

Figure 2. Deadband control strategy.

S.P. = setpoint
T.R. = throttling range
S.A. = supply air
R/C = receiver/controller

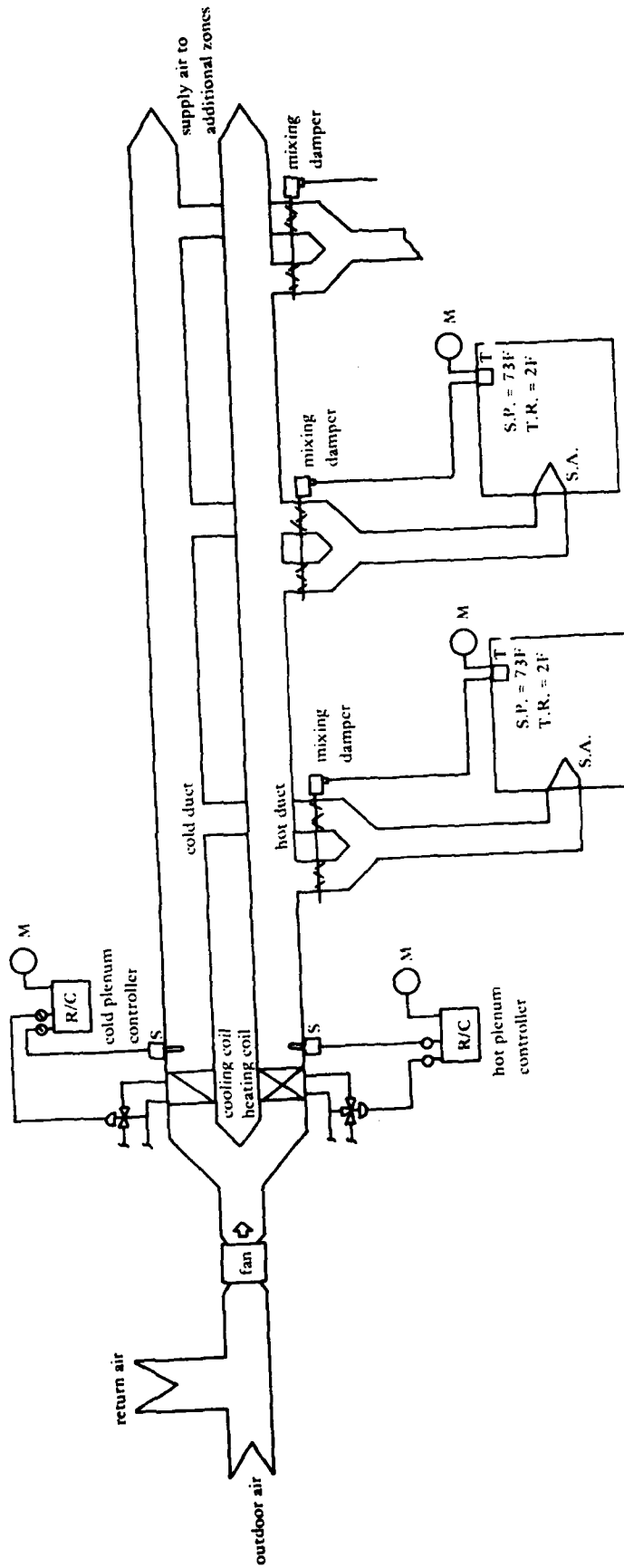


Figure 3. Conventional mixed-air HVAC system.

S.P. = setpoint
 T.R. = throttling range
 S.A. = supply air
 R/C = receiver/controller
 R/R = ratio relay
 M = control air supply
 S = sensor
 HI = highest of input pressures
 LO = lowest of input pressures

