To the Graduate Council:

I am submitting herewith a thesis written by Winston Bernard Rawlston entitled "Computer Simulated Studies for Energy-Efficient Operation of HVAC Equipment." I recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Chemical Engineering.

Charles F. Moore

Charles F. Moore, Major Professor

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COMPUTER SIMULATED STUDIES FOR
ENERGY-EFFICIENT OPERATION
OF HVAC EQUIPMENT

A Thesis
Presented for the
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Degree
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Winston Bernard Rawlston

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ABSTRACT

The International Business Machine Corporation (IBM) is currently studying the possible advantages of retrofitting heating, ventilating, and air conditions (HVAC) control systems at their Kingston, New York facilities. Steady-state computer simulations of a single-duct air handler, dual-duct air handler and perimeter radiation heating system are developed to compare existing control strategies with proposed improvements. These simulations apply mass and energy balance principles, along with appropriate semi-empirical and empirical models to describe the system fans, heating coils, cooling coils, heat exchangers and load calculations.

Studies of the single-duct air handler show that it is advantageous to apply year-round continuous operating control strategies and therefore eliminate the need for operator assistance in control and set point change summer/winter transition periods. An investigation of dehumidification operation shows that reheat set points should be placed as high as possible for optimum energy usage.

The dual-duct simulation demonstrates the large energy savings that can be accomplished by applying zone temperature feedback rather than feedforward control to determine duct temperature set points. Further savings are incurred by using outside air for free cooling whenever possible. Sensitivity studies are made to show the effect of varying the air handler's mixing box temperature set points and the lcoal zone thermostatic temperature set points. A comparison is also made to show that the zone-temperature feedback method is better than a literature suggested louvre position feedback method to determine duct temperature set points.

Simulation of the perimeter system demonstrates the advantages of a feedback method over a feedforward approach based on outside air temperature to determine the feed water temperature to the perimeter face pipe. The feedback method uses just enough energy to give the desired wall air temperature whenever physically possible. The feedforward strategy either uses too little energy and gives a cold wall air temperature or too much energy and gives a hot wall air temperature.

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INTRODUCTION

In the past decade, the world experienced the beginning of a severe energy shortage. Depletion of oil fields and the resulting increases in oil prices, coupled with past inadequate development of alternative energy sources forced leading nations to immediately concentrate on finding a solution to this shortage or face harsh social and economic damage. One readily accessible source of energy is conservation. Conservation can be accomplished by new and efficient equipment designs and by applying new techniques to older inefficient equipment. An area that is now being recognized as a savings source in industry is the heating, ventilating, and air conditioning (HVAC) equipment that is used to provide comfort.

Much of the HVAC equipment used today was designed with the emphasis on cutting capital costs (1). Energy was very inexpensive, so it was much cheaper to waste energy than to design and install an energy-efficient system. Industry is now having to reverse this attitude, not only on new systems, but in old equipment and control strategies.

There are three separate industrial HVAC systems to be studied here. The single-duct air handler, which does all heating and cooling in one duct, provides comfort-conditioned air to one zone (group of offices or one large room). The dual-duct air handler, which has separate cooling and heating ducts, serves several zones that may or may not have different energy requirements. Hot air and cold air from these ducts are mixed at each zone to provide the desired conditions. The third system studied is a hot-water radiator used to provide perimeter heating during the winter months. The three system configurations and the associated original control strategies studied are based on existing HVAC equipment at an IBM facility in Kingston, New York. Computer simulated studies of each of these old design systems will show areas for potential improvements in energy usage reduction.

In keeping with the current push toward energy conservation, it is the goal of the HVAC engineer to continue providing the same comfortable, healthy air as in the past while attempting to minimize energy usage (2). Through proper simulation it is possible to achieve a better understanding of the process and equipment involved and the techniques necessary to meet this goal.

CHAPTER I

STATE OF THE ART

An accurate model will provide a quick method for testing various equipment and control configurations for energy savings. There are several techniques mentioned in the literature for both the overall system simulation and the simulation of the component parts of the system.

Basic Energy Management Principles

Spethman (3) gives four basic principles of HVAC equipment control for energy conservation that the facility engineer should be striving toward:

1. Run equipment only when needed. The equipment should only be used during occupied periods; outside air ventilation should not be used during winter nights to maintain minimum space temperatures; morning warmup should start as late as possible; and, equipment startup should be staggered to eliminate peak demands.
2. Sequence heating and cooling. The overlap of heating and cooling is wasted energy. Simultaneous heating and cooling is especially a problem with separate HVAC units, but can be eliminated by connecting all units to one signal.
3. Provide only the heating and cooling needed. Supply air temperatures for separate heating and cooling ducts should be kept as close as possible. These supply temperatures should be reset on the basis of zone need, not outside air temperature.

4. Provide heating and cooling from most efficient source. Use outside air to provide free cooling whenever possible.

While the HVAC engineer is striving for a more efficient operation, he must always remember that he cannot minimize energy at the expense of the comfort and health of the people being served (2). Each individual may have a different opinion as to what defines comfortable air, but ASHRAE defines comfort standards that provide a range of acceptable temperatures and humidities. The current comfort standards are shown in the Figure 1-1 psychrometric chart (4). The engineer has most control over the factors of dry bulb temperature and humidity, but he must keep in mind that this comfort region on the psychrometric chart is also a function of air movement, clothing and activity (5).

HVAC Systems Considered

Although there are many systems of interest to the HVAC engineer, this work will deal only with the single-duct air handler, the dual-duct air handler, and the radiation perimeter heating system. The single-duct and dual-duct are the two common types of all-air systems. ASHRAE (6) defines an all-air system as "a system providing complete sensible and latent cooling capacity in the cold air provided by the system" requiring no additional cooling at the zone. ASHRAE further states a list of advantages and disadvantages for all-air systems. Advantages include such things as centralized location of major equipment, use of outside air for free cooling, design freedom for optimum air distribution, suitability to abnormal exhaust make-up

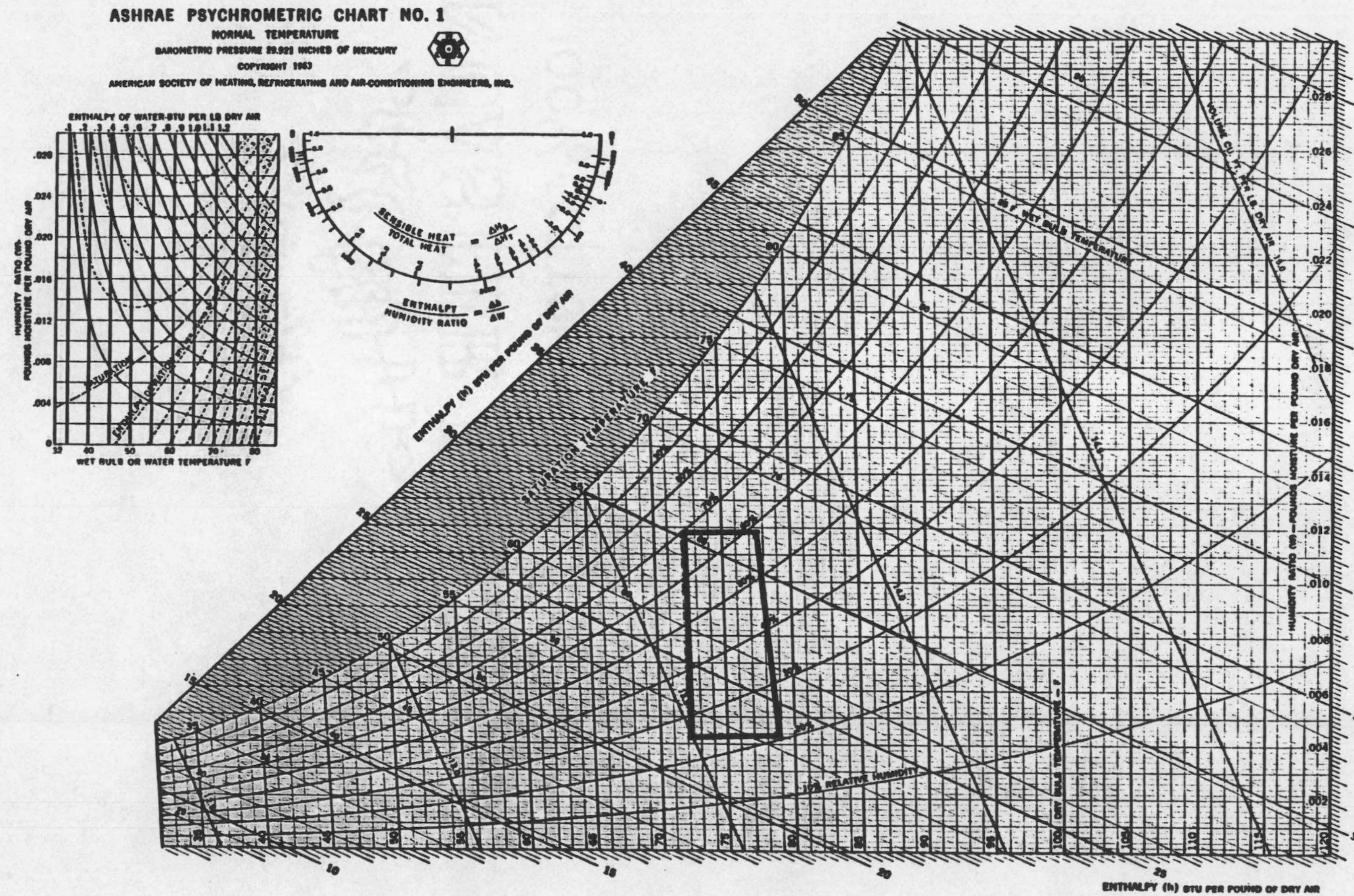


Figure 1-1. ASHRAE defined comfort standards.

applications, and adaptability to automatic season changeover. Disadvantages cited were additional duct clearance requirements and difficulty in air balancing. They also state that perimeter air-heating systems require longer fan operating hours during unoccupied periods than perimeter radiation-heating systems.

The single-duct air handler provides comfortable air to satisfy a single zone. All heating or cooling is done in the air handler, with the air leaving the handler passing through a single-duct to the zone. ASHRAE (6) calls the single-duct air handler the simplest form of the all-air system. It can maintain close control over temperature and humidity because it can be responsive to the needs of only one space.

The dual-duct air handler is designed to handle a multiplicity of zones or different thermostatic requirements (6). The air handler supplies air to both a hot duct and a cold duct. The hot air and cold air are mixed at the zone to satisfy the particular zone's requirements. Haines (7) reports a typical way of operating dual-duct systems in the past. There was no control on the chilled water valve, but the valve could be shut off in winter. The hot duct temperature was reset on the basis of outside air temperature. Haines points to the lack of feedback from the zones as a very inefficient operation of the air handler. Shinskey (2) describes an improved dual-duct operation whereby the zone requiring the most heating (or least cooling) draws entirely from the hot duct, and the zone requiring the most cooling (or least heating) draws entirely from the cold duct. The duct supply temperatures are then set on the basis of position feedback from the louvres on the mixing box dampers at each zone.

The third system being studied here is a radiation perimeter heating system. As suggested by its name, this system consists of a hot water or steam radiator which runs along the perimeter wall of the building. The radiator may be of the type to use steam or hot water. ASHRAE (8) states that these radiators are "placed at the points of greatest heat loss of the space, so as to counteract these losses." By providing the insulating layer of air, the perimeter system can many times eliminate the need for steam in interior air handlers.

Once the type of system has been defined, the HVAC engineer must decide what type of simulation he is going to perform. Stoecker (9) defines the system simulation as

> . . . predicting the operating quantities within a system (pressures, temperatures, energy- and fluid-flow rates) at the condition where all energy and material balances, all equations of state of working substances, and all performance characteristics of individual components are satisfied.

System Simulation

In predicting the system operation, the engineer must choose whether to use a dynamic or steady state simulation. Although it is recognized that the air handler never achieves steady state, Stoecker (9) points out that assuming a steady state situation is appropriate for most energy calculations, because the dynamics of the building are so much slower than those of the air-handling system. The steady-state simulation is used to determine the system operation for one hour, realizing that the operation at that hour will be different than the next. The use of dynamics is necessary only for predicting the thermal loads on the building.

The next question to be answered when approaching the simultation is whether to arrive at a solution to the operating conditions by solving simultaneous equations or by sequential calculations. Stoecker (9) writes that while sequential calculations are much easier to follow, the system simulation approach generally requires the solution of simultaneous equations. These simultaneous equations may assume a solution at one point and iterate to the correct solution (9, 10). Stoecker (11) states that the presence of recycle loops in the air handling systems makes the iterative, or successive substitution, approach seem natural.

One problem found with successive substitution is the possibility of divergence. While chemical plants have had no severe divergence problems, the ASHRAE Energy Task Group (11) encountered divergence problems in preliminary air-handling simulation work. Their suggestion was to use the Newton-Raphson method to achieve convergence. Their computer program provided for using the Newton-Raphson method requires the user only to provide equations for evaluation of function derivatives.

Once the equation solving approach to the simulation is chosen, the engineer must be certain that he has the individual component descriptions necessary to characterize his system. Crall et al (12) suggests a simple two step calculation procedure that should be followed to determine the energy consumption. First, calculate the load as a function of exterior conditions, and second calculate the performance and energy consumption of a given control system to dissipate this load. This procedure points out the need for an adequate load determination, an accurate description of air properties to determine the system consumption, and a sound understanding of the system equipment and processes. Crall et al had a test facility where they could accurately describe an HVAC system and concluded that proper data can produce an accurate simulation, but they pointed out that without this data significant errors can result.

Load Representation

Crall et al's first requirement for the simulation was an adequate load representation. There are several sources for the load on an HVAC system (10, 13, 14, 15, 16, 17, 18). They include heat transfer from the outside; solar radiation; body heat from occupants; heat from lights, power and machines; and infiltration and exfiltration through small cracks and holes. Down (13) suggests using basic heat transfer calculations to arrive at thermal transmittance coefficients. Transfer coefficients are very helpful if they can be experimentally measured, but ASHRAE (15) maintains that detailed energy balances are too time consuming. Instead, ASHRAE provides computer subroutine algorithms that give the step by step calculation procedures to assist the engineer in accounting for the essential elements of the heating and cooling loads. They point out that the thermal storage effect must be counted when determining the building load. Three suggested methods to guarantee including this storage effect are the Carrier method, which includes thermal storage factors; the time averaging technique, which uses an arithmetic average of previous hours' operation to determine storage loads; and the weighting factor, which includes a transfer function concept to determine the loads from past operation. Mitalis (16) supports the weighting factor method with experimental work.

Spielvogel (18) states that the simulations must account for the fact that "only a small fraction of annual energy use occurs during the extremes of weather." Figure 1-2 shows that for most of the year in Washington D.C. the outside air temperature is between 30°F and 70°F, and this is where the maximum amount of energy is consumed. This also means that HVAC equipment operates at part load most of the year. To account for this, Spielvogel proposes using the "Bin" method, which goes through the range of outside air temperatures for the geographic region being studied and assigns a weighting factor number according to the number of annual hours that that temperature occurs.

Psychrometric Properties

The next requirement for an effective simulation is an accurate description of the psychrometric properties of air. The psychrometric properties are necessary for all mass and energy balance calculations for an air system, such as defining the air properties after the mixing of two air streams or with heat and water loads on single air stream. Although there are many charts describing the air properties, these must be converted to algorithms to be useful in a simulation. Goff and Gratch (19) present accurate and extensive properties for moist air based on statistical mechanics, dealing with pressures from one to three atmospheres and temperatures from 150 to 400 degrees Kelvin. Goff (29) presents the algorithms used for the calculation of saturated air properties, but "the procedure is not easily adaptable to computer calculation." (21) Ksuda (22) gives a set of computer oriented algorithms for calculation of saturated air properties based on the methods of Goff and Gratch.

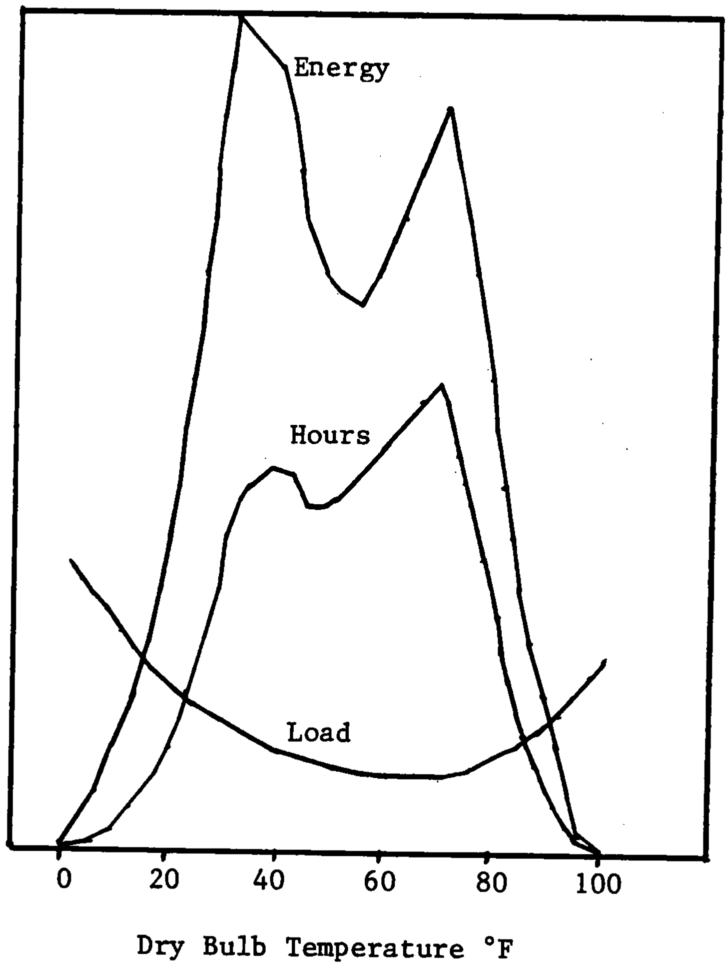


Figure 1-2. Annual energy consumption for a house in Washington D. C. as a function of load.

ASHRAE (15) presents a set of computer algorithms for approximate calculation of psychrometric properties that were developed by the National Bureau of Standards. These algorithms are a simplified version of the more rigorous Goff and Gratch method. They use curve-fit expressions to describe the wet bulb temperature and dewpoint, and replace the statistical mechanics method with the ideal gas law and Dalton's law of partial pressures. Marcev (21) tests this method against actual psychrometric data over a range of dry bulb temperature from 20°F to 80°F and wet bulb temperatures from 16°F to 80°F and found negligible error.

Spofford (23) also gives a computer program for the calculation of psychrometric properties. This algorithm again uses the ideal gas law, Dalton's law, and other refinements from Goff and Gratch. However, Spofford's routine is only accurate between 32°F and 100°F. The method is less exact above 100°F and does not apply below 32°F.

Component Models

Another element required for a good simulation is a sound understanding of the system equipment. One of the most difficult pieces of HVAC equipment to describe is the cooling coil. Kimura (10) points out that because of the simultaneous heat and mass transfer, the cooling coil requires special consideration. The cooling coils described by Kimura are simply heat exhangers where air is cooled as it passes through chilled water coils. Water vapor condenses on the coil fin surface as the surface temperature drops below the air dew point. This condensation prevents using only an energy balance to determine cooling coil operation.

Stoecker (9) offers a sixth order polynomial expression to describe the cooling coil. The user is required only to give the entering water and air thermal conditions and flow rate, data on coil surface area, and the number of rows of coils. The method is simple to use, but its use is restricted because the polynomial expression is discontinuous at low chilled water flow rates.

Marcev (21) presents a computer program subroutines for describing the cooling coil based on the work of Elmahdy and Mitalis (24). The coil surface may be completely wet or dry, or the surface may be just partially wet. The method used applicable heat transfer equations along with log mean temperature differences for the all-dry case and log mean enthalpy differences for the all-wet case. Trial and error is required to determine the boundary for the part wet/part dry case.

The other pieces of equipment in an HVAC system are much easier to describe. Heating coils, fans, humidifiers, and mixing boxes are all described by simple mass and energy balance equations (8, 25). ASHRAE's (9) alternative method is the curve fitting of existing data to polynomial expressions. However, as previously stated, these equations are simple to use but tend to have stability problems (26, 27).

A final need in providing an accurate simulation is understanding the processes involved in the system operation. Free cooling from outside air, cooling by chilled water coils, heating by steam or hot water coils, and humdifiying by spray humidifiers are processes that require only meeting the system load. Dehumidification is a special process. The conventional approach (18, 4) is to first cool the air. As described for the cooling coils this will cause condensation of the water. Although the air will contain less water, by the nature of the psychrometric properties it will have a higher relative humdiity. To decrease the relative humidity, heat is then added to the air by the heating coils. This procedure is shown in the Figure 1-3 psychrometric chart.

Oglesby (28) and Keane (29) simulate air handlers in attempts to determine methods of decreasing energy consumption at an IBM plant in Kingston, New York. Oglesby collects data from a dual-duct air handler and uses the computer routine PATERN (30) to determine the parameters for dynamic models of the system components. He then develops a dynamic computer simulation to test both the stability of the system under different operating conditions, and to study possible improvements. Oglesby notes that there was a significant energy loss because the hot-duct air temperature was set on the basis of outside air temperature. He states that this generally causes overheating in the hot duct, which consequently presents an additional cooling load. The solution is to reset the duct temperature on the basis of zone temperature. Other studies by Oglesby determine that energy savings would be realized by lowering the minimum outside requirement for ventilation, raising the summer zone temperature set point, and lowering the winter zone temperature setpoint. He also concludes that blocking off the steam flow could give significant savings but points out that this prevents the air handler from adequately meeting its winter load.

Keane uses ASHRAE component models to describe a single-duct air handler. He calculates the total energy consumption for the air handler during each of the four seasons. He then applies various control strategies to determine system improvement. A test of one degree set point change gives an average yearly energy savings of almost 10%.

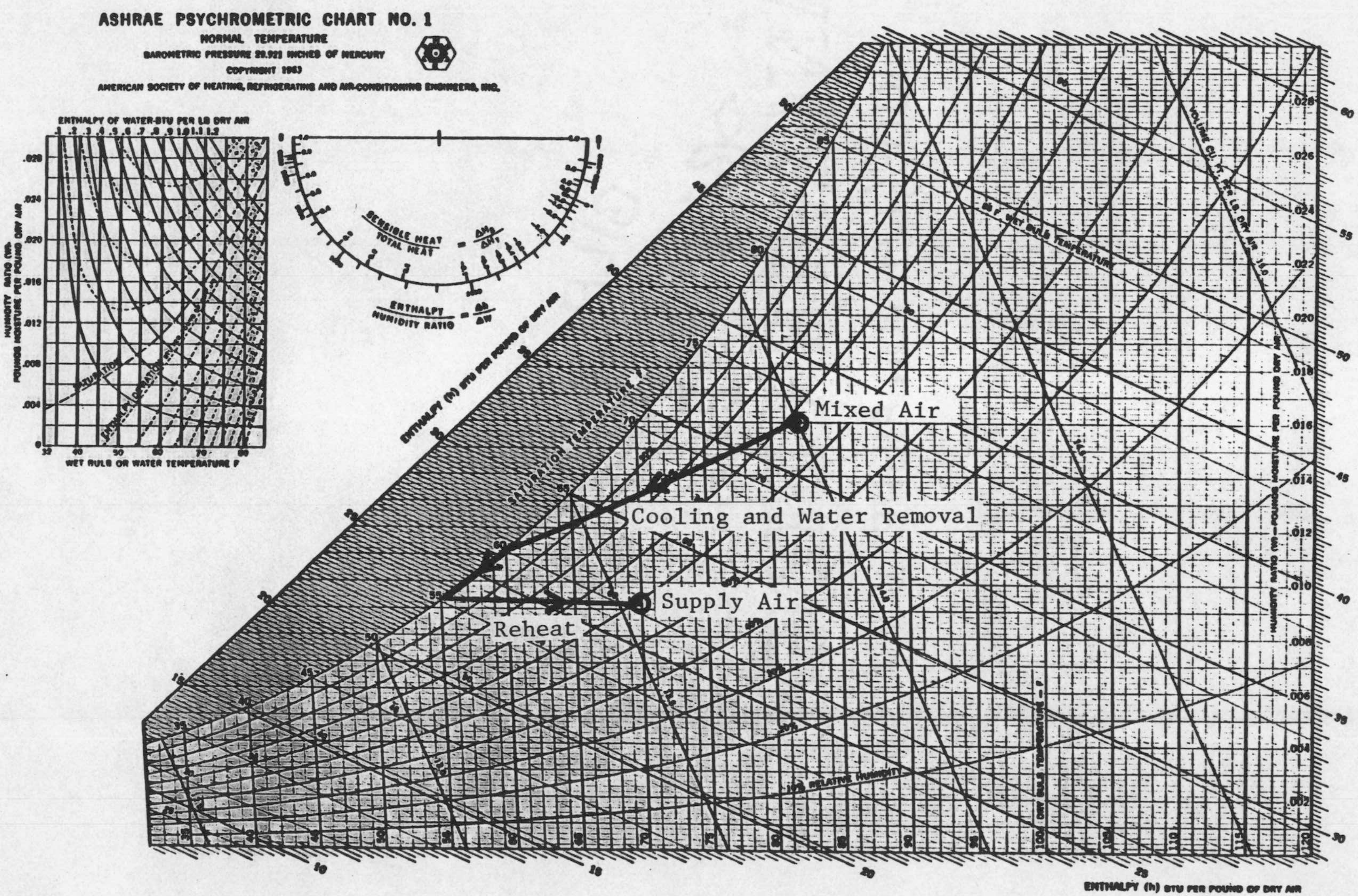


Figure 1-3. Typical dehumidification procedure.

An average energy reduction of 29% is calculated by using outside air for free cooling when needed if its enthalpy is lower than the return air. This improves on the old strategy which set outside air damper positions on the basis of outside air temperature. Keane's largest energy savings of 51% was found with a strategy that combined the outside air free cooling with gap control. The gap control applied here would shut off both steam and chilled water when the zone temperature was between 68°F and 75°F.

As these studies show, a simulation can be a helpful tool in determining proper energy saving techniques, but most authors are quick to point out that the simulation must be well planned if it is to be useful. A proper understanding of the entire HVAC process involved is required before simulation can begin. The engineer must then choose a simulation that describes that process. This includes choosing whether to use a steady state or dynamic simulation, and whether to use simultaneous or sequential calculations to converge on the system operating conditions. He next writes routines that will follow this technique in describing each of the simulation components, such as load calculations, psychrometrics, and system equipment. If the processes have been properly described, the engineer must then rely on accurate system data to provide credible results.

CHAPTER II

PROCESS DESCRIPTION

There are three different HVAC systems to be simulated for determination of possible energy-saving improvements. The single-duct air handler, dual duct air handler, and perimeter radiation heating system are described below in terms of operating methods implemented at the IBM facility that will be used as a base for comparison with proposed operating approaches.

Single-Duct Air Handler

The single-duct air handler is a large piece of HVAC equipment used for providing air to a single building zone at a comfortable temperature and humidity. As implied by its name, the single-duct air handler does all heating or cooling in one duct.

The constant volume single-duct air handler discussed here provides 15,000 cubic feet per minute of comfortable air to an office area of approximately 19,500 square feet (29). As shown in Figure 2-1, the air in the air handling system is blown by a large supply fan through a duct to the zone. Convective currents mix the supply air with existing air in the zone where heat is gained from lights and equipment; water vapor and heat are gained from the occupants; and heat is either gained or lost to the outside. This mixed air is exhausted from the zone to an overhead return area, much like the attic of a house. It should be noted that the physical configuration of this air handler has the uninsulated supply duct passing through the return air area. Therefore, when a temperature difference between the supply and return air exists, some heat transfer will take place.

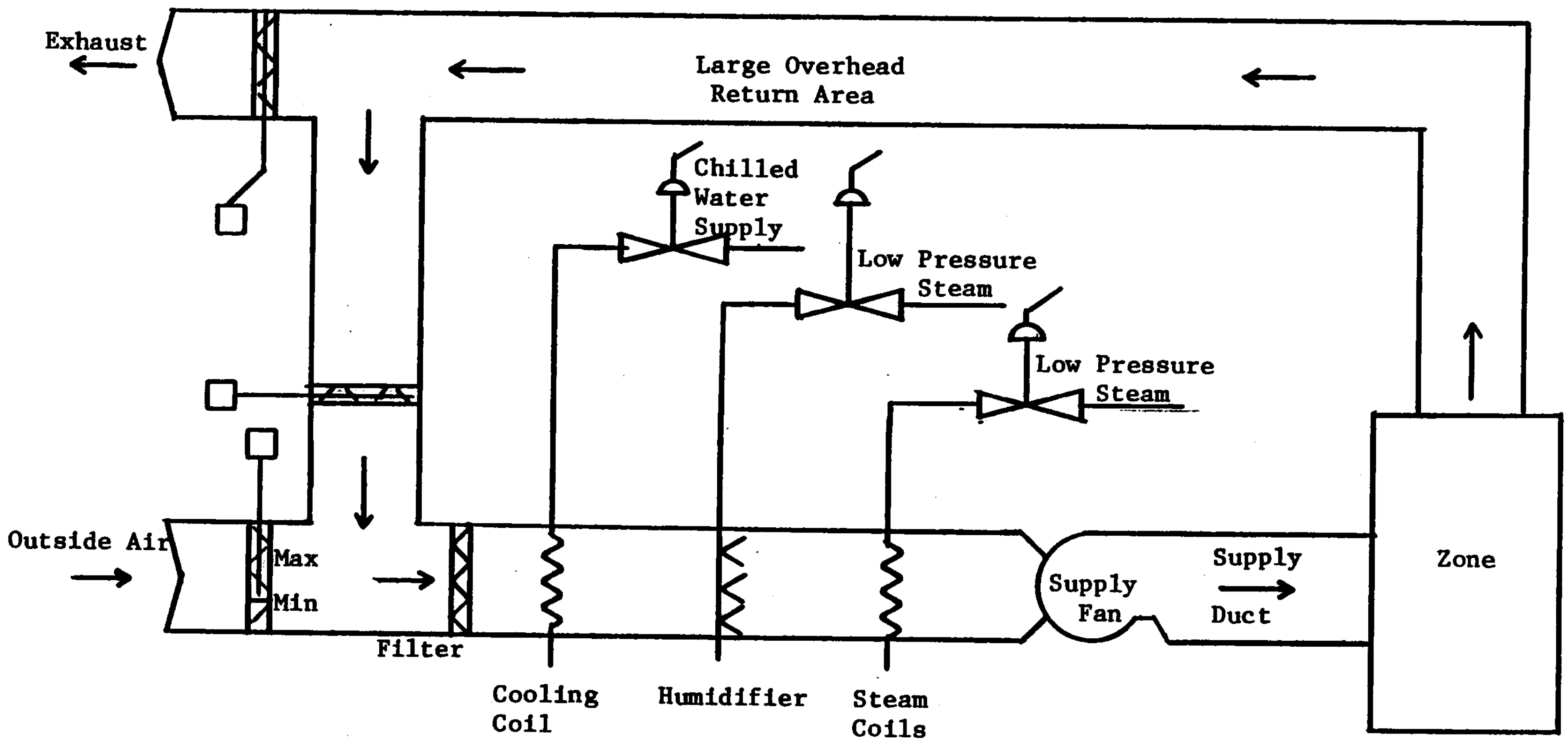


Figure 2-1. Single-duct air handler system.

From the return air area the mixed air is either exhausted to the outside of the building, returned to the air handler, or used in some combination of the two. Whether the return air is exhausted or recycled depends upon the relative positions of the return air damper of the building. These dampers are linked so that when one is fully open the other is fully closed. Another damper linked to these two is the outside air intake damper. This intake damper also operates in inverse manner to the return air damper. Inverse operation of these dampers will provide an approximately constant flow of air through the system. The positions of these dampers will be dictated by the control strategy. It is also important to note the minimum outside air damper. This damper must remain open, according to local code, during all operating hours of the air handler. In this case a significant fifteen per cent of the supply fan flow rate is used for minimum outside air. To maintain the constant air flow, an equilavent amount of air must also be exhausted to the outside at all times.

The air from the return and outside air dampers is mixed in the air handler's large (approximately 1200 cubic feet) mixing box. The mixed air is then pulled by the supply fan across the chilled water coils for cooling, the steam coils for heating, and the spray coils for humidification. Under normal circumstances only one, and not more than two, of these coils will be used. The air cycle is completed as the conditioned air from these coils flows to the supply fan.

The original single-duct air handler control strategy is implemented with pneumatics. The operating strategy is determined by a manual summer/winter switch. The winter strategies, shown in Figures 2-2 to 2-4,

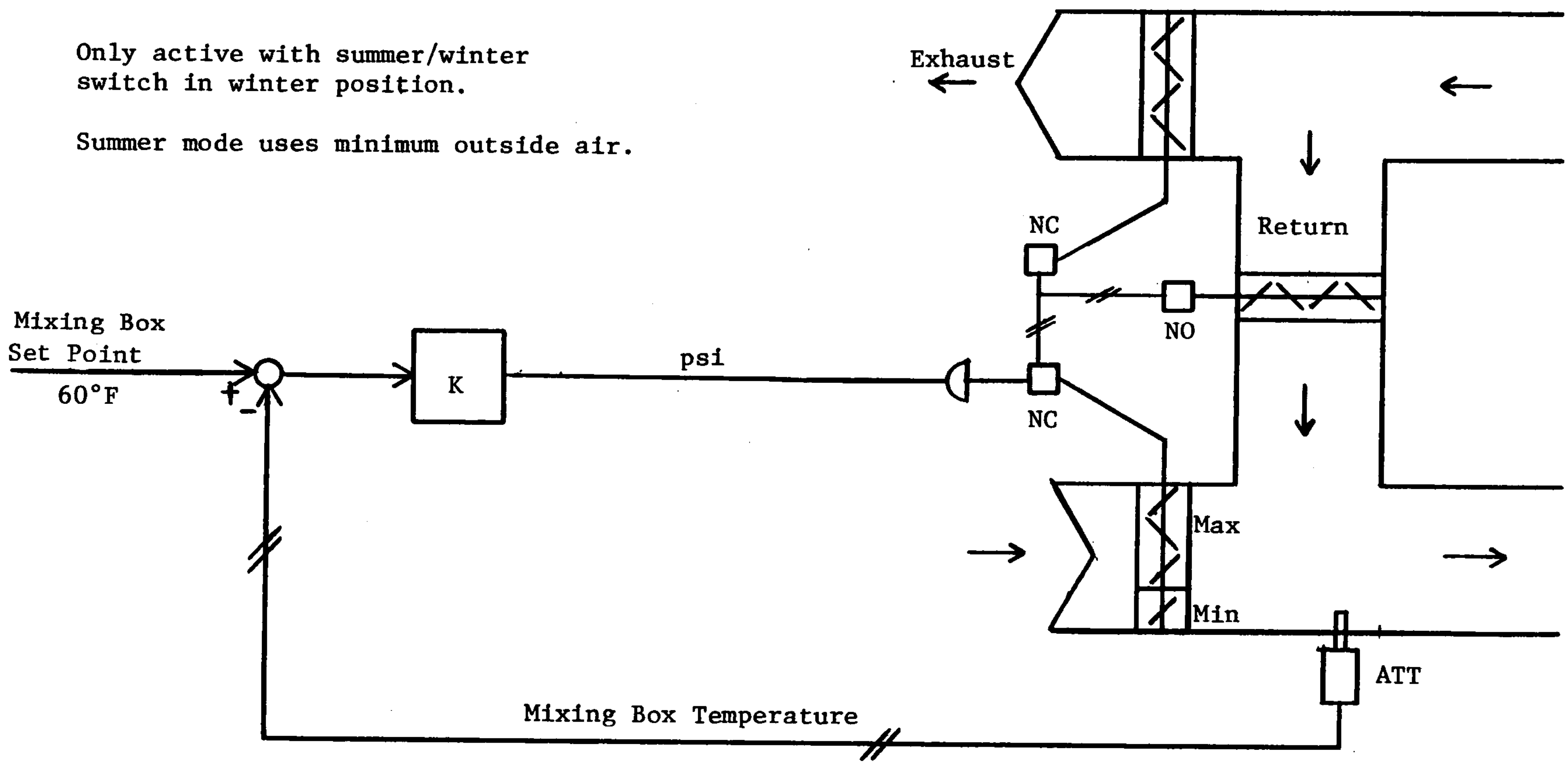


Figure 2-2. Single-duct winter mode free cooling by original air handler mixing box control strategy.

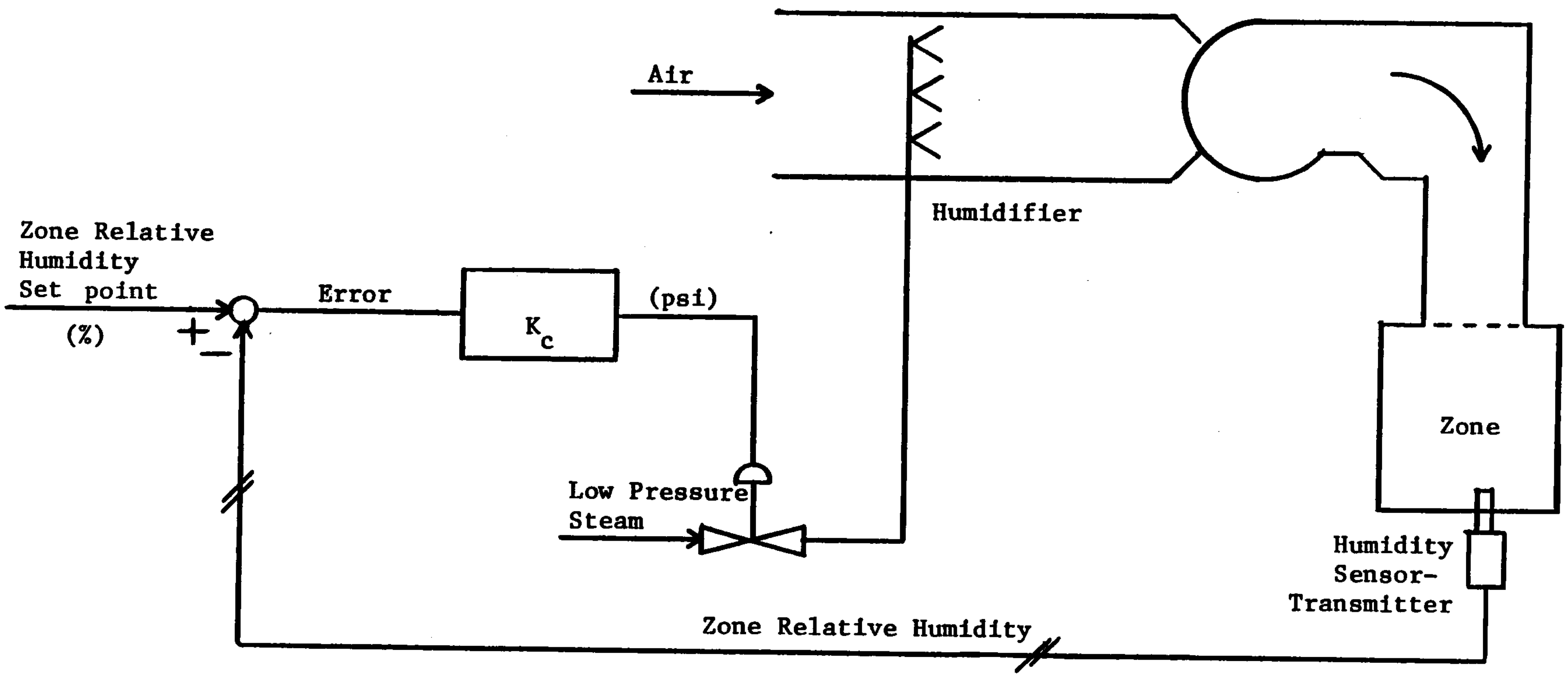


Figure 2-3. Winter mode single-duct humidification control loop.

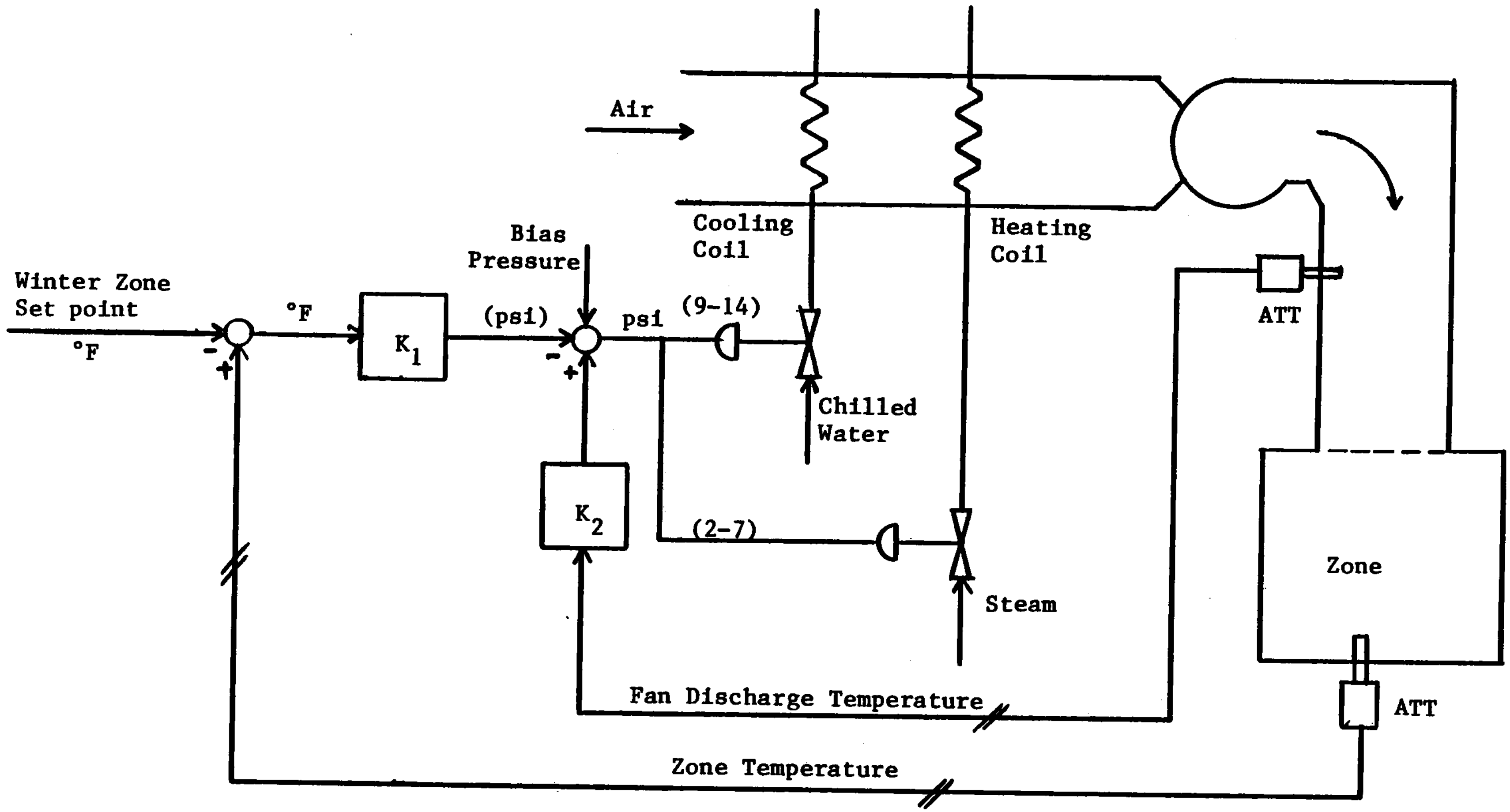


Figure 2-4. Single-duct winter mode temperature control loop.

include (a) free cooling from the outside air, (b) humidifying by steam stray coils, and (c) heating and cooling by the respective steam and chilled water coils. The summer mode is concerned only with controlling the steam and chilled water valves to meet the zone temperature and high summer humidity set points.

Figure 2-2 shows the winter operation of free cooling from the outside air. The outside and return air dampers are modulated to try and meet a mixing box air temperature set point of 60°F. In the summer mode, these dampers are always positioned for full return air and minimum outside air.

The process of humidification is required when the zone relative humidity is below the low winter set point of 25%. This 25% relative humidity is insured by controlling the valve position of the steam spray coils, as seen in Figure 2-3.

The zone temperature winter mode control loop, shown in Figure 2-4, uses split-range pneumatic feedback to determine the position of the steam and chilled water valves. As the feedback pressure to the valves increases above 9 psi, the chilled water valve begins opening, until it reaches its maximum at 14 psi. As the feedback pressure decreases below 7 psi, the steam valve begins to open until it reaches its maximum at 2 psi.