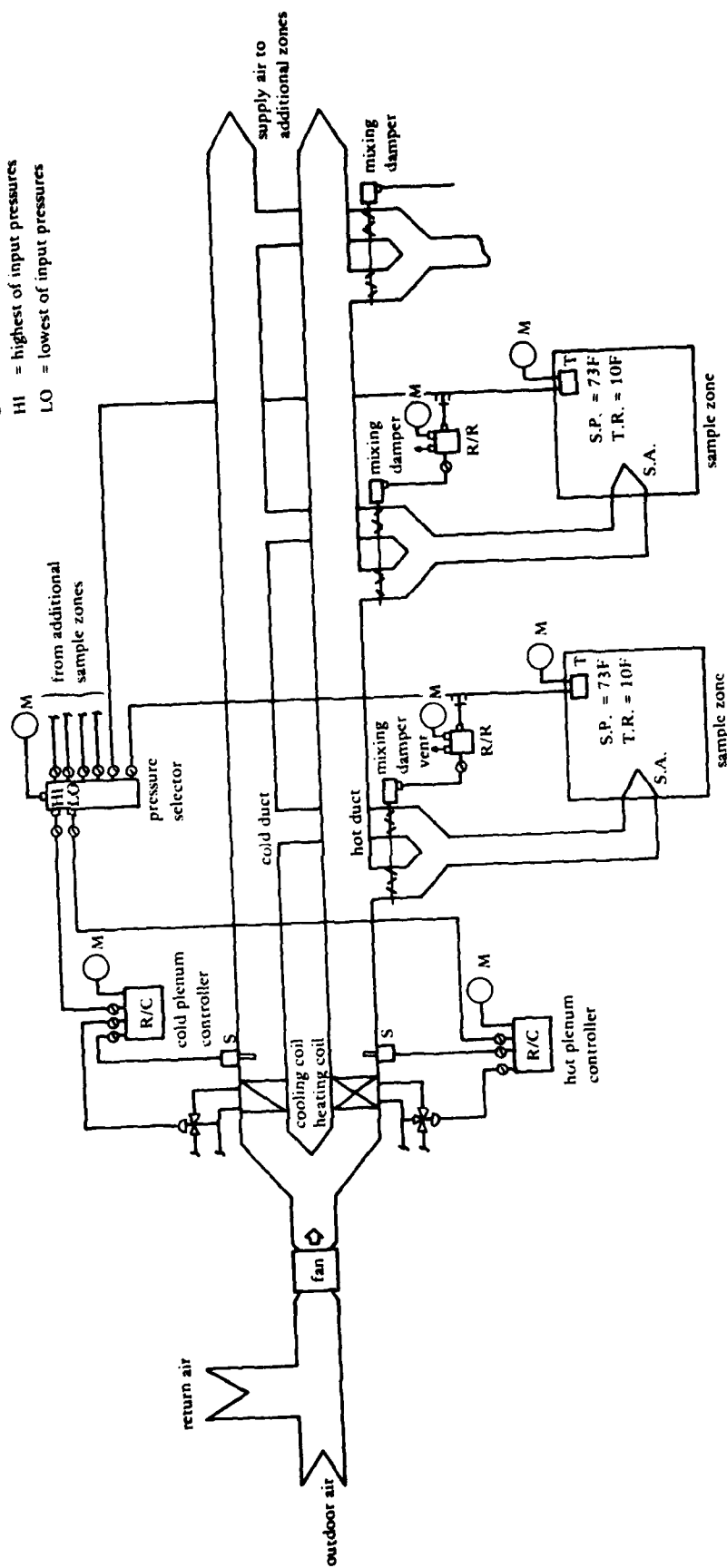


Figure 4. Deadband control of mixed-air HVAC system.

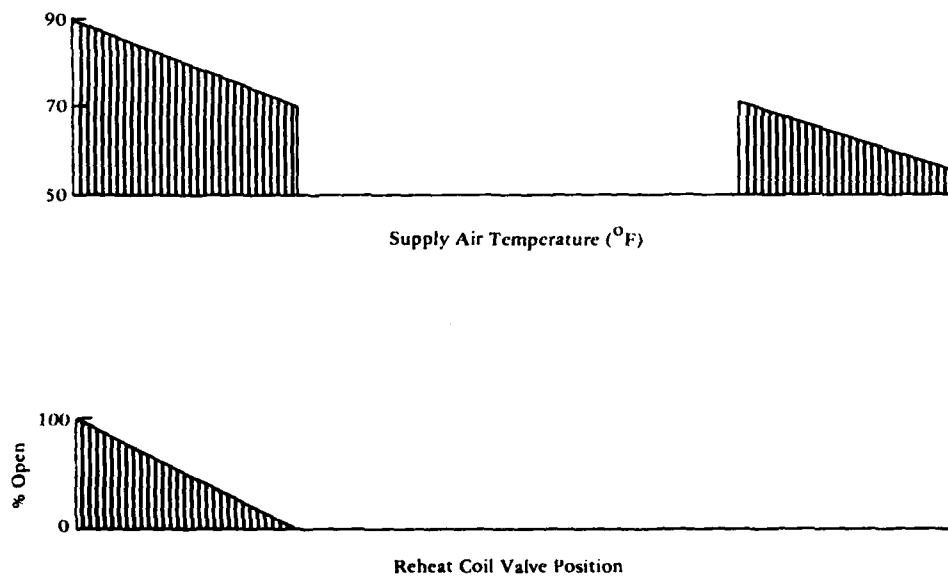
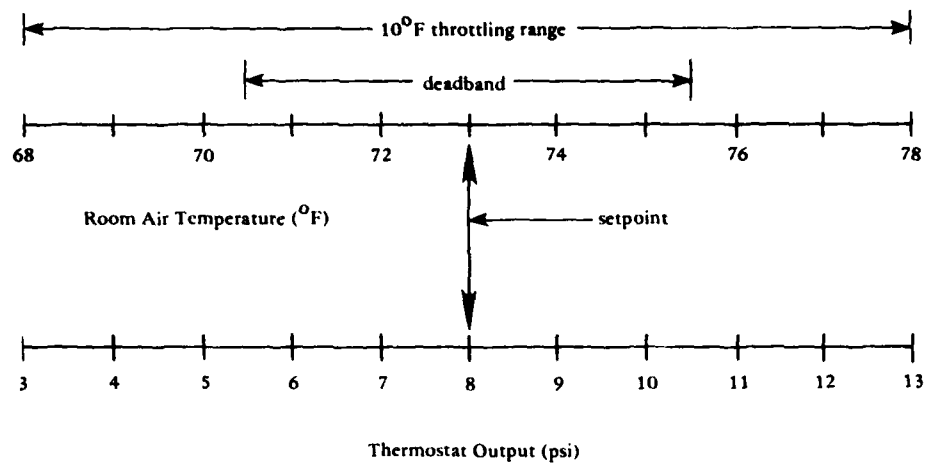
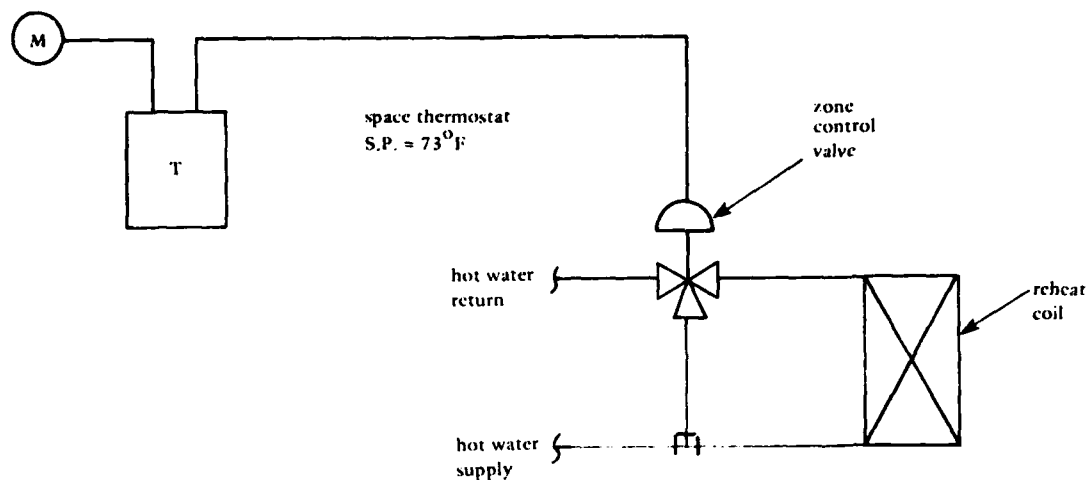
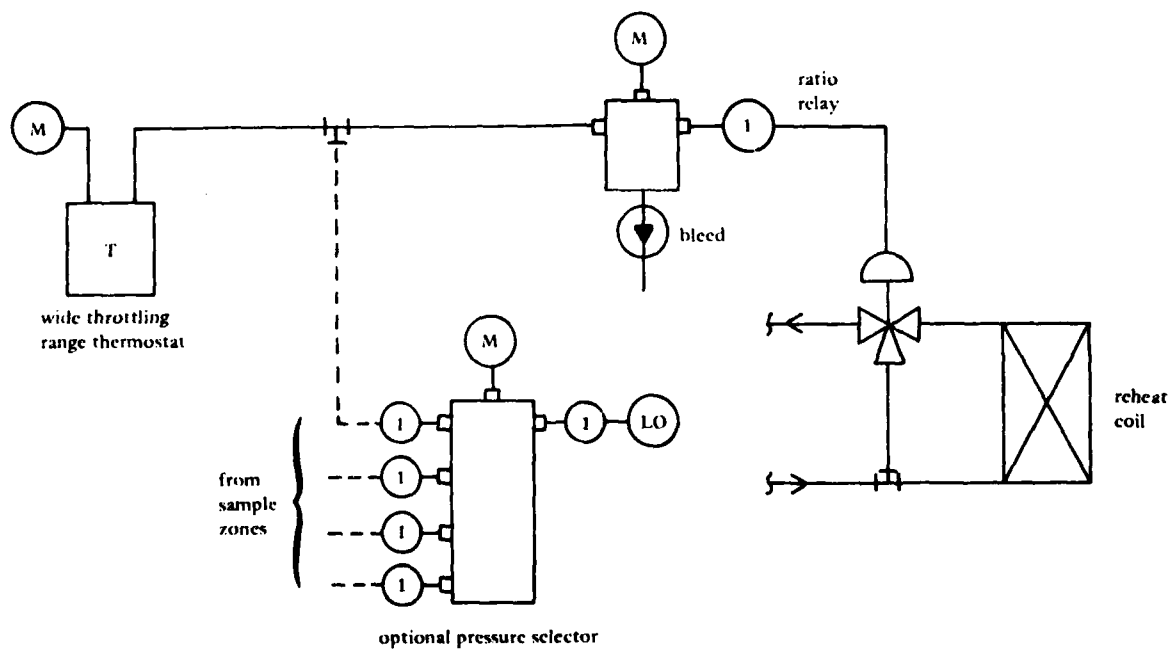


Figure 5. Deadband control strategy for reheat HVAC system.



a) Existing zone controls.



b) New zone controls.

Figure 6. Zone controls for deadband control of reheat HVAC systems.

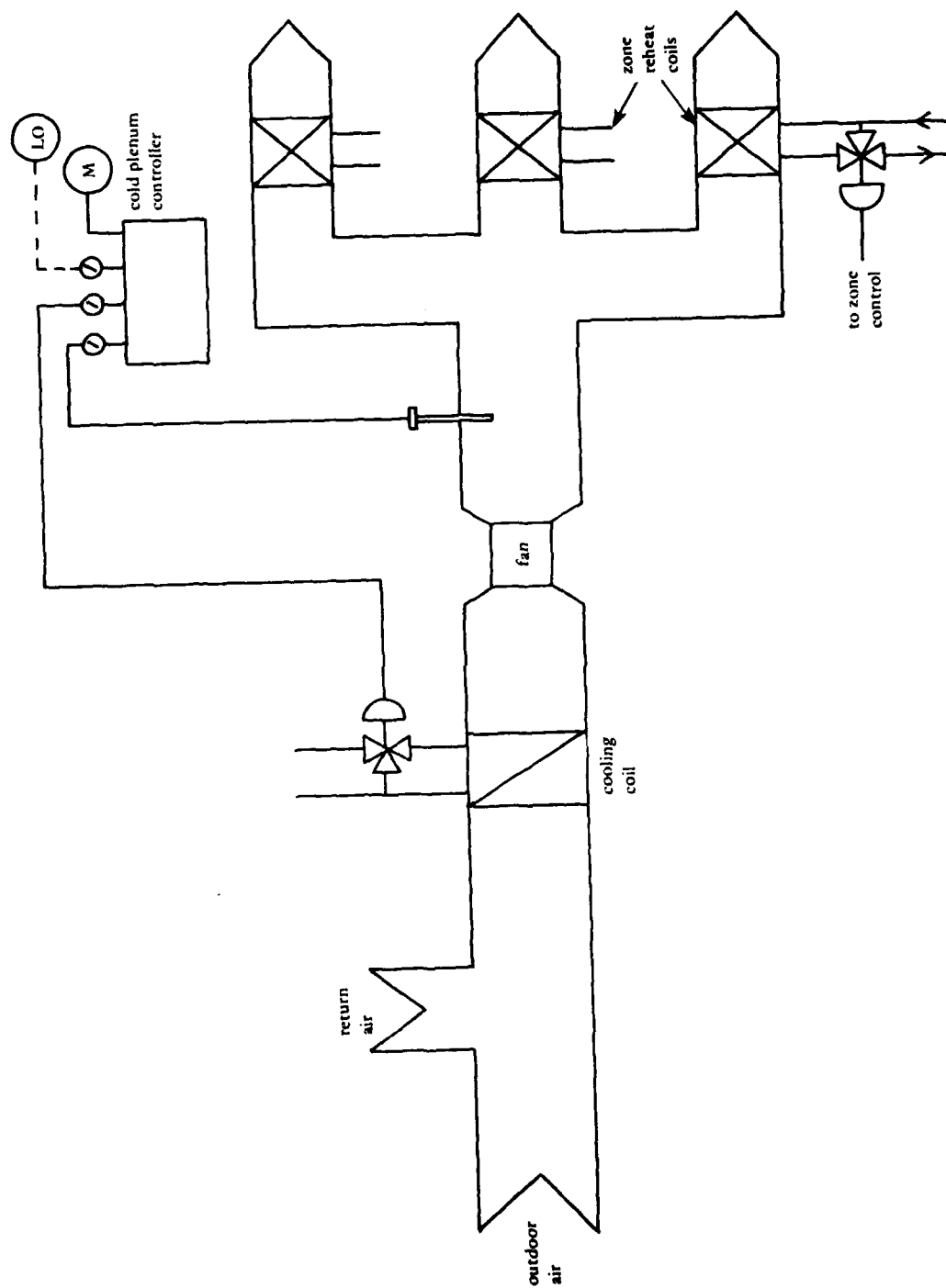


Figure 7. Apparatus controls for deadband control of reheat HVAC systems.