The only control loops in operation for the summer mode are shown in Figure 2-5. As with the winter mode, the heating coil is modulated fully closed to fully open by a feedback pressure from 7 psi to 2 psi, and the cooling coil goes fully closed to fully opened with a feedback pressure from 9 psi to 14 psi. However, for the summer mode, the zone temperature set point will be manually set higher than for the winter

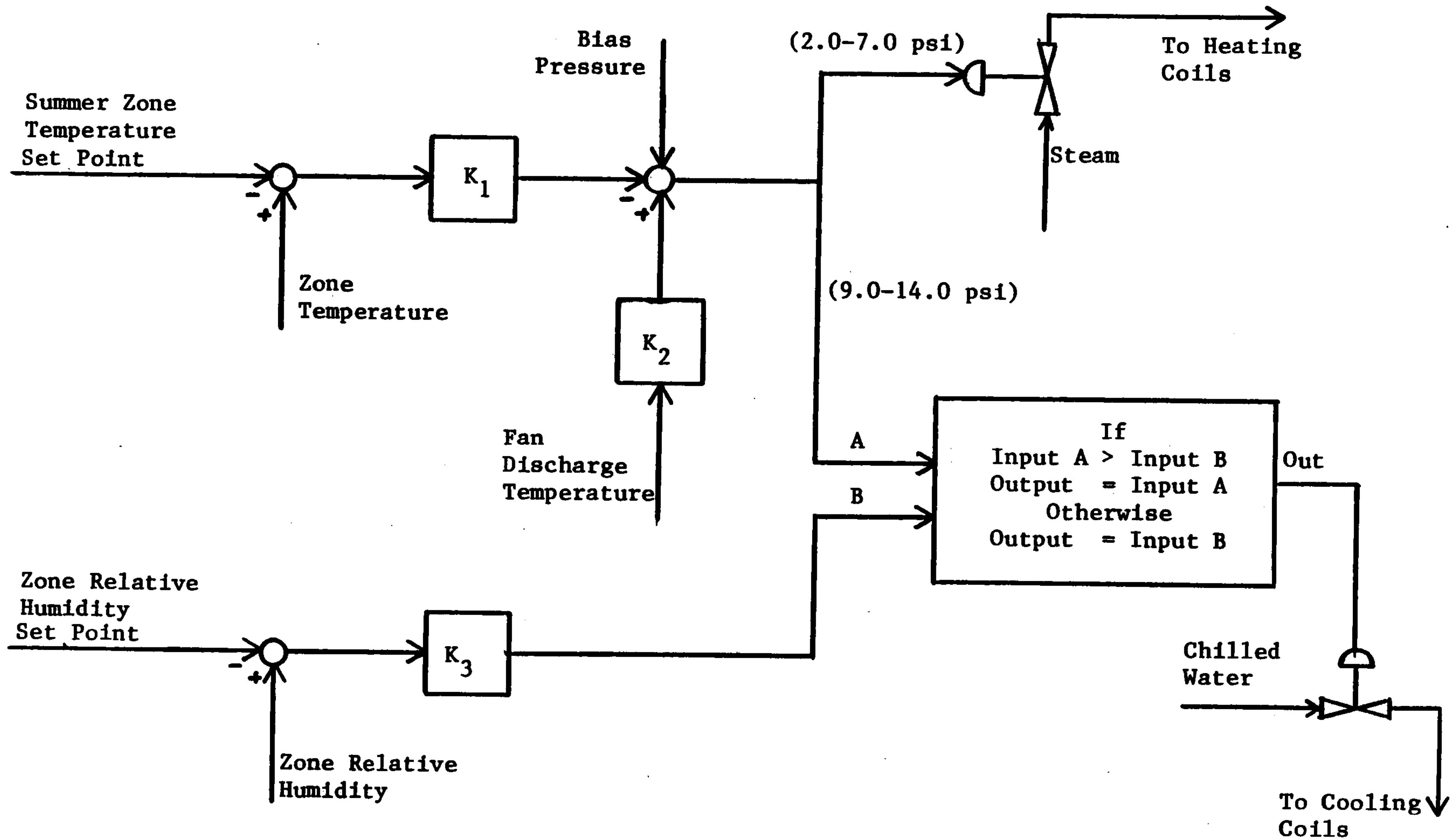


Figure 2-5. Single-duct summer mode temperature and dehumidification control loops.

mode. The big difference for the summer control is for dehumidification. If the zone humidity should increase above the high summer set point, a feedback pressure signal is sent to a high select relay, and compared with the feedback pressure signal from the temperature loop. If dehumidification is required, the humidity feedback pressure signal controls the chilled water valve, otherwise, the temperature feedback pressure signal controls the valve. If the cooling associated with dehumidification should drive the zone temperature low enough to cause the pressure feedback signal to drop below 7 psi, the steam heating coil will be opened. The cooling process of dehumidification serves to condense water from the air, but also increases the relative humidity. The adding of sensible heat back to the air will decrease the relative humidity. Therefore, by simultaneous cooling and heating, the air is dehumidifying.

Dual-Duct Air Handler

The constant volume dual-duct air handler is also a large piece of HVAC equipment. However, its purpose it to provide comfortable air to many zones that may have different thermostatic requirements. For this reason, the dual-duct air handler has both a hot and cold duct, and uses mixing at each zone to provide the necessary air.

The dual-duct air handler selected here also has a total air flow of 15,000 cubic feet per minute, to allow use of cooling coil information known for the single duct air handler. Although many more zones can be served, this dual-duct will provide air to only three zones, enabling a look at high and low extremes, along with one intermediate temperature zone. The dual-duct system, shown in Figure 2-6, uses local thermostats

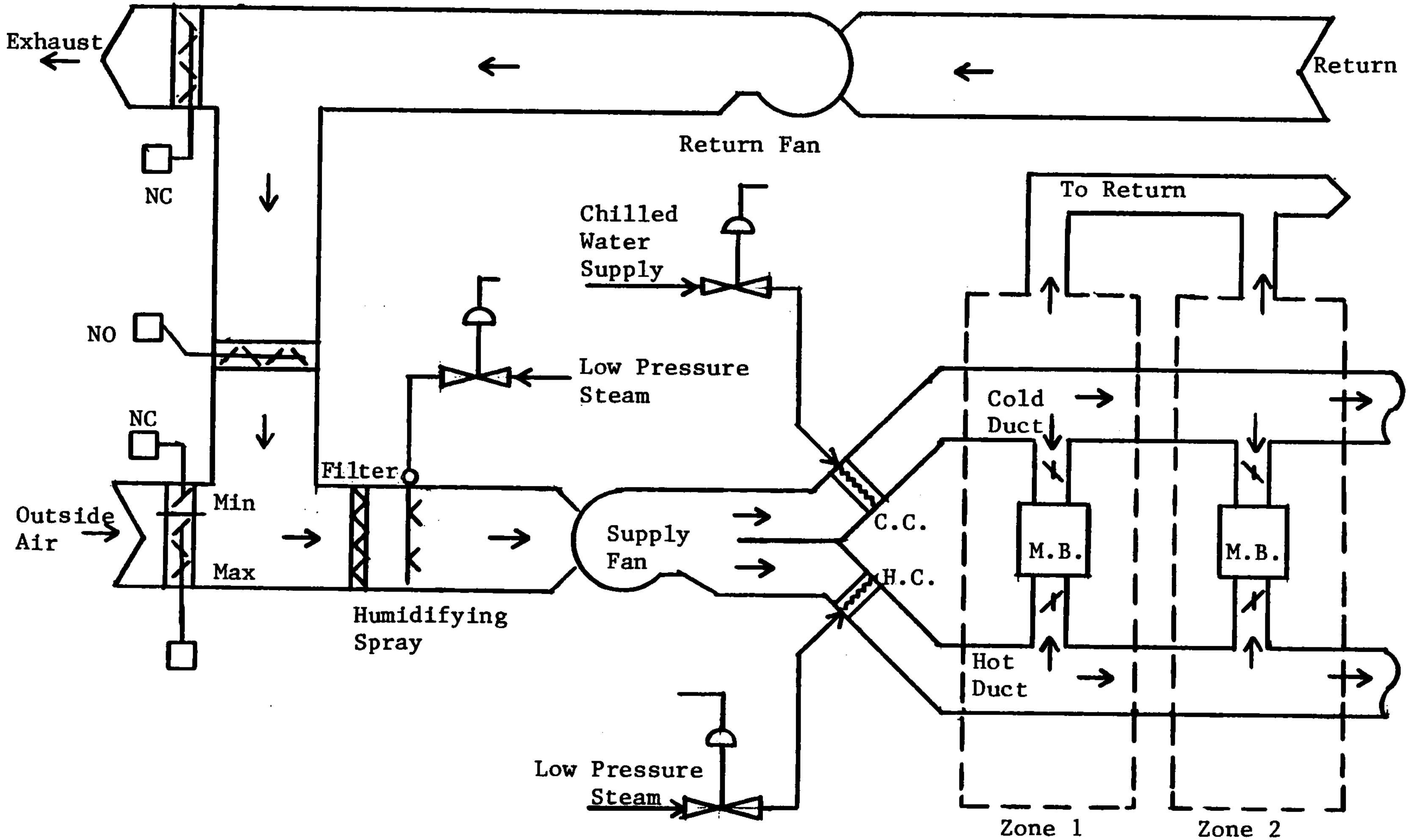


Figure 2-6. Dual-duct system.

to determine the necessary mixture of hot duct air and cold duct air to meet their individual needs. These local zone mixing box dampers are linked such that a fully open damper to the hot duct will cause a fully closed damper to the cold duct. The relative positions of these dampers determine the flow rates of air through the hot and cold ducts. As with the single-duct air handler, convective currents mix the air from the ducts with existing air in the zone, and additional heat and or water is gained from lights, equipment, people, and the outside. The air from each of these zones is exhausted to a return air area overhead. This return air is exhausted or recycled depending upon the relative positions of the return air damper of the air handler and the exhaust damper from the building. The dual-duct outside air and return air dampers operate in the same linked fashion as the single-duct air handler. It also has a 15% of total flow requirement for minimum outside air. The outside and return air are combined in the large mixing box before passing throught he supply fan. The air leaving the supply fan then goes to either the hot or cold duct, according to the relative positions of the separate zone mixing box dampers. Air going through the cold duct passes over the cooling coils for any necessary cooling, and air going through the hot duct passes over the steam coils for any necessary heating. The cycle is completed as the air moves through the hot and cold ducts to the zones.

The original pneumatic control strategies designed for the dual-duct air handler, shown in Figures 2-7 to 2-10, are those associated with (a) local zone mixing boxes, (b) chilled water valve, (c) steam coil valve, and (d) air handler mixing box (28).

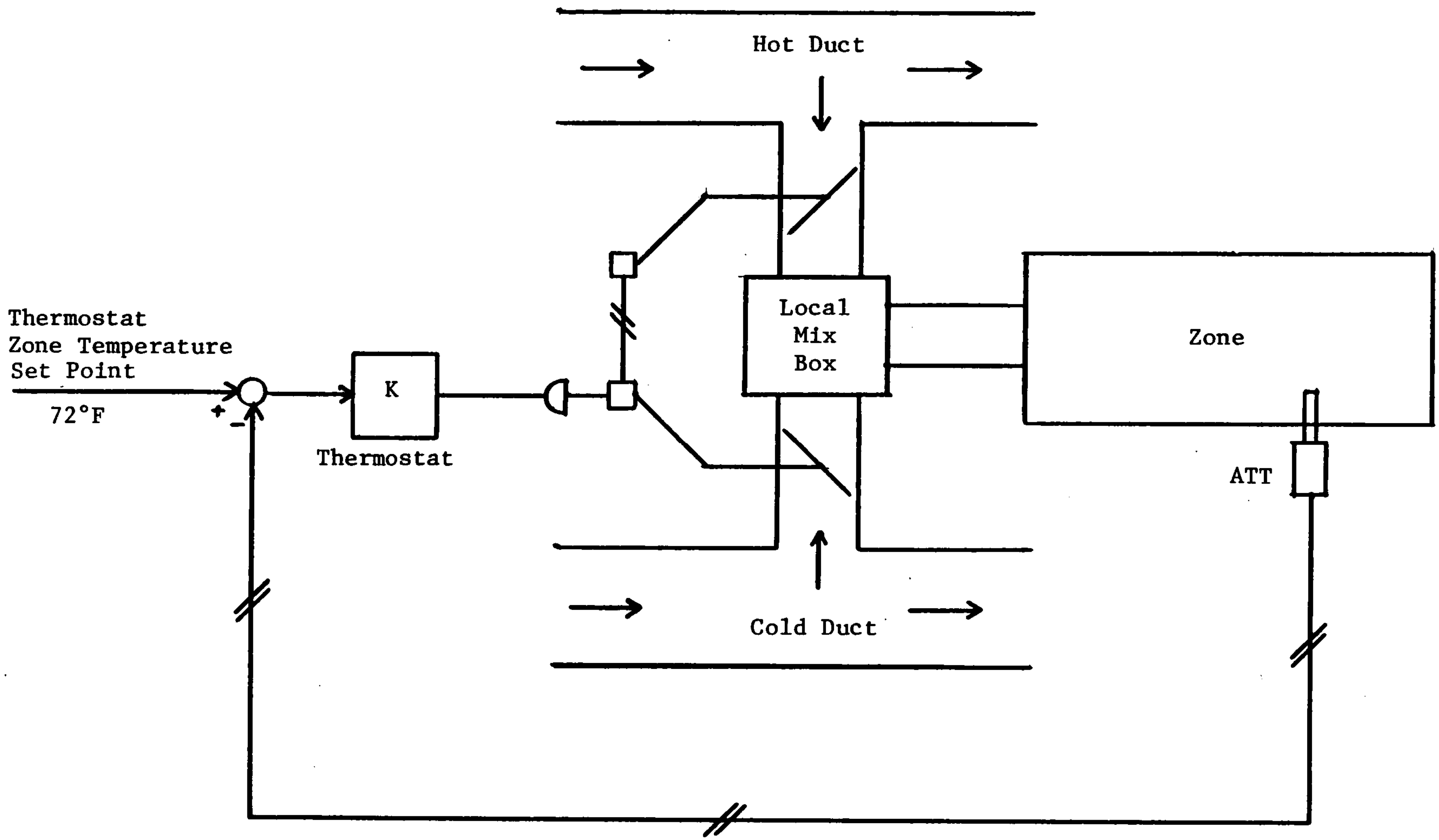


Figure 2-7. Dual-duct local zone mixing box control.

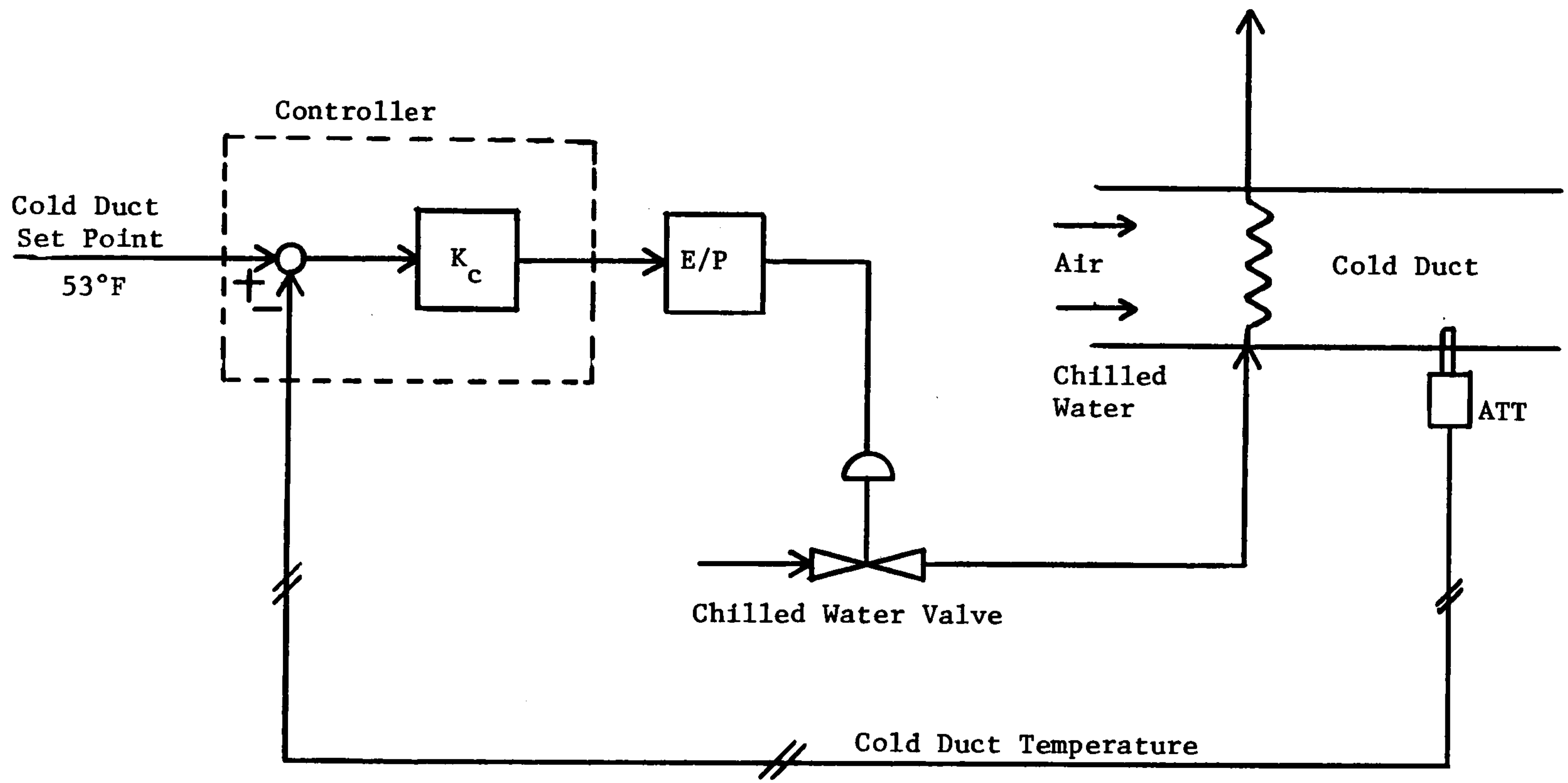
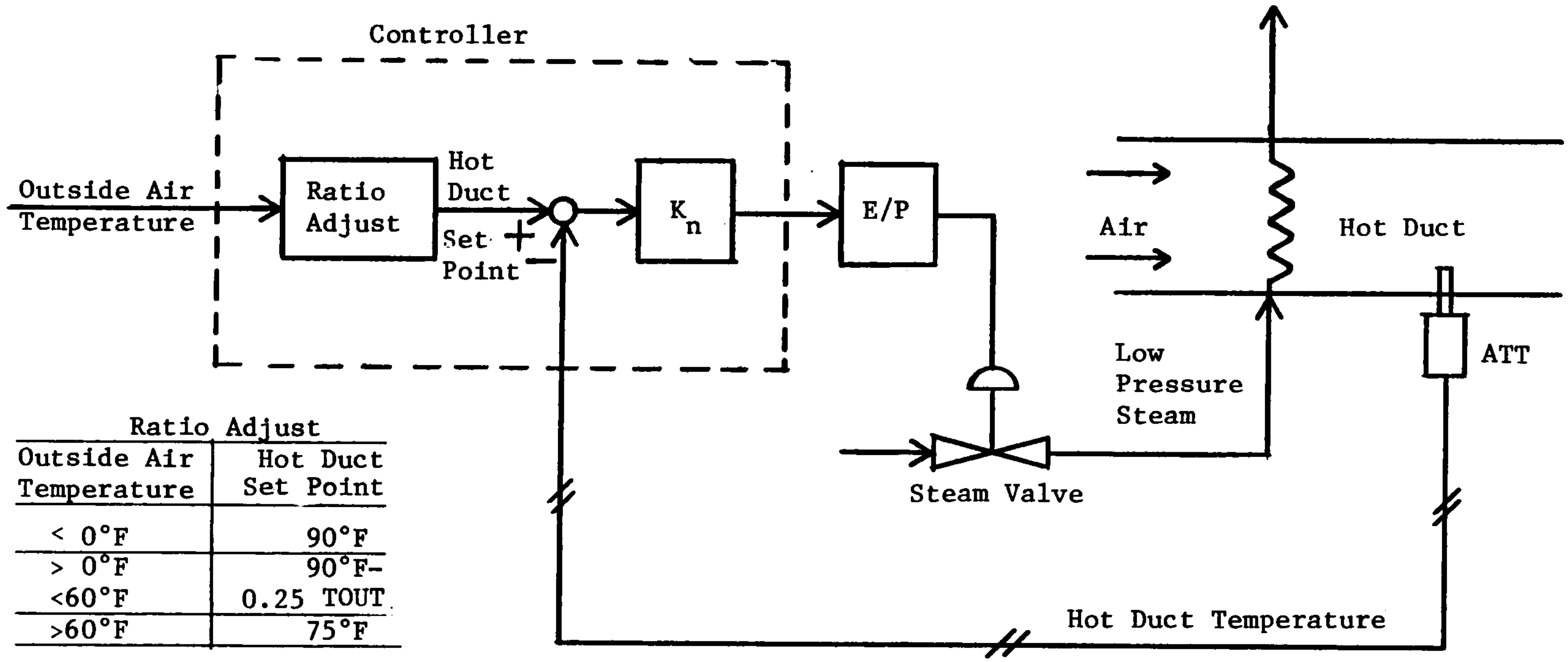


Figure 2-8. General-dual duct, cold duct temperature control loop.



Ratio Adjust

Outside Air Temperature	Hot Duct Set Point
< 0°F	90°F
> 0°F <60°F	90°F- 0.25 TOUT
>60°F	75°F

Figure 2-9. General dual-duct, hot duct temperature control loop.

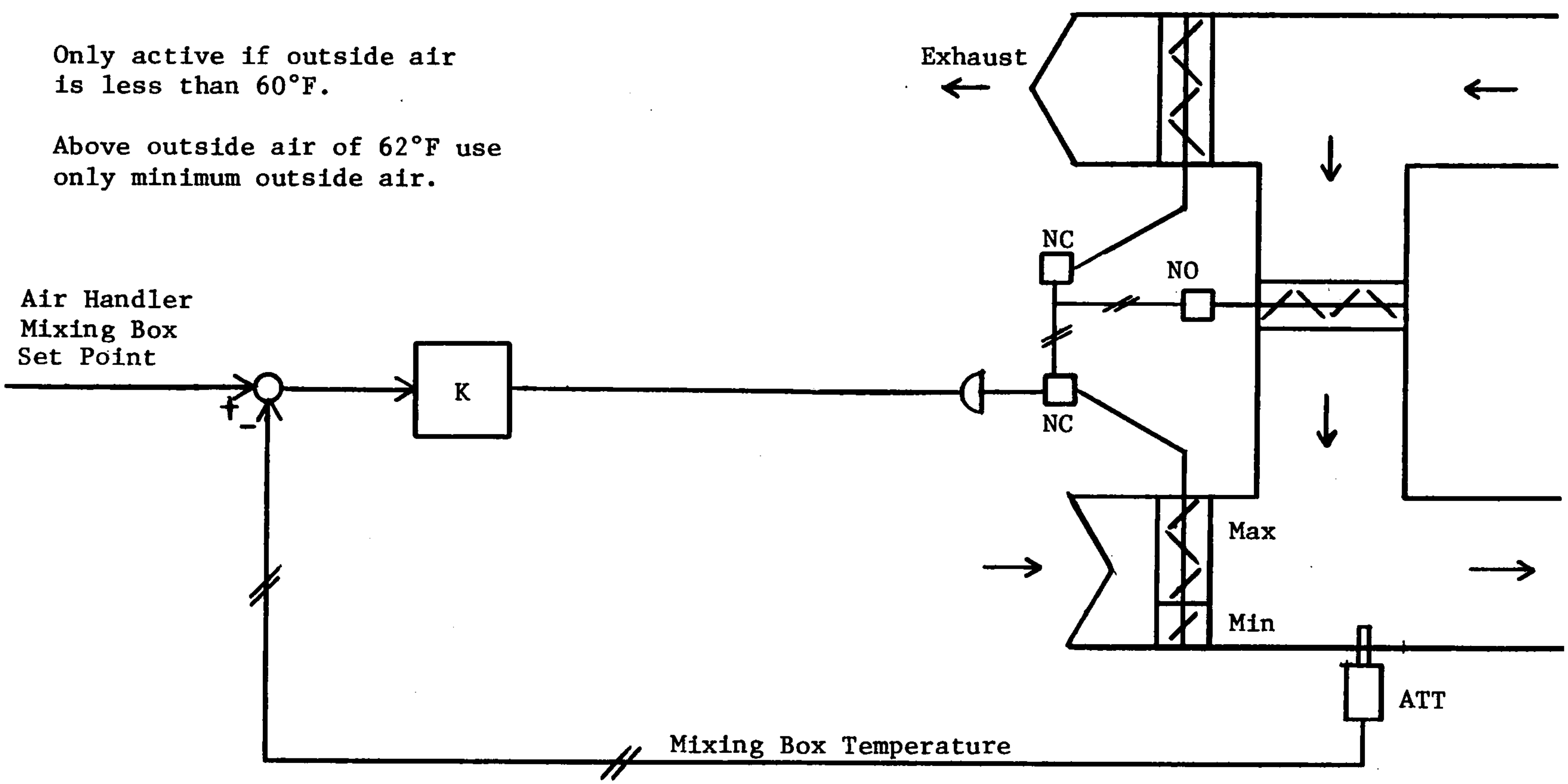


Figure 2-10. Dual-duct original air handler mixing box control strategy for free outside air cooling.

As stated above, the zone mixing boxes are controlled by local thermostats which determine the necessary damper positions to mix hot duct and cold duct air to meet their set point.

The chilled water valve position is based on a simple feedback strategy that requires the cold duct to maintain a constant air temperature set point of 53°F.

The steam valve is adjusted as required to meet the hot duct set point. The hot duct set point is a variable function of the outside air temperature according to the following relationships (with TOUT = outside air temperature):

Set Point = 90°F	TOUT $\leq$ 0°F
Set Point = .25 (TOUT) + 90°F	0°F < TOUT < 60°F
Set Point = 75°F	TOUT $\geq$ 60°F

The air handler mixing box is operated by an automatic strategy that closes the outside air damper, meeting only the minimum outside air requirement, when the outside air temperature is above 62°F. If the outside air temperature falls below 60°F, the mixing box dampers are modulated to meet a mixing box set point of 55°F.

It should be noted that this description does not include a separate strategy for dehumidification. Dehumidification is typically a summer problem. Under summer conditions, much of the air will pass through the cold duct which has a set point of 53°F. Cooling a large percentage of the air to 53°F will always condense enough water to satisfy the upper limit humidity constraint.

Perimeter Radiation Heating System

The final piece of HVAC equipment under consideration is the perimeter radiation heating system. This winter system is used to provide a warm insulating layer of air around the perimeter wall of a building.

The perimeter system consists of a steam heat exchanger which provides hot water to north, south, east, and west building exposures as shown in the Figure 2-11 piping and control valve diagram (31). This diagram shows that the north exposure receives hot water directly from the exchanger. The south, east, and west exposures have three-way control valves which allow mixing of return water with supply water, thus providing a lower temperature water to these exposures. This also reduces the flow of water that must pass through the steam heat exchanger. The system pumps provide a constant flow of water to their respective perimeter faces where the water passes through the perimeter pipes and heat is released.

Figure 2-12 shows the pneumatic control strategy for the north exposure, which sets the steam valve position to the heat exchanger. The temperature of the water leaving the exchanger is modulated according to the following schedule (with TOUT = outside air temperature, and TW = water temperature out of the exchanger):

$TW = 200°F$	$TOUT \leq 0°F$
$TW = 200°F - 1.667\ (TOUT\ F)$	$0°F < TOUT < 60°F$
$TW = 100°F$	$TOUT = 60°F$

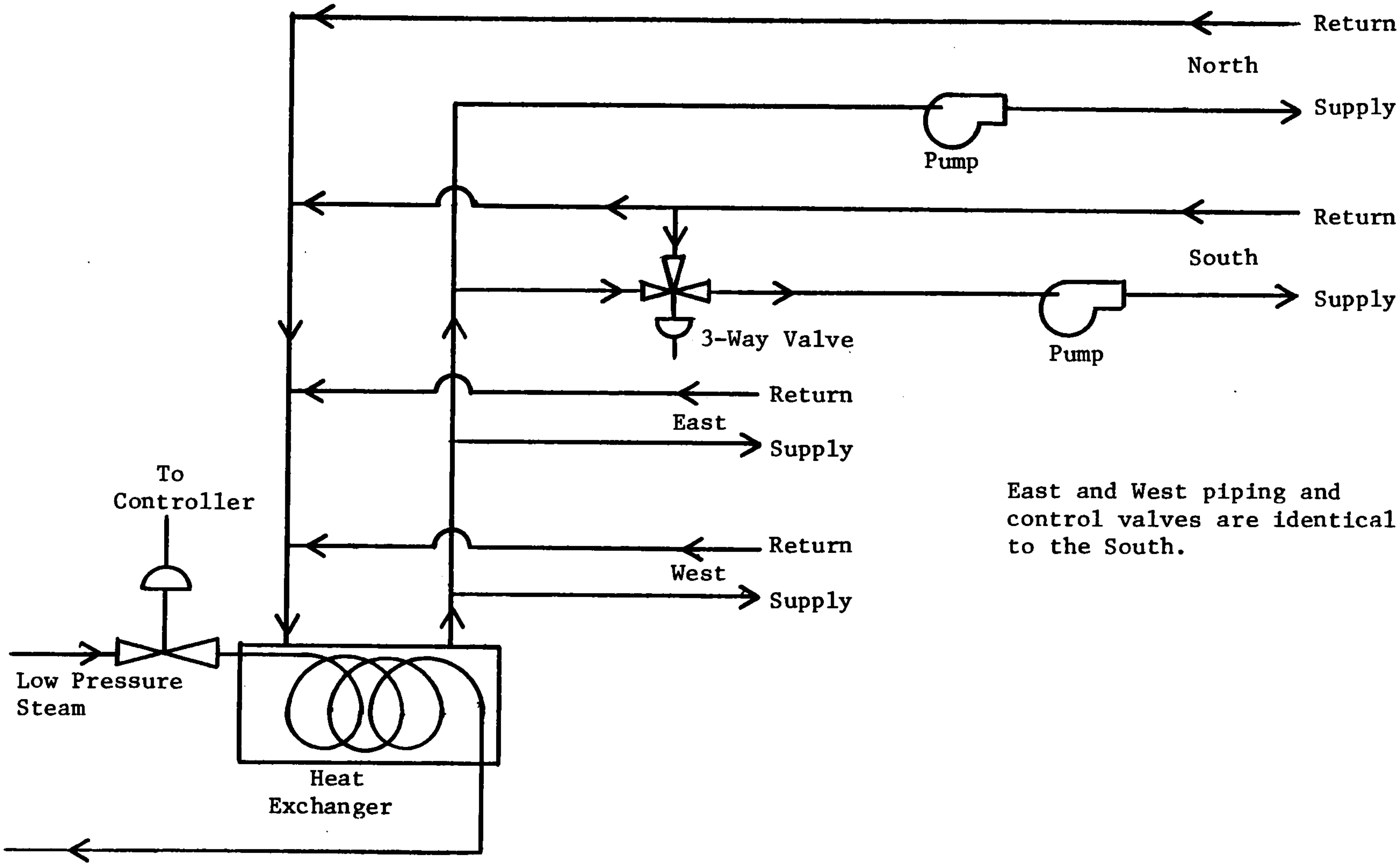
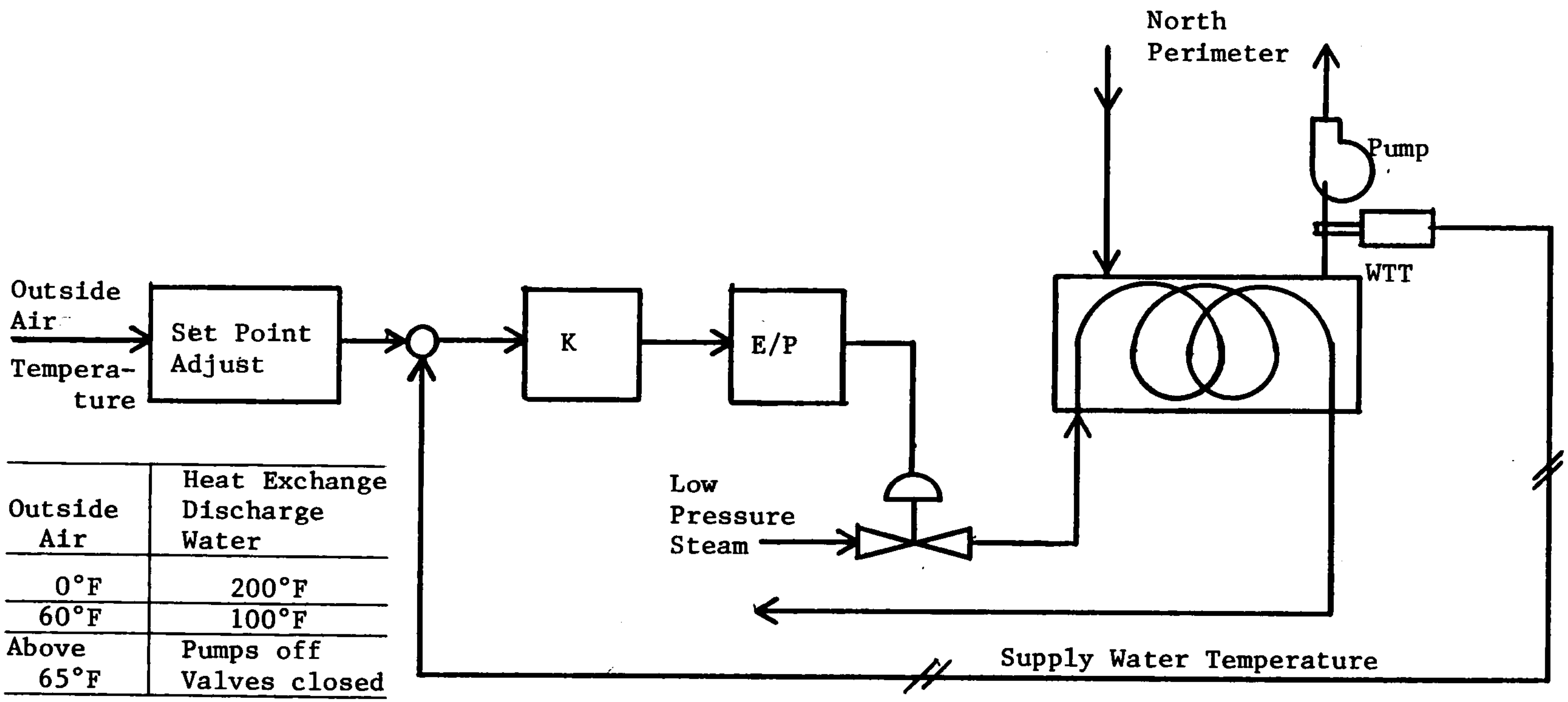


Figure 2-11. Piping and control valves for perimeter heating system.



Outside Air	Heat Exchange Discharge Water
0°F	200°F
60°F	100°F
Above 65°F	Pumps off Valves closed

Figure 2-12. Existing control system for main hot water supply.

The other three exposures were designed, as shown in Figure 2-13, with strategies to allow decreasing their desired water temperature set point with a main bias and a solar correction. However, for the system under study this bias is set so that all exposures receive water at the exchanger temperature, with no use of the three-way valves.

The description given in this chapter for each of these systems will be used to set up the simulation for determination of a base system energy requirement. Any modifications made for the purpose of simulations will be noted.

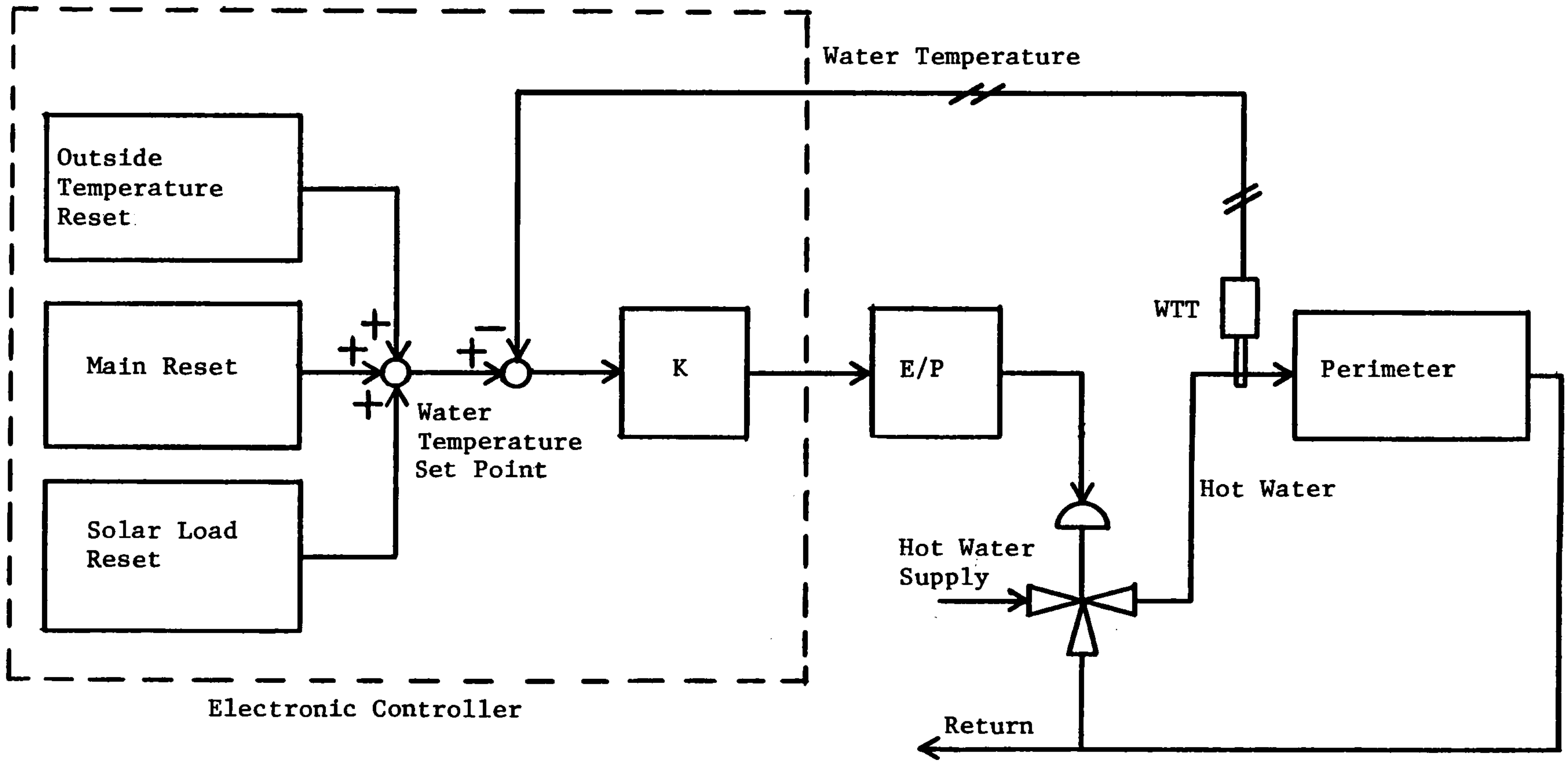


Figure 2-13. Existing control system for south, east and west perimeter system.

CHAPTER III

SYSTEM SIMULATION APPROACH

One of the most important aspects of HVAC computer studies is an effective simulation. The goal in these simulations is to be complex enough to adequately describe the equipment involved, yet simple enough to be easily implemented in a larger program. All models discussed are used for steady state energy calculations.

Many of the routines written for the air handling equipment are applicable to both the single-duct and dual-duct systems. These routines are intentionally made general to provide easy interchange-ability. These include simulation of the psychrometric properties of air, air mass and energy balances, air handler fans, heating coils and cooling coils.

Psychrometric Properties of Air

The routines used for describing the psychrometric properties of air are those written by Marcev (21). Given the atmospheric pressure, PATM, along with any two conditions of DB, WB, RH, DP, H, W, or V, the routines will calculate the remaining properties where,

DB = dry bulb temperature

WB = wet bulb temperature

RH = relative humidity

DP = dew point

H = enthalpy

W = humidity ratio

V = specific volume of moist air

For future reference, a list of variable names and units applicable is given in Appendix A.

A simplification of Marcev's work provides alternate routines which calculate only selected properties of air, eliminating time consuming calculations for useless information. The original Marcev program is entitled PSYCHO and the simplified version if SSYCHO. A list of program names has been provided in Appendix B, and the complete program listings in Appendix C.

Air Balances

A common requirement for many of the large component simulations is a simple mass and energy balance for air streams. This may involve the summing of two streams, such as in a mixing box, or the addition of heat and moisture, such as in a zone or through a duct. The Figure 3-1 diagram is used to make a mass and energy balance around two summing air streams, using the notation:

F = air flow

T = air dry bulb temperature

W = air water content

H = air enthalpy

LW = water vapor load

DLH = latent heat of water vapor

LQ = heat load

The general energy balance gives Equation 3.1.

FI(HI)+FJ(HJ)+QLI+WLI(DLH)=WLO(DLH)+QLO+FO(HO) (3.1)

The total mass balance on dry air gives Equation 3.2.

FO=FI+FJ (3.2)

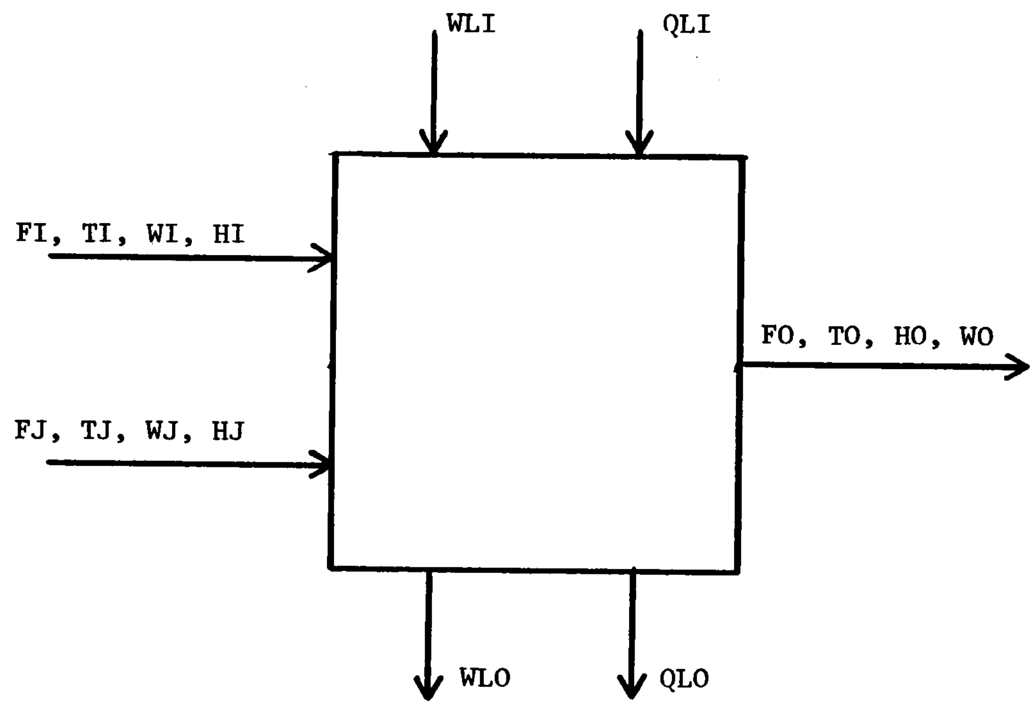


Figure 3-1. Air mass and energy balance diagram.

The total water mass balance gives Equation 3.3.

$$WLO+FO(WO)=FI(WI)+FJ(WJ)+WLI \qquad (3.3)$$

These equations are then solved to give two general cases for air balances. The first case, subroutine AISRM, is with both input streams given and the output stream condition calculated. The second case, subroutine AIRSM2, finds the conditions of one input stream given the condition of the other input stream and the output stream. Both of these routines may be used to solve a single stream case, where load calculations are desired, by zeroing one input stream.

Fan

Another common piece of HVAC equipment is the large system fan. The heat gained by the air as it passes through a fan is given by Equation 3.4 (29)

$$DELH = \frac{2546.(WF)}{EFF(FEX)} \qquad (3.4)$$

where,

DELH = change in air enthalpy across the fan

WF = brake horsepower of the fan

EFF = fan efficiency

FEX = air flow rate

Equation 3.5 then gives the change in temperature across the supply fan, DELTT,

$$DELTT = \frac{DELH}{CPAIR} \qquad (3.5)$$

where,

CPAIR = heat capacity of air

Subroutine SFAN is used for calculations where the air conditions leaving the fan are given and subroutine SFAN2 for conditions where the air conditions entering the fan are given.

Heating Coils

Subroutine HEACOL gives the amount of steam required for a given heat rise across the heating coils. Since the effective heat from steam is its latent heat, the steam flow may be calculated by Equation 3.6.

$$FST = \frac{FAIR(HAHC-HBHC)}{STLH} \tag{3.6}$$

where,

FST = steam flow

FAIR = air flow

HAHC = Air enthalpy after heating coils

HBHC = air enthalpy before heating coils

STLH = steam latent heat

Cooling Coils

One of the most difficult pieces of HVAC equipment to simulate is the cooling coils, because cooling air across cooling coils involves a simultaneous mass and energy transfer. The simulation approach chosen is the one given by Stoecker (9). This approach provides a general model with readily available parameters such as cooling water temperature, coil face area, number of rows of coils, number of parallel circuits, and coil inner diameter. Given these parameters along with air and cooling water flow rates, a large polynomial expression is used to calculate coil operation.