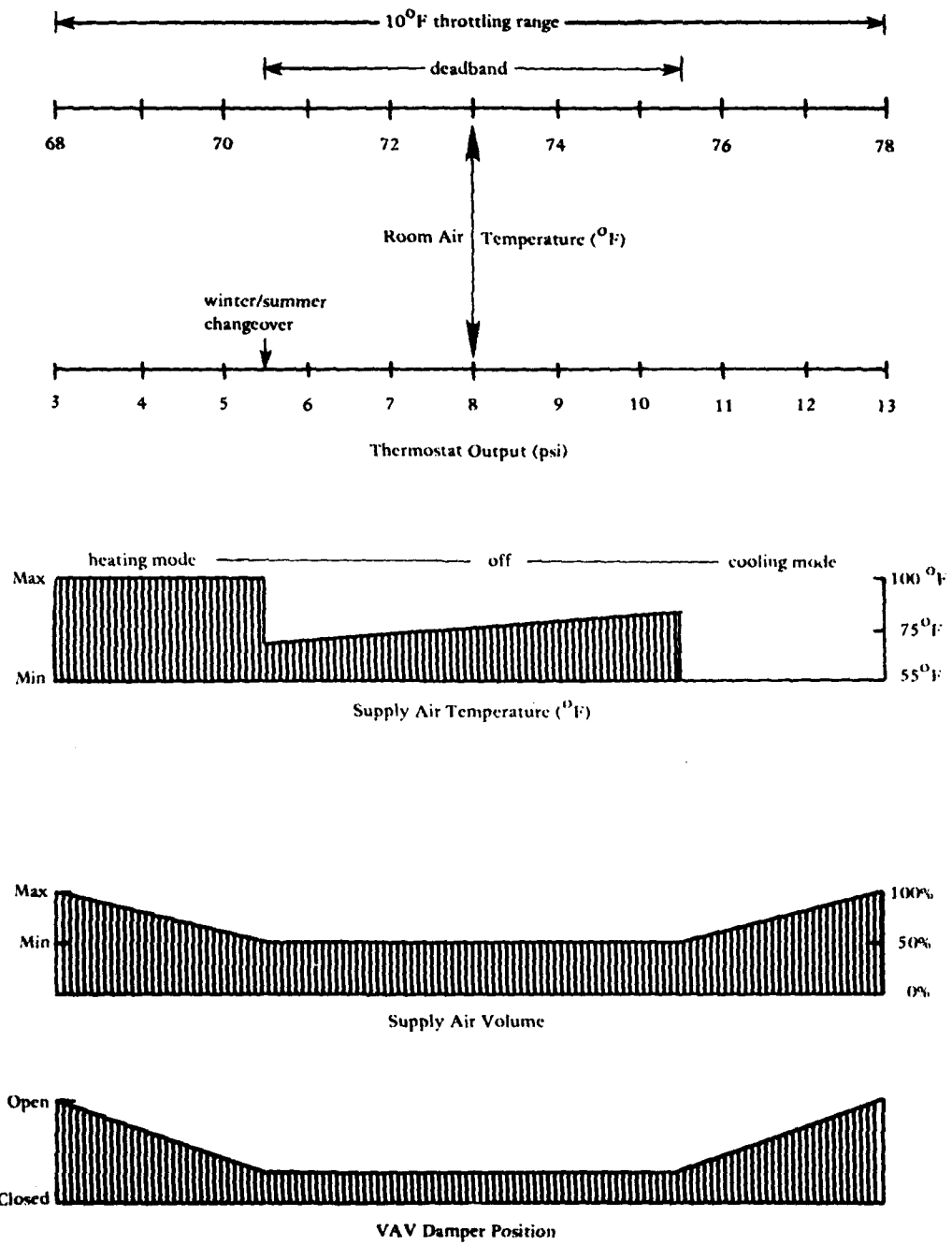


Figure 8. Deadband control strategy for variable air volume systems.

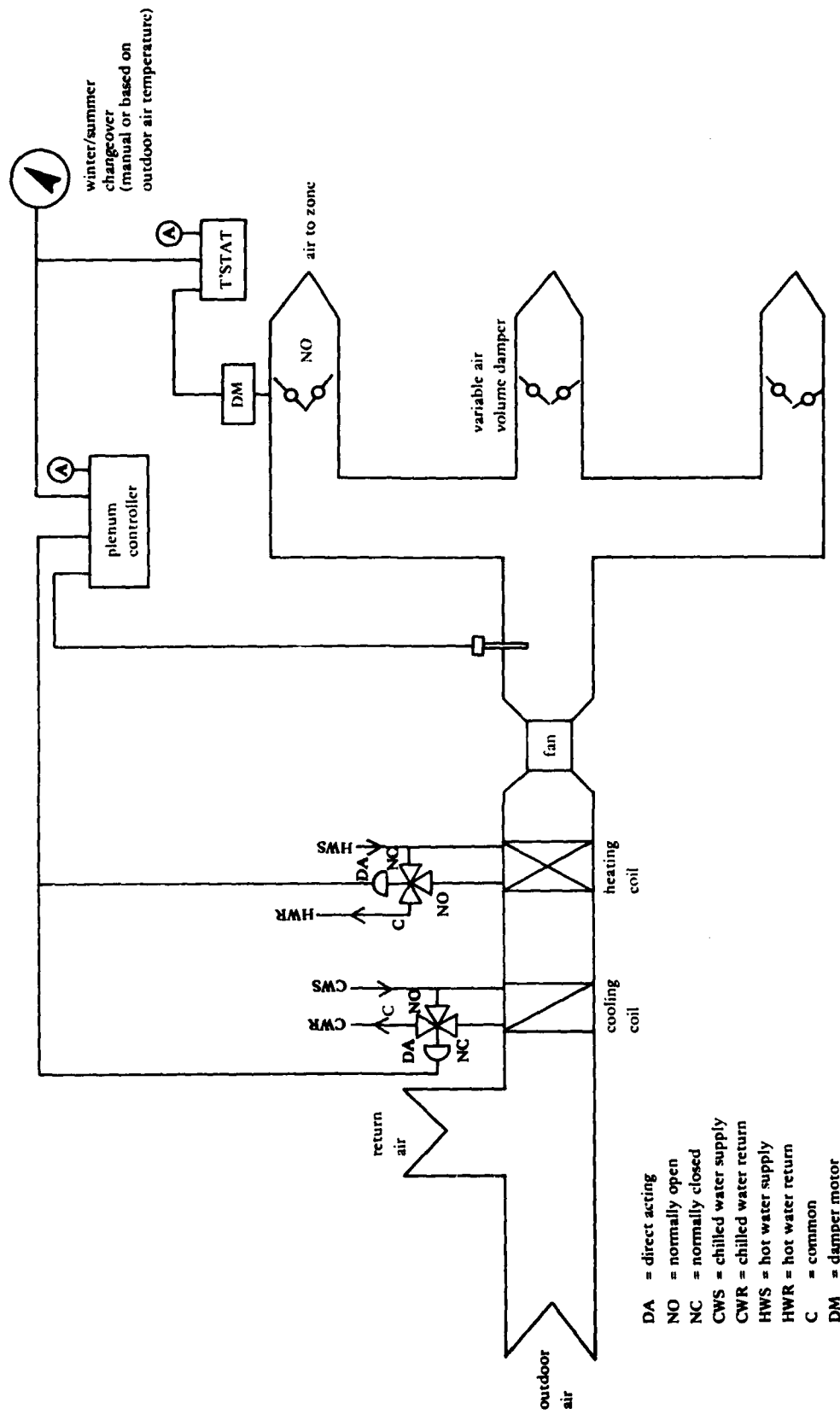


Figure 9. Apparatus controls for deadband control of variable air volume systems.

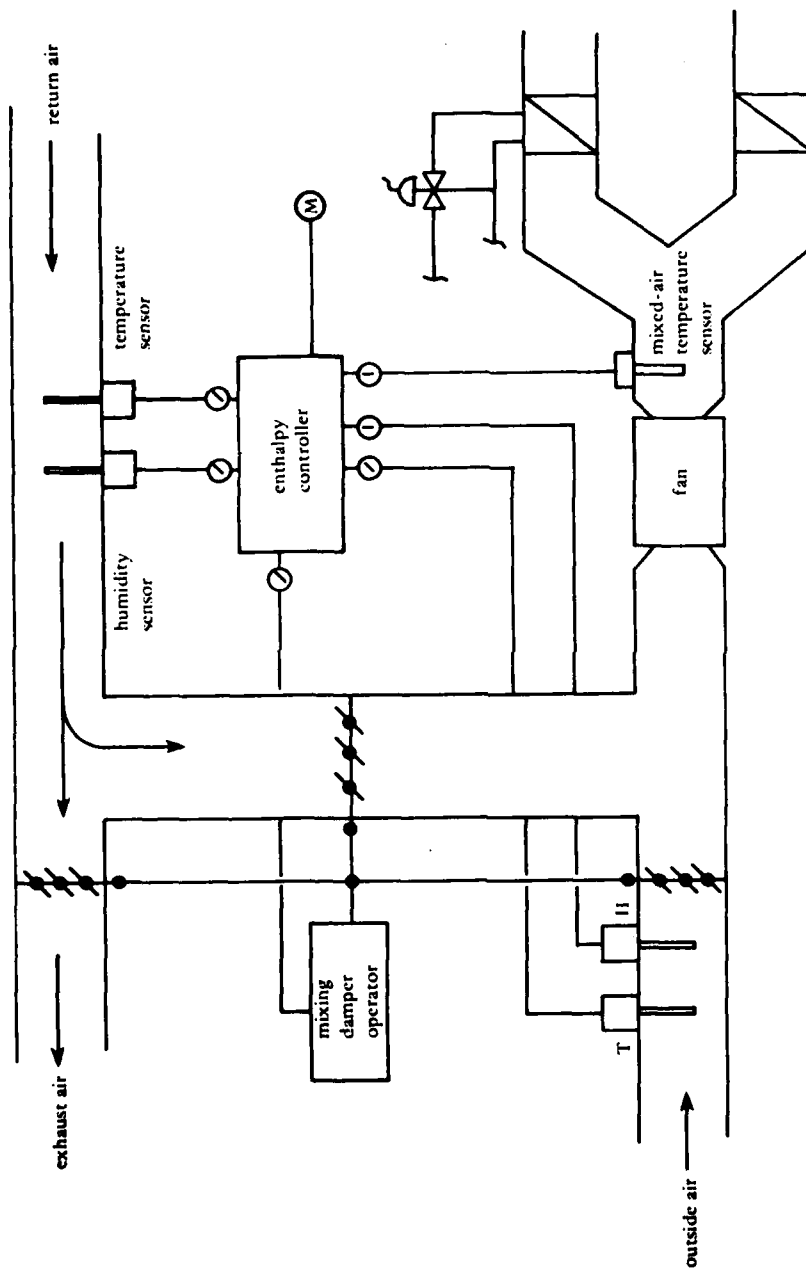


Figure 10. Installation of enthalpy controller.