The first requirement is calculation of the air and water flow rates by Equations 3.7 and 3.8.

$$FPM = \frac{FAIR}{DAIR(3600.)(FA)} \tag{3.7}$$

$$FPS = \frac{GPM(7.48)}{60.(\pi/4)(ID^2)(NC)} \tag{3.8}$$

where,

FPM = air-side face velocity

FPS = water-side velocity

FA = coil face area

FAIR = air flow rate

DAIR = air density

GPM = water flow rate

ID = coil inner diameter

NC = number of parallel circuits

From these values a coil base rating can be calculated by Equation 3.9.

$$BRCW = \frac{1000}{C1+\frac{C2}{FPM}+\frac{C3}{FPS}+\frac{C4}{(FPS)^2}+\frac{C5}{(FPM)^3}+\frac{C6}{(FPS)^2\,(FPM)^4}} \tag{3.9}$$

Here BRCW is the coil base rating in BTU/hr per square foot of face area per row of tubes deep per log mean temperature difference. The coefficients fit to this equation by Stoecker (9) from general cooling coil performance data are given in Table 3-1.

The next calculation for this model is the wetted surface factor given in Equation 3.10. This factor is used to account for the effect of moisture on coil performance.

TABLE 3-1

CONSTANTS FOR COOLING COIL EQUATION 3.9
GIVEN BY STOECKER (9)

Coefficient	Value
C1	0.22074304×10^{1}
C2	0.14178455×10^{4}
C3	0.33925943×10^{1}
C4	$-0.55238537 \times 10^{0}$
C5	$-0.11071271 \times 10^{8}$
C6	$-0.39288579 \times 10^{9}$

$$WSF=b1+b2(PW)+b3(PW)(BW)+b4(PW)^2+B5(BW)^2(PW)+b6(BW)(PW)^2$$
$$+b7(BW)^3(PW)+b8(BW)^3(PW)^2+b9(BW)^3(PW)^3 \qquad (3.10)$$

where PW and BW are given by Equations 3.11 and 3.12.

$$PW=EDP-EWT \qquad (3.11)$$

$$BW=EDB-EWT \qquad (3.12)$$

where,

EDP = entering air dew point temperature

EDB = entering air dry bulb temperature

EWT = entering water temperature

For Equation 3.9 the coefficients, again fit by Stoecker (9) to general cooling coil performance data, are given in Table 3-2.

The procedure for using this model requires a numerical search to determine the temperature of the water leaving the coil. Given the leaving water temperature, the heat transfer across the coils is given by Equation 3.13.

$$QTW=(CWFLO)(CPW)(LWT-EWT) \qquad (3.13)$$

where,

QTW = water enthalpy change in BTU/hr

CWFLO = chilled water flow rate in lbs/hr

CPW = heat capacity of water

LWT = leaving water temperature

When the heat gained by the water is equal to the heat lost by the air, the guessed leaving water temperature is correct. The heat lost by the air is calculated by Equation 3.14.

$$QTCA=(BRCW)(WSF)(NR)(FA)(MLTD) \qquad (3.14)$$

where,

QTCA = heat lost by air across cooling coils

TABLE 3-2

CONSTANTS FOR COOLING COIL EQUATION 3.10
GIVEN BY STOECKER (9)

Coefficient	Value
b1	0.10007596×10^{1}
b2	$0.42652727 \times 10^{-1}$
b3	$-0.22742525 \times 10^{-2}$
b4	$0.16232278 \times 10^{-2}$
b5	$0.41384645 \times 10^{-4}$
b6	$-0.24564063 \times 10^{-4}$
b7	$-0.24092761 \times 10^{-6}$
b8	$-0.11066765 \times 10^{-8}$
b9	$0.54302870 \times 10^{-10}$

MLTD = log mean temperature difference

NR = number of rows of coils

The value of MLTD is given by Equation 3.15.

$$MLTD = \frac{D1-D2}{\ln(\frac{D1}{D2})} \tag{3.15}$$

where D1 and D2 are given by Equations 3.16 and 3.17.

$$D1=EDB-LWT \tag{3.16}$$

$$D2=LDB-EWT \tag{3.17}$$

The only value not previously determined is the leaving air dry bulb temperature, LDB. LDB is found by subtracting the heat gained by the water from the air entering the coils. It is important to note that the heat lost by the air may be either sensible or latent. Stoecker (9) suggests letting all cooling be sensible until a relative humidity of 90% is reached, and then allow the cooling to follow the 90% relative humidity line as shown in Figure 3-2. However, ASHRAE (8) states that typical cooling coil operation has a sensible cooling to total cooling ratio between 0.6 and 1.0. In most cases there will always exist some latent cooling.

An empirical equation is devised in this work to account for expected latent cooling that will occur before the 90% relative humidity line is reached. The two factors most responsible for high latent cooling are the amount of moisture in the air and the length of cooling coil surface that is at a temperature cold enough to condense moisture from the air. To account for the moisture in the air, parameter A is introduced into the empirical equation as given in Equation 3.18.

$$A = \frac{EDP}{EWT} \tag{3.18}$$

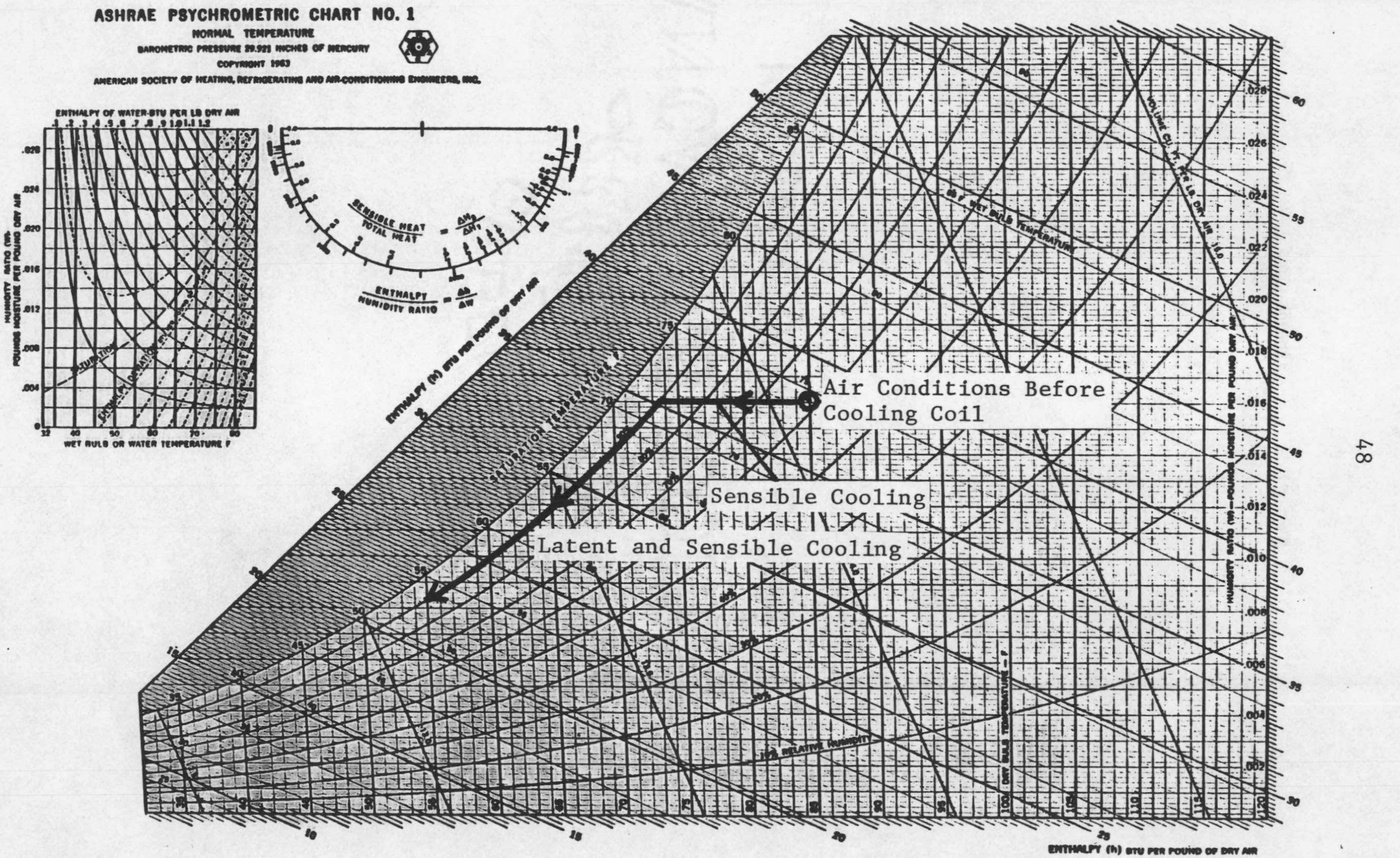


Figure 3-2. Suggested (9) sensible-latent heat removal approach for cooling coil model.

Parameter B, given in Equation 3.19 is introduced to account for the length of cold cooling coil surface.

$$B = \frac{CWFLO}{CWMAX} \tag{3.19}$$

Where,

CWMAX = maximum chilled water flow

An adjustable parameter, C, is introduced to enable the ratio runction, RATO, to be held in the ASHRAE range of 0.6 to 1.0. The function devised is given in Equation 3.20.

$$RATO=1.0-C(B)(A-1.0)^2 \tag{3.20}$$

Testing shows that the form of this equation follows the intuitive concept that some latent cooling will occur if there is a high moisture content and the chilled water flow is high. However, as the air becomes dry, with EDP close to EWT, or the chilled water flow goes very small causing only a short length of the pipe to be cold enough to condense moisture from the air, the cooling becomes all sensible. Bounds are required to set RATO equal to 1.0 if EDP is less than EWT. The sensible to total heat ratio function is applied to the air across the cooling coils only until the 90% relative humidity line is reached. The model then follows the 90% line as suggested by Stoecker.

The cooling coil model described above began with the assumption that the flow rate of the chilled water is given. In actual use, the values normally known are the entering and leaving air temperatures. A search technique must then be used outside the already described model to find the correct flow of chilled water to achieve a desired amount of cooling. A flow chart of the cooling coil routine, CCCALL, is given in Figure 3-3.

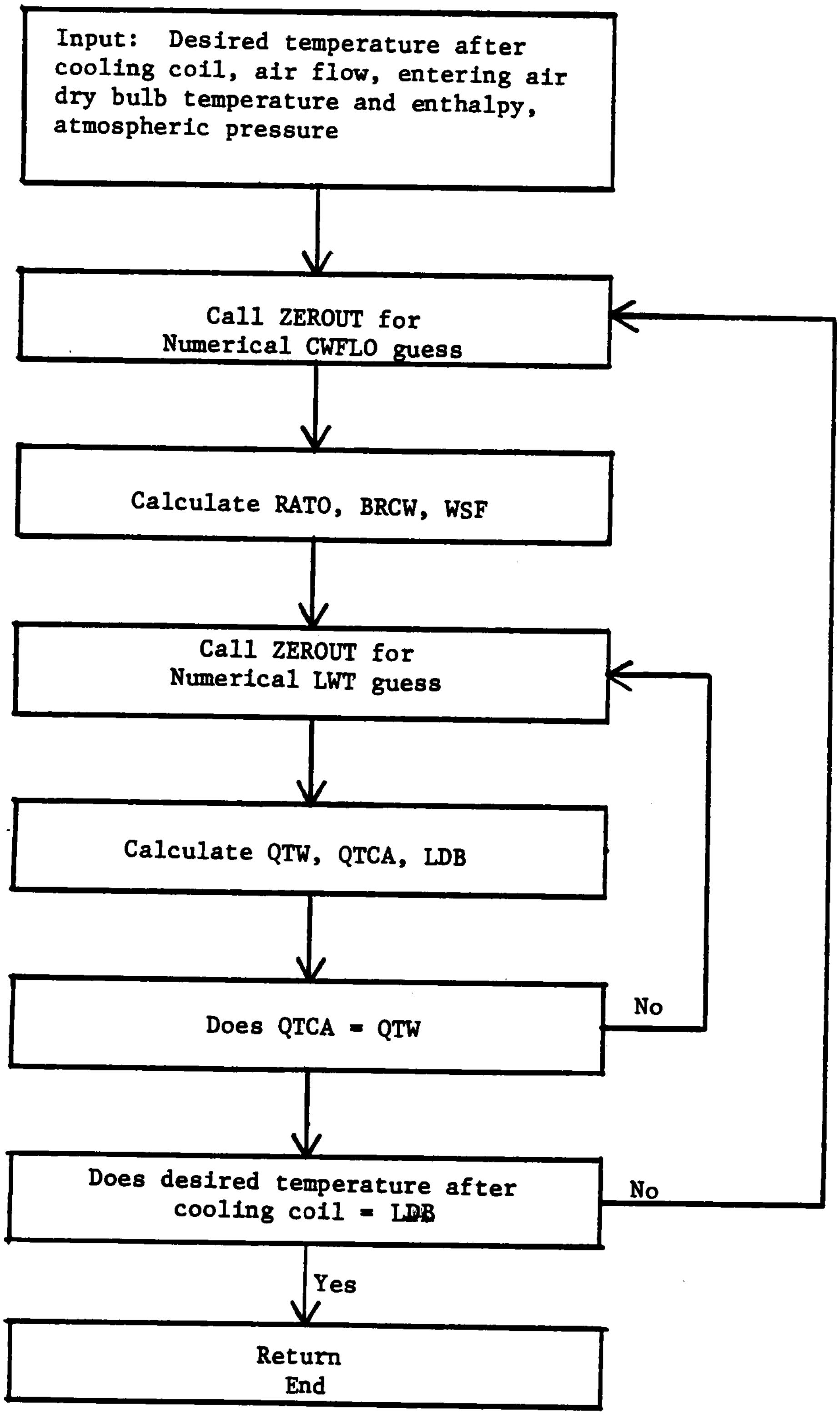


Figure 3-3. Cooling coil routine, CCCALL, flow chart.

Actual implementation of this model presents some difficulty at low chilled water or air flow. Due to the high order of the terms with coefficients C4, C5, and C6 in Equation 3.9, this equation became numerically unstable. To force this equation to be stable these three terms are separated from the first three as given in Equations 3.21 and 3.22.

$$BRCW1 = C1 + \frac{C2}{FPM} + \frac{C3}{FPS} \tag{3.21}$$

$$BRCW2 = -\left(\frac{C4}{(FPS)^2} + \frac{C5}{(FPM)^3} + \frac{C6}{(FPS)^2(FPM)^4}\right) \tag{3.22}$$

with BRCW given by Equation 3.23.

$$BRCW = \frac{1000}{BRCW1 - BRCW2} \tag{3.23}$$

A test of BRCW1 and BRCW2 over the range of data given by Stoecker shows that the maximum ratio of BRCW2/BRCW1 is 0.172 occurring at the minimum data values of air flow and chilled water flow used to determine the coefficients C1 to C6. However, at physically realizable chilled water flows less than this minimum, the ratio goes to infinity. To prevent these terms from controlling the equation at lower values, the model is set in this work such that the maximum ratio of BRCW2/BRCW1 is 0.172. This is done by first calculating values for BRCW1 and BRCW2 by Equations 3.21 and 3.22. If BRCW2/BRCW1 is less than 0.172, the value for BRCW is calculated by the original Stoecker (9) equation shown in another algebraic form in Equation 3.23. If BRCW2/BRCW1 is greater than 0.172, the value for BRCW is calculated by Equation 3.24.

$$BRCW = \frac{1000}{BRCW1-0.172(BRCW1)} \tag{3.24}$$

This provides a smooth continuous curve for BRCW as a function of either chilled water flow or air flow, with BRCW going to an expected value of zero as either flow approaches zero.

Single-Duct Air Handler

Overall simulation of the single-duct air handler makes use of all the general routines previously mentioned along with some specialized routines to describe other operations. The basic idea behind the air handler simulation is to break all the calculations into small unit operations. A possible large set of equations is therefore simplified to a set of much easier calculations.

Figure 3-4 demonstrates the subroutine blocks for each of the unit operations. The starting point for the single-duct simulation, shown in the Figure 3-5 flow chart, is the building zone. This starting point is used since the ultimate goal of the control strategy is to meet a desired zone temperature set point. To start the simulation, the zone temperature is assumed to be set. Since the choice of any two psychrometric variables sets all others, the relative humidity, which must also be checked for set point constraints, is assumed as the search variable in a numerical analysis search. Given a set of load conditions, a mass and energy balance is performed around the zone to determine the entering air conditions using routine AIRSM.

The next calculation is made with routine DUCTEX to determine the amount of heat transferred between the supply air duct and return air area. DUCTEX uses a heat exchanger approach and requires the user to

*Unit Operations Blocks Enclosed by Dashed Lines

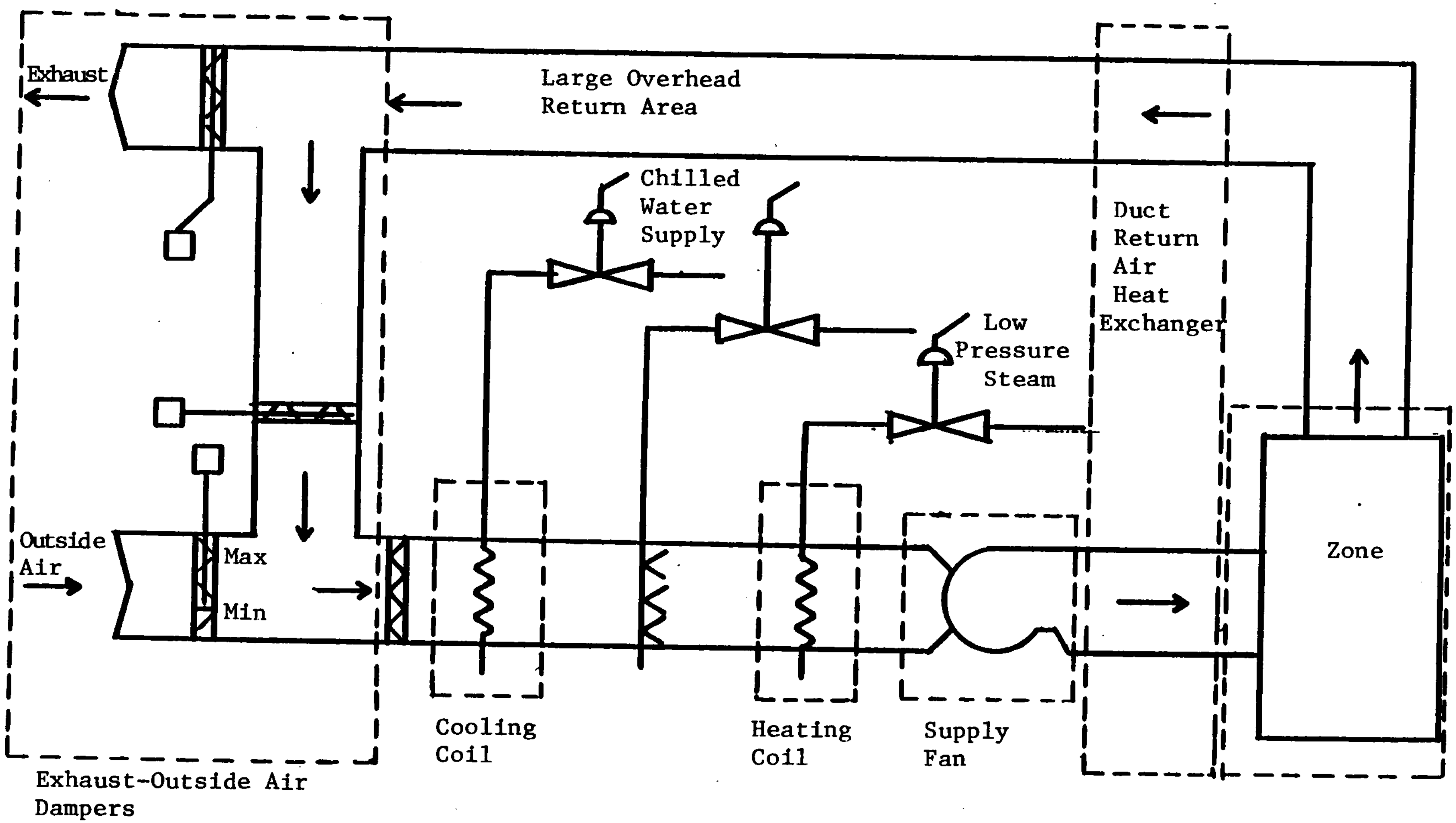


Figure 3-4. Single-duct air handler unit operations* that are represented in program subroutines.

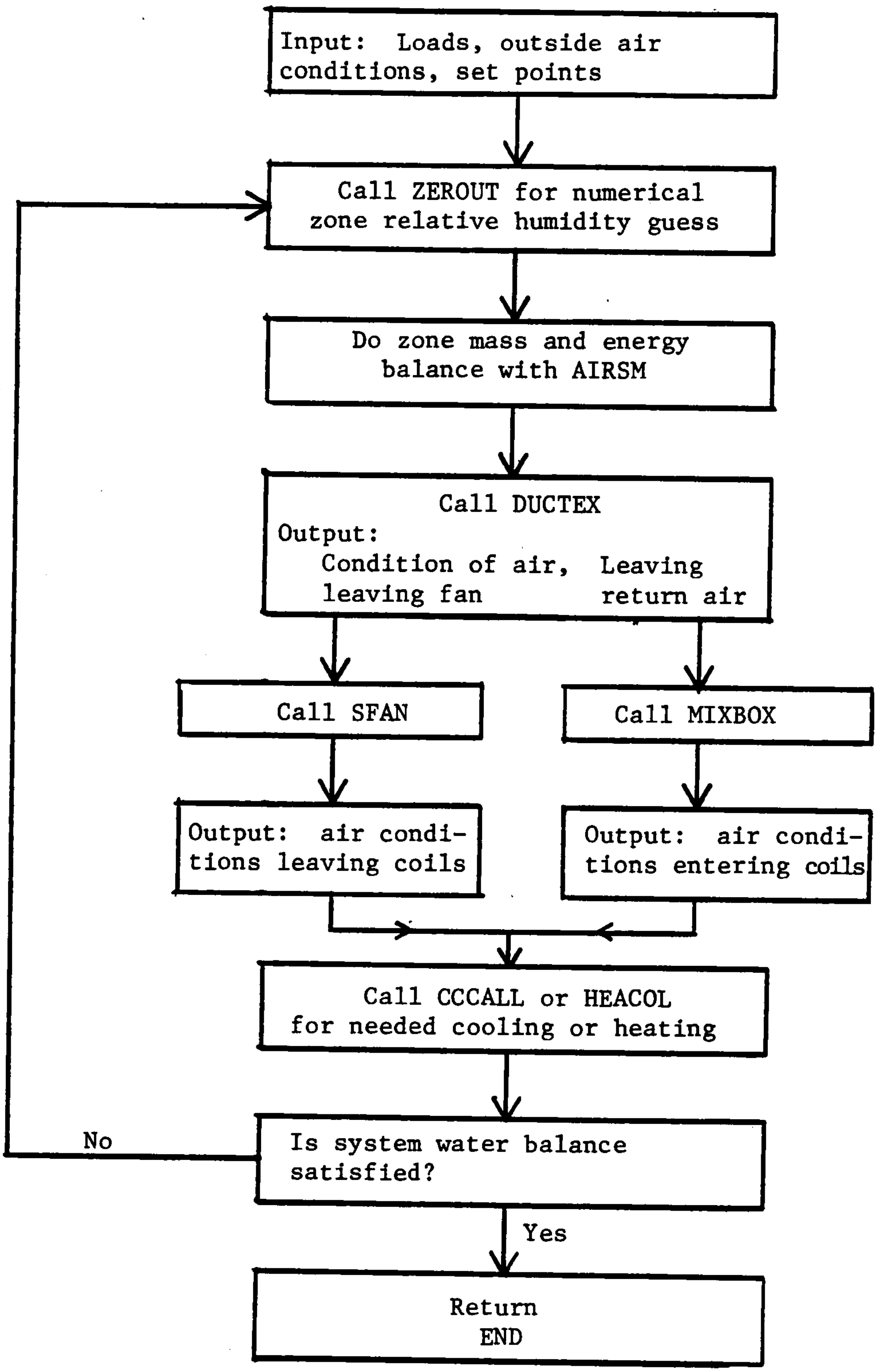


Figure 3-5. Single-duct air handler simulation flow chart.

supply a heat transfer coefficient between the supply and return air. Knowing the conditions of air entering the return air area and leaving the supply duct, DUCTEX gives the condition of air leaving the supply fan and the condition of air available to recycle or exhaust.

The state of the air entering the supply fan is set by applying routine SFAN to the air leaving the supply fan. Parallel to this the condition of air leaving the mixing box is determined by using routine MIXBOX knowing the return air and outside air conditions. MIXBOX applies the given mixing box strategy to the exhaust, return, and outside air dampers. For the original strategy this means a set point of 60°F in the winter and full return air plus minimum outside air during the summer operation.

The temperature and humidity are now known for air entering the cooling and heating coil sequence and for air leaving this sequence. If necessary the cooling coil or heating coil routine is called to lower or raise the air temperature across the coils.

The set of calculations from the zone to coils is performed until a zone relative humidity is found such that the overall air handler water balance described by Equation 3.25 is satisfied.

$$WOD+WLZ-WEX-WDEH=0 \tag{3.25}$$

where,

WOD = water gained through outside air damper

WLZ = water load on zone

WEX = water lost through exhaust damper

WDEH = water lost by dehumidification across cooling coils

Smith (32) uses the secant method proposed by Forsythe (33) in writing subroutine ZEROUT for a single variable search. ZEROUT is used for all single variable searches in these HVAC simulations. In this search ZEROUT is used to find a value of relative humidity that will allow Equation 3.25 to be satisfied. It is important to remember that convergence is normally a serious problem in many HVAC programs as noted by ASHRAE (11). However, no convergence problems are found in single variable searches ran here with ZEROUT.

Since all dry air mass balances and all energy balances are correct at each given unit operation, convergence of Equation 3.25 for the water system means the entire system is balanced. Testing of the single-duct simulation proves this to be correct. When Equation 3.25 is satisfied the temperature and relative humidity leaving the coils, calculated from the zone to the coils in the direction of air movement, are equal to the temperature and relative humidity entering the supply fan, calculated from the zone to the fan, opposite the direction of air movement.

The procedure for applying the original control strategy on top of the air handler simulation is given in flow chart form in Figure 3-6. It involves a stepwise set point check to determine the proper operation. When chilled water is required to achieve the high summer set point or steam is required to achieve the low winter set point, the control strategy is satisfied. It is also satisfied if the system can remain between these set points without use of steam or chilled water. Note that it is necessary to check the relative humidity constraint and apply the appropriate cool and reheat action if dehumidification is needed.

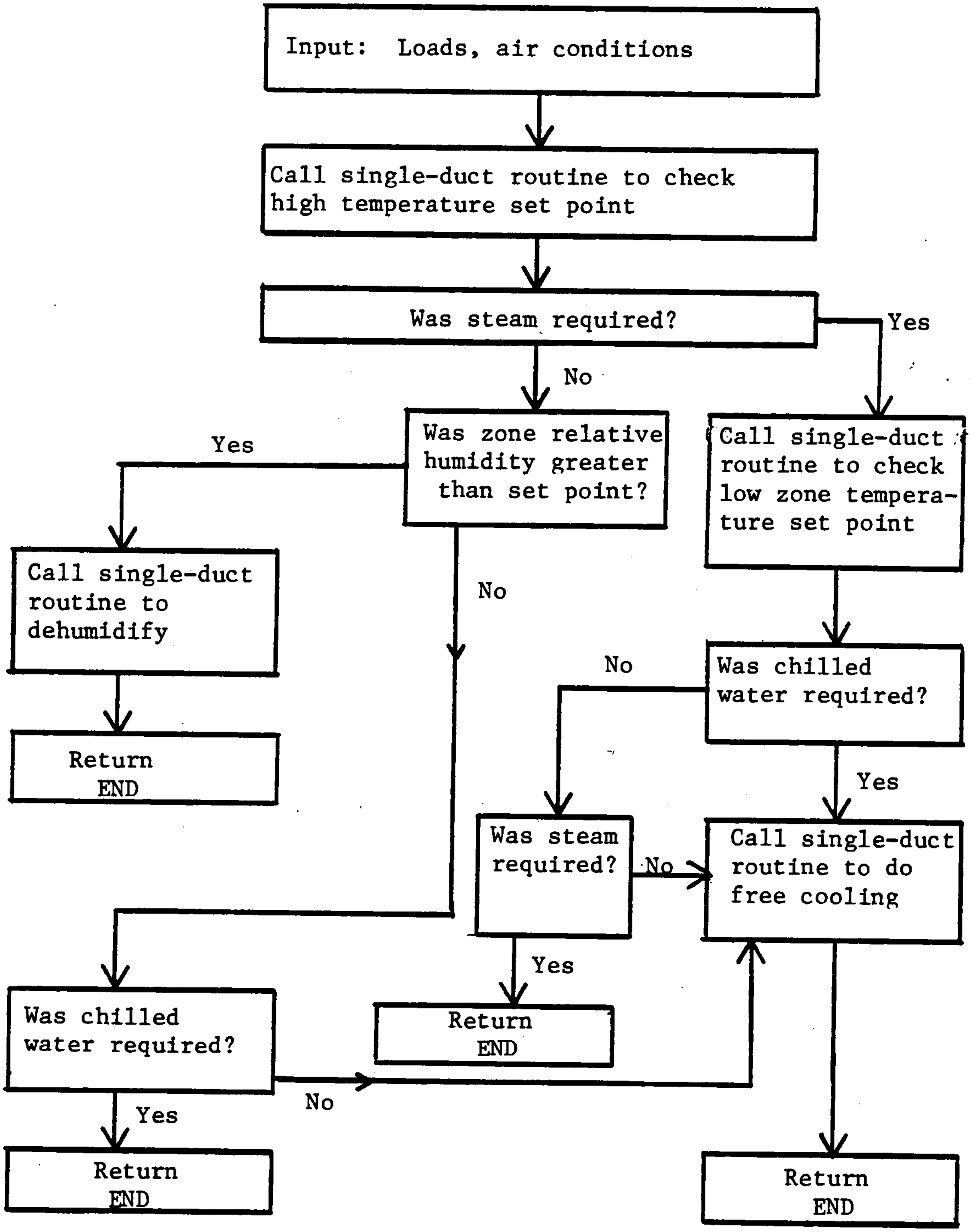


Figure 3-6. Single-duct air handler flow chart to describe original control strategy.

Dual-Duct Air Handler

As with the single-duct air handler simulation, the dual-duct is best presented in terms of unit operations. Figure 3-7 presents the dual-duct unit operations which are treated as subroutines in the simulation. Due to the multiple zones, the zone set point conditions can not be used as the starting point in the numerical search as is done for the single-duct air handler. The starting point for the dual-duct simulation, shown in Figure 3-8, is at the air leaving the supply fan, to allow searching about a common point for the air before splitting it into the hot and cold ducts. Since there is no set point for air at this point, both temperature and relative humidity leaving the supply fan must be searched to guarantee system convergence. The choice of this location as the simulation starting point does have an advantage over the single-duct simulation in that it has much more flexibility at system constraints. Set points may exist, but since the simulation does not start by assuming they are met, as in the single-duct case, constraints may be applied without the simulation search collapsing. The air leaving the supply fan passes through the ducts, zones, return air area, mixing box and fans. Mass and energy balances are performed at each of these locations. When the temperature and relative humidity leaving the supply fan after the balances are equal to the original guesses, the balances are converged. The two variable search routine used to solve for the temperature and relative humidity leaving the fan is ZSCNT borrowed from the IMSL Library routines (33).

Before any cooling coil or heating coil calculations can be performed on the air leaving the supply fans, the portion of air which

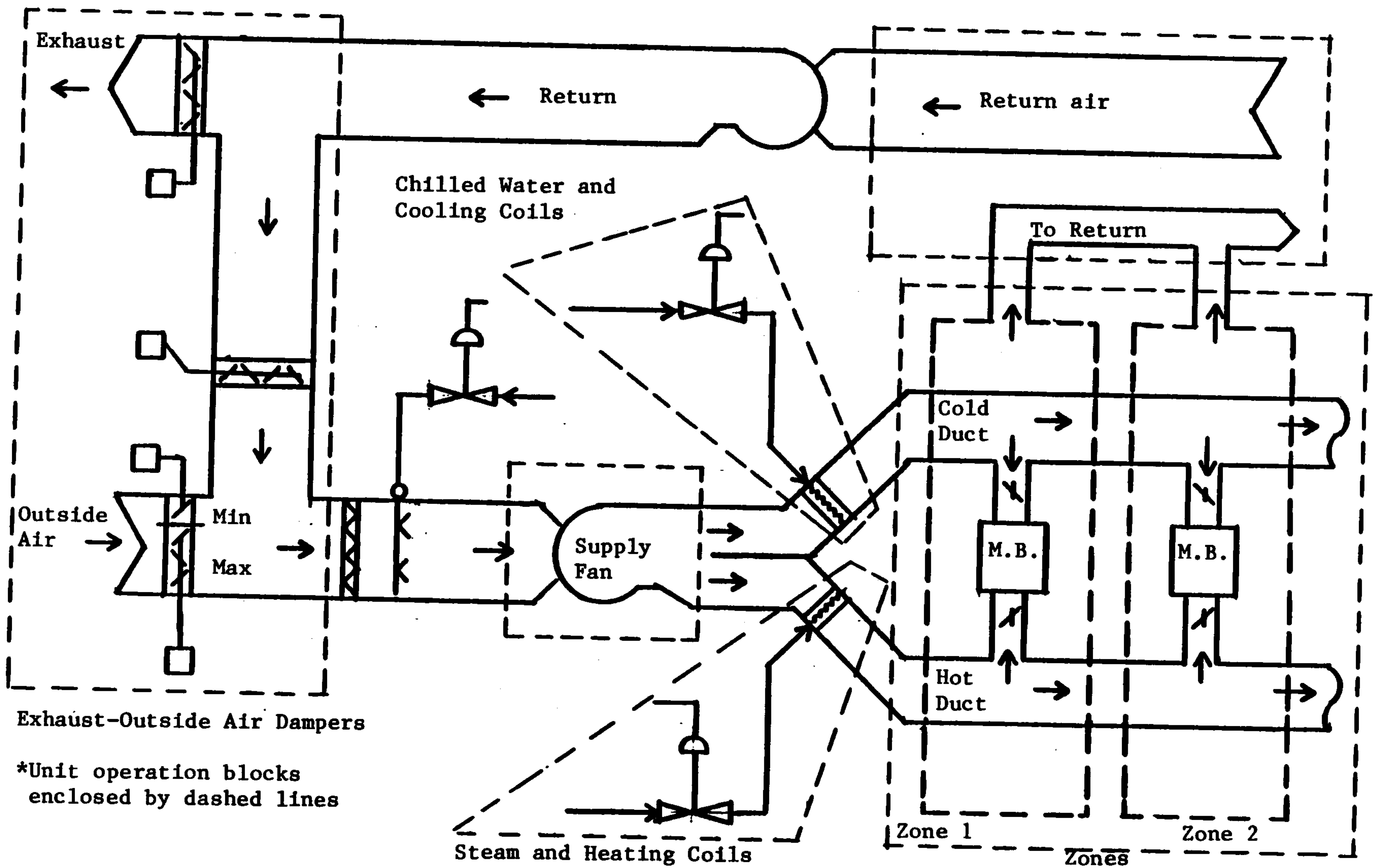


Figure 3-7. Dual-duct air handler unit operations* that are represented in program subroutines.

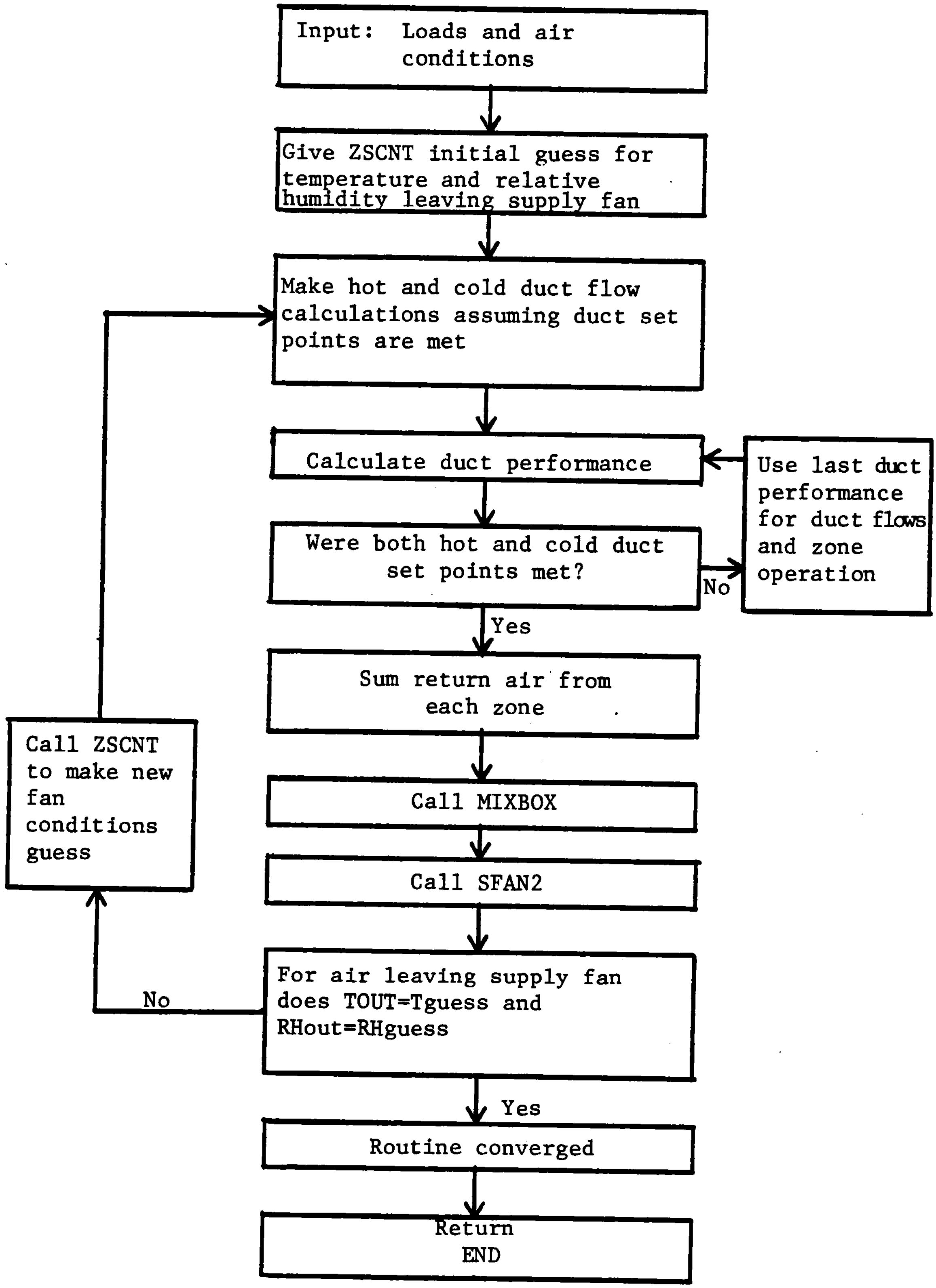


Figure 3-8. Dual-duct simulation flow chart.

flows down each duct must be determined. For a first pass guess, it is assumed that both ducts contain air at the provided duct temperature set points. Mixing box calculations are then performed by routine ZTMXBX to determine the appropriate mix of hot duct air and cold duct air to meet each zone thermostatic set point. This determines the amount of air flowing through each duct. Heating and cooling coil calculations are then made to determine if the hot and cold duct temperature set points can be met. If system physical limits prevent these set point from being achieved, the ZTMXBX calculations are done again using the new calculated values of hot and cold duct temperature. This iterative procedure is done until the temperature values calculated agree with those used in the zone mixing box calculations.

Air flows from each of the zones and is combined by routine AIRSM. It them flows through the return air duct, where it is either exhausted or returned to the system according to the control strategy contained in MIXBOX, which is similar to that used for the single-duct calculations. The mixed air is then pulled through the supply fan, where the heat gained is calculated by routine SFAN2. As stated previously, the temperature and relative humidity of the air leaving the supply fan are then checked to see if they match the initial guesses, giving system convergence.

The original control strategy is easily applied to the dual-duct system simulation. The cold duct temperature set point is constant, and the hot duct temperature set point is calculated as a function of outside air temperature. The original air handler mixing box strategy has a constant mixed temperature set point if the outside air is below 62°F, and allows only minimum outside air if the outside air is above 62°F.

Perimeter System

Simulation of the perimeter radiation heating system involves setting up a model for the heat transfer process which exists in the zone air, and a model for the heat exchanger and three-way valves which supply the appropriate hot water to the pipes at the perimeter faces.

The model chosen to represent the heat transfer process which takes place in the zone air is a one-dimensional steady-state moddel with a mixing term to represent heat transfer from the wall to the perimeter bulk air phase, and a lumped transfer coefficient for heat transfer between the outside air and air at the wall plane. The model could have been more complicated with a three-dimensional analysis with convection terms and radiation terms to represent the flow of heat into the room bulk air phase. A temperature profile would have been found not only perpendicular to the wall plane, as done in the one-dimensional analysis, but in the vertical and horizontal directions of the plane parallel to the wall. However, the goal of this model is to provide a representative process with which the original control strategy can be tested. The simpler model provides this representative process. In fact, it is realized before any simulations are ran that the less complications that are seen in this model, the better a feed-forward strategy, such as the original strategy, will appear.

The one-dimensional representation used for the perimeter air heat transfer is shown in Figure 3-9. The model assumes that values are known for To, T2, UW, and Km, where,

To = temperature of the outside air

T2 = bulk phase room temperature

UW = lumped transfer coefficient from outside air through the perimeter wall to the pipe plane

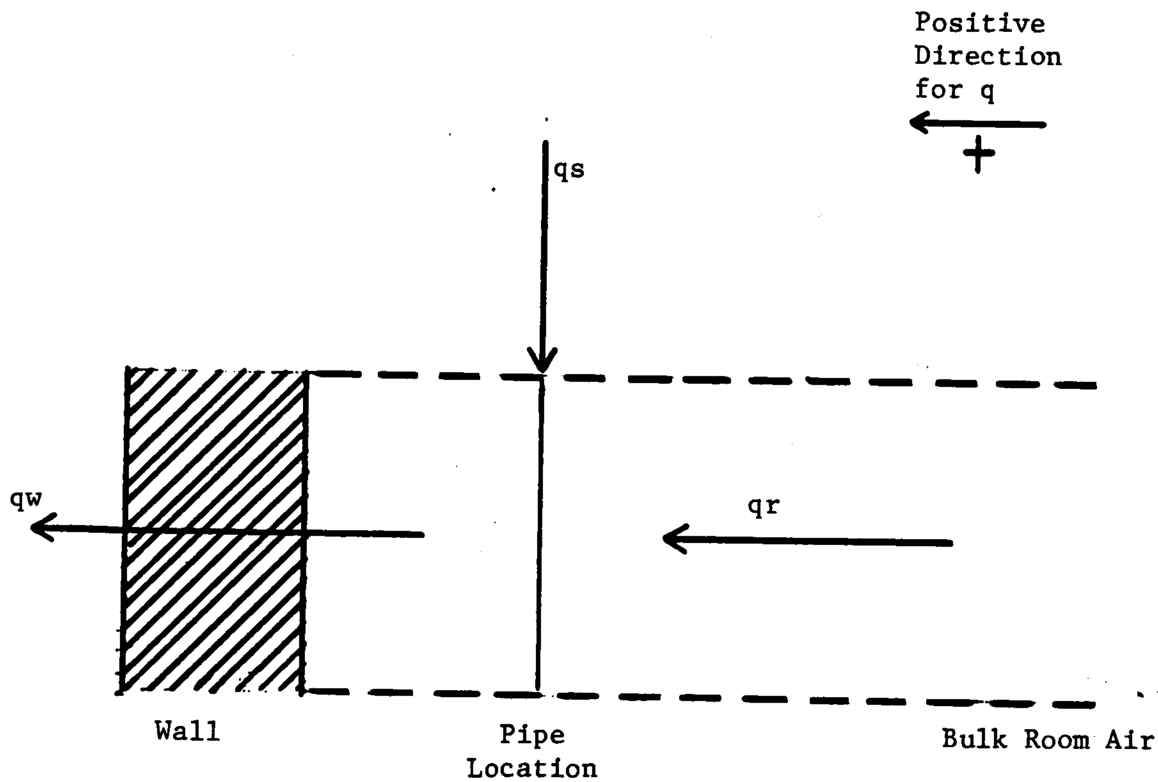


Figure 3-9. One-dimensional representation of wall, pipe and room for perimeter air heat transfer.