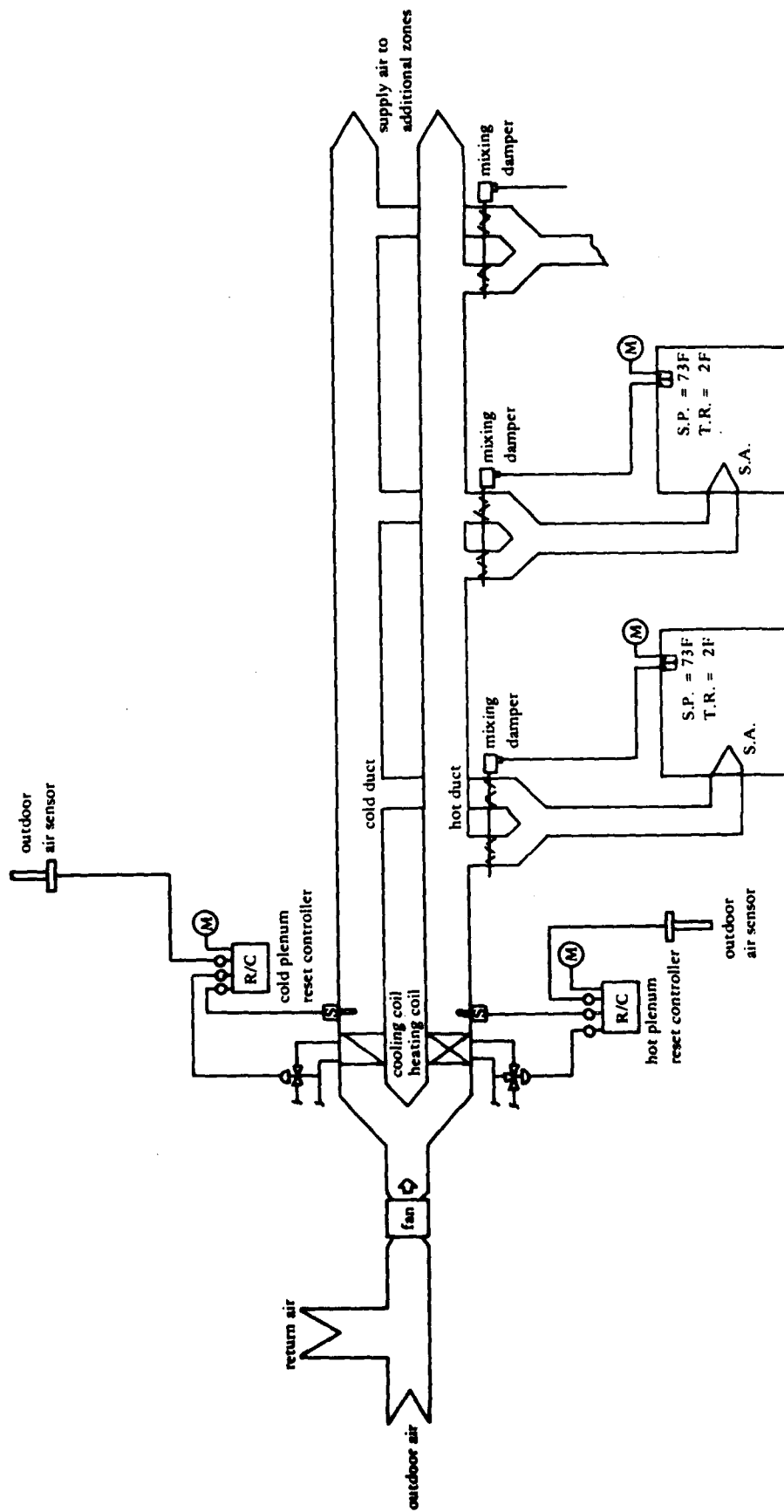
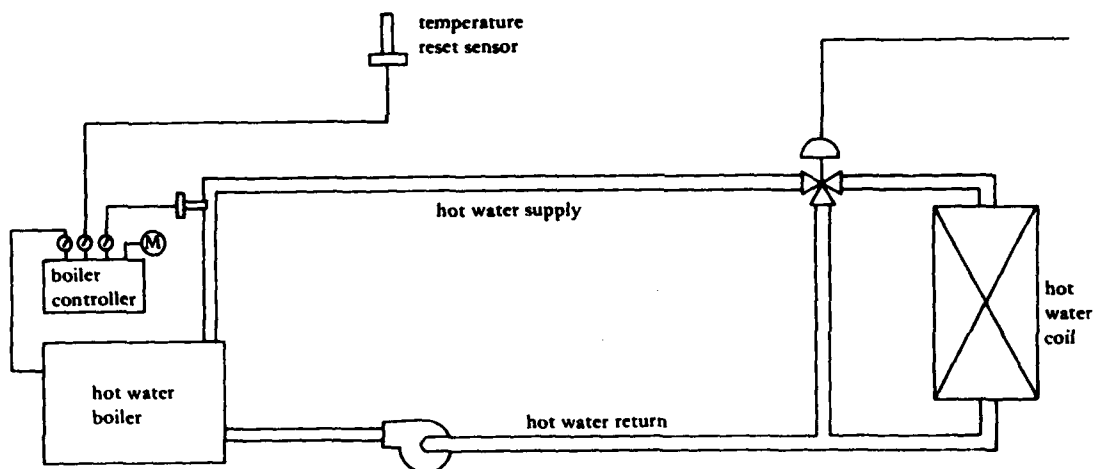
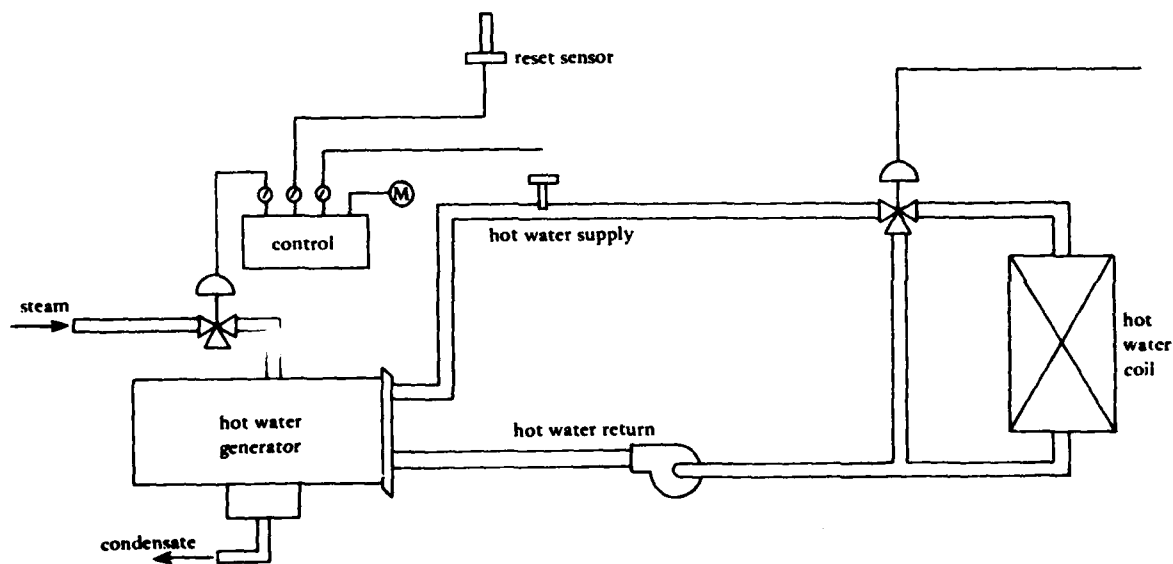


Figure 11. Mixed-air HVAC system with hot duct/cold duct temperature reset.