Km = coefficient describing the heat transfer which takes place by room mixing

The unknowns for the model on a per square foot of perimeter wall basis are T1, qw, qr, and qs, where,

T1 = perimeter temperature measured in the pipe plane

qw = heat transferred between air at pipe plane and outside air

qr = heat transferred between air at pipe plane and bulk room air by mixing

qs = heat gained from the perimeter pipe at pipe plane

The heat that is gained from the perimeter pipe is assumed to enter as a point source at the pipe. It is also assumed that this model holds for any line perpendicular to the perimeter plane. Thus, when expanded to the entire room, the heat gained from the pipe uniformally distributed over the plane of the pipe parallel to the perimeter wall. Equations 3.26, 3.27, and 3.28 are written to describe this system where heat flowing from the inside to the outside is considered positive.

$$qw = -\ UW(To-T1) \qquad (3.26)$$

$$qr = -\ Km(T1-T2) \qquad (3.27)$$

Solving these three equations for T1 gives Equation 3.29.

$$T1 = \frac{UW(TO)+Km(T2)+qs}{Km+UW} \qquad (3.29)$$

Since the purpose of the perimeter system is to provide a compensating heat source for heat loss through the wall, the value of T1, the wall temperature, and qs, the heat input to achieve T1, are the most important variables to be considered. Equation 3.29 for T1 requires knowing the value of qs. Kern (34) graphically gives the combined convection and radiation heat transfer coefficients for various sized horizontal pipes to air at 70°F as a function of the temperature difference between the pipe fluid and the air. The temperature range present in the perimeter system gives a linear relation for the heat transfer coefficient. For the one inch pipe used in these simulations the heat transfer coefficient from Kern is given by Equation 3.30.

$$hp=2.08+.005(\overline{T}p-T1) \tag{3.30}$$

where,

hp = pipe transfer coefficient in BTU/hr-ft^2-°F

$\overline{T}p$ = average water temperature in pipe

where $\overline{T}p$ is given by Equation 3.31.

$$\overline{T}p = \frac{(Tw_i+Tw_o)}{2} \tag{3.31}$$

where,

Tw_i = supply water temperature

Tw_o = water temperature at end of room length

For Equation 3.31 Tw_i is given by the control strategy being applied, but Tw_o must be calculated by energy balance according to Equation 3.32.

$$Tw_o = \frac{(FW)(Cp)(Tw_i)-Qs}{(FW)(Cp)} \tag{3.32}$$

where,

Fw = water flow rate

Cp = water heat capacity

Qs = heat transfer from pipe for entire perimeter face

Qs is given by Equation 3.33.

$$Qs = (qs)(Lr)(Hr) \tag{3.33}$$

where,

Lr = perimeter wall and pipe length

Hr = perimeter wall height

Solution of Equations 3.29 to 3.33 requires another equation to find the value of qs for heat gained from the perimeter pipe on a per square foot of perimeter wall basis. Kern gives the relation for transfer from a pipe in the form of Equation 3.34.

$$qs = (hp)(\pi)(Dp)(\overline{T}p - T1) \tag{3.34}$$

where,

Dp = pipe diameter

A check of Equations 3-29 to 3-34 shows that these six equations contain the six unknowns Tp, Qs, qs, Tw_o, T1, and hp. These equations are solved analytically for the desired unknowns in subroutine PERIMZ to provide the model for the heat transfer process which exists in the zone.

The model to describe the heat exchanger and three-way valves requires only knowing Tw_i desired for each face according to the control strategy, and a resultant Tw_o after application of PERIMZ. (For this discussion an additional subscript N, S, E, or W will be added to distinguish the perimeter face.) To describe the heat required by the exchanger, the resultant temperature of the mixed water being returned from the faces must be known. This is given by Equation 3.35.

$$TWM = \frac{(FRN)(Tw_oN)+(FRS)(Tw_oS)+(FRE)(Tw_oE)+(FRW)(Tw_oW)}{FRN+FRS+FRE+FRW} \tag{3.35}$$

where,

TWM = return water temperature

FR = return water flow to heat exchanger from each face

Since the temperature of the water leaving the exchanger is the supply temperature for the north face, the heat exchanger load, QEX, is given by Equation 3.36.

$$QEX=(FRN+FRS+FRE+FRW)(Cp)(Tw_iN-TWM) \quad (3.36)$$

The steam flow, STFL, is given by Equation 3.37.

$$STFL = \frac{QEX}{STLH} \quad (3.37)$$

where,

STLH = latent heat of steam

The equations to model the south, east, and west face three-way valves are those seen in common recycle stream problems as shown in Figure 3-10. The question is how much water must be recycled to give the appropriate mixed water temperature, assuming values are known for F (where F = constant flow through perimeter pipe leaving three-way valve), Tw_iN, Qs, and Tw_iS (looking here at the south face recognizing that the east and west faces are similar).

Knowing the amount of heat lost through the pipe and the temperature drop encountered, the amount of supply water which is allowed through the three-way valve is given by Equation 3.38.

$$FRS = \frac{Qs}{(Cp)(Tw_iN-Tw_oS)} \quad (3.38)$$

The amount of water recycled, FREC, is then given by Equation 3.39.

$$FRECS=F-FRS \quad (3.39)$$

Applying the original control strategy to the perimeter air heat transfer model, and the exchanger and three-way valve models requires only the calculation of Tw_iN, Tw_iS, Tw_iE, and Tw_iN as a function of outside air temperature. Thus given the outside air temperature, the

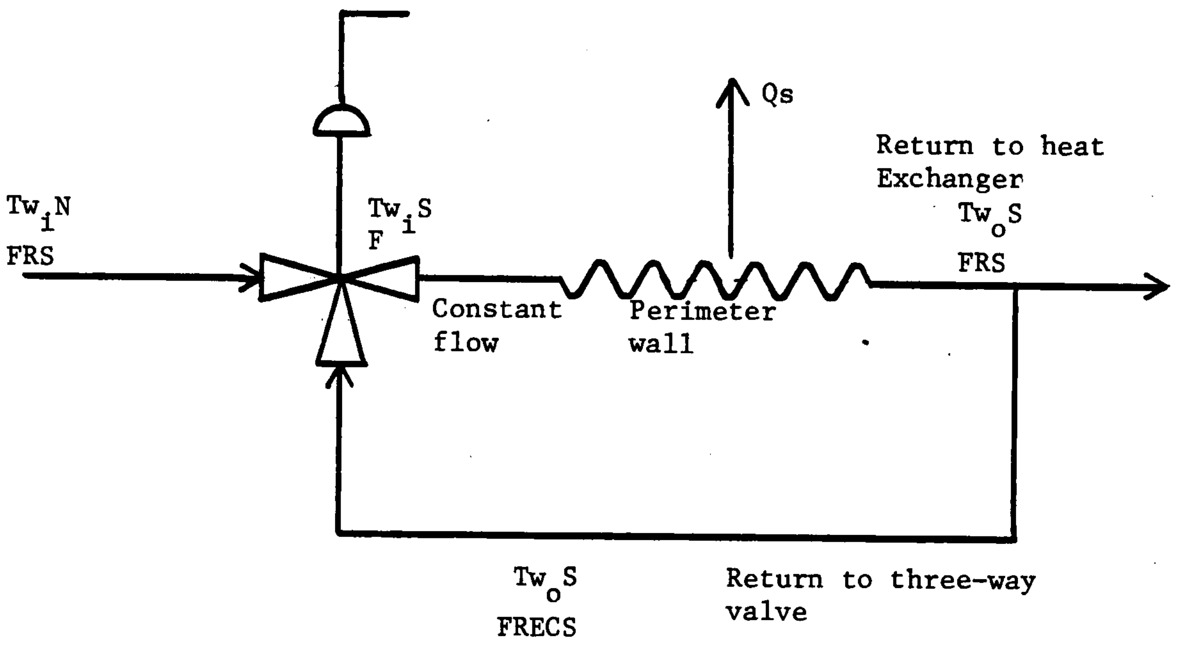


Figure 3-10. Perimeter system south face recycle flow diagram.

bulk room temperature, the room dimensions, the pipe size, the total water flow rate, and the lumped wall transfer coefficient, these perimeter models will calculate the heat transfer from the pipe, perimeter wall temperature, heat exchanger load, steam flow, and three-way valve recycle requirements.

CHAPTER IV

SINGLE-DUCT STUDIES AND RESULTS

The first HVAC system studied is the single-duct air handler. This air handler serves a single zone, enabling it to do all heating or cooling in a single duct to provide the desired zone temperature and relative humidity. Steady-state simulations are used to investigate a problem with the original mixing box operation, and to test a possible improvement. Another study looks at the problem of determining the optimum reheat set point for the dehumidification operation. The single-duct simulation uses the numerical search routine ZEROUT to converge the overall water balance, given heat and water loads and the temperature and relative humidity of the outside air.

The following load values are estimations in keeping with suggested literature values (2, 13, 25, 28, 29). System sizing is based on values given by Keane (29) and Oglesby (28). The simulations use a constant air mass flow rate of 70,000 lbs/hr through the supply fan. The zone is considered square with an area of 19,500 square feet, a 12 foot high ceiling, and only one wall exposed to the outside. The transfer coefficient through this wall to the outside air is assumed to be 0.1 BTU/ft^2-hr-°F, with a working perimeter system eliminating this heat transfer below an outside air temperature of 66°F. There are 78 people in the zone, assuming 1 person for every 250 square feet of zone space, with each person contributing 250 BTU/hr of sensible heat and .241 lbs/hr of water vapor. Equipment loads are computed at 100 watts/person and electric lighting at 2 watts/ft^2. A transfer coefficient through the roof to the return air is assumed to be 0.15 BTU/ft^2-hr-°F, with the area of the roof equal to the zone area.

For the air handler, the maximum chilled water flow rate is 110,124 lbs/hr at a temperature of 44°F, and the maximum steam rate is 850 lbs/hr at a pressure of 25 psig. The simulation supply fan is rated at 18.5 HP with an efficiency of 70%.

For both the single-duct air handler, and later the dual-duct air handler, yearly energy consumption is calculated by the "Bins" method suggested by Spielvogel (18). This involves making hourly energy calculations at 5°F increment temperatures and multiplying by the number of hours per year to get the yearly energy consumption assuming 24 hours/day operation. ASHRAE (25) provides the "Bins" data distribution for Albany, N.Y., shown in Figure 4-1, with temperatures in 5°F increments from -18°F to 97°F. This data will be used for all yearly consumption calculations.

Base Case Studies

Steady-state simulation of the single-duct air handler requires the assumption that the original pneumatic control strategy can meet its set points within physical system limitations. The original control strategy simulation cools the air when the zone temperature is above 78°F and heats the air when the zone temperature is below 66°F. This strategy provides feedback of the temperature of the single zone being served to set the desired temperature of the duct.

One of the suspected problems with the original single-duct air handler strategy involves the summer/winter switch. This manual switch has the greatest effect on the mixing box strategy. In the winter

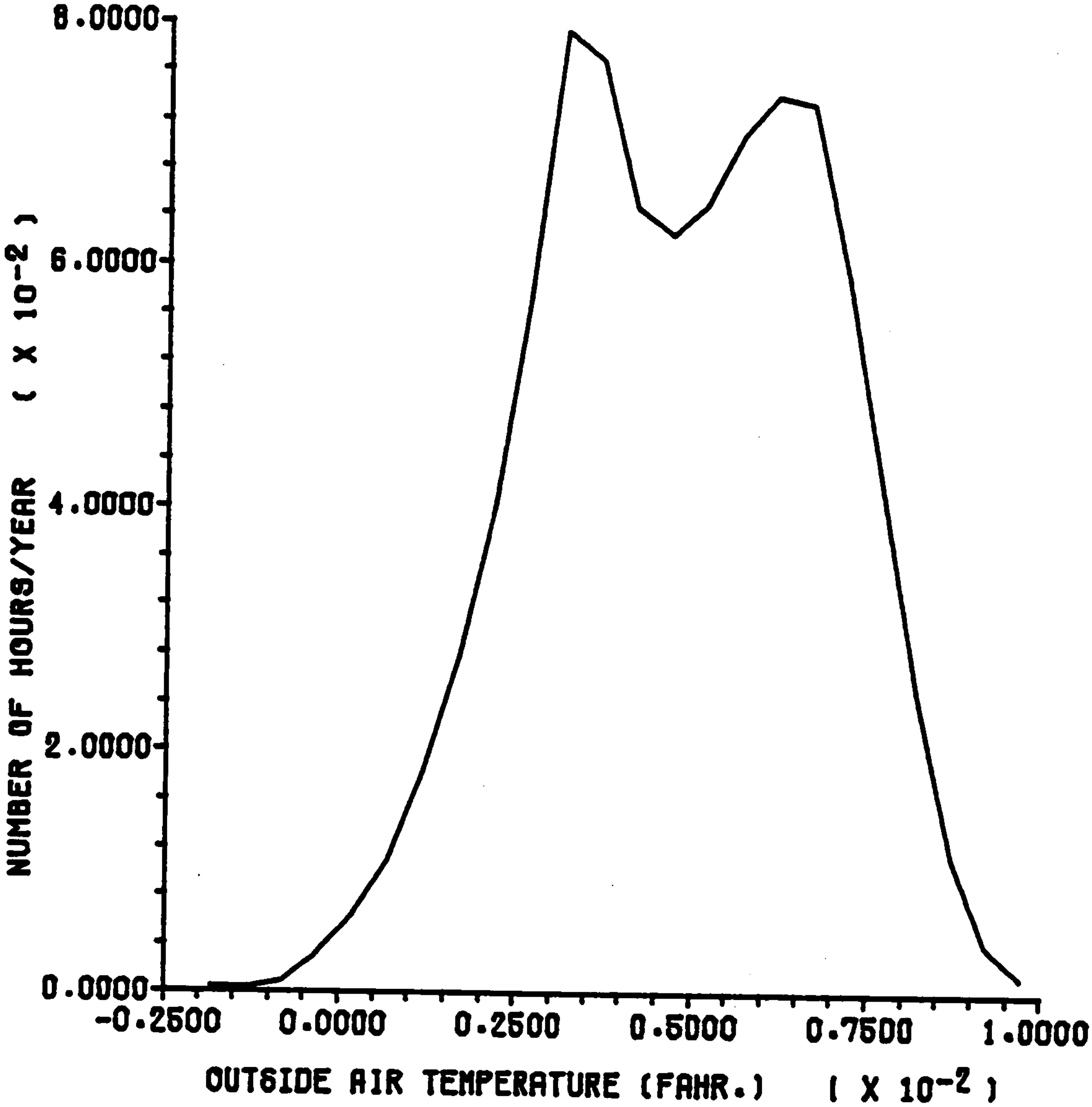


Figure 4-1. Bins data for Albany N.Y. giving the number of hours per year that 5°F increment temperatures occur from -18°F to 97°F.

position the control strategy allows the use of free cooling from the outside air by meeting a mixing box set point of 60°F. However, in summer use the mixing box control strategy provides only minimum outside air. It is not known what criteria is used to determine the timing for the summer/winter switchover. The base case results are expected to be very dependent on the division of summer and winter hours. To test this summer/winter switch, the simulation is run at three base case conditions, covering the two expected extremes of this changeover timing, and an intermediate case. Case 1 involves running in the winter mode for all hours seen in the "Bins" data at or below 47°F, and in the summer mode for all hours at or above 52°F. The 5°F gap from 47°F to 52°F is due to the 5°F increment for temperatures in the "Bins" data. Case 2 uses the winter mode for all hours at or below 67°F and the summer mode for all hours at or above 72°F. Case 3 runs the simulations at a more expected crossover distribution of summer/winter hours, as seen in Figure 4-2.

The hourly steam requirement for each of the simulation temperatures is shown in Figure 4-3. Note that the steam requirement goes to zero at 18°F as the perimeter system and zone loads provide all heating, and is therefore nonexistant in the temperature range for summer/winter switchover cases. Also, since the switchover involves only a change in mixing box strategy, which is used for free cooling, the steam curve for Figure 4-3 is not altered, and therefore applies to all three cases. By multiplying the steam BTU/hr curve by the "Bins" distribution curve for hrs/yr, the BTU/yr curve shown in Figure 4-4 is calculated. Figure 4-4 shows that the temperature ranges of highest yearly consumption, which should be concentrated on for conservation

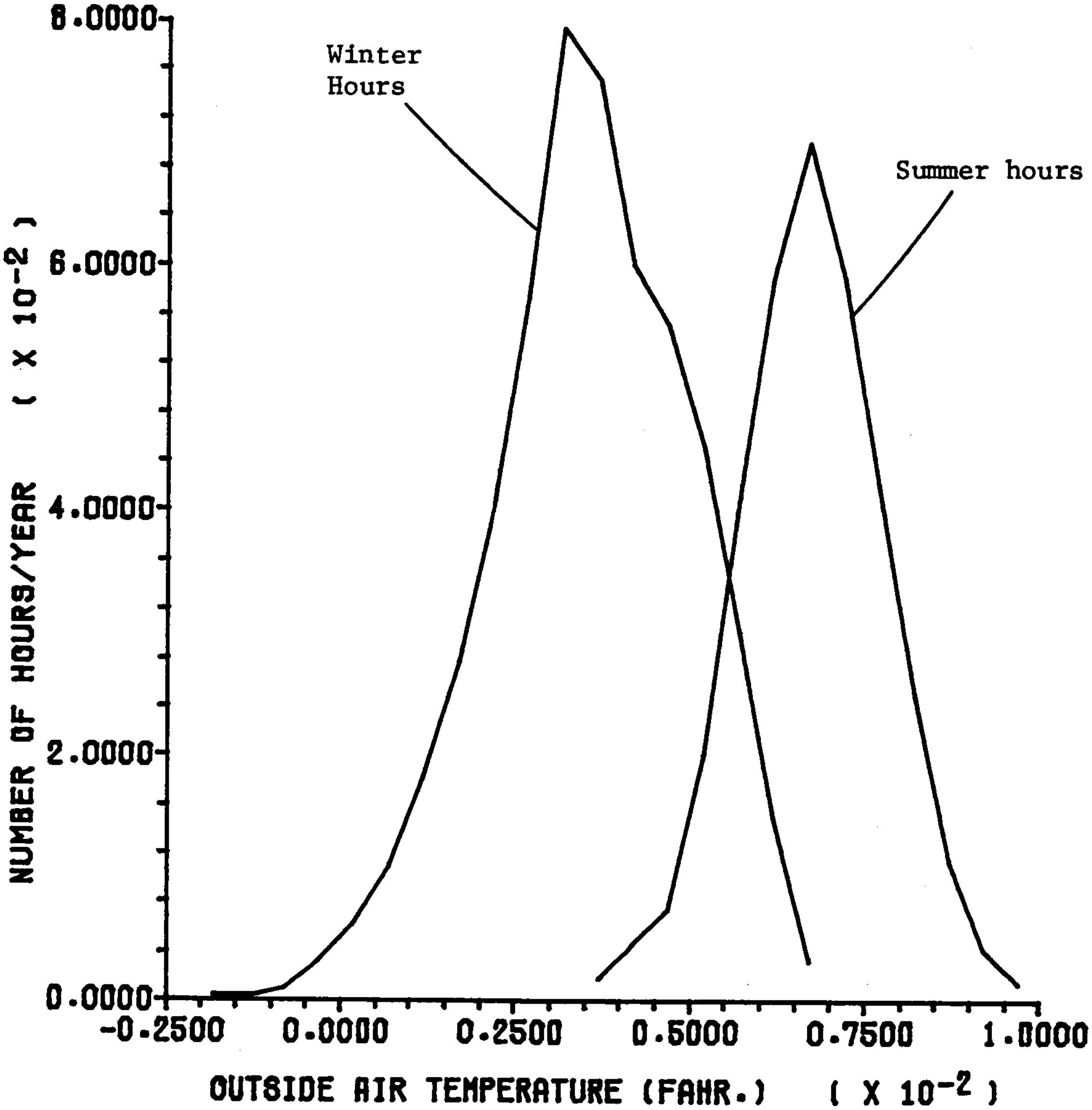


Figure 4-2. Overlapping summer/winter "Bins" distribution for base case 3.

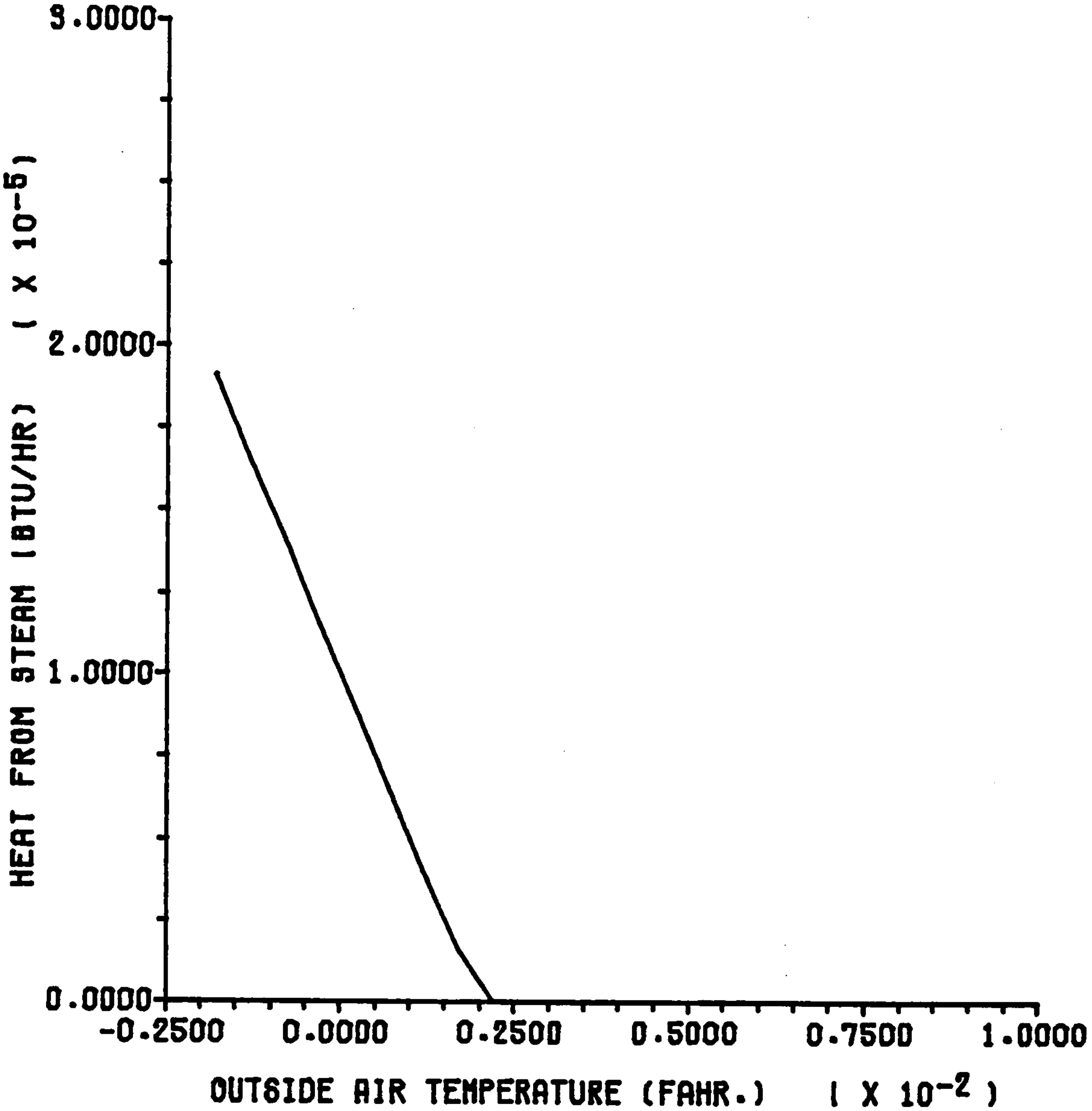


Figure 4-3. Single-duct hourly steam use for base cases 1,2, and 3.

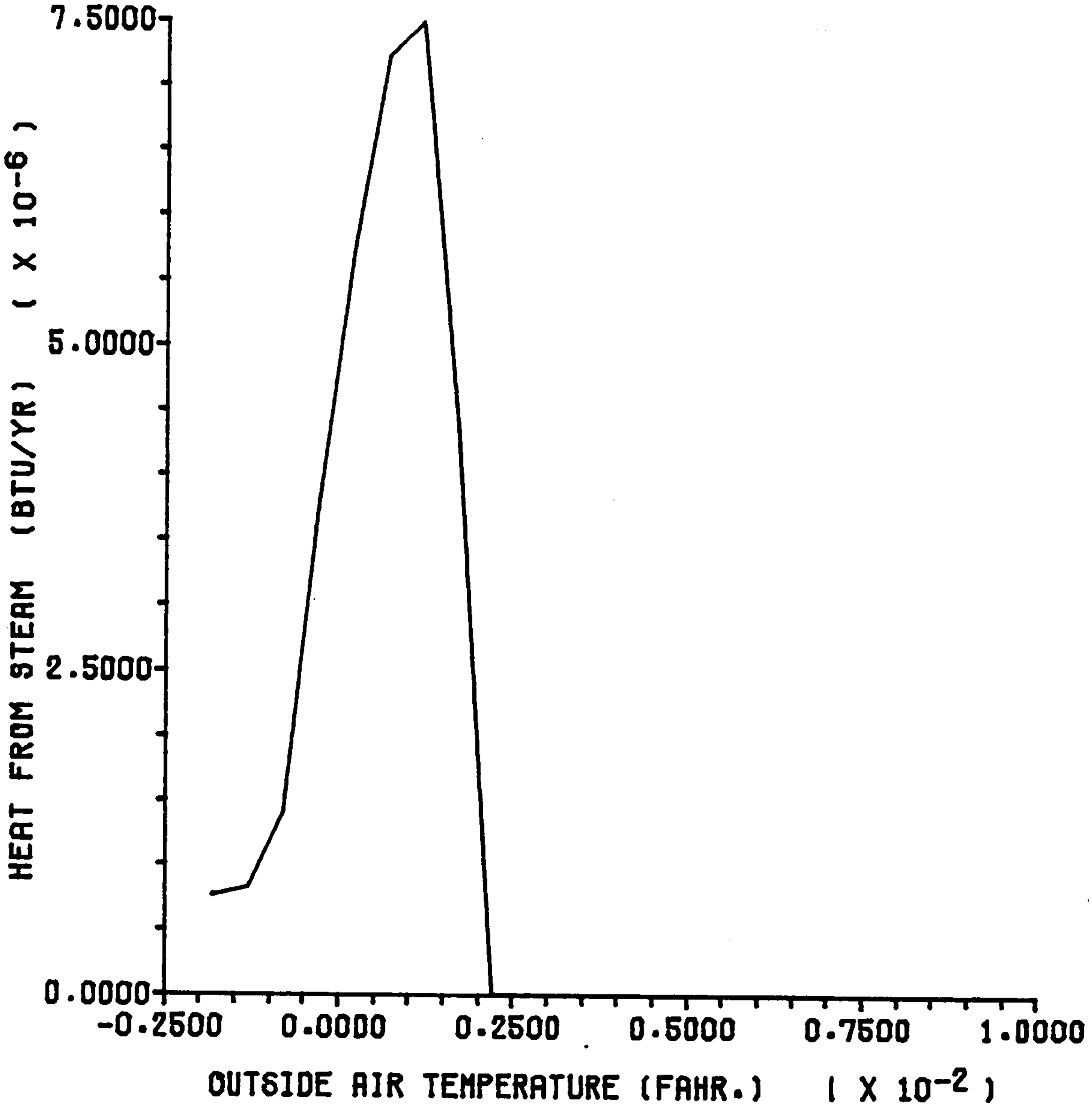


Figure 4-4. Single-duct yearly steam use for base cases 1, 2, and 3.

measures, are not necessarily the same as the high hourly consumption areas.

The curves for hourly chilled water energy required are given in Figures 4-5 to 4-7 for the three respective switchover cases. A look at Figure 4-5 shows that due to the loss of free cooling, an early switch to the summer mode causes a chilled water load down to 52°F, where the switch is made. Comparing to Figure 4-6 shows that switching to the summer mode at a higher temperature of 72°F demonstrates a large difference in chilled water load for the two modes. Even in the free cooling winter mode, some chilled water is required at 67°F. However, it is much less than the cooling required for the summer mode of operation seen at that temperature in Figure 4-5. A look at Figure 4-7 shows the result of overlapping the switchover as in case 3. This graphic is most important in the 37°F to 67°F overlap region. In this region, even with identical loads, the free cooling mode requires cooling energy only at the higher temperature part of its distribution, while the summer mode requires cooling at the lowest temperature end of its distribution at 37°F.

The yearly cooling energy consumptions for these three cases are compared in Figure 4-8. These curves show that for these particular zone loads and outside air conditions, the higher temperature is best for summer/winter switchoever. Due to the splitting of hours for its distribution, the case 3 cooling load is less than the low temperature case 1 in the 46°F to 67°F region. They have nearly the same hourly loads in this region, but have different hours/year distributions. A comparison of total cooling and heating energy consumption for these three cases is given in Table 4-1.

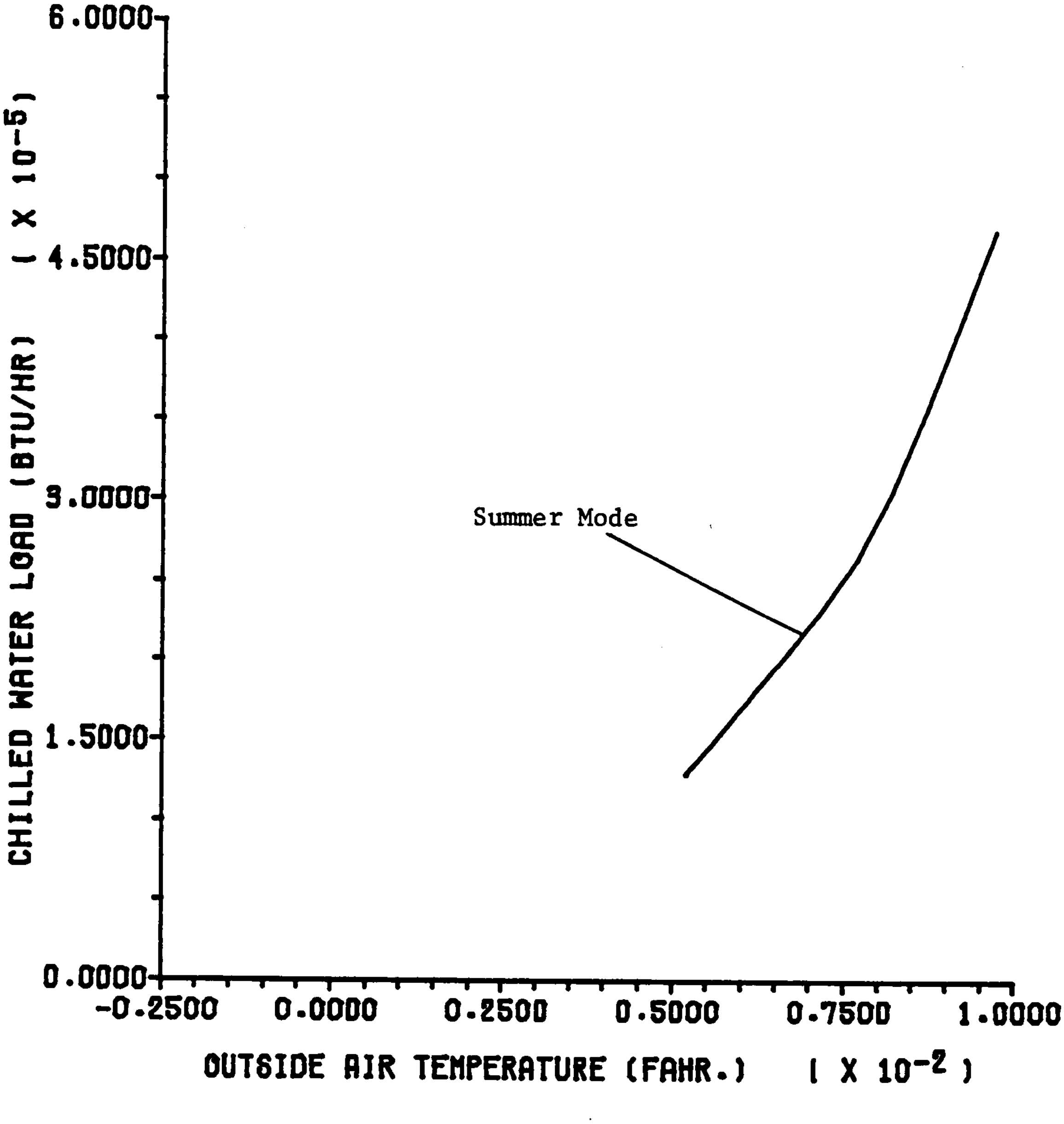


Figure 4-5. Single-duct base case 1 hourly chilled water load.

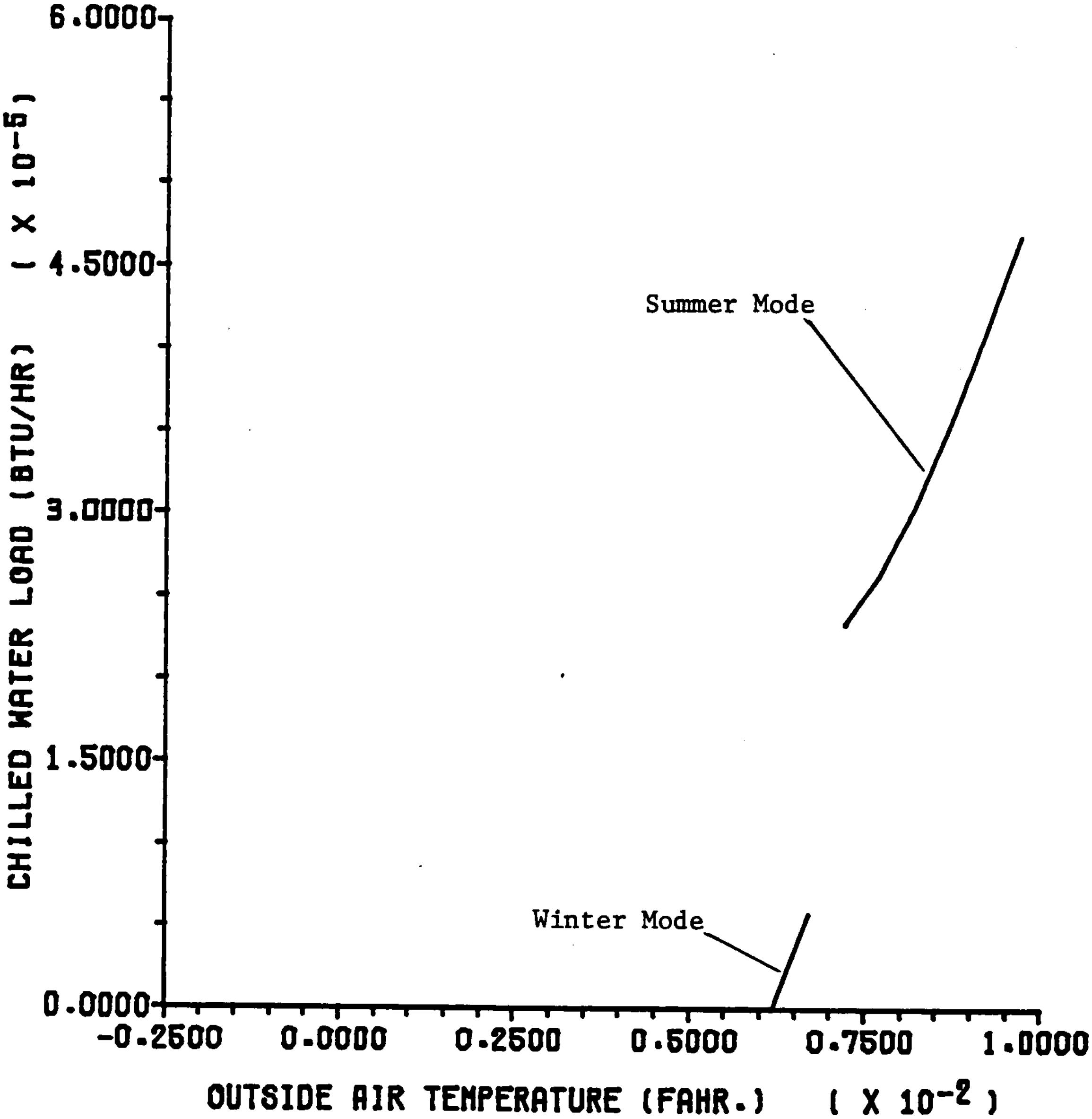


Figure 4-6. Single-duct base case 2 hourly chilled water load.

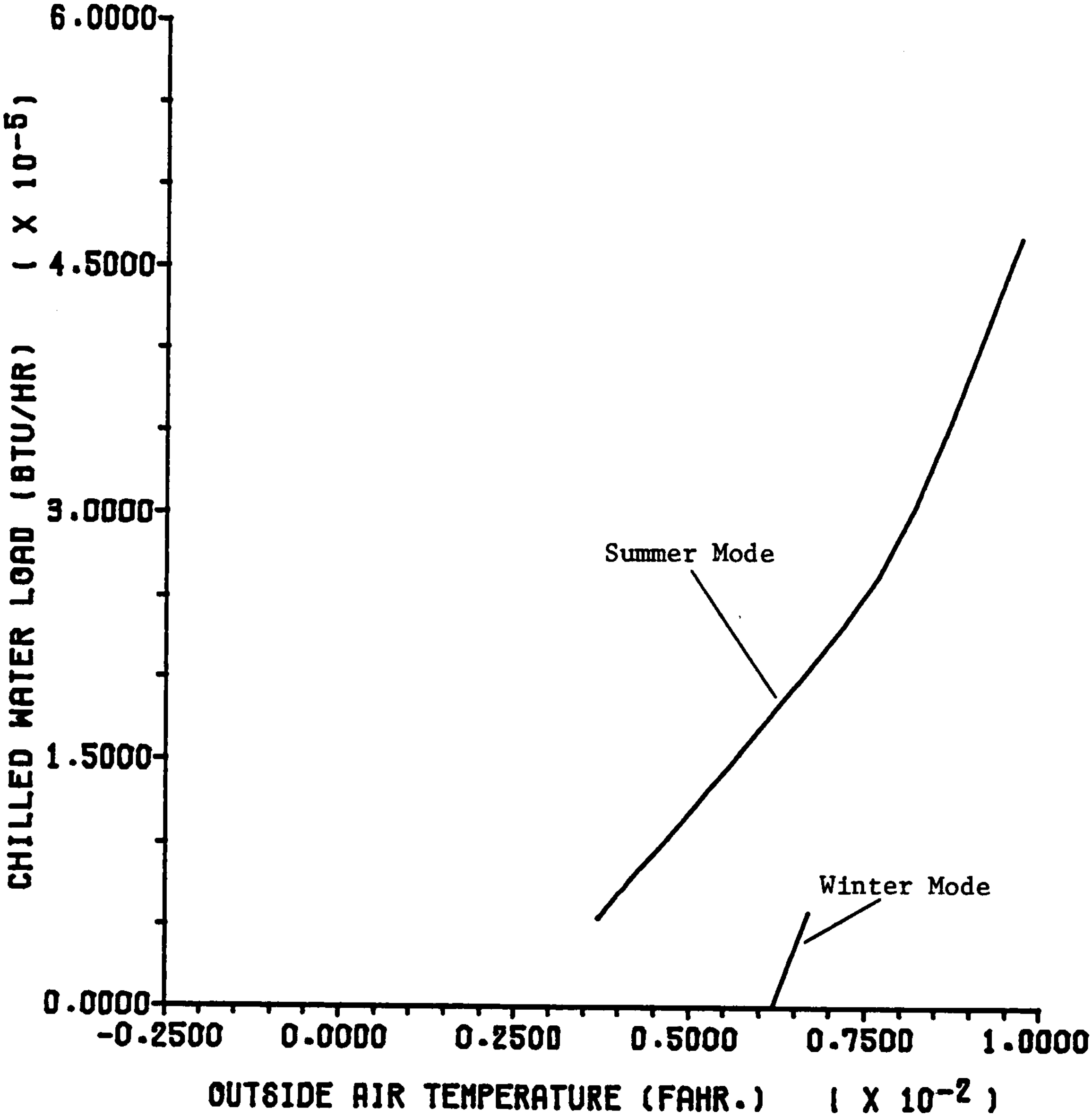


Figure 4-7. Single duct base case 3 hourly chilled water load.

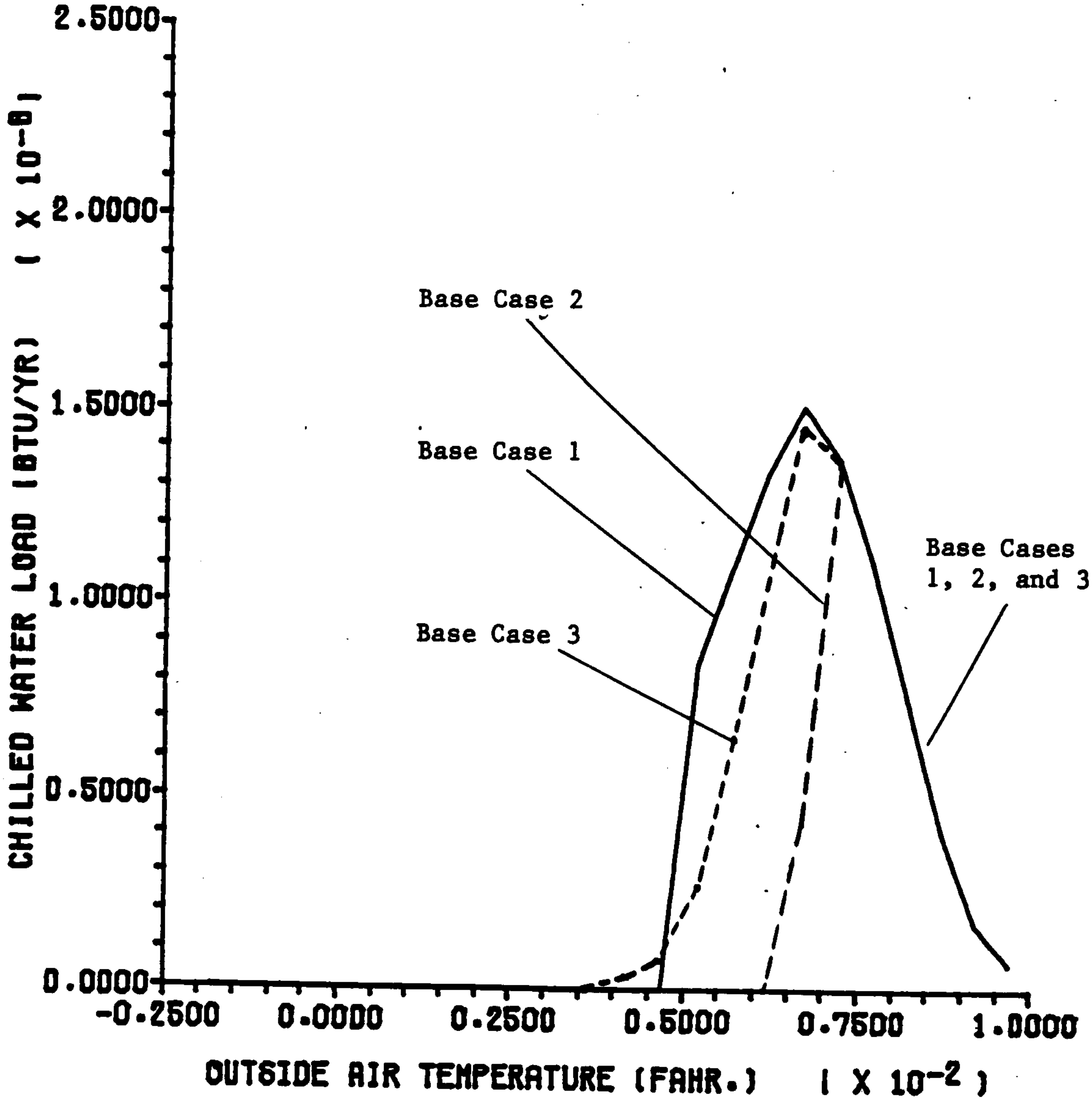


Figure 4-8. Single-duct yearly chilled water load for base cases 1, 2, and 3.

TABLE 4-1

COOLING AND HEATING LOAD FOR THE THREE SUMMER/WINTER SWITCHOVER DISTRIBUTIONS OF BASE CASE CONTROL STRATEGY

Case	BTU/hr Cooling Load	BTU/hr Heating Load
1	8.68×10^8	3.15×10^7
2	3.90×10^8 (45%)*	3.15×10^7 (100%)*
3	7.42×10^8 (85%)*	3.15×10^7 (100%)*

*Relative to Base Case 1

This table shows that timing of the summer/winter switchover can have dramatic impact on chilled water requirements.

Steam and Chilled Water Strategy Change

The original single-duct strategy is a pneumatic system, with efficiency limitations due to the summer/winter switch. This switch can be eliminated entirely with the installation of a digital computer based control strategy which would also eliminate the need for any set point changes.

The suggested cascaded digital computer control strategy for the steam loop is given in Figure 4-9. This strategy sets the set point for the temperature in the duct by a feedback controller on the zone temperature. If the temperature of the zone is above 66°F, the strategy will call for a temperature leaving the supply fan that is lower than the existing air temperature, thus shutting the steam valve. If the zone temperature goes to 66°F, this loop will force the steam valve to open to prevent the zone temperature from dropping below 66°F.

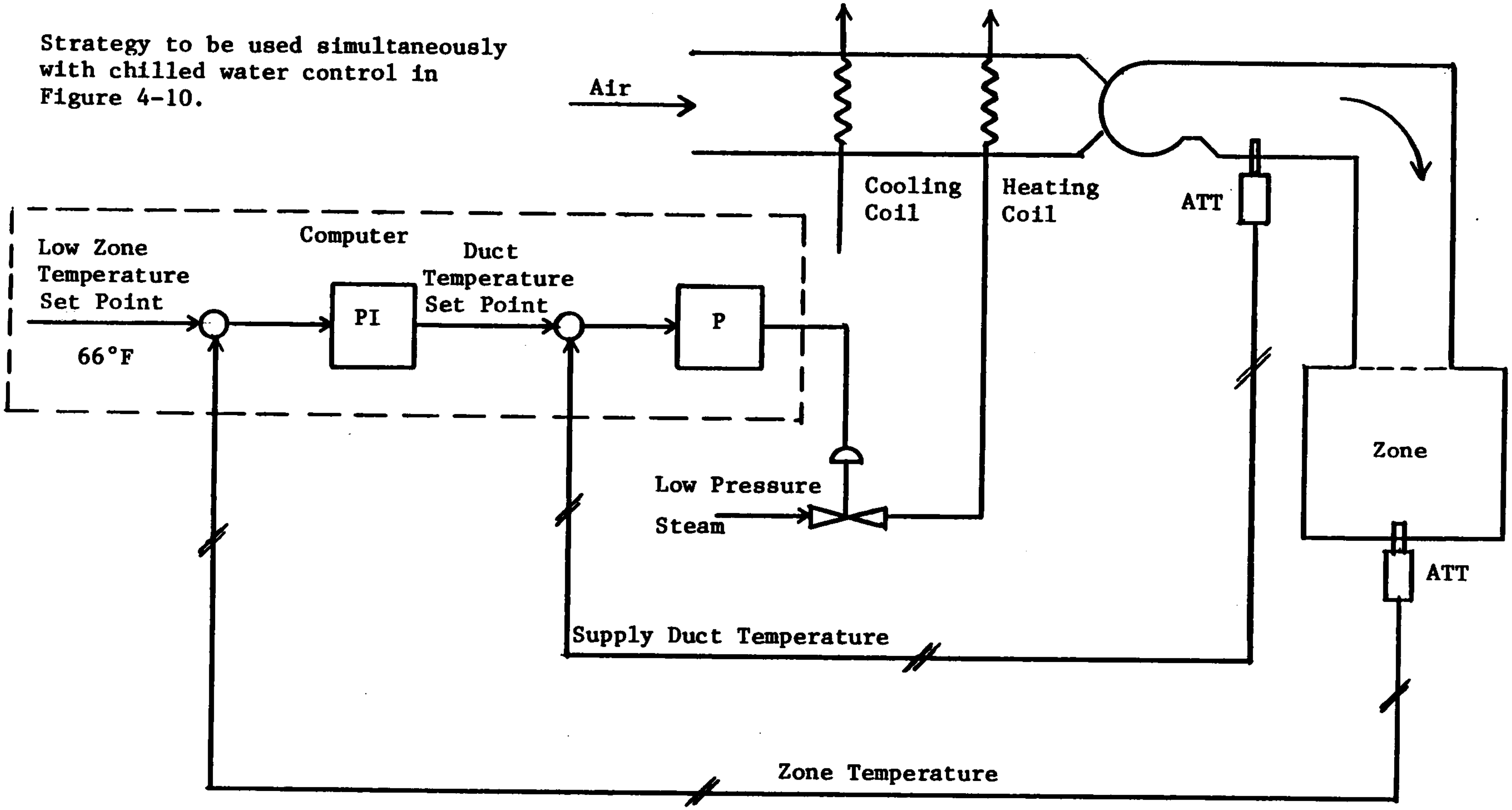


Figure 4-9. Proposed all year single-duct steam loop control.

The suggested strategy shown in Figure 4-10 for the cold duct loop is slightly more complex. The outermost loop on this strategy feeds back the zone relative humidity. If the zone relative humidity is less than 65%, the output of this outer controller is limited to a zone temperature set point of 78°F. If the zone relative humidity goes to 65%, the zone temperature set point is lowered, thus requiring additional cooling, which in turn dehumidifies the air. The process of reducing the relative humidity also requires reheating the air as discussed in Figure 1-3 (pg. 15). Therefore, if the outer control loop in the chilled water control system is active, the zone temperature set point for the steam loop is changed from its normal value of 66°F to some higher reheat set point which is addressed below. Upon deactivation of this outer loop, this set point is returned to 66°F. If no dehumidification is required and the zone temperature is below 78°F, the chilled water valve will be closed. If the zone temperature reaches 78°F, the chilled water valve is controlled to meet the 78°F set point.

Because of the steady-state simulations used here it is not possible to get an adequate numerical comparison of the original pneumatic strategy and the suggested cascaded strategies. Both strategies work on the principle of heating only at a certain low zone temperature set point and cooling at a certain high temperature set point except at humidity constraints. However, comparisons, such as shown with the summer/winter switch for the mixing box, would indicate that the manual requirements for the original pneumatic system can be costly.

Mixing Box Change

The other suggested strategy change, which can be investigated numerically, involves the mixing box. The original strategy only

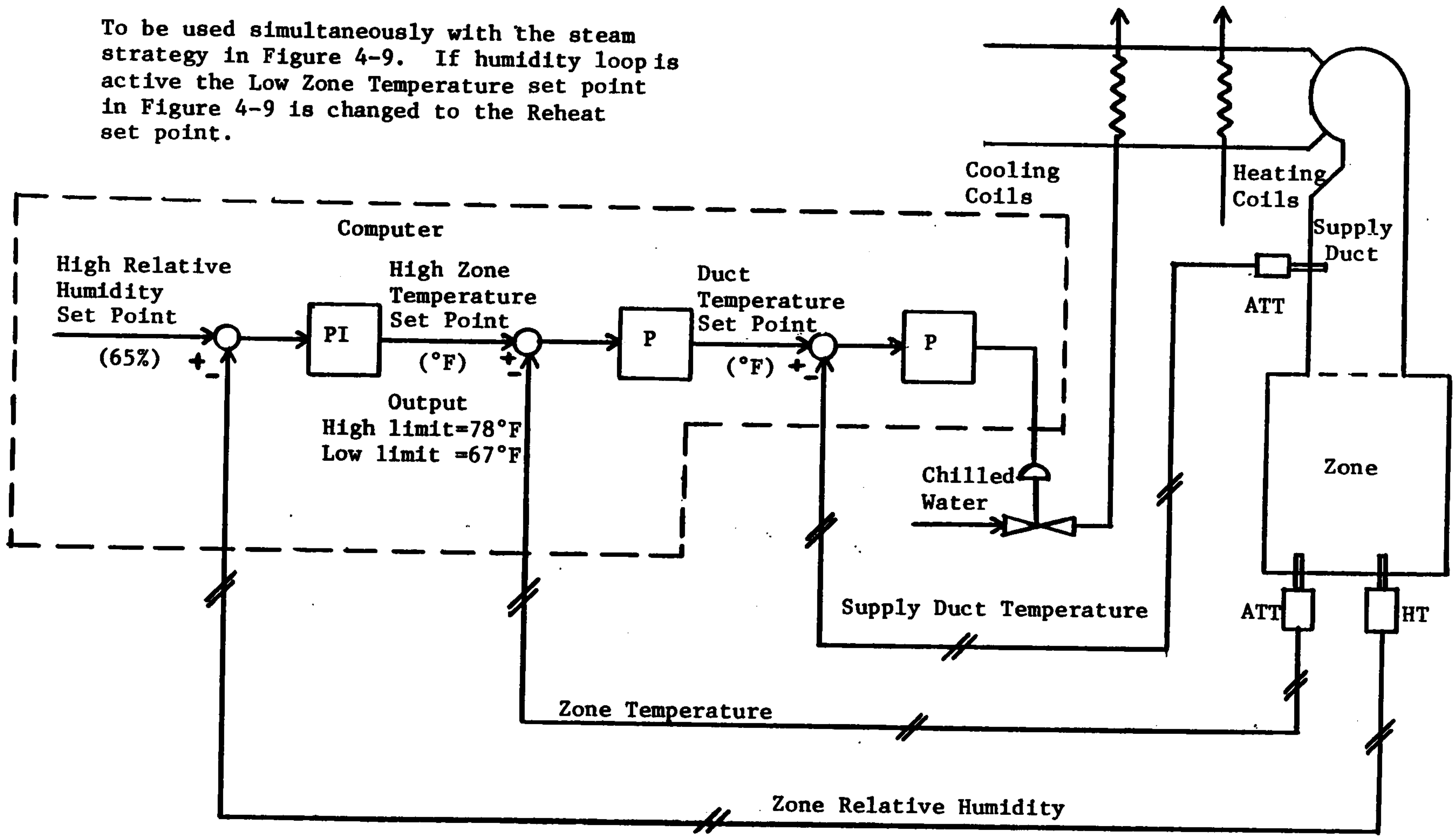


Figure 4-10. Proposed all year single-duct chilled water control loop.

allowed free cooling in the winter mode. It was shown that this could be very inefficient, with the suggestion that free cooling could be beneficial to higher outside air temperatures than allowed by the original strategy. This is true for the loads simulated, however, there are cases in this same temperature range where outside air is of poorer cooling quality than allowed by the original strategy. Figure 4-11 presents the suggested mixing box strategy change which says that outside air should be used for free cooling when it is of better quality for cooling than the return air. Since the psychrometric properties of air allow the possibility for the dry bulb temperature of one air stream to be lower than that for another stream, but still have a higher enthalpy due to water content, the condition for a better quality cooling air requires checking both dry bulb temperature and enthalpy. If the temperature and enthalpy of the outside air are less than the temperature and enthalpy of the return air, outside air may be used for free cooling. The mixing box set point is set by cascading from a zone temperature set point of 72°F. Above a zone temperature of 72°F all the free cooling possible will be used. Below 72°F no free cooling is used because it is not necessary.

The simulation of this mixing box strategy change at the same load conditions as the "Base Case" studies shows no change for the steam curve. This is because the mixing box change is only for free cooling. The Figure 4-12 comparison of the yearly chilled water energy requirement curve for the mixing box change with the best of the "Base Case" runs shows that an even further savings can be accomplished in the 67°F to 77°F range. Above 77°F the outside air is of poorer cooling quality than the return air and is only minimally used. The total yearly

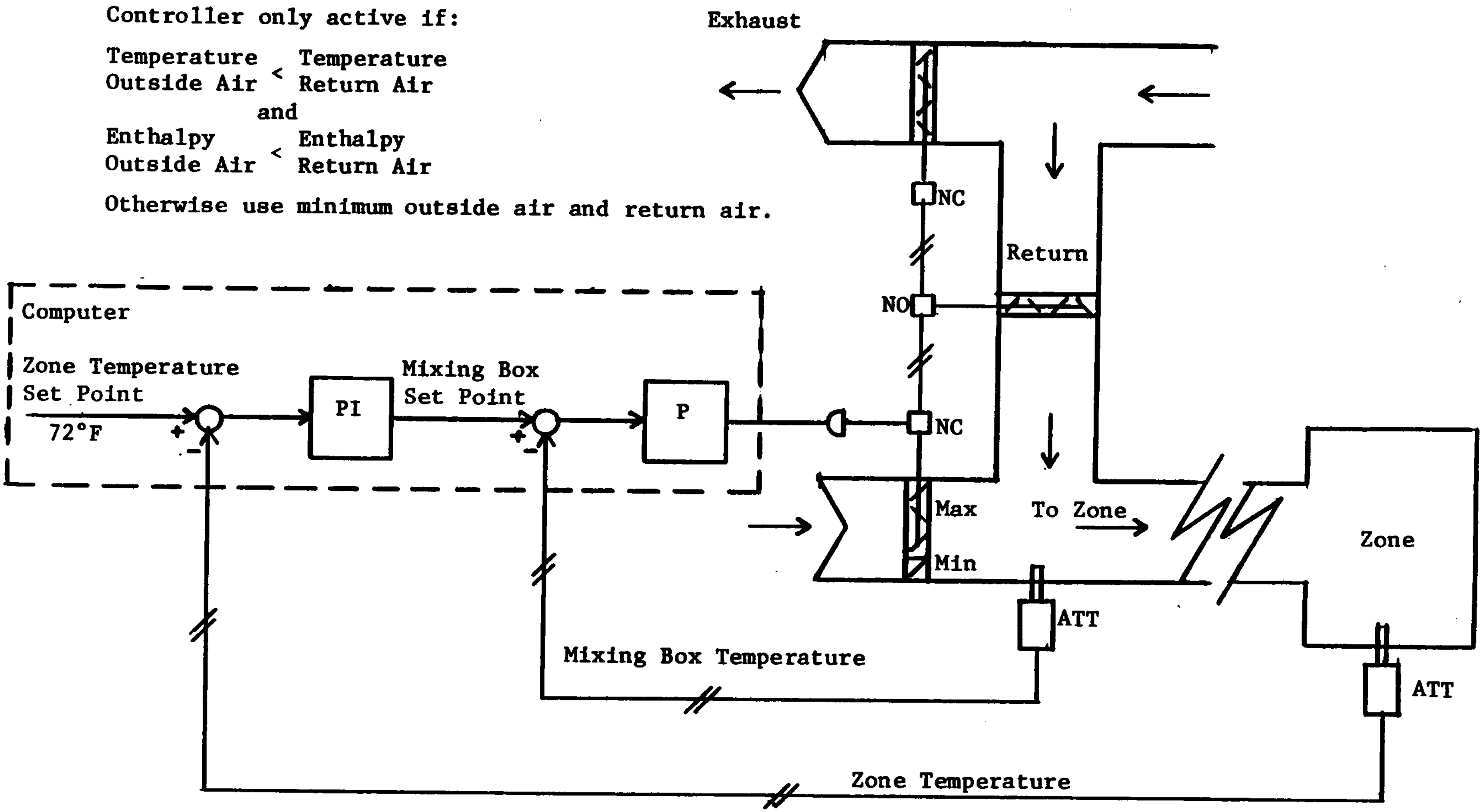


Figure 4-11. Proposed single-duct mixing box strategy.

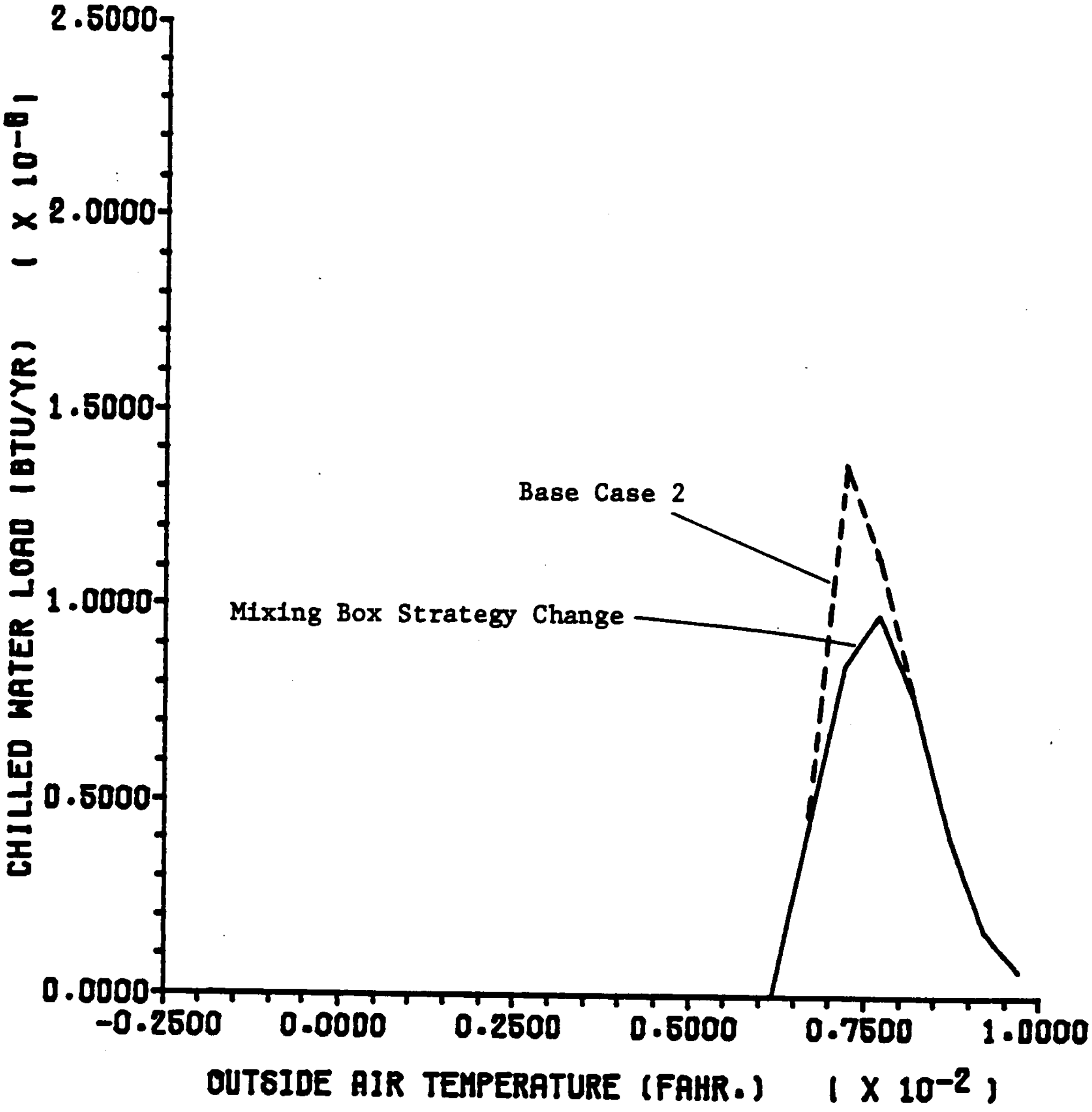


Figure 4-12. Single-duct yearly chilled water load for mixing box strategy change compared with base case 2.

chilled water requirement with the new mixing box strategy is 3.67×10^8 BTU/yr which is 94% of the cooling energy required for the best base case. The important factor for this strategy is that at all times free cooling from the outside air can be utilized, but outside air is only used when it provides an advantage.

Reheat Set Point Study

The final study made on the single-duct air handler involves choosing an appropriate reheat set point for the steam loop of the dehumidification strategy. As stated earlier, the cooling of air reduces the water content, or humidity, but increases the relative humidity. It is therefore necessary to add dry heat back to the air to decrease the relative humidity. The position of the reheat set point determines how much heat will be added back to the air, and in turn how much cooling must be done. For comparison of reheat set points, two outside temperatures with humidity problems are chosen. For an outside temperature of 80°F and 50% relative humidity, it is found that with a zone dry heat load equal to half the value used in the base case, a water load of 30 lbs/hr will require a very small amount of dehumidification to meet the 65% set point. A zone water load of 141.0 lbs/hr requires enough dehumidification to run the chilled water coils near maximum capacity. These two cases are studied along with a midrange load of 85.0 lbs/hr. To investigate the reheat set point, the above loads are used while varying the reheat set point in 0.5°F increments from 70°F to 77.5°F, and hourly cooling and steam requirements calculated. The reason no higher set point is investigated, is that the normal cooling set point is 78°F. If the reheat set point is at or above this, a possible fighting of cooling and heating loops exists.

The Figure 4-13 cooling load curves show that for all three cases as reheat set point is increased, the cooling load decreases. The heating requirements to enable dehumidification are shown in Figure 4-14. For the lowest water load case there is a slight increase for the heating requirements, but for the higher water load cases, the heating requirements are actually reduced as reheat set point is increased. This slight reduction is due to the nonlinear relationship between dry bulb temperature and relative humidity seen on the psychrometric chart. At these high water load cases, the optimum reheat set point would have to be 77.5%F for the range investigated, because it represents the lowest energy needs for both heating and cooling. Analysis of the low load case shows that going from a 70°F reheat set point to 77.5°F reheat set point decreased the cooling load from 2.67×10^5 BTU/hr to 2.16×10^5 BTU/hr, while the steam load increased from 2.3×10^4 BTU/hr to 2.47×10^4 BTU/hr. Since these lines appear linear, a simple cost analysis can be used to show that a BTU of steam would have to cost more than 30 times as much as a BTU of chilled water for any reheat set point other than the highest of 77.5°F to be economical. This is unreasonable, therefore for the given conditions, the optimum set point would be 77.5°F.

For verification, this reheat study is also performed with an outside air temperature of 95°F and relative humidity of 50%. The three water loads used are 1.0 lb/hr, 50.0 lbs/hr, and 102.0 lbs/hr, with 102.0 lbs/hr pushing the chilled water to a maximum flow. For this study, the trends are the same as for the 80°F case, with both chilled water and steam decreasing as reheat set point is increased for the two high water load cases, indicating an optimum of 77.5°F. For the

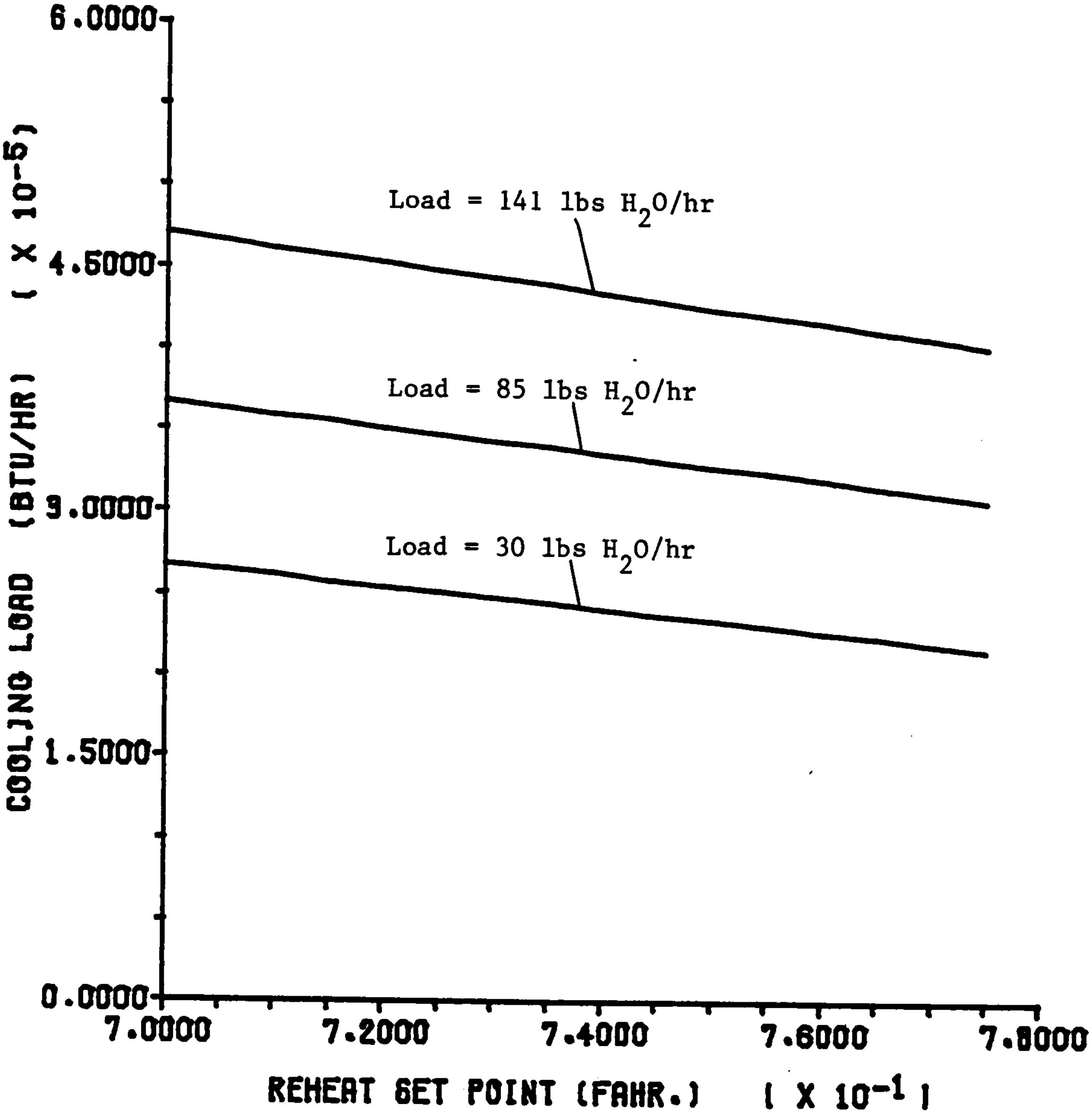


Figure 4-13. Single-duct cooling requirements for dehumidification at 85°F outside air temperature for a range of zone water loads at varying reheat set points.

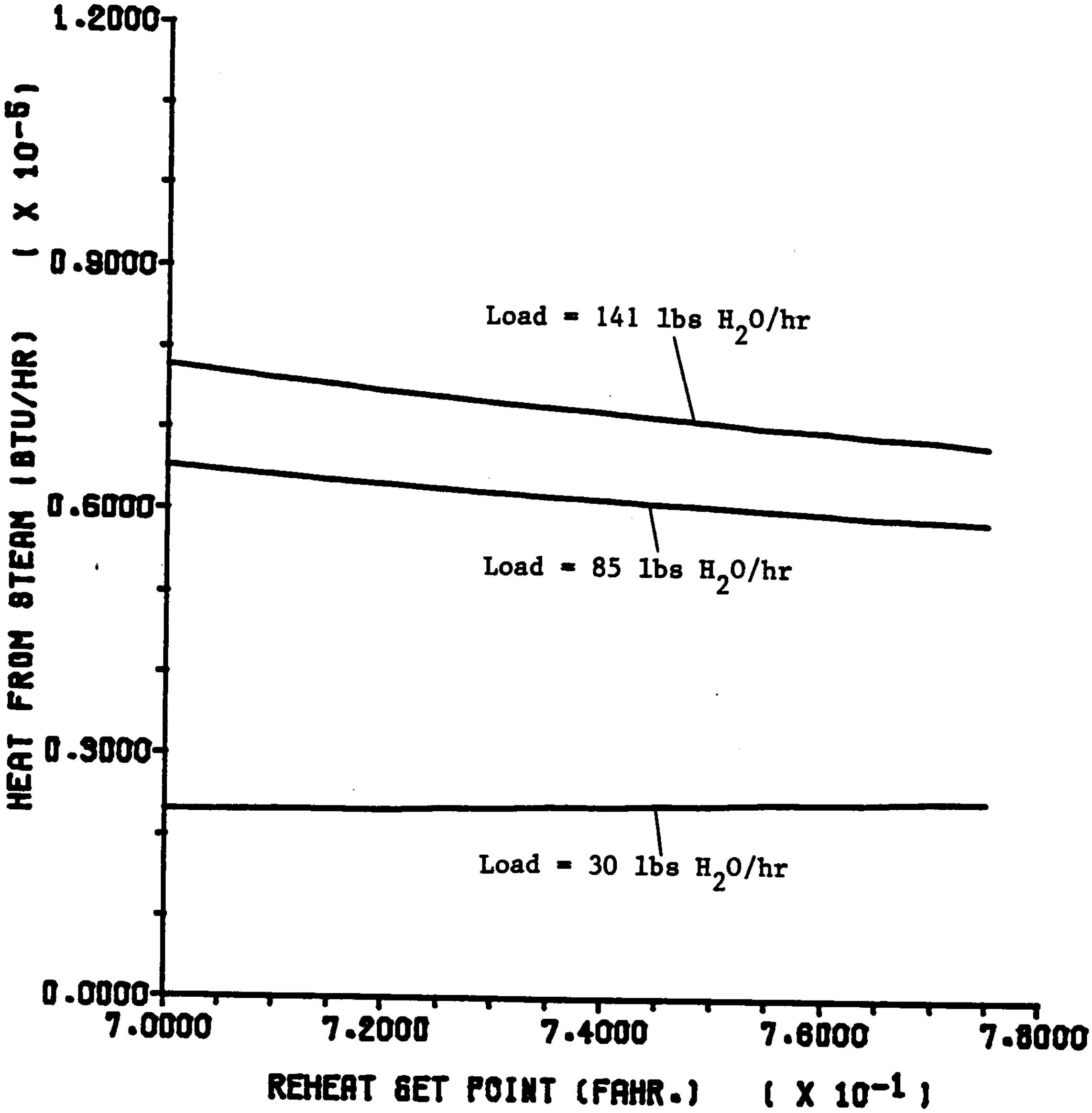


Figure 4-14. Single-duct heating requirements for dehumidification at 85°F outside temperature for a range of zone water loads at varying reheat set points.

low water case, as reheat set point is varied from 70°F to 77.5°F, the chilled water requirement drops from 3.72×10^5 BTU/hr to 3.24×10^5 BTU/hr while the steam requirement increases from 5.67×10^2 BTU/hr to 6.09×10^3 BTU/hr. An economic analysis shows that for this case the most economical reheat set point is 77.5°F unless steam costs 8.7 times as much as chilled water per BTU. In most cases chilled water costs will be no better than equivalent to the steam cost. Therefore, for both cases the highest realistic reheat set point will be the most economical. In actual implementation, this reheat set point could not be within 0.5°F of the normal cooling set point because of dynamics. Knowing the normal oscillation of the cooling mode determines how high this set point may be placed.

CHAPTER V

DUAL-DUCT STUDIES AND RESULTS

The next HVAC system studied is the dual-duct air handler. Local zone mixing boxes mix an appropriate amount of hot and cold air from this air handler's hot and cold ducts to provide the desired zone temperature. Steady-state simulations are used to demonstrate control strategy improvements, to investigate sensitivity problems, and to compare a conventional feedback strategy approach to one suggested here. As discussed earlier, the dual-duct simulation involves converging a set of mass and energy balance equations, given zone heat and water loads, and the temperature and relative humidity of the outside air. The strategy comparison simulations will use the "Bins" method described for the single-duct simulation, making hourly energy calculations at 5°F increment temperatures and multiplying by the number of hours per year to get the yearly energy consumption. Unless otherwise specified the outside relative humidity is set at a constant 50%.

For all cases studied here, the simulation uses a constant flow through the air handler fans at a rate of 70,000 lbs/hr. This is split among the three zones supplied in this simulation, called zone 1, zone 2, and zone 3, at respective rates of 20,000, 24,000, and 16,000 lbs/hr according to zone sizes. The areas of these zones are for zone 1, zone 2, and zone 3 respectively 8350, 6700, and 4450 square feet. The rooms are considered square with 12 foot ceilings and only one wall exposed to the outside. The dual-duct loads are chosen the same as for the single-duct air handler studies. The transfer coefficient through this wall to the outside air is assumed to be 0.1

BTU/ft^2-hr-°F, however, below a 66°F outside air temperature a working perimeter system is used to prevent heat loss at the zone. The simulation credits each zone with one person for every 250 square feet, with each contributing 250 BTU/hr of sensible heat and .241 lbs/hr of water vapor. Equipment loads are computed as 100 watts/person and electric lighting at 2 watts/ft^2. These are used as the base load values for each zone. It should be noted that they will give an equivalent BTU/(lbs air flow) load for each zone.

For the air handler, the maximum chilled water flow rate is 110,124 lbs/hr at a temperature of 44°F, and the maximum steam flow is 850 lbs/hr at a pressure of 25 psig. The simulation includes only one of the fans, the supply fan, and it is assumed to be rated at 18.5 HP with an efficiency of 70%.

Base Case Results

Base case dual-duct energy consumption is determined by running the simulation with the above loads and applying the original control strategy given in Figures 2-7 to 2-10 on pages 28 to 31. The hourly heating load requirements are given in Figure 5-1 and the hourly cooling load requirements in Figure 5-2. As seen in Figure 5-1, the heating load requirement shows little decrease until the outside air temperature reaches about 60°F. It is important to remember that below 60°F a working perimeter system is assumed to insulate the zone. Therefore, the slight heat load change below 60°F is due only to a change in the temperature of the required minimum outside air, and to the change in hot duct set point according to the original feedforward strategy. The Figure 5-2 hourly cooling load plot shows a cooling requirement even at low temperatures. This is due to the constant cold duct set point of

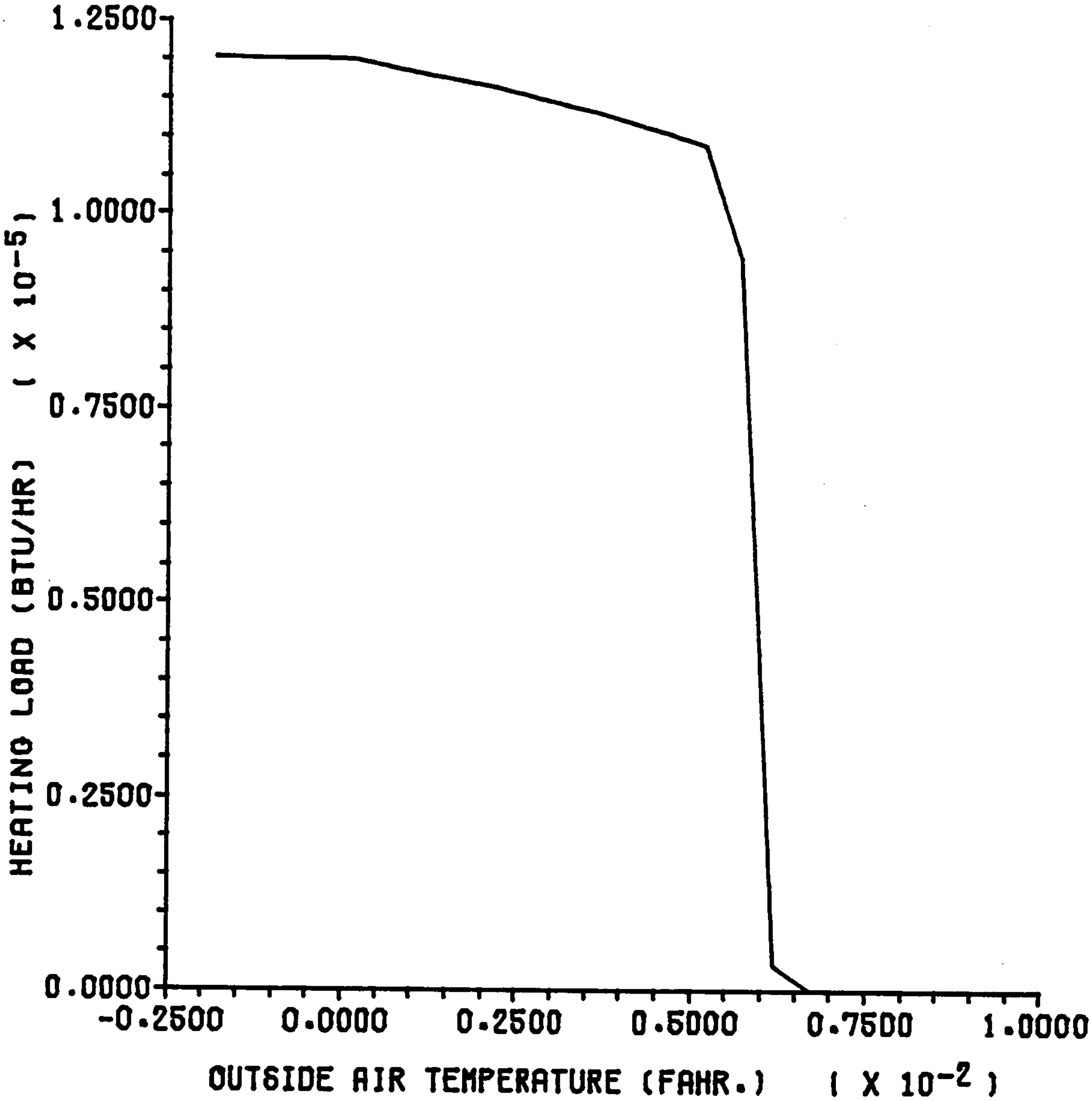


Figure 5-1. Dual-duct hourly heating load with original control strategy.

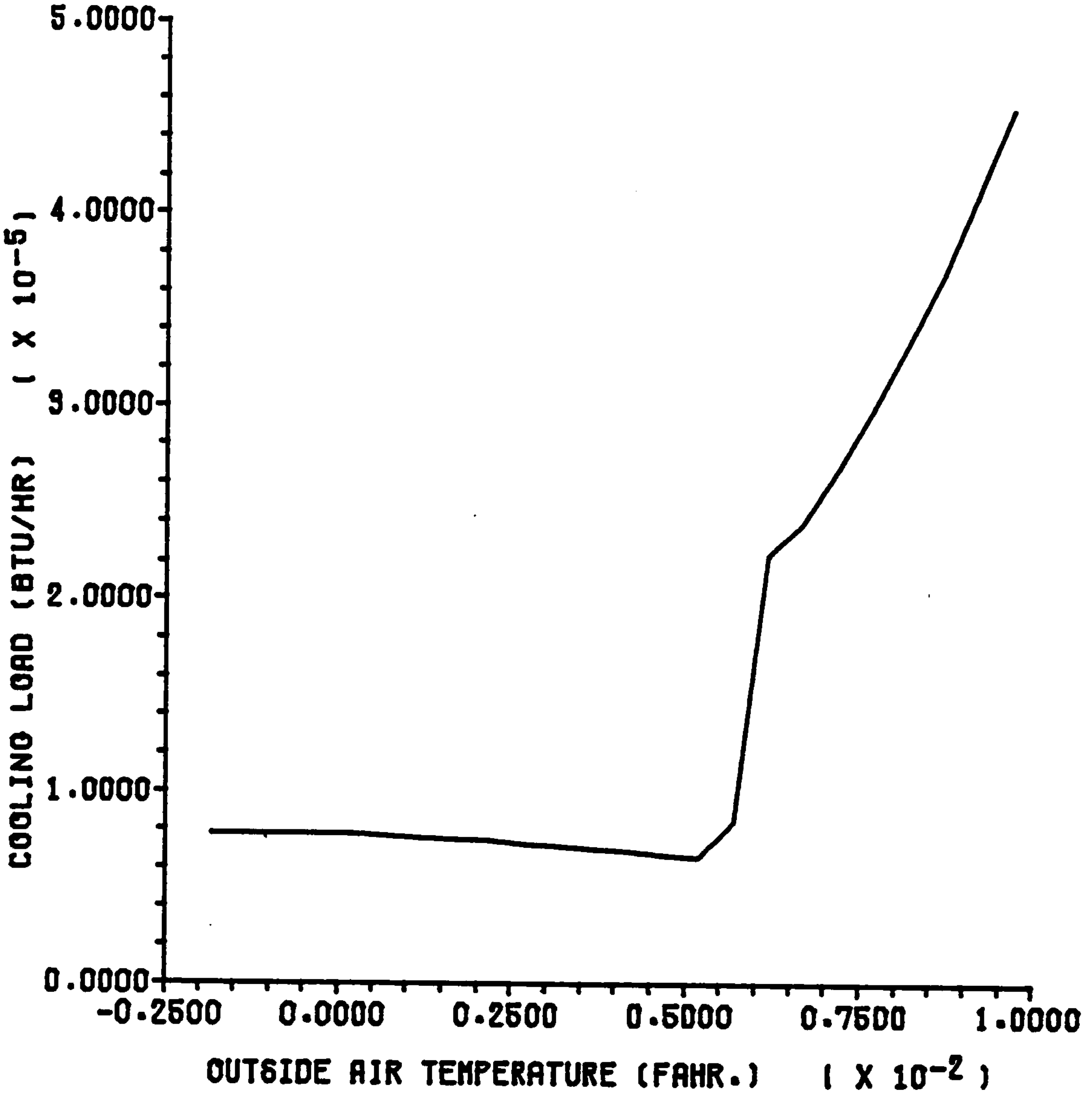


Figure 5-2. Dual-duct hourly cooling load with original control strategy.

53°F. Since the hot duct and cold duct both have firm set points for a given temperature, the air must always be either heated or cooled, unless the supply air is already beyond these set points. For example. at high outside air temperatures the supply air to the ducts is already above the 75°F hot duct set point and heating is eliminated. At these high temperatures, Figure 5-2 shows a fast increasing cooling requirement as zone loads increase.

Figure 5-3 and 5-4 show the yearly heating and cooling requirements, found by multiplying the Figure 4-1 (pg. 72) "Bins" curve by the hourly energy requirements. Figure 5-3 shows that the heaviest yearly heating consumption with the original strategy occurs around 27°F to 42°F. The heaviest yearly cooling requirements occur around the 62°F to 77°F range. Therefore, it should be expected that slight hourly improvements in these areas are much more important than other areas in reducing overall yearly consumption.

<u>Zone Temperature Feedback</u>

The first attempt at control strategy improvement is to provide zone temperature feedback to set the hot and cold duct temperature set points. The idea is to use heating or cooling only as needed by the zones rather than heating on the basis of ourside air temperature of cooling to a constant set point. Figures 5-5 and 5-6 present the suggested hot duct and cold duct cascaded feedback strategies. The Figure 5-5 hot duct strategy uses a low temperature select device to feedback the temperature of the coldest zone. This temperature is compared with a low temperature set point of 66°F. If this lowest temperature is above 66°F, the strategy will shut off the steam valve. As the

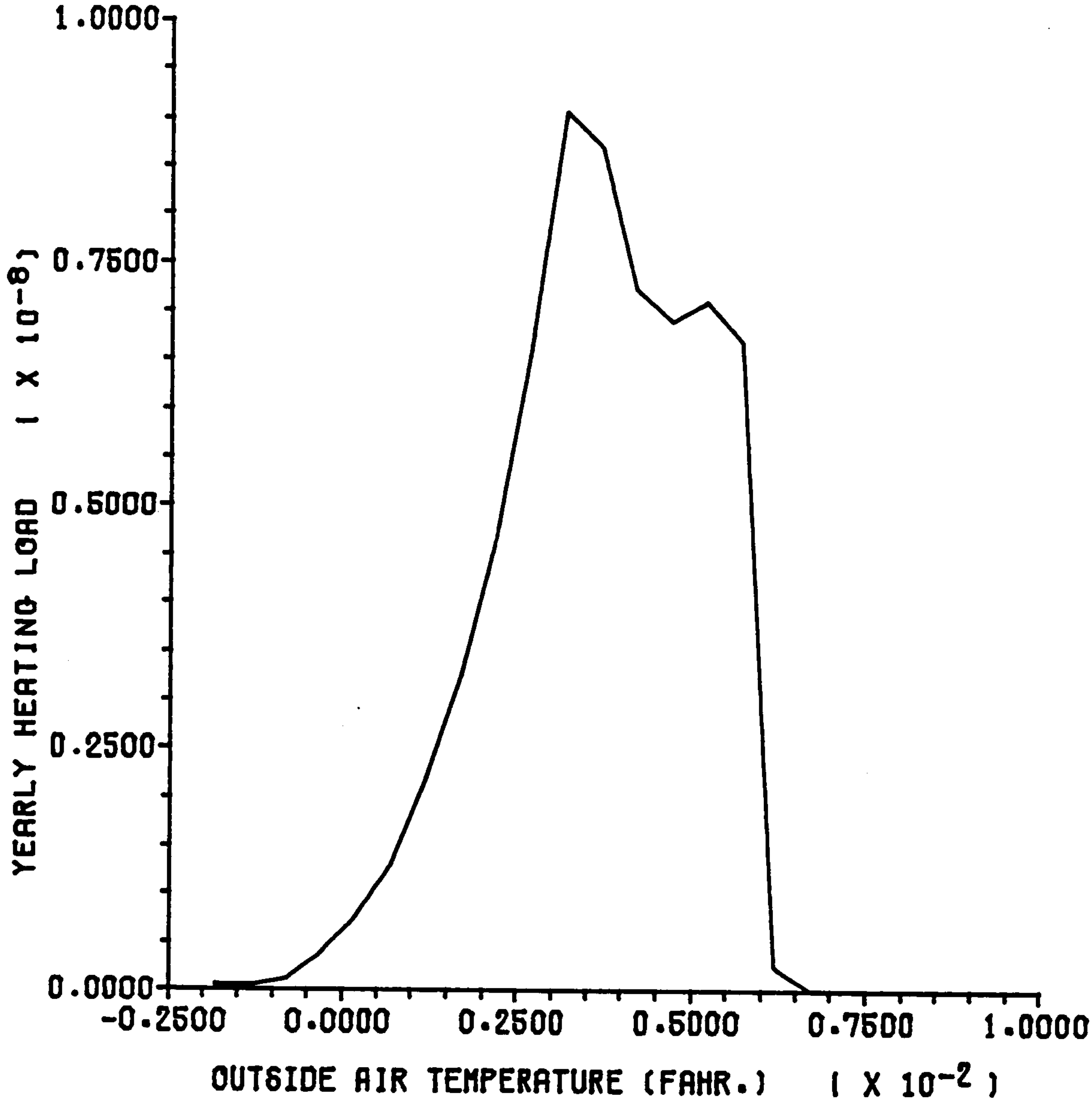


Figure 5-3. Dual-duct yearly heating load with original control strategy.