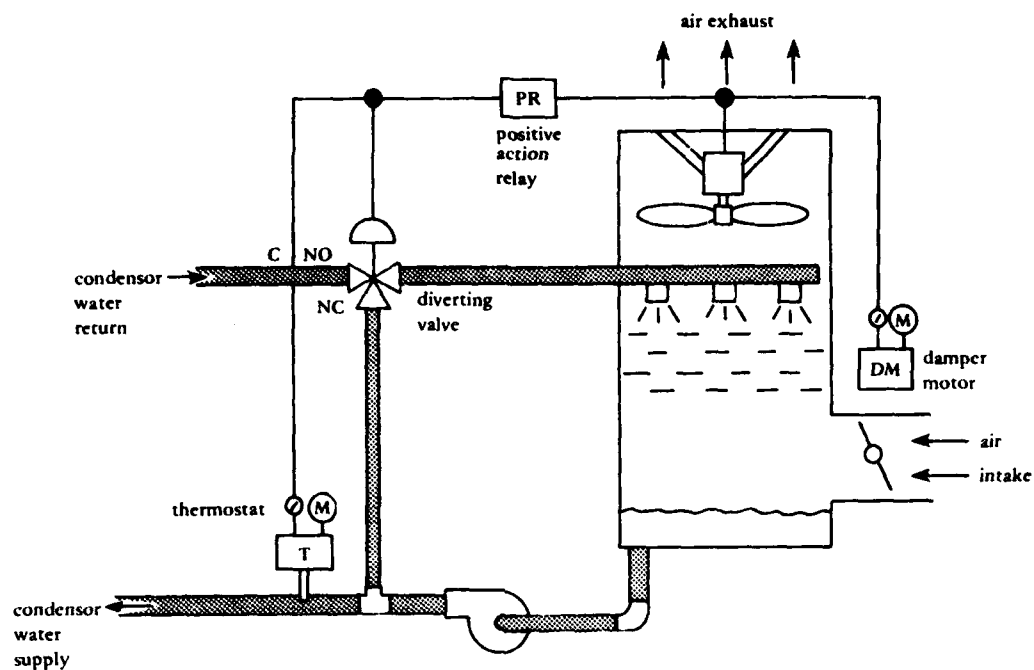


a) Hot water boiler with temperature reset.

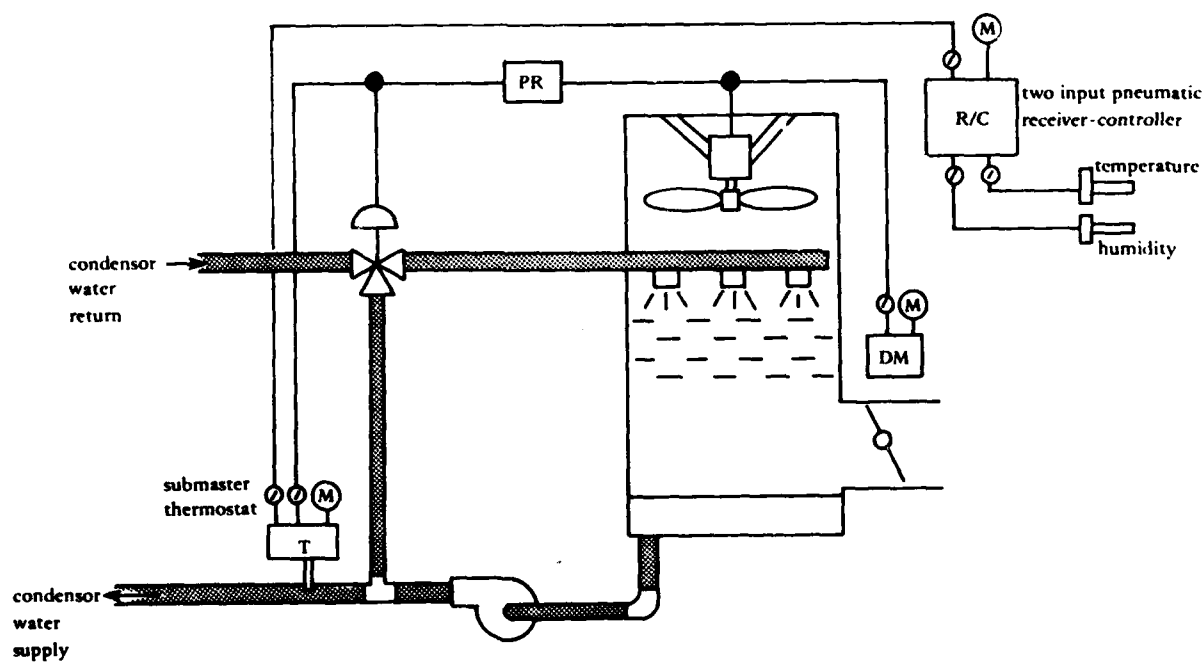


b) Hot water generator with temperature reset.

Figure 12. Hot water reset controllers.



a) Conventional cooling tower control system.



b) Cooling tower control with temperature reset.

Figure 13. Condenser water temperature control systems.