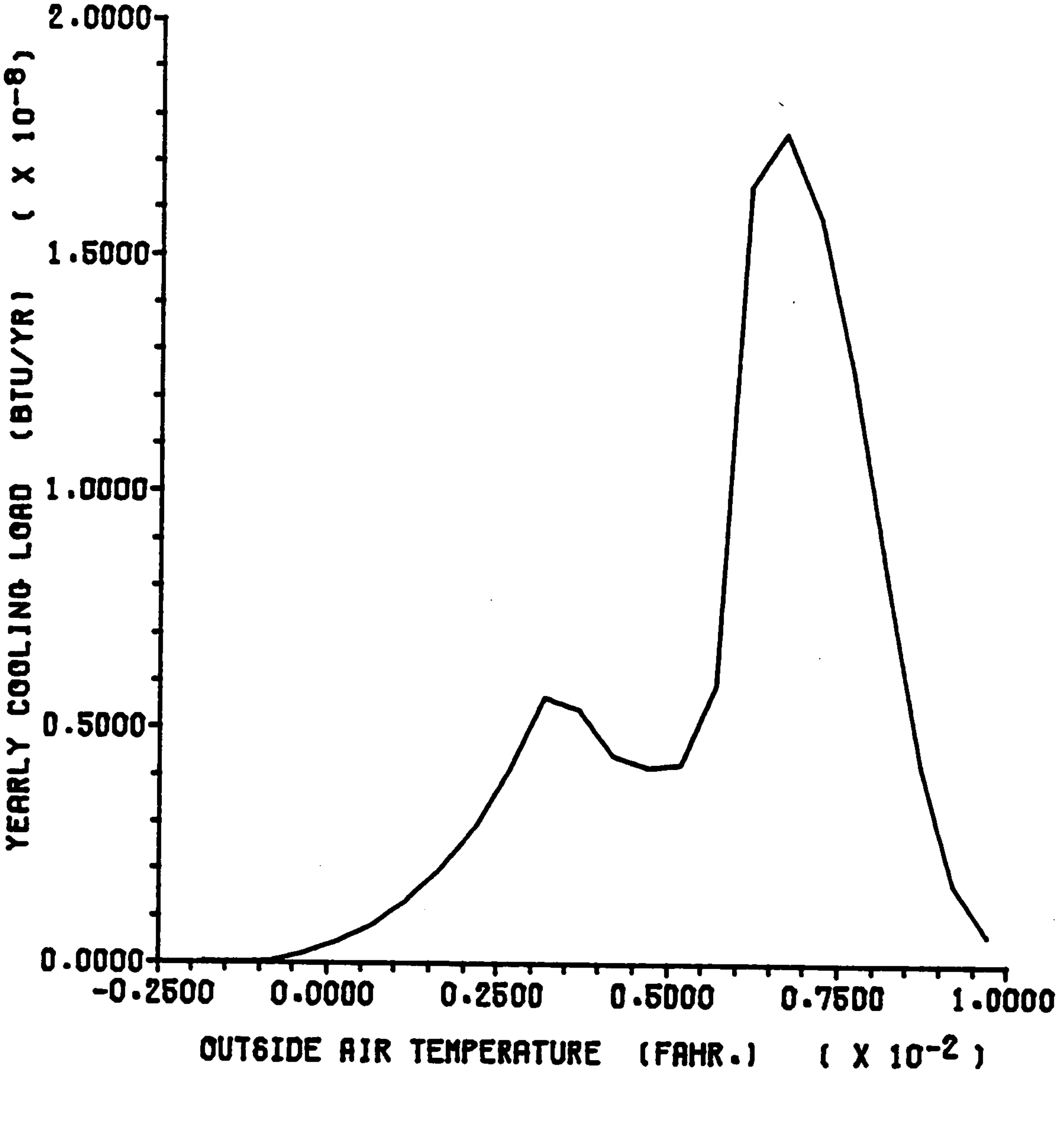


Figure 5-4. Dual-duct yearly cooling load with original control strategy.

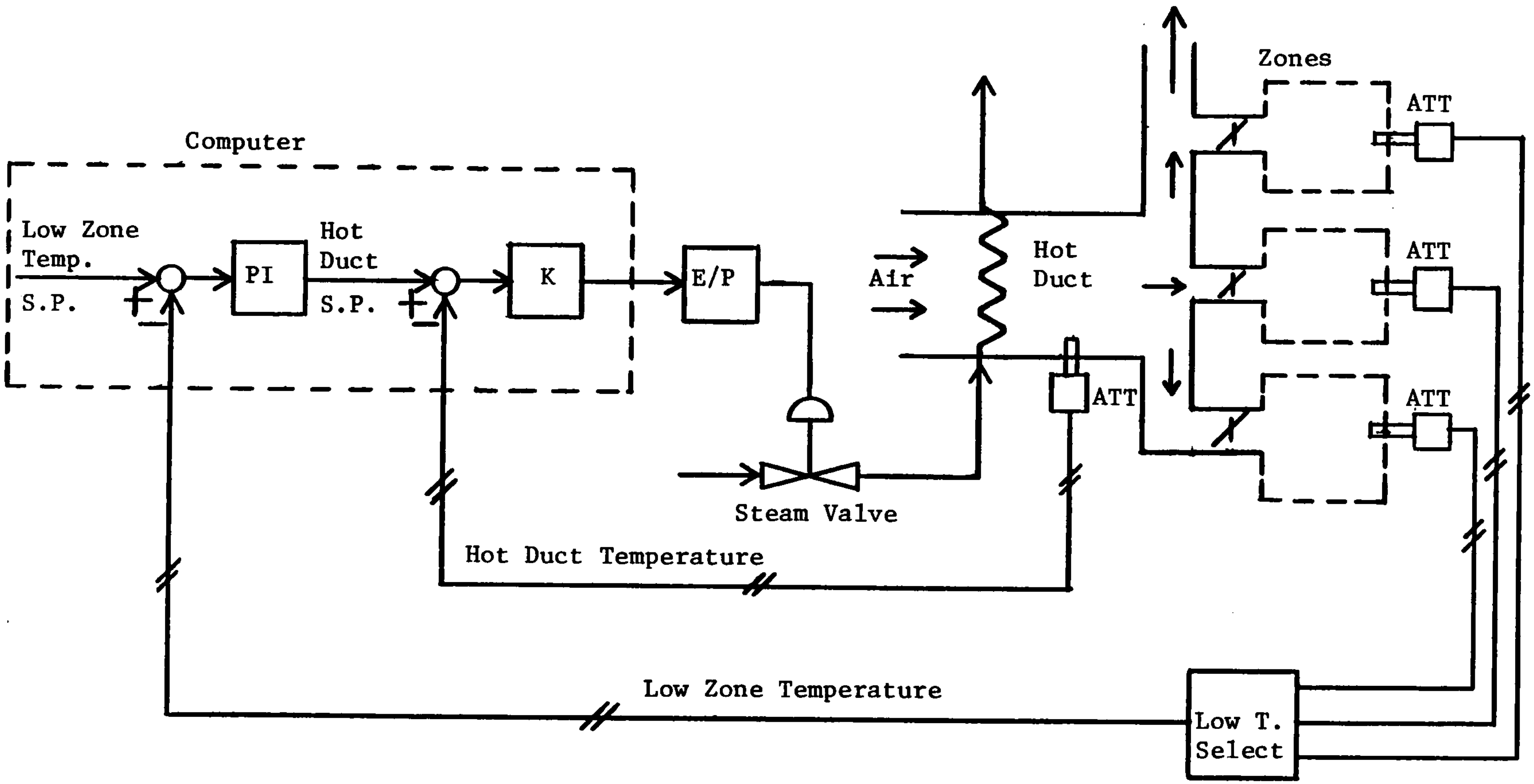


Figure 5-5. Suggested hot duct cascaded feedback strategy.

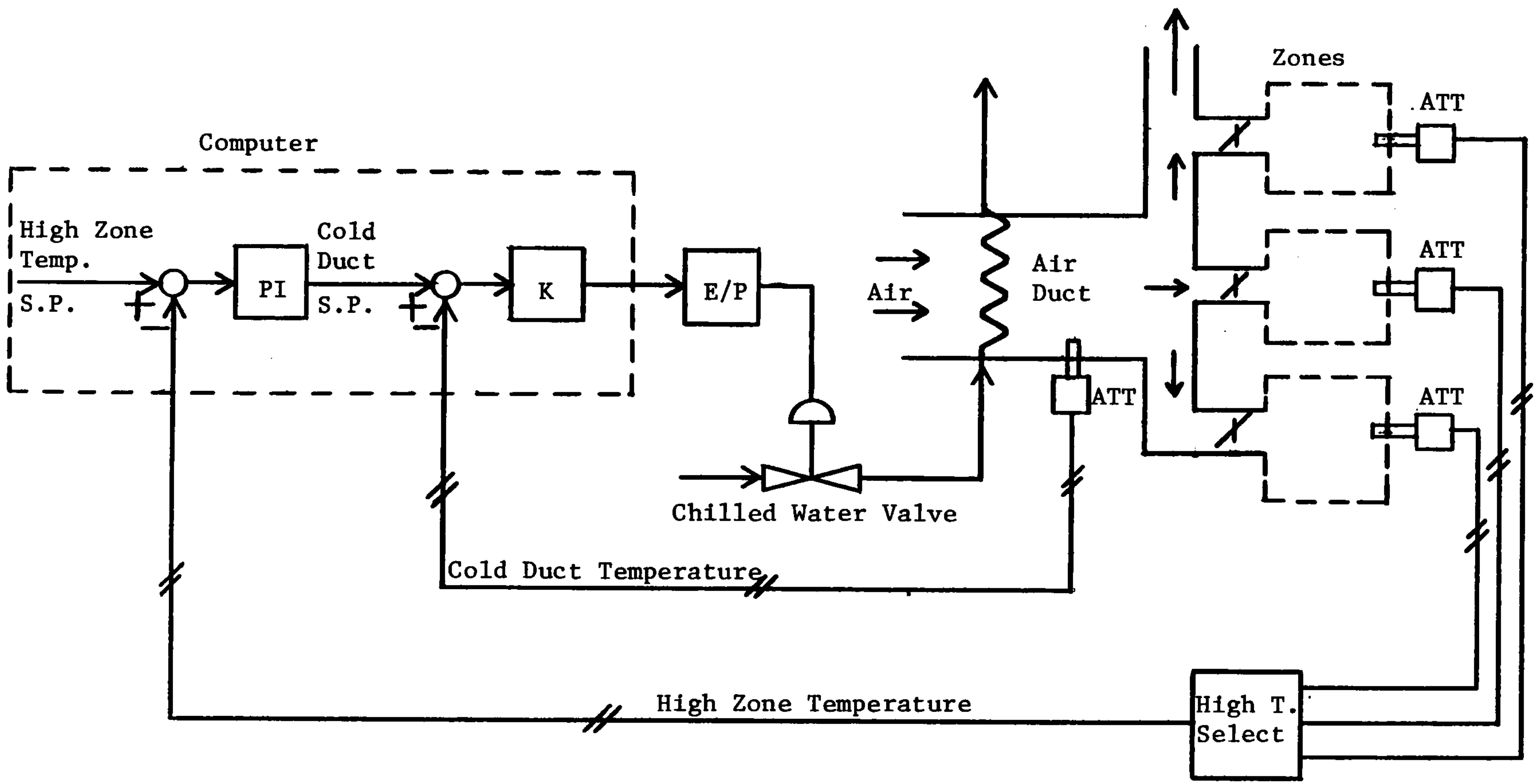


Figure 5-6. Suggested cold duct cascaded feedback strategy.

lowest temperature drops to 66°F, the hot duct set point is increased and the steam valve is opened to hold this low temperature zone at 66°F. The cold duct control strategy, shown in Figure 5-6, uses the exact inverse of the hot duct. This strategy uses a high temperature select to feedback the temperature of the warmest zone. If the highest temperature is below the high set point of 78°F, no cooling is required, and the chilled water valve is shut off. As the highest temperature rises to 78°F, the cold duct set point is decreased to hold the warmest zone at 78°F. In combination, these two strategies will do no heating or cooling as long as all zone temperatures remain between 66°F and 78°F. Heating or cooling is used only to prevent any zones from leaving these bounds. By feeding back the worst case temperatures, adequate heating and cooling will always be available to satisfy intermediate temperature zones. Applying these feedback strategies with the same loads as used in the base case gives the Figures 5-7 and 5-8 hourly energy requirements. Figure 5-7 shows that at these loads, the feedback strategy eliminates all heating needs. The original base strategy hot duct set points are therefore putting an additional false heating load on the system. The hourly cooling loads for the feedback strategy, given in Figure 5-8, show that the base cooling needs below 57°F have been elminated. Above this temperature the cooling load curve is seen to be slightly below the base curve. By providing cooling and heating only when necessary, the feedback strategies have eliminated all heating requirements, eliminated all cooling requirements below 57°F, and decreased the cooling requirements above 57°F.

The yearly energy consumption curve is needed only for the cooling load, since the heating load has been completely eliminated

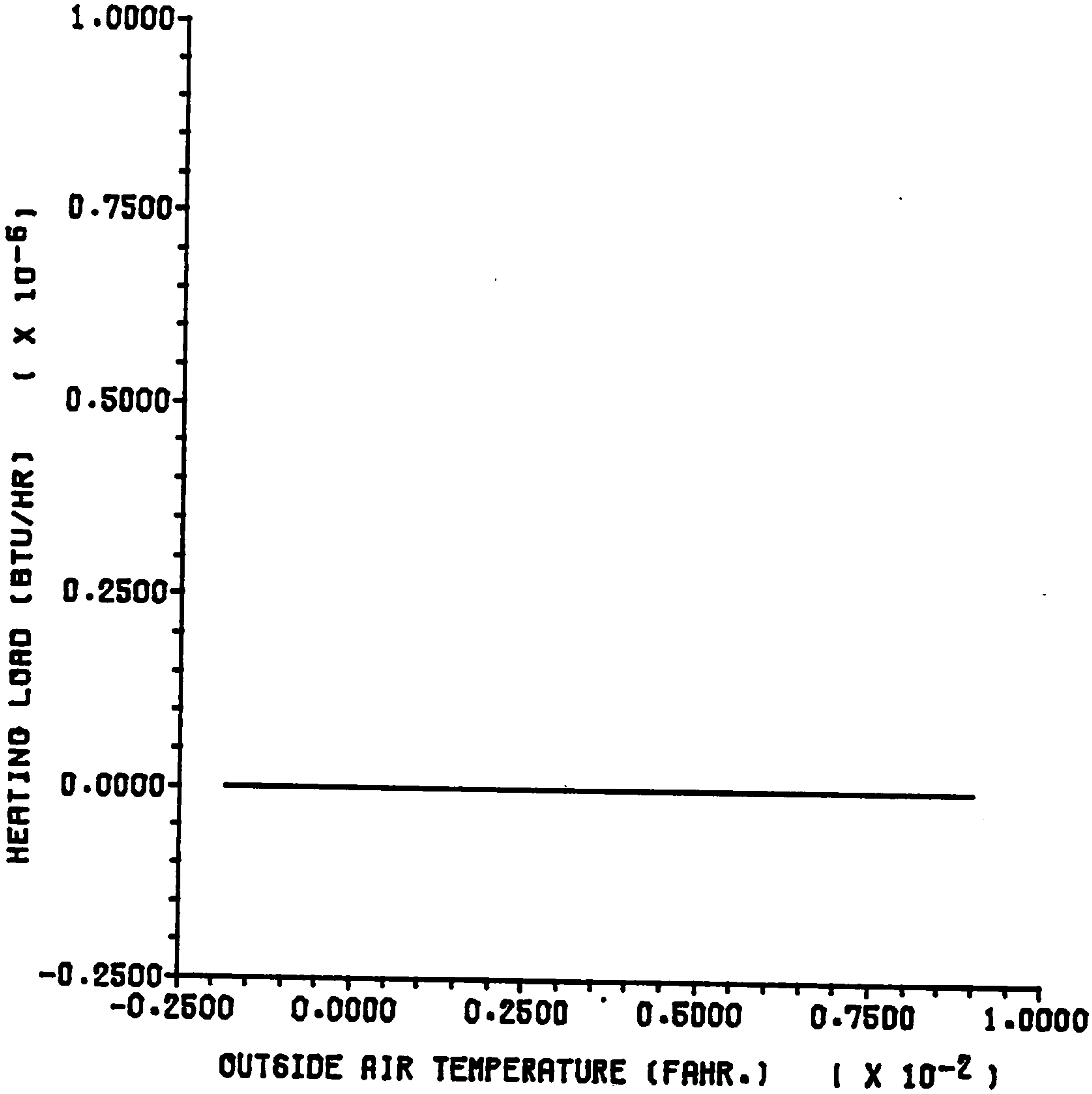


Figure 5-7. Dual-duct hourly heating load with zone temperature feedback.

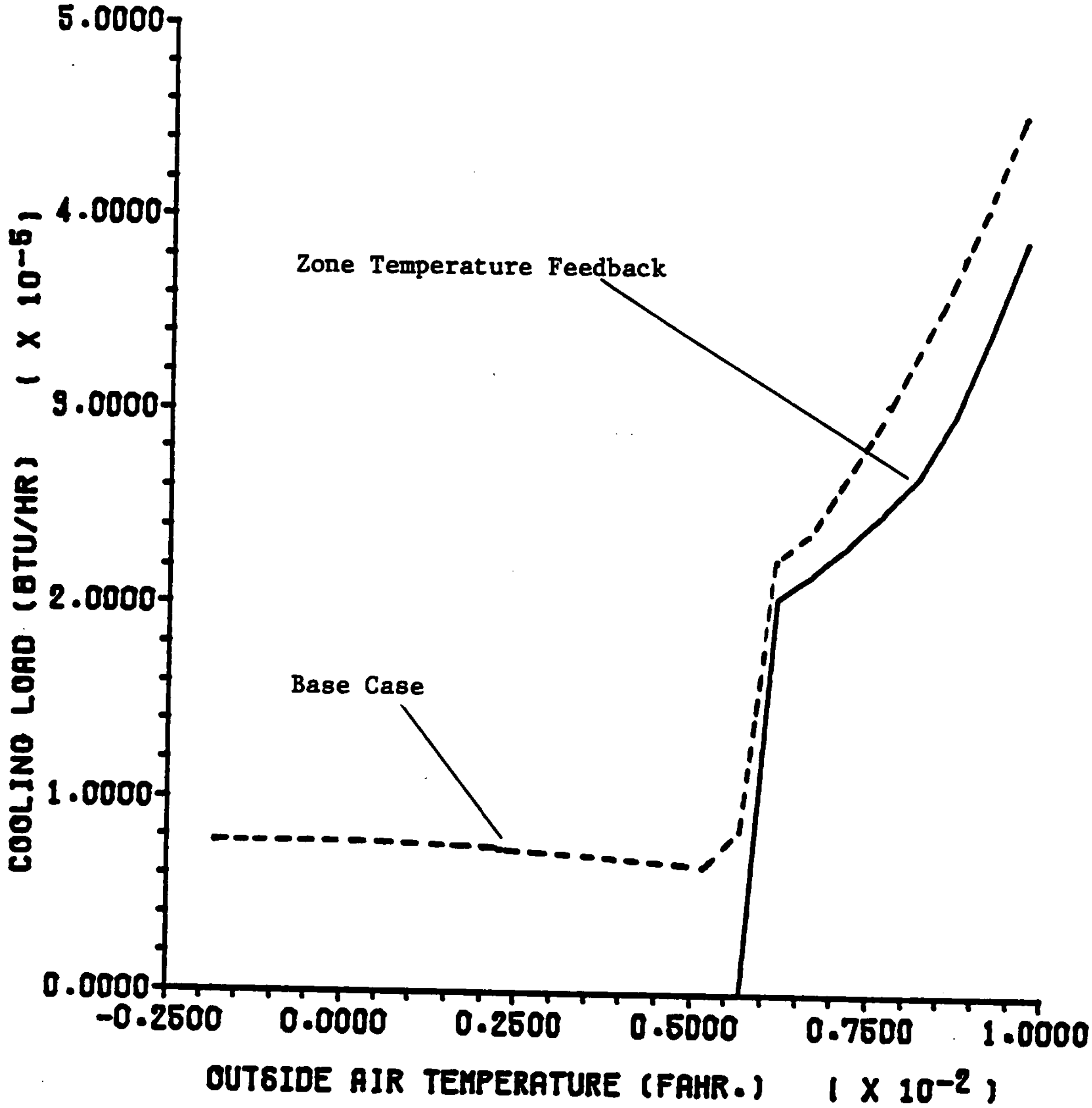


Figure 5-8. Dual-duct hourly cooling load with zone temperature feedback.

with the feedback strategy. Figure 5-9 shows a diminished yearly cooling requirement in comparison with the base case. The feedback yearly cooling load is only 57% of the original base yearly cooling load.

Economizer Mixing Box

Another area where control strategy change is expected to yield significant improvement is the operation of the air handler mixing box. The original air handler strategy allows use of free cooling from the outside air only when the outside air temperature is below 60°F. Above this temperature only minimum outside air is allowed. Here, free cooling indicates reducing the temperature and enthalpy of the mixing box air. The proposed improvement is to allow the use of free cooling from the outside air anytime it is of better quality for cooling than the return air, as was done for the single-duct air handler. Figure 5-10 shows the suggested control strategy for the air handler mixing box, which says that if the temperature and enthalpy of the outside air are less than the temperature and enthalpy of the return air, use outside air for free cooling to meet the original control strategy 55°F set point.

This control strategy change is added to the proposed zone temperature feedback strategy change to determine the overall energy consumption improvement. This strategy change will have no effect on heating load requirements since it is used only to provide free cooling. For these particular runs, the heating load is already zero due to the feedback strategy. The Figure 5-11 hourly cooling load curve shows that the mixing box strategy change decreased the cooling load requirements

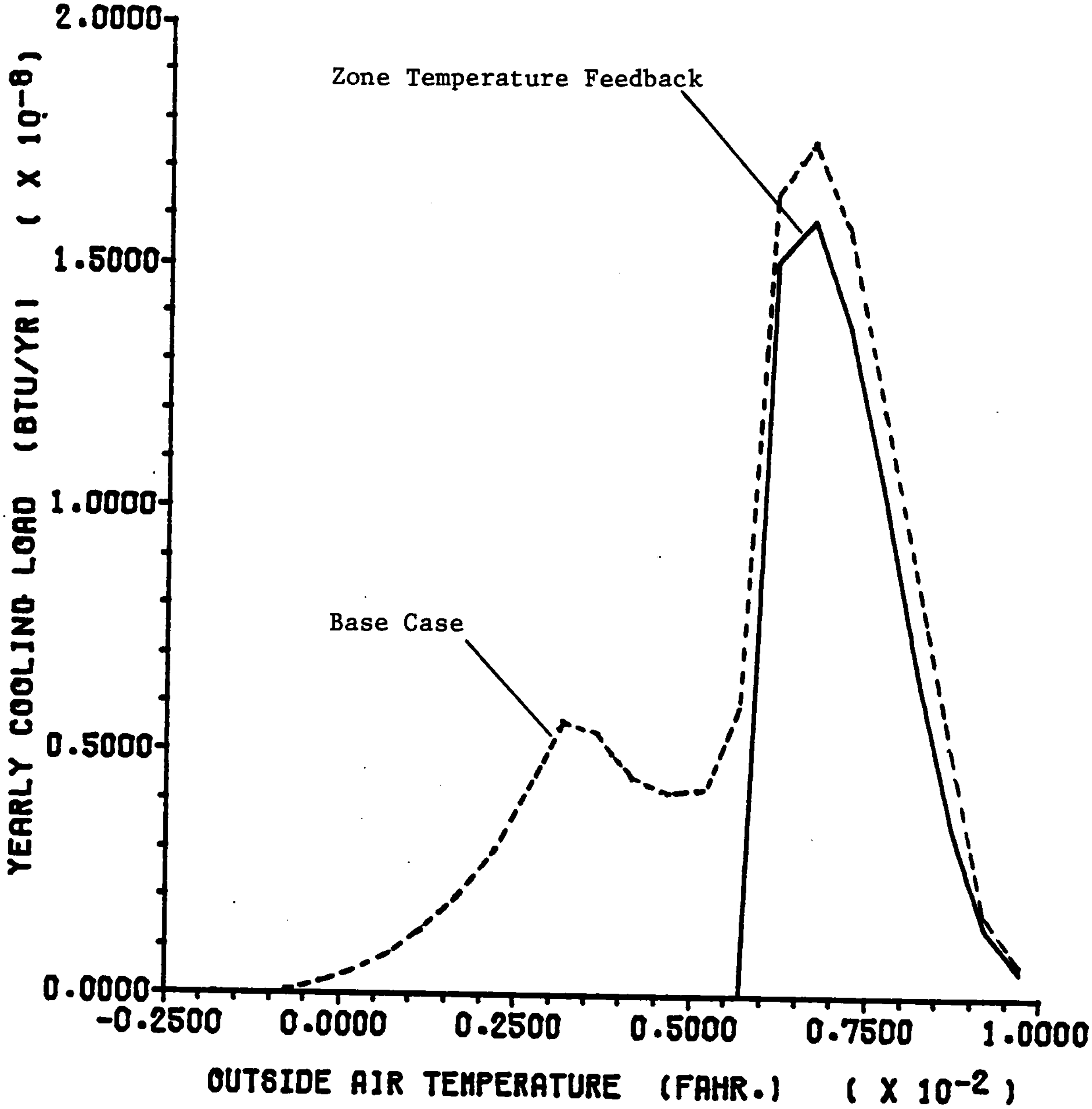


Figure 5-9. Dual-duct yearly cooling load with zone temperature feedback.

Only active if: Temperature Outside Air < Temperature Return Air

and

Enthalpy Outside Air < Enthalpy Return Air

Otherwise use minimum outside air.

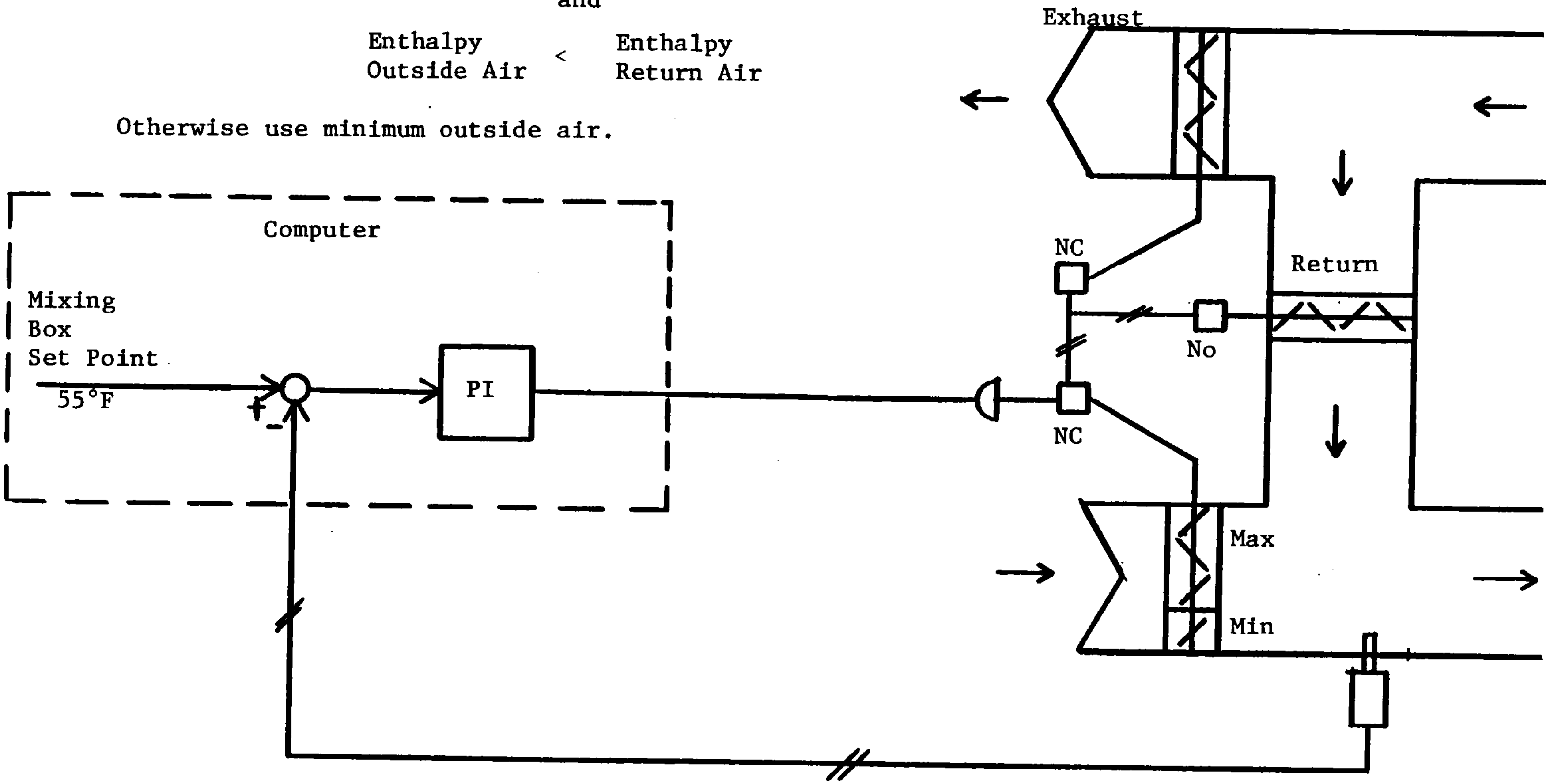


Figure 5-10. Proposed dual-duct mixing box strategy.

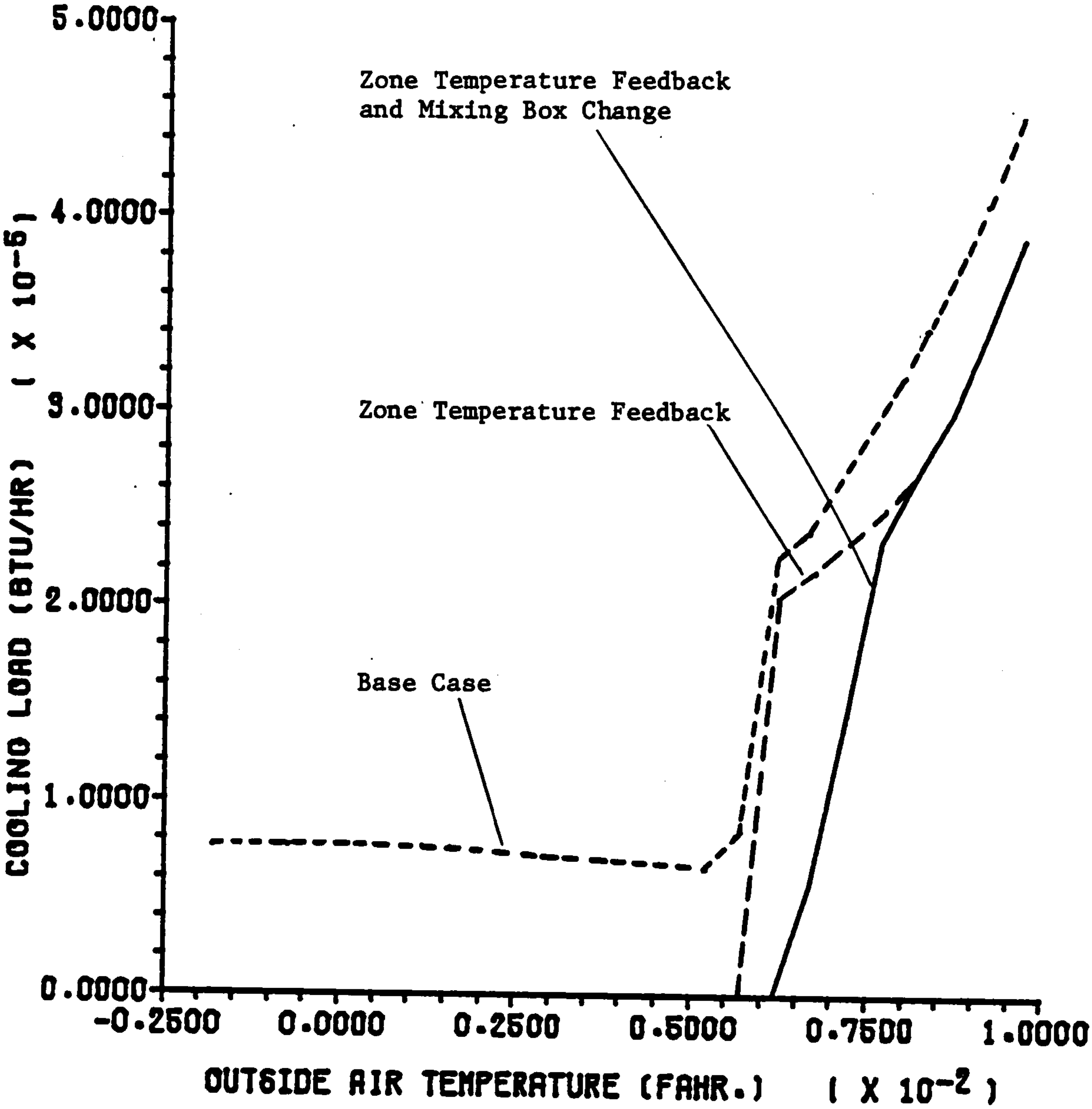


Figure 5-11. Dual-duct hourly cooling load with zone temperature feedback and mixing box strategy change.

significantly in the 57°F to 77°F range. At 82°F and above the mixing box strategy shows no improvement over the feedback strategy because the outside air is of worse cooling quality than the return air and both strategies use identical minimum outside air.

The mixing box cooling load improvement in the 57°F to 77°F range is very important on a yearly curve, as seen in Figure 5-12, due to the large number of yearly hours at these temperatures. The mixing box strategy in combination with the zone temperature feedback strategy uses only 29% of the cooling energy that is used with the original strategy. Table 5-1 compares the yearly heating and cooling total energy usage for each of the strategies discussed.

Another set of simulation runs shows in Table 5-2 the effects of varying the zone heat loads on the potential energy saving. For these runs the load on zone 1 is held at a value 50% greater than described for the equal load runs. Zone 2 is held at its same load value, and zone 3 is decreased to 50% its original value. This simulation projects a possible wide zone load condition case that might be expected in the multiple zone-serving-dual-duct air handler. Table 5-2 shows that for this case the heating requirements are not eliminated, but are reduced to a value that is only 16% of the original strategy requirement. The chilled water load is reduced to 60% by the zone temperature feedback and to 40% by the combination feedback-mixing box strategy change. This unequal zone heat load case demonstrates that the savings seen for the equal zone heat load case carries over to other zone load conditions and are therefore not restricted to a narrow operating range.

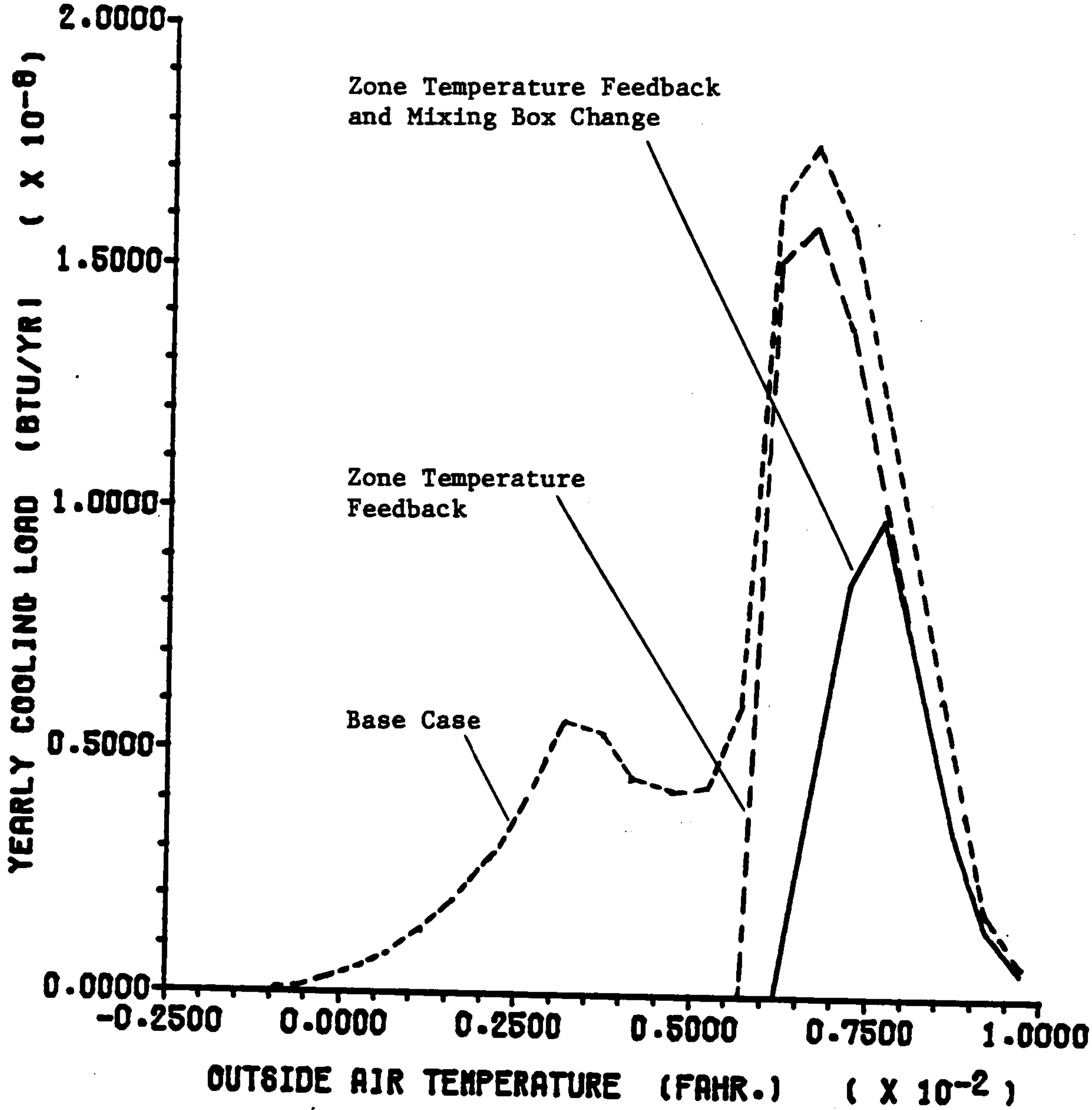


Figure 5-12. Dual-duct yearly cooling load with zone temperature feedback and mixing box strategy change.

TABLE 5-1

COMPARISON OF ENERGY REQUIREMENTS FOR THREE DUAL-DUCT STRATEGIES WITH EQUAL BTU/(lbm AIR FLOW) ZONE LOADS

Strategy	Steam BTU/yr	Chilled Water BTU/yr
Original	6.5×10^{8}	12.0×10^{8}
Zone Temperature Feedback	0.0 (6%)*	6.8×10^{8} (57%)*
Zone Temperature Feedback and Economizer Mixing Box	0.0 (0%)*	3.5×10^{8} (29%)*

*Percentages relative to original strategy use

TABLE 5-2

COMPARISON OF ENERGY REQUIREMENTS FOR THREE DUAL-DUCT STRATEGIES WITH UNEQUAL* BTU/(lbm AIR FLOW) ZONE LOADS

Strategy	Steam BTU/yr	Chilled Water BTU/yr
Original	$5.7x10^8$	$12.7x10^8$
Zone Temperature Feedback	$0.9x10^8$ (16%)**	$7.6x10^8$ (60%)**
Zone Temperature Feedback and Economizer Mixing Box	$0.9x10^8$ (16%)**	$7.6x10^8$ (40%)**

*Zone 1 load increased 50% over original value
Zone 2 load equal to original value
Zone 3 load decreased 50% from original value

**Percentages relative to original strategy value

Mixing Box Set Point Sensitivity

For the mixing box strategy just presented the set point was held constant at its original value of 55°F. The question of whether this is the optimum fixed set point location can best be investigated by running the dual-duct simulation with varying values of this mixing box set point. This is done at varying zone load conditions to also determine the effect of load on the desired set point. Rather than make unnecessary runs at temperatures that require no mixing box manipulation, these runs are made with varying outside air temperature from 26°F to 86°F in 12°F increments. This range of outside temperature is checked with mixing box set points varying from 52°F to 64°F at 3°F increments. The first case assumes loads equal to the original base case values discussed earlier. The second case makes runs at load values equal to 25% of the original values, and the third case is run with the load on zone 1 increased 50%, zone 2 held constant, and zone 3 decreased 50%.

Figure 5-13 shows that for the 100% equal load case that all mixing box set points have the same chilled water requirement except at a set point value of 64°F. At this value the mixing box temperature is high enough at all outside air temperatures to require cooling the zones to keep them at 78°F. This demonstrates that at certain loads, too high a mixing box set point can require cooling at all temperature conditions. For this 100% equal load case, with the zone temperature feedback strategy, no steam is required at any of the mixing box set points. The zone temperature trend for this 100% case, given in Figure 5-14 shows that up to the outside temperature of 62°F, increasing the mixing box set point will increase the zone temperatures by an

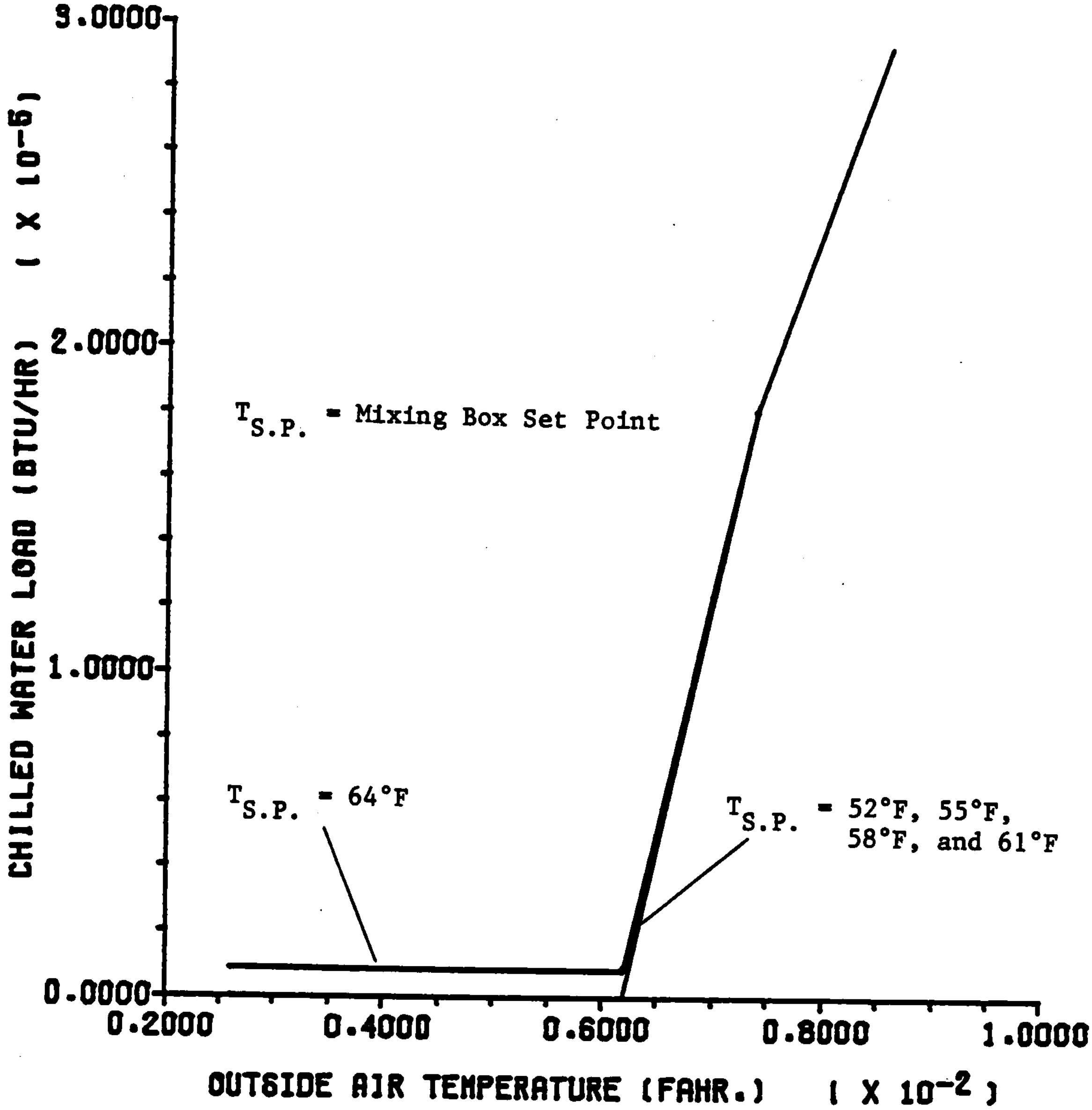


Figure 5-13. Dual-duct mixing box sensitivity study chilled water loads for 100% equal zone loads.

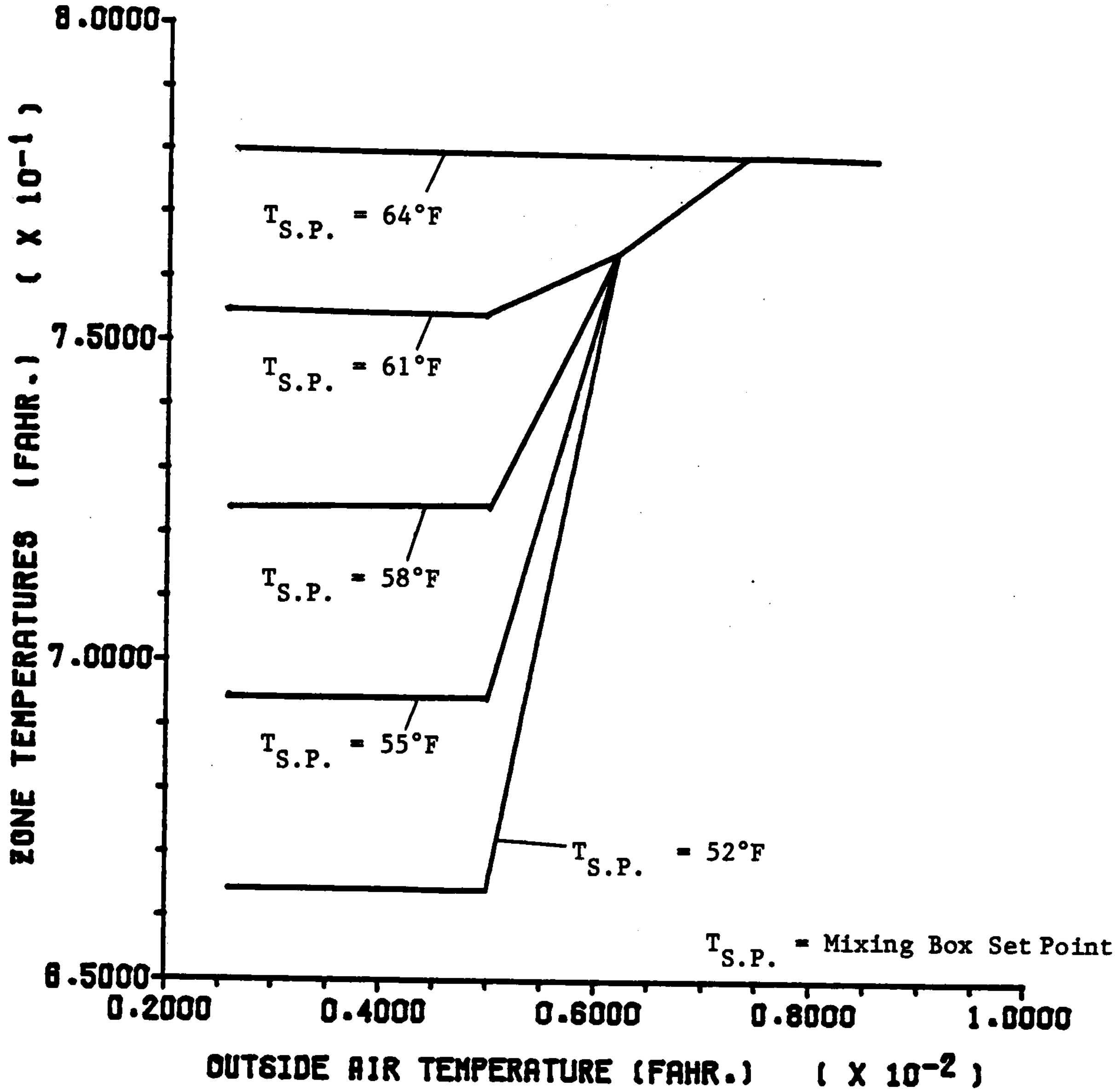


Figure 5-14. Dual-duct mixing box sensitivity study temperature trends with 100% equal zone loads.

equivalent amount. Note that at a mixing box set point of 64°F the zone temperatures remain at 78°F, thus always requiring cooling.

Similar runs made for zone loads of 25% of the original base zone loads show that changing the mixing box set point has no effect on the chilled water requirements. This is because at low zone loads no cooling is required in the medium outside temperature range, and as the outside air temperature gets to higher values, no free cooling by outside air is possible. However, for the low load case,the Figure 5-15 steam requirement curve shows that having a low mixing box set point such as 52°F will cause the system to need much more heating, while the same zone load conditions will require no heating as the mixing box set point is raised to 61°F. A check of the Figure 5-16 zone temperature profiles shows that at ourside air temperatures of 52°F and below, the mixing box set points of 52°F, 55°F, and 58°F result in zone temperatures at the low limit of 66°F. At higher mixing box set points, zone temperatures are elevated above 66°F without the use of any steam. As expected, comparison of the 25% load case with the 100% load case indicates that proper choice of a mixing box set point is very much dependent on the existing zone loads. Expected zone loads of the 100% case would indicate that a low mixing box set point is desirable, however, should the zone loads be drastically below this, as in the 25% case, the low mixing box set points will unnecessarily force heating requirements on the air handler.

The third mixing box case, which has the zone 1 load increased 50% and zone 3 load decreased 50%, provides a good example of how the proper mixing box set point can eliminate unnecessary heating or cooling requirements. Figure 5-17 shows that at a mixing box set point of 58°F or above there is no heating necessary. Conversely,

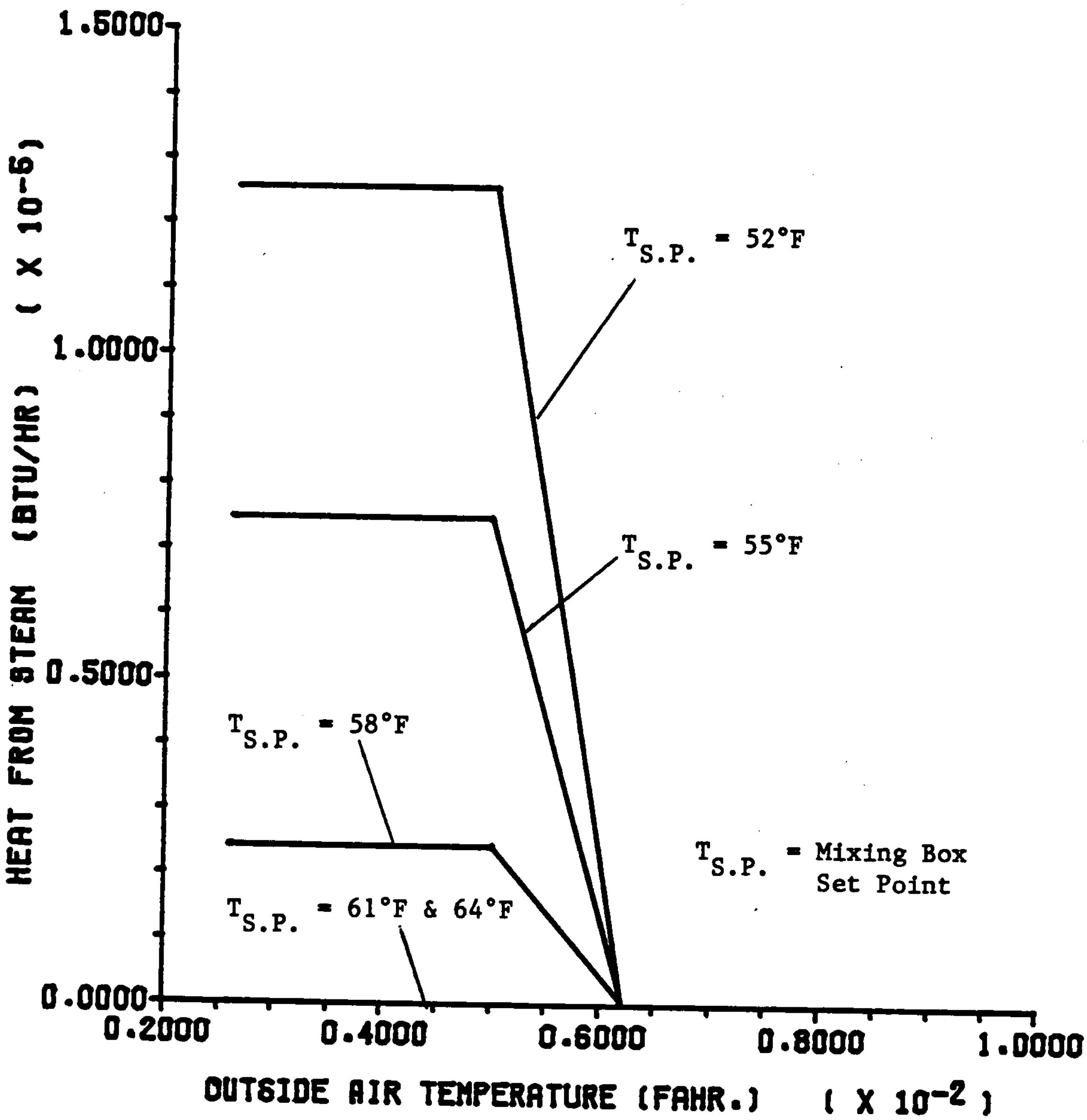


Figure 5-15. Dual-duct mixing box sensitivity study steam loads for 25% equal zone loads.

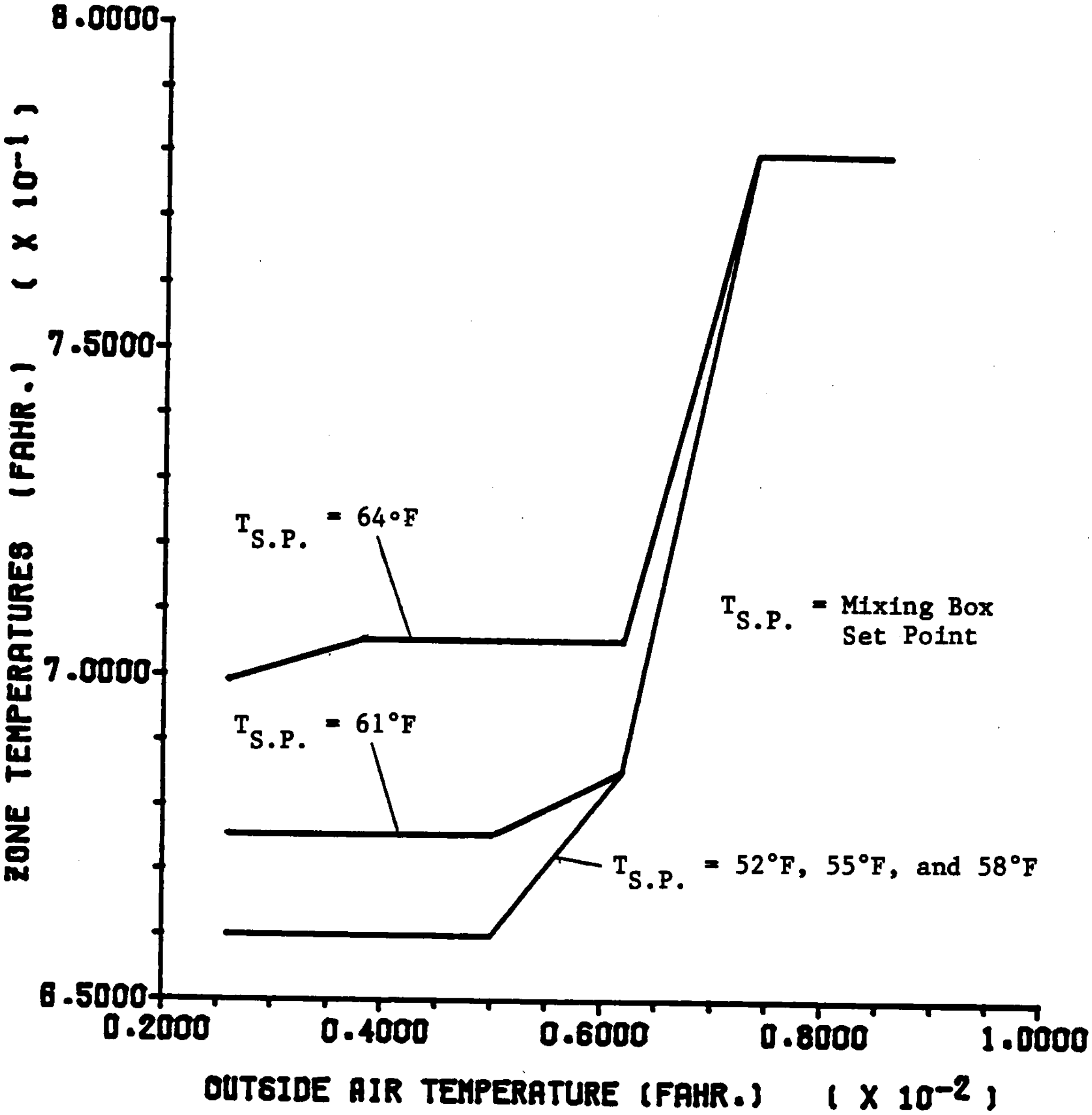


Figure 5-16. Dual-duct mixing box sensitivity study temperature trend for 25% equal zone loads.

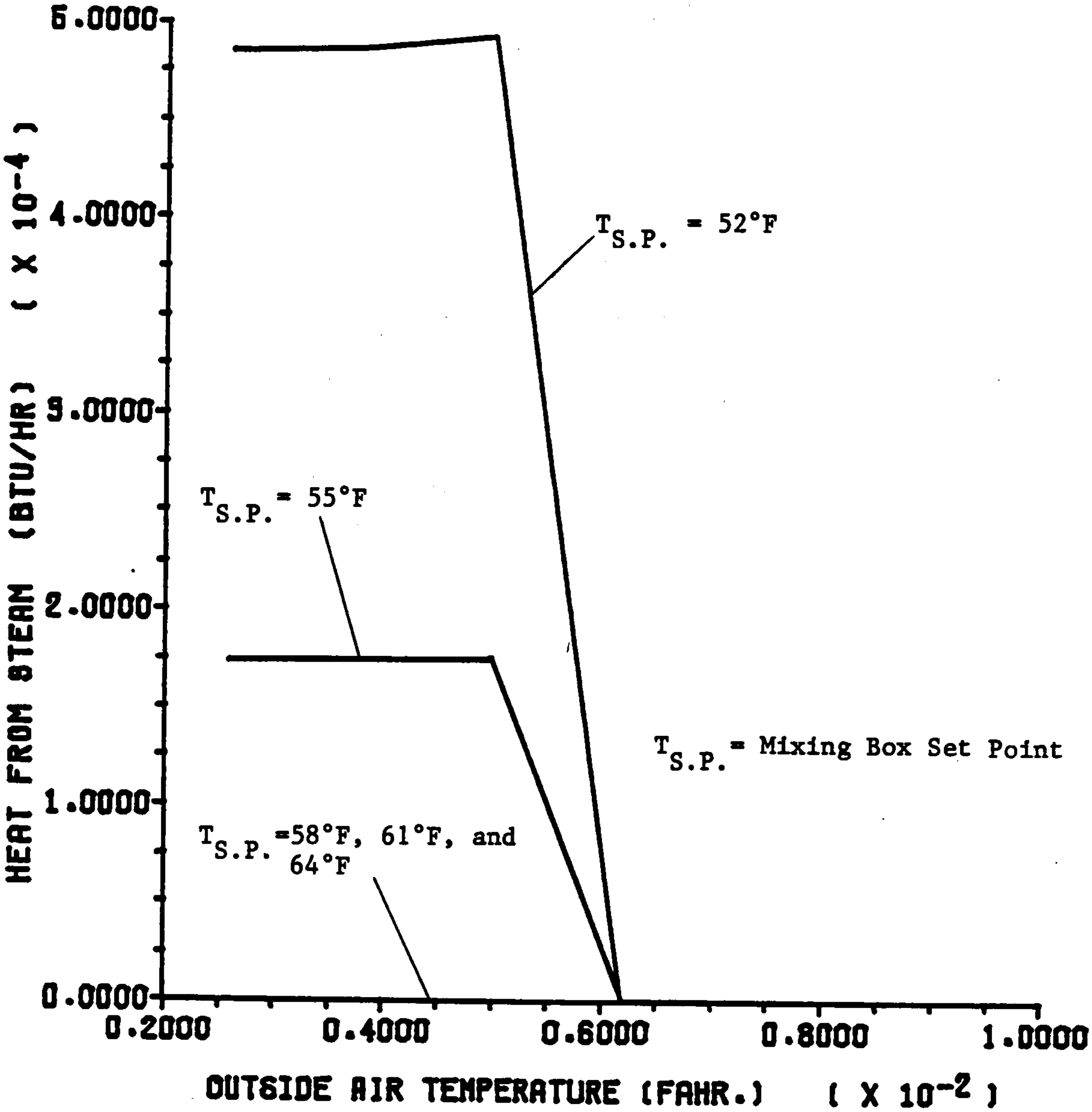


Figure 5-17. Dual-duct mixing box sensitivity study steam loads for unequal zone loads.

Figure 5-18 shows that only at a mixing box set point of 58°F or below can cooling needs be eliminated for the low outside air temperature cases. Combining these two figures shows that for the given load conditions a mixing box set point of 58°F would eliminate both the needless cooling and heating and would therefore be the optimum of the set points tested. However, it is important to remember that these are hypothetical loads, and therefore may be used only to present the principle and not for actual set point determination. The zone temperature profile of Figure 5-19, for the unequal load case, shows that zone 3, with the low load, is around 66°F at the low outside temperatures and is therefore responsible for zone heating needs. Zone 1, with the high load, gets to 78°F at low outside temperatures with a mixing box set point of 61°F or 64°F and is therefore responsible for low outside temperature cooling needs.

Zone Set Point Sensitivity

Another set of control strategy set points that are held constant, but may have an effect on energy consumption are the zone temperature set points. These set points are set on local zone thermostats at 72°F and left at that value all year. Since they are used only to control the local zone mixing boxes, these set points are usually not met. The feedback control strategy performs no cooling or heating unless the zone temperatures move to the 78°F or 66°F limits. Therefore, the local set points determine when the mixing boxes are receiving hot air or cold air. If the local zone temperature is above 72°F the dampers will be 100% open to the hot duct. The purpose of this sensitivity study is to determine whether moving the local zone

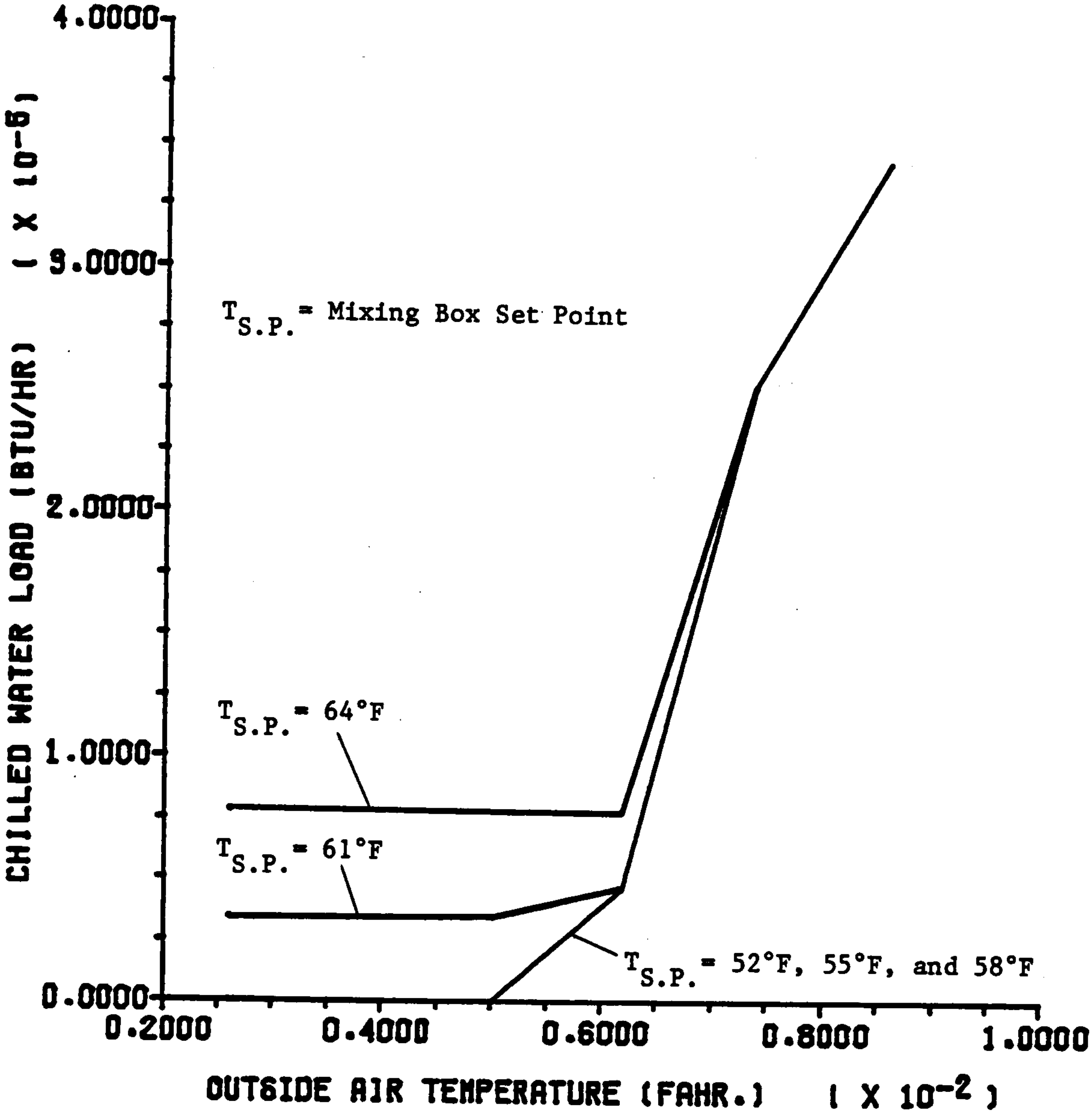


Figure 5-18. Dual-duct mixing box sensitivity study chilled water loads for unequal zone loads.

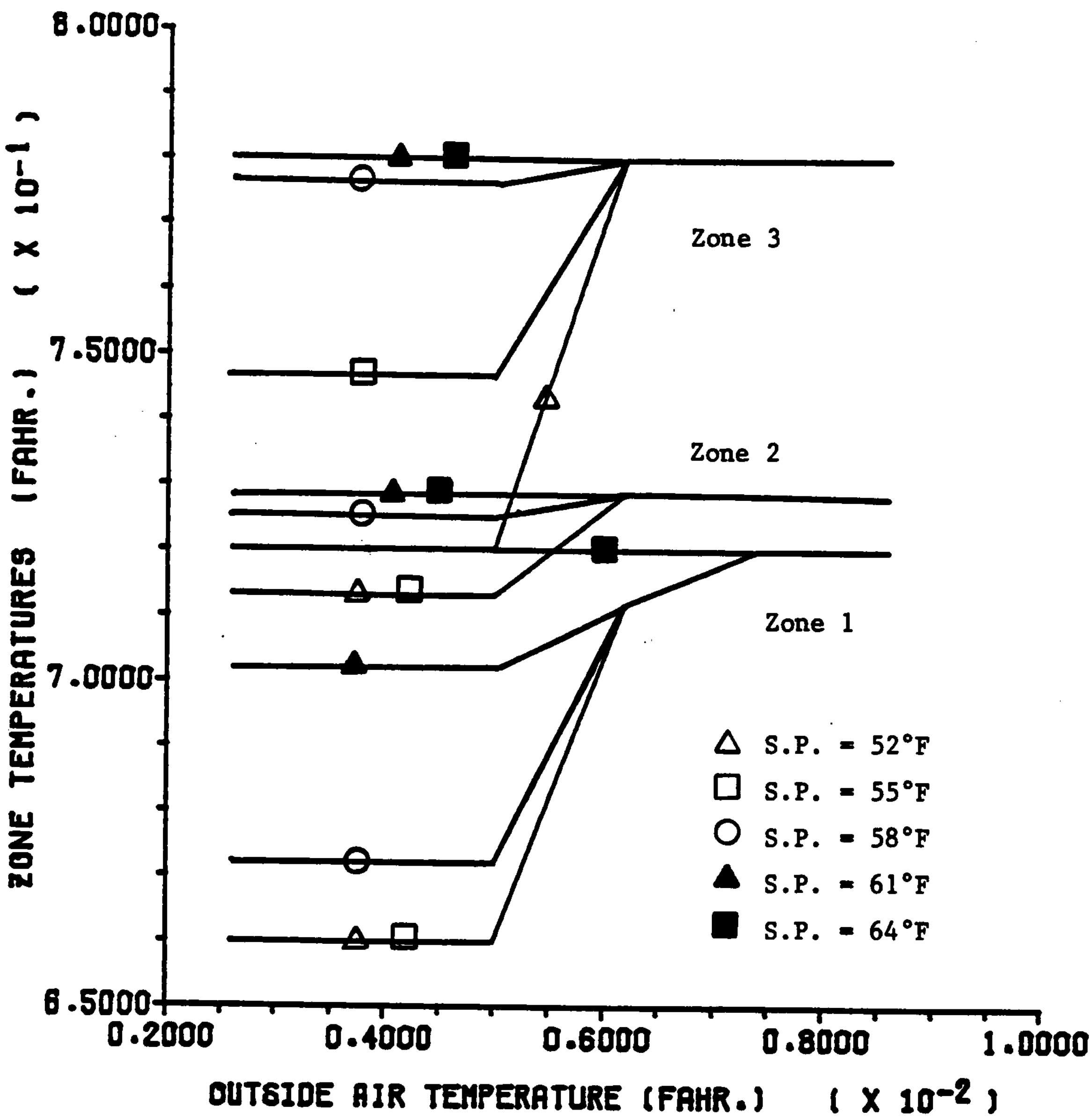


Figure 5-19. Dual-duct mixing box sensitivity studies temperature profiles for unequal zone loads.

temperature set points down to 70°F or up to 74°F will affect the energy consumption.

These simulation studies are made using the "Bins" method with equal load conditions used in the original analysis; with slight zone load variations, decreasing the zone 1 load by 10% and increasing the zone 3 load by 10% and with greater zone load variations, decreasing the zone 1 load by 50% and increasing the zone 3 load by 50%. The results of these studies are presented in Table 5-3 which show the yearly heating and cooling consumption for each set point and load condition. The indicated percentages show the relation to the 72°F set points at that particular load condition.

Table 5-3 shows that for equal loads or for slight zone load deviations, the zone set point has less than a 1% effect on the total energy consumption. This small change is due to the simplified method of calculating zone loads which includes a heat transfer coefficient based on zone temperature set point rather than actual zone temperature. However, for the heavy 50% load variation there are noticeable changes in energy consumption. A set point change to 70°F increases the chilled water energy consumption by 8.4%, but only increases the steam requirements by 0.8%. Thus, depending on the relative costs of the steam and chilled water, a set point change in one direction or the other could prove advantageous if wide load variations exist. The reason for this potential savings lies in the "starving" manner in which the feedback strategy operates. For example, if all three zones are at a temperature above 72°F with the original zone set point, but only one zone has a temperature at 78°F, all three zones will be getting 100% cold duct air that is being cooled only because of the high temperature zone.

TABLE 5-3

YEARLY HEATING AND COOLING CONSUMPTION FOR SUGGESTED FEEDBACK AND MIXING BOX STRATEGY WITH VARYING LOCAL THERMOSTAT ZONE TEMPERATURE SET POINT AND VARYING LOADS

	Local Thermostat Set Point °F		
BTU/hr	70°F	72°F	74°F
Case 1*			
Chilled Water	3.50×10^8 (100.3%)**	3.49×10^8	3.47×10^8 (99.4%)
Steam	0.0	0.0	0.0
Case 2*			
Chilled Water	4.89×10^8 (106.5%)	4.59×10^8	4.20×10^8 (91.5%)
Steam	8.12×10^7	1.22×10^8	1.23×10^8
Case 3*			
Chilled Water	3.81×10^8 (100.5%)	3.79×10^8	3.77×10^8 (99.5%)
Steam	0.0	0.0	0.0

*Case 1 Equal BTU/lbm air flow loads
Case 2 Zone 1 load decreased 50%
Zone 3 load increased 50%
Case 3 Zone 1 load decreased 10%
Zone 3 load increased 10%

**Percentages given are relative to 72°F case.

If the other zones are operating just above 72°F at this time, a set point of 74°F will allow the two cooler zones to receive 100% hot duct air that is not being heated since no zone is at 66°F. These zones would be receiving free circulated air, and the cooling coils have to meet the same cold duct set point, but with a much reduced volume of air. Therefore, a set point change will only produce noticeable results if there is a load variation in which some zones are receiving cooled or heated air when they would be satisfied with free air from the other unconditioned duct.

Damper Position Feedback Versus Zone Temperature Feedback

The remaining dual-duct study involves a comparison between a conventional dual-duct feedback method (2) and the zone temperature feedback method already used here. The conventional method involves using the position of the local zone mixing box dampers to determine the set points for the hot and cold duct. The idea is that the zone damper which is open most to the hot duct should set the set point of that hot duct, and the zone damper which is most open to the cold duct should be used to set the cold duct temperature. For this strategy to be useful the damper which is most open may not be allowed to travel to the full 100% position. If so, once the damper is at 100% open there is no way to determine if it wants to be opened more for additional cooling or heating. Therefore, the damper position set point that must be met will have to be less than 100% to allow controllability. The value chosen for this simulation is 95%. If a damper is open to the cold duct at 95% and it or another damper should open a greater amount to meet the local set point, the cold duct temperature will be reduced to bring the most open damper back to 95%. The same approach is used for the hot duct.

Comparison of these two feedback schemes is possible with only part of the entire dual-duct simulation. For this simulation the temperature leaving the supply fan is set at a constant 67°F. The zone temperature set points are all at a constant 72°F. The zone heat loads are varied from -2,000 BTU/hr in each zone to 100,000 BTU/hr by increments of 10,000 BTU/hr. Since the zones have unequal air flows, this creates an unequal BTU/(lbm air flow) situation from zone to zone. The desired hot and cold duct temperature set points are determined for both feedback strategies, and sensible heating and cooling requirements relative to the 67°F supply fan temperature are calculated. Since actual cooling requirement increases with the addition of latent cooling, the calculation of sensible cooling rather than actual cooling will conservatively favor the method requiring the most cooling.

Figure 5-20 shows the sensible heating requirements as a function of zone load for both feedback methods and Figure 5-21 shows the sensible cooling requirements. Both figures demonstrate that the zone temperature feedback uses much less energy than the damper position feedback. Further inspection shows that for the summed system zone load ranging from 30,000 to 120,000 BTU/hr the damper position feedback requires simultaneous cooling and heating.

One of the reasons for increased energy consumption for the damper position feedback is because it must have a damper position set point less than 100% open, and therefore must always allow hot and cold stream mixing. However, the major problem with not being able to control

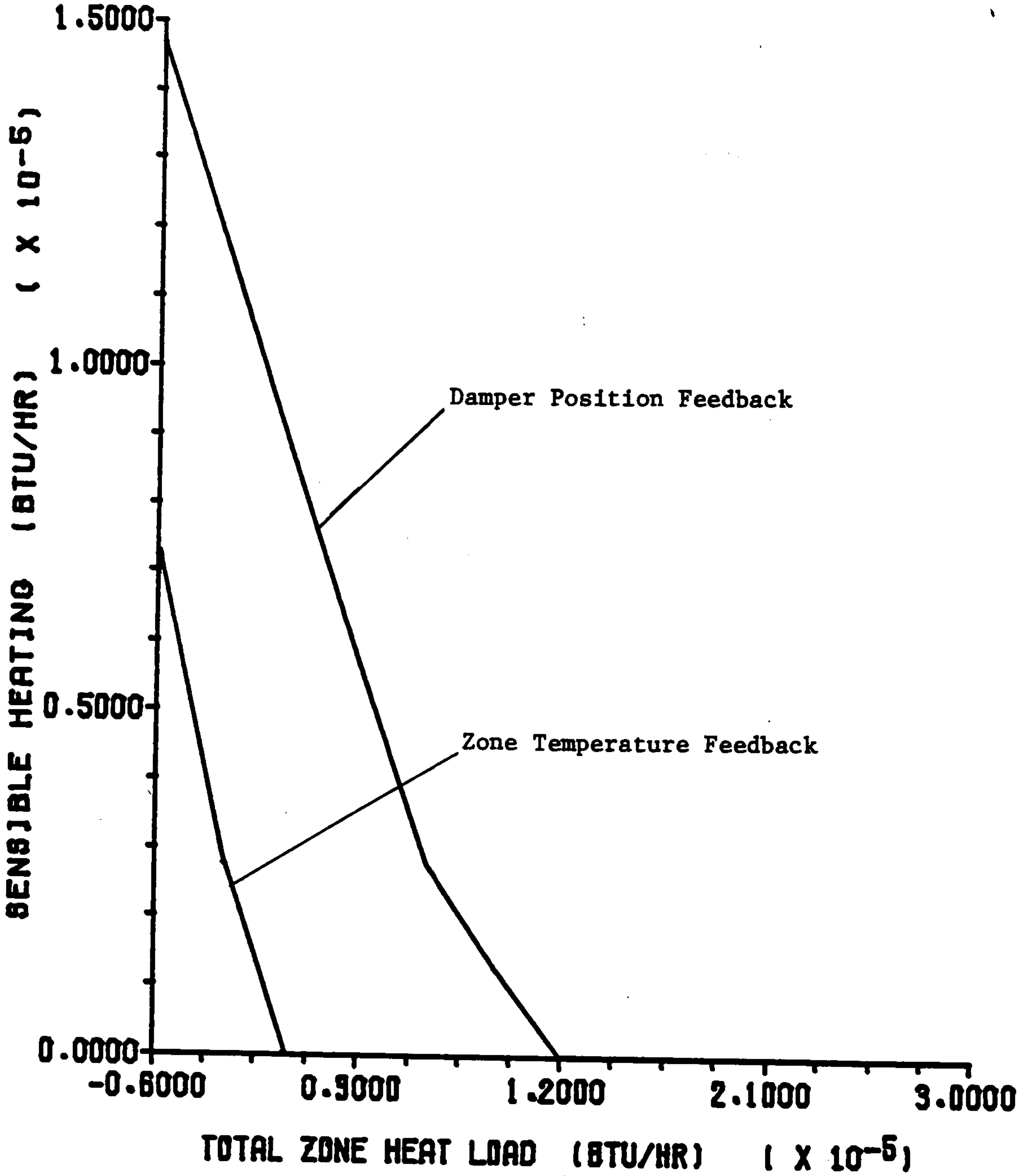


Figure 5-20. Dual-duct sensible heating loads for feedback method comparison.

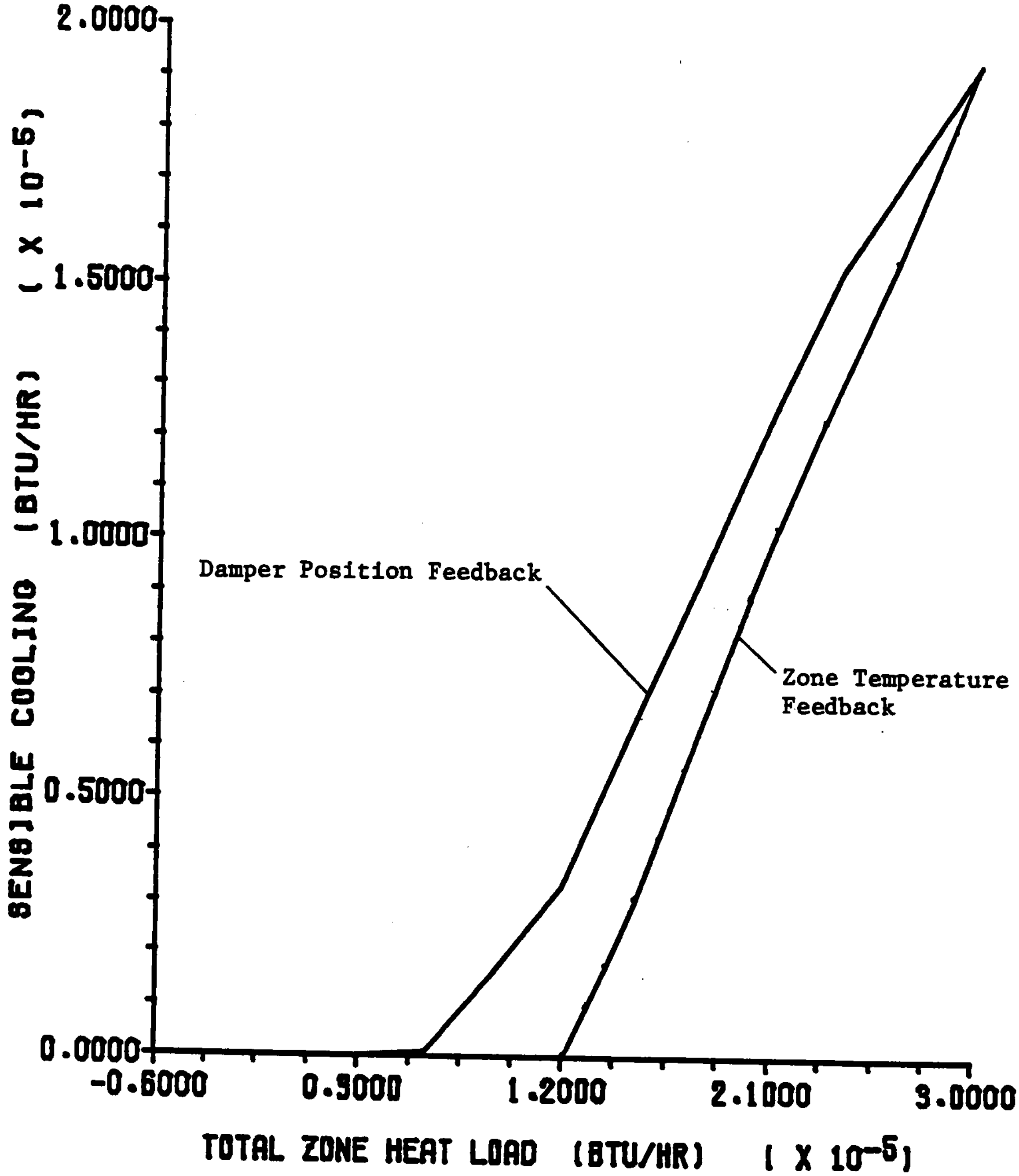


Figure 5-21. Dual-duct sensible cooling loads for feedback method comparison.

the dampers at 100% open is best demonstrated in Figures 5-22 and 5-23 which shows the zone temperatures as a function of total load. Since the dampers are not allowed to be at a 0% or 100% limit, they must be meeting the set point that is controlling them, which in this case is 72°F. However, with zone temperature feedback the dampers are at 0% or 100% and no cooling is necessary until the high set point of 78°F or low set point of 66°F are reached. The damper position feedback does not have the luxury of this zone temperature deviation.

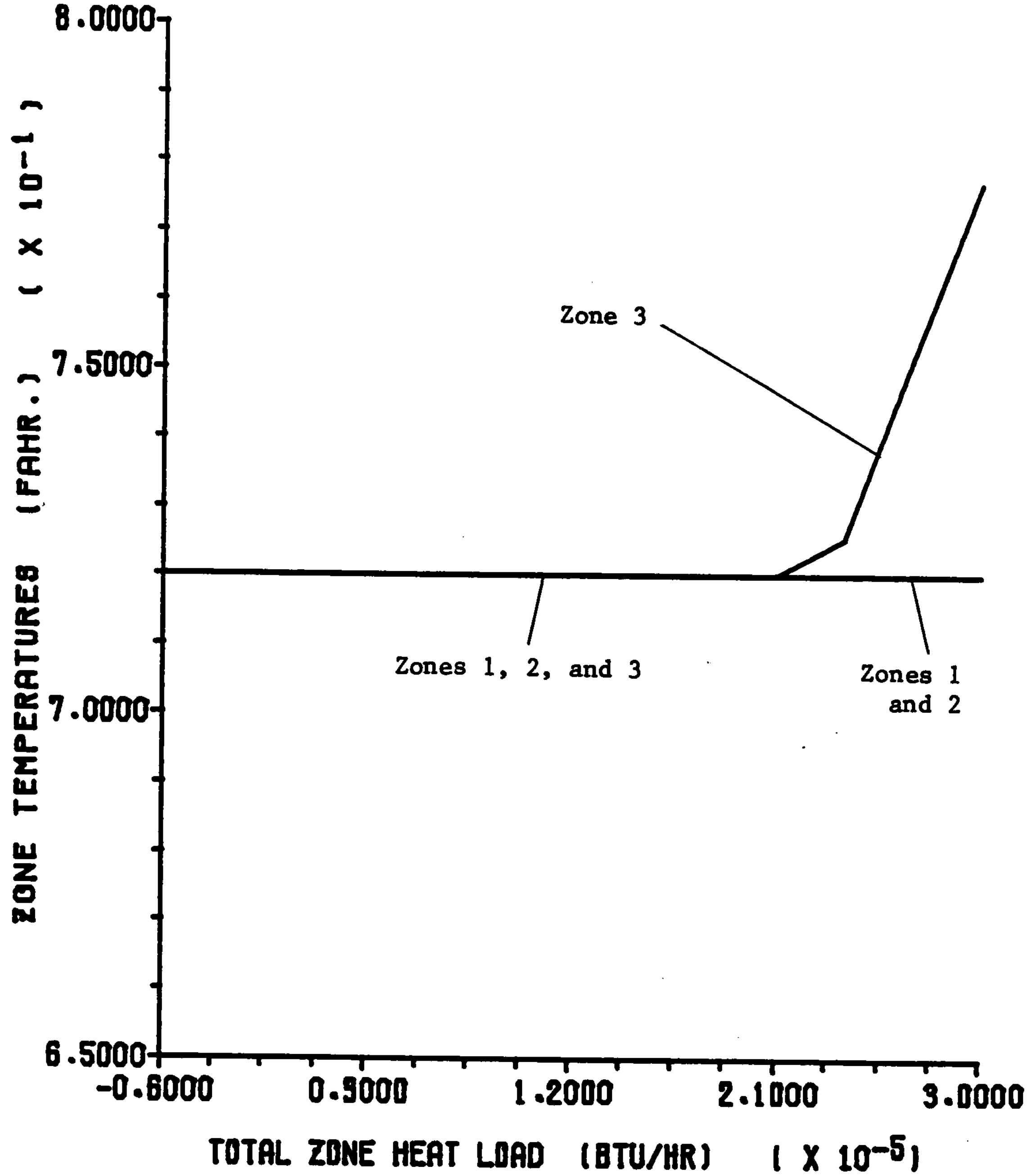


Figure 5-22. Dual-duct feedback method comparison temperature trends with louvre position feedback.

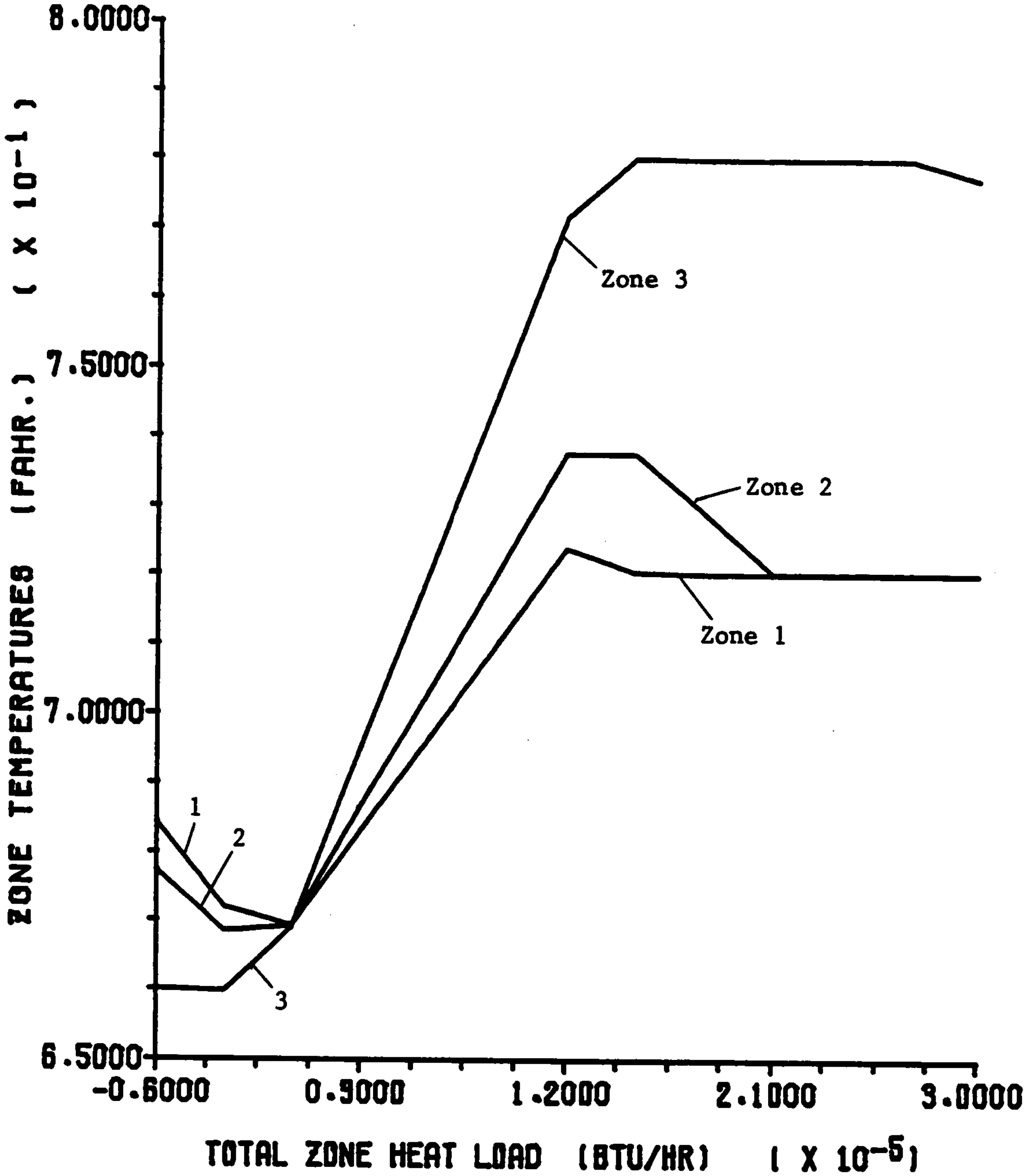


Figure 5-23. Dual-duct feedback method comparison temperature trends with zone temperature feedback.