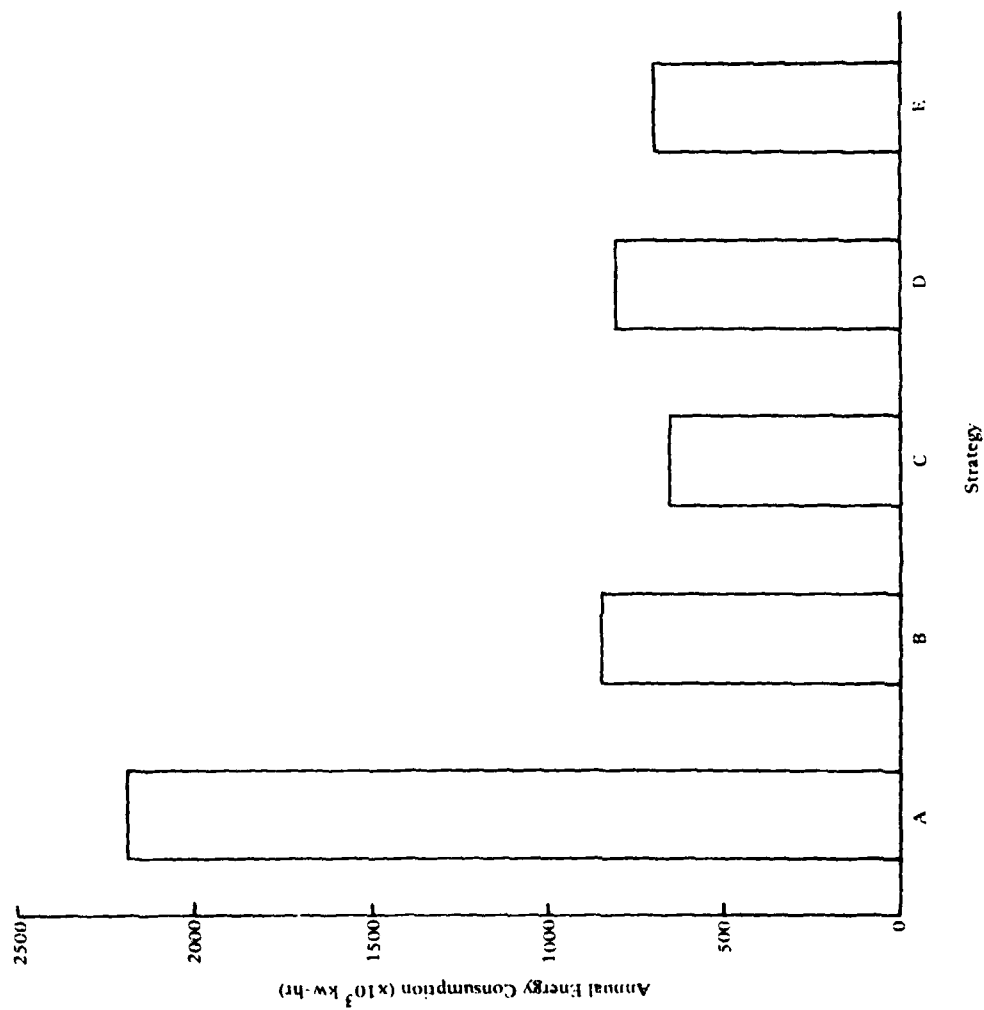


Figure 14. Comparison of energy consumption for constant volume reheat system.

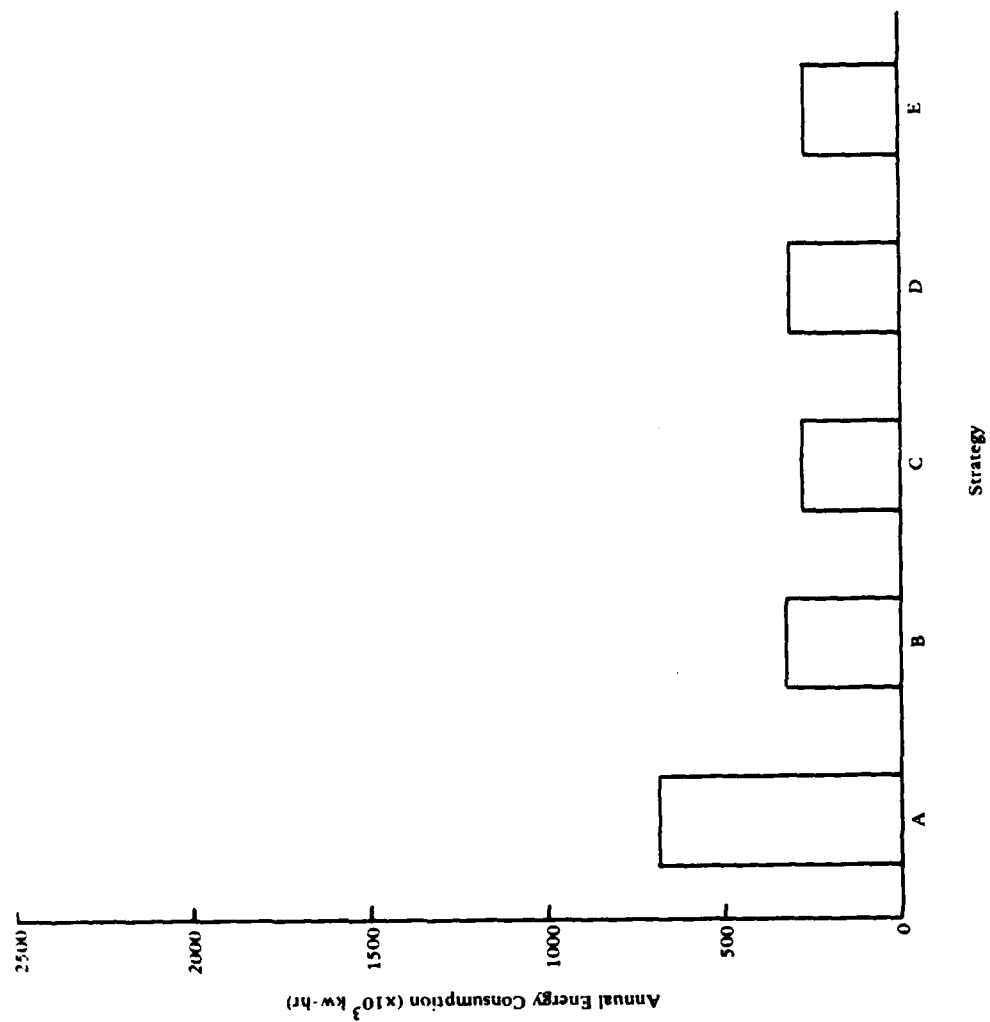


Figure 15. Comparison of energy consumption for variable volume reheat system.

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