CHAPTER VI

PERIMETER SYSTEM STUDIES AND RESULTS

The final piece of HVAC equipment studied is the perimeter radiation heating system. As noted in earlier discussions, this system involves a heat exchanger supplying hot water to pipes running along the perimeter walls of a large building. The heat from these pipes is designed to provide an insulating layer of warm air at the perimeter wall of the building. A simple one-dimension steady state analysis is used to describe the heat transfer process involving the outside air, perimeter wall, heat from the pipe, and local room air conditions. Simple heat exchanger equations and recycle equations describe the perimeter piping system.

Simulation of this system is used to demonstrate the differences between a feedforward strategy, such as the original strategy, and a proposed feedback strategy. The original strategy sets the temperature of the water supplied to each perimeter face as a function of outside air temperature, as shown in Figures 2-12 (pg. 35) and 2-13 (pg. 37). Since the purpose of this system is to provide an insulating layer of warm air at the perimeter wall, the proposed control strategies, shown in Figures 6-1 and 6-2, use feedback of the temperature at the perimeter wall to determine the desired hot water temperature. The feedback advantage is that the system adjustments are made according to changes in a desired output variable, rather than a load variable. To allow a computational analysis of these two strategies, hypothetical perimeter and piping sizes are chosen as could be expected in an office facility. A one inch diameter pipe is chosen to carry hot water to each perimeter

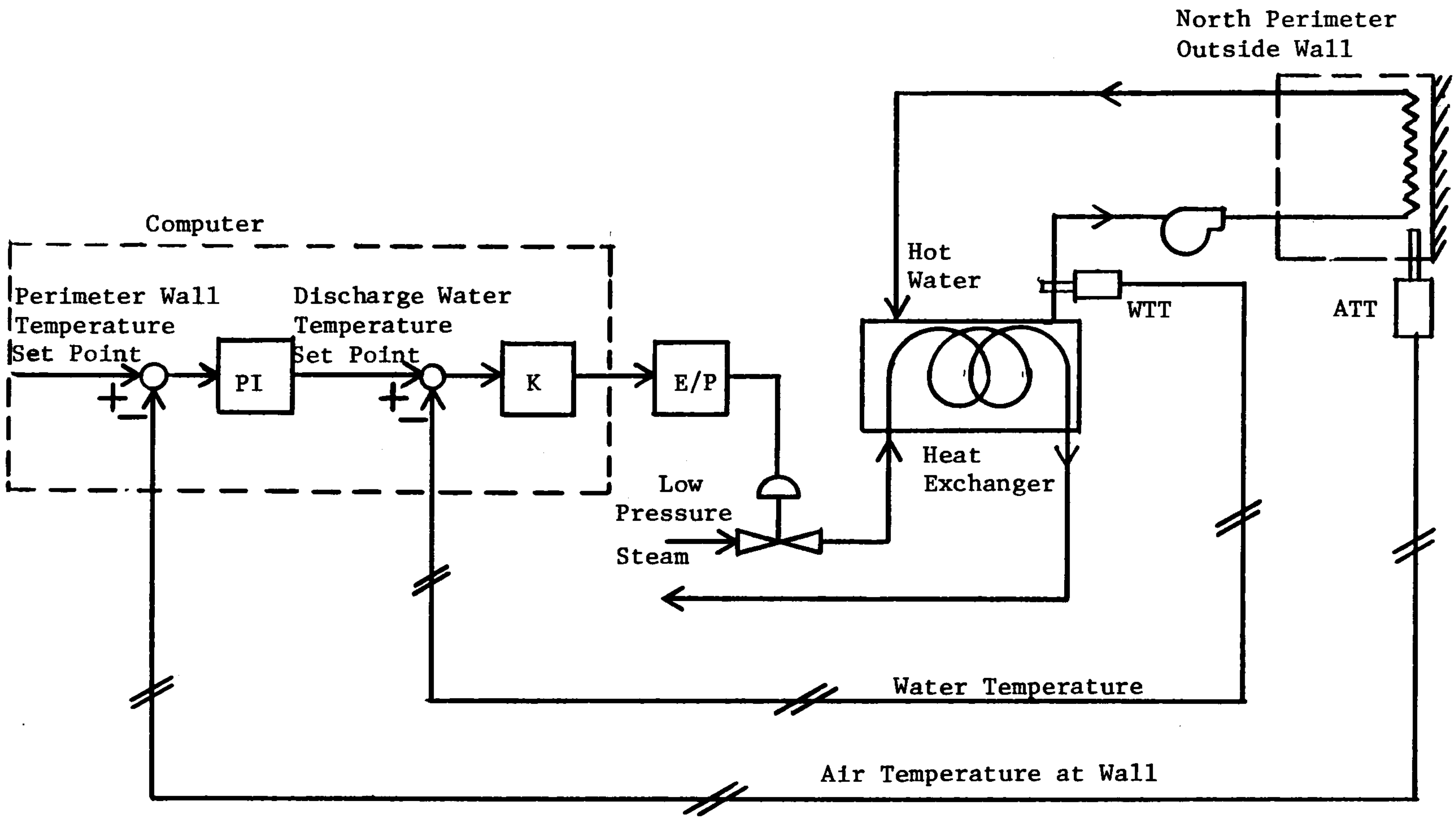


Figure 6-1. Suggested control strategy for main hot water supply.

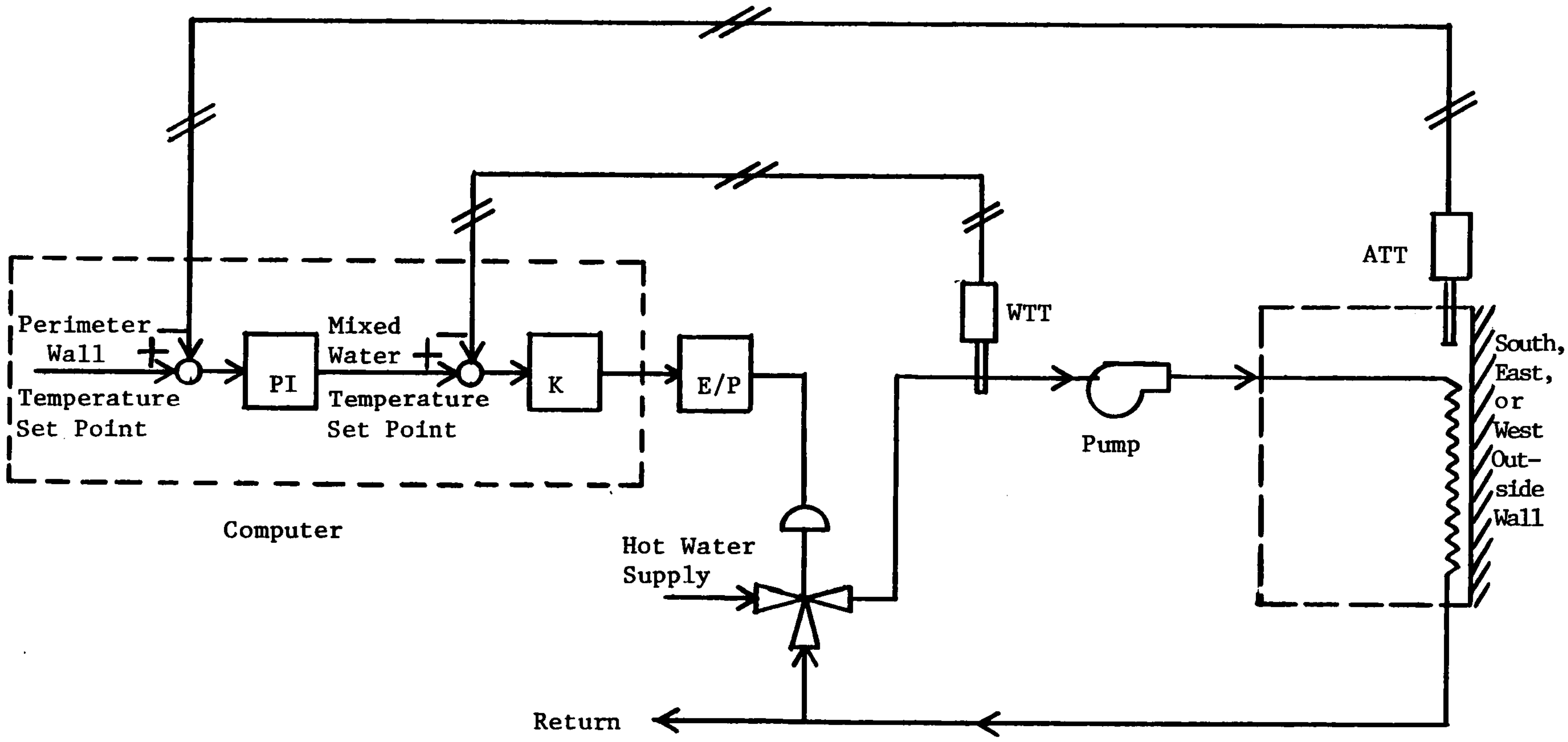


Figure 6-2. Suggested control strategy for south, east, and west perimeter system.

face. The pipe runs along a 150 foot wall that is 12 feet high. The bulk room temperature of 66°F is said to exist at a distance 25 feet from the perimeter wall. The water velocity in the pipe is held at approximately 5 ft/sec.

Values for UW, the lumped heat transfer coefficient, and TO, the outside air temperature, are needed to complete the unknowns needed for system simulation. These two variables will be used to demonstrate the two strategy differences. The original strategy varies the feed water temperature to the north face from 200°F to 100°F as the outside air temperature increases from 0°F to 60°F. Therefore, the range of outside air temperatures from 0°F to 60°F is used in this study. Since values for the lumped transfer coefficient, UW, are not known, this value is varied from 0.05 BTU/ft^2-hr-°F, representing very good insulation to 0.25 BTU/ft^2-hr-°F, representing very poor insulation. Varying UW also presents a method for comparing the two strategies under different load conditions. In reality the heat load that exists at any one temperature will vary according to such things as solar radiation and outside wind velocity. Differing values of UW will provide the same load varying effect.

The simulation runs presented here show a case where equivalent loads are seen on each perimeter face. Values given for temperature at the wall will be the same for each perimeter face. Values for heat from the steam will be for the total steam heat exchanger requirements.

Figures 6-3 and 6-4 show the temperatures that exist at the perimeter wall, remembering that the desired temperature for minimum heat transfer from a bulk room temperature of 66°F is also 66°F. Figure 6-3 shows that with the given feedforward function a wide

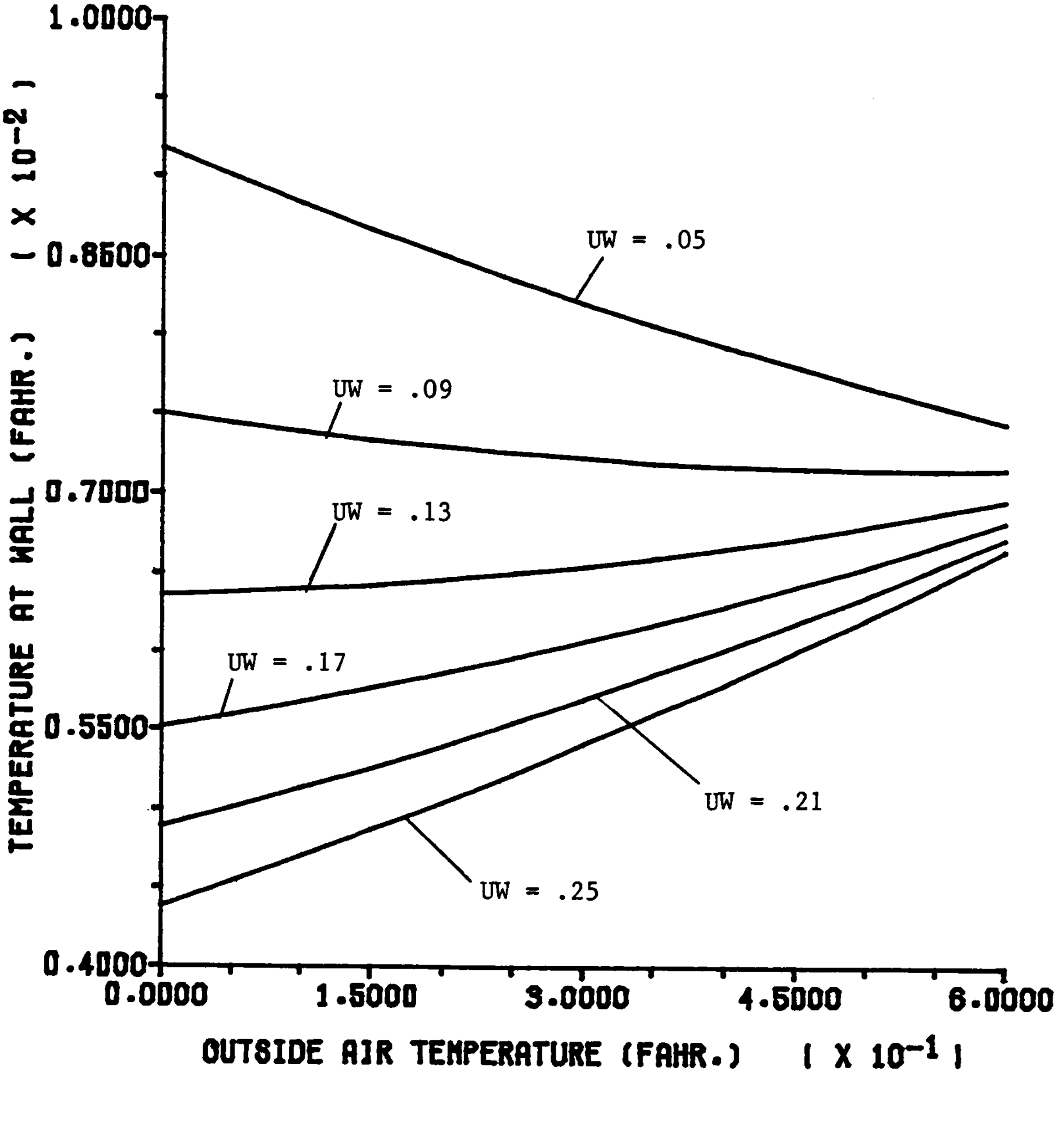


Figure 6-3. Perimeter system wall temperatures as a function of outside air temperatures with original control strategy.

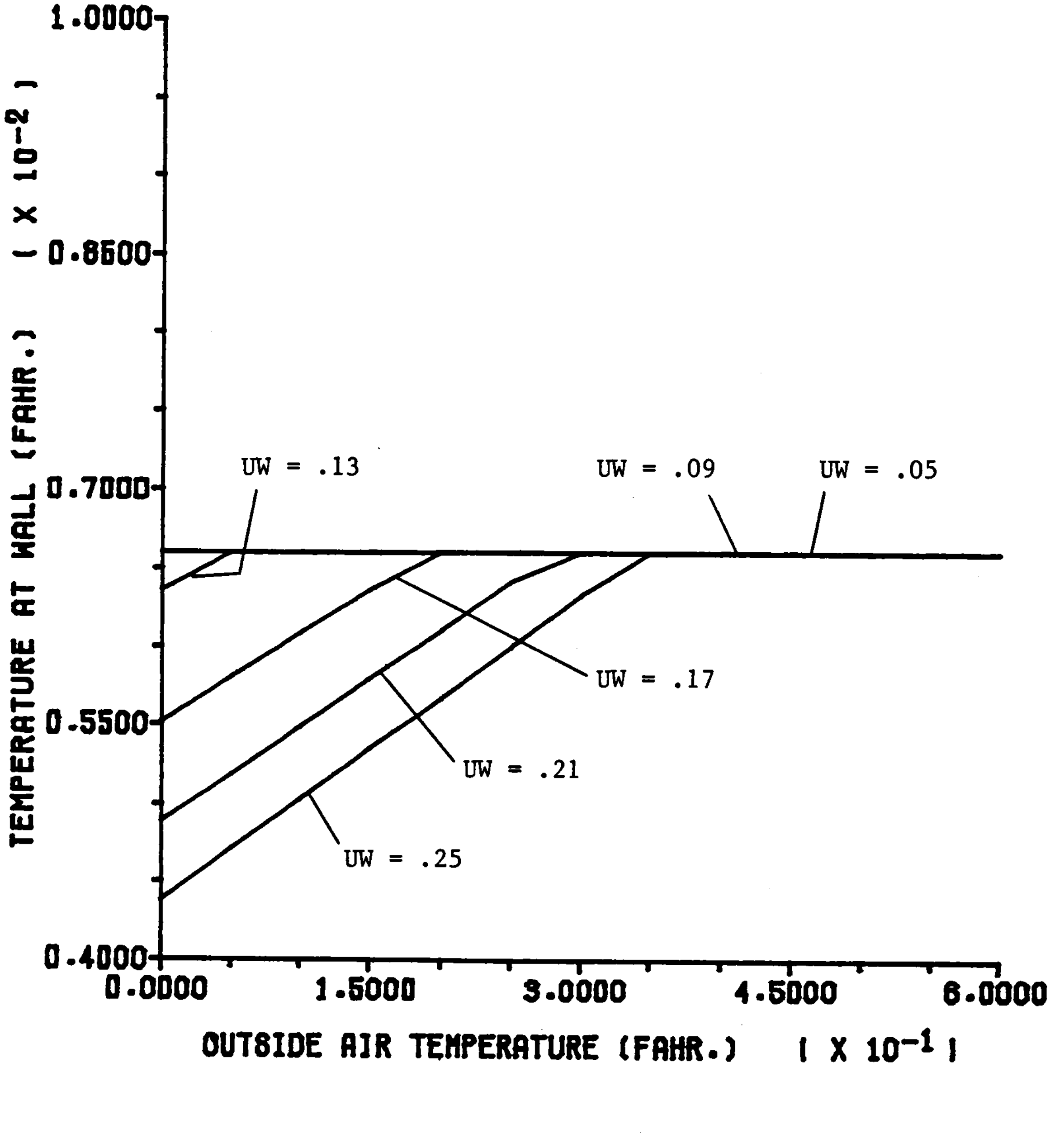


Figure 6-4. Perimeter system wall temperatures as a function of outside air temperatures with feedback control strategy.

variety of wall temperatures may exist. With very good insulation, the temperatures are too high, and with very poor insulation the temperatures are too low. For this hypothetical room a value of UW=0.13 comes closest to meeting the desired 66°F. This supports the basic idea for a feedforward strategy that an appropriate function to fit the system can give reasonable control. However, recalling that varying UW also represents a varying load we realize this strategy cannot consistently keep the wall temperature at its desired value.

A look at Figure 6-4 shows that except for low temperatures under high transfer rate conditions, the feedback strategy will always meet the desired 66°F wall temperature. The reason for the low wall temperatures for the cases seen, is that the control strategy is limited by the process. A check of the simulation data shows that for every point below this 66°F line, the hot water entering the pipes is at a high limit of 200°F. Comparing this with the original strategy shows that for each of these low wall temperature cases, the wall temperatures are identical at an outside air temperature of 0°F, with both having 200°F water entering the pipes. However, the original strategy decreases this water temperature as outside air temperature increases, while the feedback strategy holds this temperature at 200°F until the desired 66°F set point is reached.

The major reason for studying this system is to determine a more energy efficient operation. However, it must be done without sacrificing comfort. A comparison of the steam requirements for these strategies is seen in Figures 6-5 and 6-6. These figures show that at very low transfer coefficients, much less steam is required using the feedback strategy, which is also giving the desired wall temperature. At high

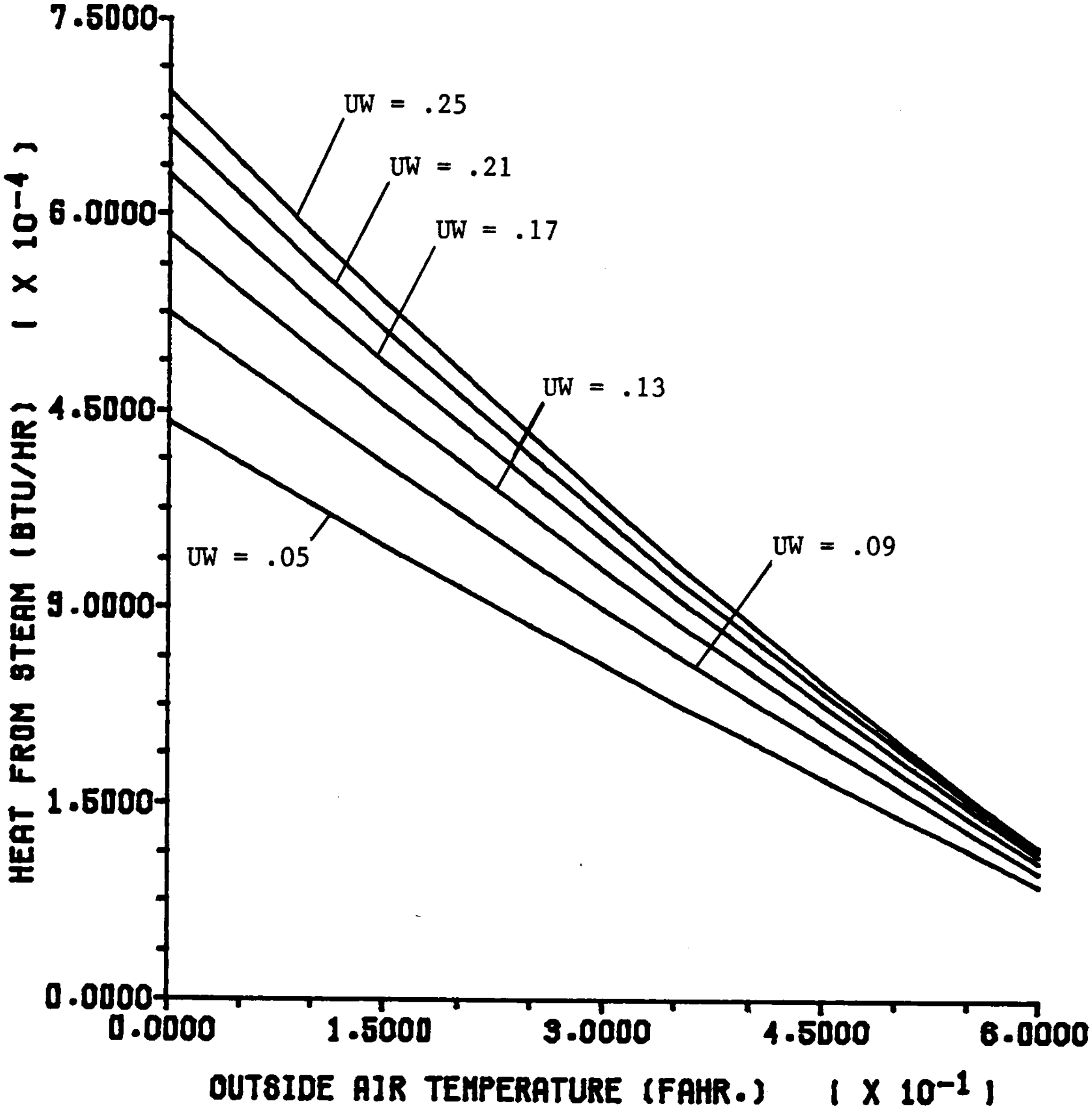


Figure 6-5. Perimeter system heating load as a function of outside air temperature with original control strategy.

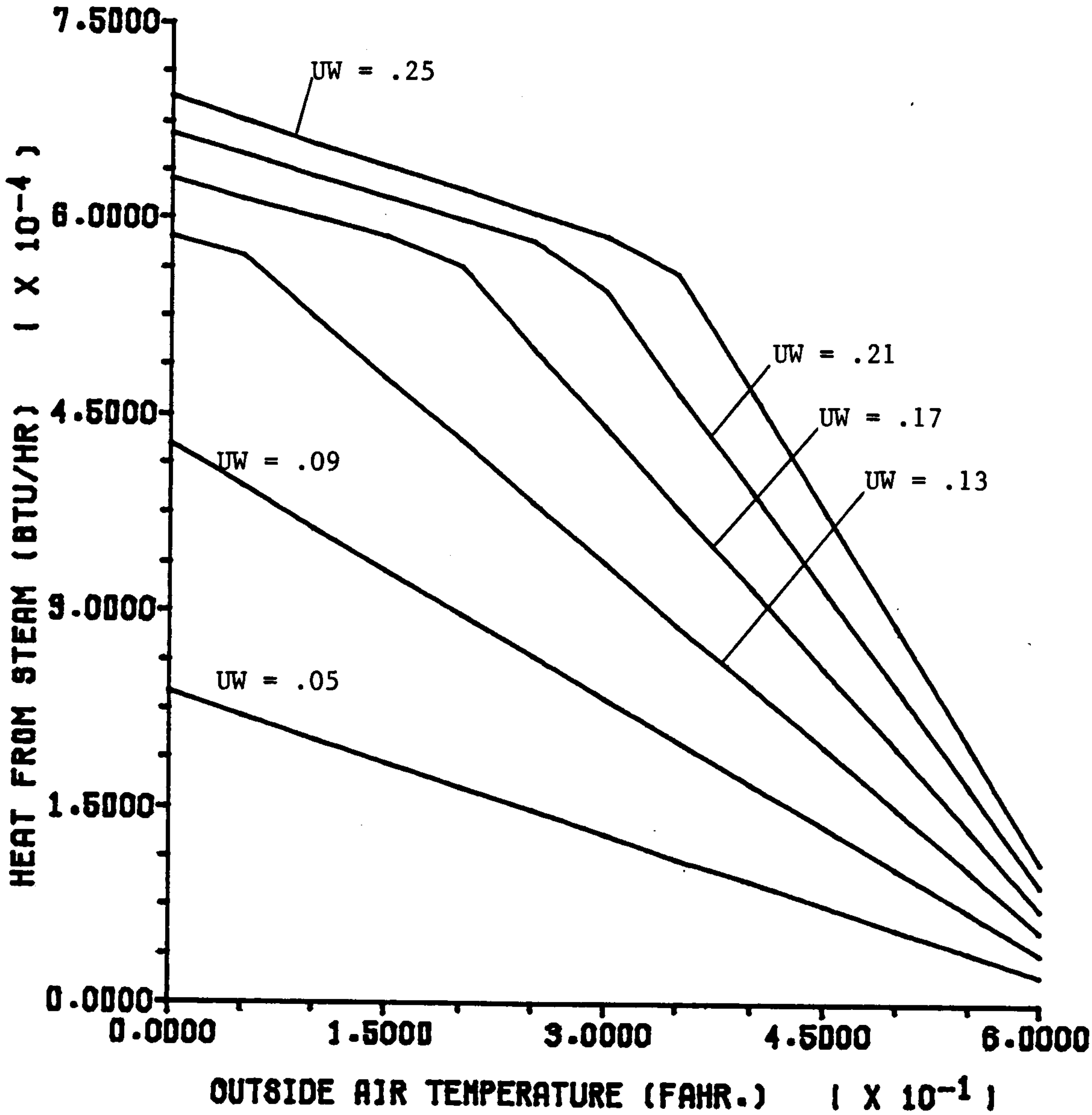


Figure 6-6. Perimeter system heating load as a function of outside air temperature with feedback control strategy.

transfer coefficients the original strategy uses less steam, but a check of the temperature plots shows that it never reaches the desired wall temperature. At these high transfer rates the feedback strategy pushes whatever it can get out of the heat exchanger to attempt to meet the wall set point. The bends that are seen in the steam loads for the high transfer coefficient cases for the perimeter system represent the points where the exchanger is no longer running at maximum capacity and has begun to meet set point.

The important point to remember in this feedforward-feedback comparison is that the feedback strategy will push as much as possible to meet the desired wall temperature, and meeting set point in the number one goal for most control systems. However, the feedback strategy will use no more energy than is necessary, as evidenced by no wall temperatures above 66°F. Conversely, the feedforward strategy meets set point in only a small number of cases, and is either wasting energy by making the temperature at the wall too hot, or not using enough energy to allow the system to perform its designed function of providing the insulating layer of air.

CHAPTER VII

CONCLUSIONS AND RECOMMENDATIONS

The purpose of investigating these three HVAC systems is to determine more energy-efficient methods of operating or controlling them. The results of the steady-state simulations show several areas where this may be accomplished.

For the single-duct air handler an investigation shows the possible energy waste due to manual operation of a summer/winter control strategy select switch. If the switch is put in summer mode too early, there is a loss of free cooling from outside air. If the switch is left in winter mode too long, the strategy allows warm outside air into the system while exhausting better quality return air. It is suggested that a cascaded computer control strategy, which has both a cooling and heating control loop running continuously, with a mixing box strategy allowing free cooling from the outside air when it is desired and possible, will provide a more energy-efficient operation. This strategy will only heat when the zone drops to 66°F and only cool when the zone rises to 78°F.

A further investigation on the single-duct air handler shows that the optimum reheat set point in a dehumidification strategy is at as high a temperature as possible. This is limited only to being lower than the high zone temperature set point of 78°F to eliminate possible control loop fighting.

Study of the dual-duct air handler shows that change of the hot and cold duct temperature control loops from feedforward and straight set point to a cascaded strategy, which determines these duct temperature set points according to the feedback temperatures of the hottest and coldest zones, can eliminate heating requirements and reduce cooling coil energy needs to 57% of the base strategy value for the investigated loads. A further reduction of cooling energy to only 29% of the original base strategy value can be accomplished by the addition of an air handler mixing box strategy which allows free cooling from outside air whenever possible.

A study of mixing box set point sensitivity shows that the energy usage of the dual-duct air handler is dependent on the mixing box temperature set point. However, the actual desired value can only be determined according to the system load. High loads require a low mixing box set point, and low loads a high mixing box set point. In other words, if free cooling is desired, as in the high load case, allow as much as possible. A suggested study from this mixing box investigation would be to determine a possible higher level strategy that would set the mixing box set point on the basis of zone needs, much as is done for the single-duct air handler, except there must be provisions for the multiple zone serving feature of the dual-duct air handler. Another suggested study is to determine the possible energy savings that can be realized by the addition of a hot gas bypass duct. The idea is to recycle warm return air directly to the hot duct as needed, without suffering the energy loss associated with the mixing with cold air in the air handler mixing box.

A zone temperature set point sensitivity study for the dual-duct air handler shows that changing this set point is only effective if there is a wide variation in the zone loads. Otherwise, the change in energy requirements is less than 1% for the investigated cases.

Comparison of zone temperature feedback versus a literature suggested louvre position feedback for determination of the dual-duct hot and cold duct temperature set points shows that the zone temperature feedback uses less cooling and heating energy. Most of this savings is because the zone temperature feedback does no cooling or heating between zone temperatures of 66°F and 78°F while the louvre position feedback requires all zones to operate at their thermostatic set point.

The final system studied is the perimeter radiation heating system. It is shown that a strategy which sets the pipe water feed temperature as a feedback function of the desired perimeter wall temperature provides desired system operation with minimal energy usage. The original strategy uses a feedforward function of outside air temperature to set feed water temperature and therefore meets the desired wall temperature in only a small number of cases. It either uses too much steam and produces a hotter wall than needed, or too little steam and a cooler wall than needed.

A common statement for all three HVAC systems is that there are more energy-efficient ways to operate these systems than the original control strategies. In many cases the savings appear to be very significant. The question is whether the suggested changes will be energy-efficient enough to warrant the capital expenditures necessary for implementation. It is therefore recommended that these suggested strategies be further investigated using a more comprehensive actual load analysis and the results applied to a rigorous economic analysis to determine installation feasibility.

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APPENDICES

APPENDIX A

LIST OF SYMBOLS AND ABBREVIATIONS

A - Parameter in Equation 3.20 used to account for the moisture in the air in determining the ratio of sensible to latent cooling for the cooling coils.

ATT - Air temperature transmitter

B - Parameter in Equation 3.20 used to account for the temperature of the cooling coil in determining the ratio of sensible to latent cooling for the cooling coil.

BRCW - Base performance rating for cooling coils defined in Equation 3.9 (BTU/hr per square foot of face area per row of tubes deep per log mean temperature difference).

BRCW1, BRCW2 - Redefined base performance rating terms for cooling coils defined in Equations 3.21 and 3.22.

BW - Parameter in Equation 3.10 giving difference in entering air dry bulb temperature entering chilled water temperature for cooling coil calculations (°F).

b1 to b2 - Coefficients in cooling coil Equation 3.10 for wetted surface factor.

C - Adjustable parameter in cooling coil equation.

CPAIR - Heat capacity of air (BTU/lb°F).

Cp, CPW - Heat capacity of water (BTU/lb°F).

CWFLO - Chilled water flow rate for cooling coils (lbs/hr).

CWMAX - Maximum cooling coil chilled water flow (lbs/hr)

C1 to C9 - Coefficients for cooling coil Equation 3.9 for base rating.

DB - Dry bulb air temperature (°F).

DELH - Change in air enthalpy across supply fan (BTU/lb).

DELTT - Change in air temperature across supply fan (°F).

DLH - Latent heat of vaporization for water vapor in zone water loads (BTU/lb).

Dp - Pipe diameter in perimeter system.

DP	- Dew point of air (°F).
D1	- Temperature difference in cooling coils between entering air dry bulb and leaving water (°F).
D2	- Temperature difference in cooling coils between leaving air dry bulb and entering water (°F).
EDB	- Cooling coil entering air dry bulb temperature (°F).
EDP	- Cooling coil entering air dew point temperature (°F).
EFF	- Fan efficiency (%).
E/P	- Electronic to pneumatic signal transmitter.
EWT	- Cooling coil entering water temperature.
F, FI, FO, FJ	- In Equations 3-1 and 3.2 represent air stream flows (lbs/hr).
F	- In Equation 3.9 represents constant total water flow through perimeter system pipes.
FA	- Cooling coil face area (ft^2).
FAIR	- Air flow rate across cooling coils (lbs/hr).
FEX	- Air flow through supply fan (lbs/hr).
FPM	- Air-side face velocity across cooling coils (ft/min).
FPS	- Water-side velocity in cooling coils (ft/sec).
FR	- Water flow returning to heat exchanger from perimeter system pipes (lbs/hr).
FREC	- Recycle water flow back to perimeter system pipes (lbs/hr).
FST	- Steam flow in air handler heating coils (lbs/hr).
GPM	- Cooling coil chilled water flow rate (gallons/min).
H	- Air enthalpy (BTU/lb).
HBHC	- Air enthalpy before heating coils (BTU/lb).
HAHC	- Air enthalpy after heating coils (BTU/lb).
hp	- Perimeter pipe heat transfer coefficient (BTU/ft^2-hr-°F).
Hr	- Height of perimeter wall (ft).
ID	- Inner diameter of coils in cooling coils (ft).

K, Kc, K1, K2, K3, Kn - Proportional controller gains.

Km - Mixing heat transfer coefficient for mixing air between perimeter wall and bulk room air (BTU/ft^2-hr-°F).

LDB - Leaving air dry bulb temperature from cooling coils (°F).

Lr - Length of perimeter wall (ft).

LWT - Leaving water temperature from cooling coils (°F).

MAX - Maximum air flow dampers

M.B. - Mixing box.

MIN - Minimum air flow dampers.

MLTD - Log mean temperature difference for cooling coils (°F).

NC - In Figures = normally closed control actions.

NC - In Equation 3.8 = number of parallel circuits in cooling coils.

NO - Normally open control action.

NR - Number of rows of cooling coils.

P - Proportional controller.

PI - Proportional-integral controller.

psi - pressure signal in lbs/in^2.

PW - Parameter in Equation 3.10 giving difference in entering air dew point and entering water temperature for cooling coil calculations (°F).

QEX - Heat requirement for perimeter system heat exchanger (BTU/hr).

QL, QLI, QLO - Heat loads for zones (BTU/hr).

qr - Heat transfer by mixing between room and perimeter wall air (BTU/ft^2-hr).

Qs - Heat transfer from perimeter pipe to surrounding air (BTU/hr).

qs - Heat transfer from perimeter pipe to surrounding air (BTU/ft^2-hr).

QTCA - Cooling coil heat transferred from air to water (BTU/hr).

QTW - Cooling coil heat gain by chilled water from air (BTU/hr).

qw - Heat transferred through perimeter wall (BTU/ft^2-hr).

RATO - Ratio of sensible to latent cooling for cooling coil model.

RH - Air relative humidity.

RHSP - Air relative humidity set point.

S.P. - Set point.

STFL - Steam flow in perimeter system heat exchanger (lbs/hr).

STLH - Latent heat of steam (BTU/lb).

T - Air dry bulb temperature (°F).

TACC - Desired air dry bulb temperature after cooling coils (°F).

To.A., TOUT, TO - Outside air temperature (°F).

$\overline{T}p$ - Average water temperature in perimeter pipe (°F).

Ts.P. - Set point temperature (°F).

TWM - Mixed water temperature from all perimeter faces returning to system heat exchanger (°F).

Tw_i - Water temperature entering perimeter face pipe (°F).

Tw_iN - Water temperature leaving perimeter system heat exchanger (°F).

Tw_o - Water temperature leaving perimeter face pipe (°F).

T1 - Perimeter wall plane air temperature (°F).

T2 - Bulk room air temperature in perimeter simulation (°F).

UW - Lumped heat transfer coefficient through perimeter wall (BTU/ft^2-hr-°F).

V - Air specific volume (ft^3/lb dry air).

W - Air water content (lbs water/lb sry air).

WB - Air wet bulb temperature (°F).

WDEH - Water removed from air by cooling coil operation in single-duct simulation (lbs/hr).

WEX - Water leaving with exhaust air in single-duct simulation (lbs/hr).

WF - Work of supply fan (HP).

WLZ - Zone water load in single-duct simulation (lbs/hr).

WOD - Water entering with outside air intake in single-duct simulation (lbs/hr).

WSF - Wetted surface factor rating for cooling coil model given by Equation 3.10.

WTT - Water temperature transmitter.

APPENDIX B

COMPUTER ROUTINE NAMES AND PURPOSE

AIRSM - Called by both single-duct and dual-duct routines to make mass and energy balances given loads and two input air streams. Calculates condition of output air stream.

AIRSM2 - Called by both single-duct and dual-duct routines to make mass and energy balances given loads, one input air stream and one output air stream condition. Calculates condition of other input stream.

BRCWCL - Called by CWCOIL to calculate the base rating function of the chilled water coil.

CALALL - Single-duct air handler routine that calls all the component subroutines. This routines uses ZEROUT to find zone relative humidity that allows convergence of system water balance.

CWCCAL - Chilled water coil routine that uses ZEROUT to find chilled water flow that allows meeting desired air temperature.

CWCOIL - Routine called by CWCCAL that gives chilled water coil performance for a given chilled water flow.

CONTRO - Single-duct air handler routine that applies control strategy. Calls CACALL to evaluate different set points.

DDBC - Main calling routine for dual-duct base case. Provides all system loads.

DDLFCN - Routine providing dual-duct balancing equations to be used by ZSCNT for system convergence.

DDLOOP - Routine which calls alll dual-duct unit operation routines when provided loads and set points.

DUCTEX - Single-duct air handler routine using heat exchanger approach to describe heat transfer between zone supply air and return air.

FREECL - Single-duct routine called to find zone conditions when no steam or chilled water is required.

HEACOL - Heating coil subroutine.

MAXMIN - Dual-duct routine used to determine the high and low feedback valves from a set of zone temperatures.

MIXBOX - Air handler routine used to calculate return, exhaust, and outside air flows for a given mixing box control strategy.

NEWPER - Main routine for suggested feedback perimeter simulation.

OLDPER - Main routine for original feedforward perimeter simulation.

PSYCHO - Psychrometric routine calculating all other psychrometric properties given any two properties plus atmospheric pressure.

PSOAT - Routine containing original perimeter system control strategy.

PSWT - Routine containing suggested feedback perimeter system control strategy.

QCWCAL - Routine doin energy balances required by CWCOIL.

REHEAT - Single-duct routine doing cool and reheat if required for dehumidification.

SDBC - Main calling routine for single-duct base case. Provides all system loads.

SFAN - Routine calculating input fan air conditions given fan size, fan efficiency and output air conditions.

SFAN2 - Routine calculating output fan air conditions given for size, fan efficiency and input air conditions.

SSYCHO - Simplified version of PSYCHO which calculates only selected air psychromatic properties.

ZEROUT - Single variable search routine written by C. Russell Smith.

ZSCNT - Multiple variable search routine used from IMSL Library Routines.

ZTMXBX - Dual-duct routine describing action of local zone mixing boxes.

APPENDIX C

COMPUTER PROGRAM LISTINGS

```
C
C
C                SINGLE-DUCT AIR HANDLER SIMULATION
C
C
C
C
C
C                SINGLE DUCT BASE CASE
C
C
C
C       MAIN SINGLE DUCT ROUTINE
C       THIS ROUTINE PROVIDE LOADS AND THEN CALLS ROUTINE "CCNTRO"
C
C       THIS ROUTINE USES THE SAME VARIABLE NOTATICN AS "CALALL"
C
        DIMENSICN OAT(100),OURS(1C0)
        NAMELIST/CNE/OADB,OARH,ZOCB,PATM,WLIZ,WLOZ,QLIZ,QLOZ,
       1ZFO,DLHI,DLHO,QOARA,UAFAC,CPAIR,WFAN,EFFAN,OAMIN,FOA,FRA,
       2WATEXH,WATINP,WATREM,NUM,RCOMRH,ZTI,ZHI,TDI,HDI,TRAC,HRAC,
       3TINFAN,HINFAN,TAC,HAC,FST,FCHW,HHEAT,HCOOL,TMB,HMB,IFLAG,DWATBL,
       4TSPHI,TSPLO,RHSPHI,TRHSP,NCAT,HS,HC,FS,FC,HSYR,HCYR,FSYR,
       5FCYR
        COMMON/SMIX/MXSWIT
C
C SINGLE DUCT PARAMETERS
C
C
C IF SUMMER SET MXSWIT=2
C IF WINTER SET MXSWIT=1
C
        MXSWIT=2
        ZFAC=.5
        FRESP=72.
        TRHSP=76.
        TSPHI=78.
        TSPLO=66.
        RHSPHI=65.
        DLHI=1037.9
        DLHO=1037.9
        ZFO=70000.
        OAMIN=0.15*ZFO
        FSMAX=855.
        CWMAX=110145.
        CWEWT=44.
        PATM=29.921
        WFAN=18.5
        EFFAN=.7
        CPAIR=.24
        ASTLH=962.2
```

```
      ZCDB=78.
      CARH=50.
C
C
C READ IN BINS DATA
C
C
      DO 50 J=1,100
      READ(20,*,END=60)OAT(J),OURS(J)
      NDAT=NDAT+1
 50   CONTINUE
C
C GIVE VALUES FOR LOADS
C
 60   AREA=19500.
C ZONE WALL OUTSIDE EXPOSURE AREA WITH 12 FOOT CEILING-SQUARE ROOM
      FAREA=((AREA)**.5)*12.
C TRANSFER COEF. TO OUTSIDE
      UAZ=.1*FAREA
C PEOPLE IN ZONE 1/250  SQUARE FEET
      PEOPZ=78.
C EQUIPMENT GAINS ASSUME 100 WATTS PER PERSON
      EQUIPZ=PEOPZ*100.*3.4122
C ELECTRIC LIGHT GAINS 2 WATTS/FT**2 FLOOR AREA
      ELITE=AREA*2.*3.4122
C PEOPLE SENSIBLE LOAD  250 BTU/PERSON
      PEEPS=PEOPZ*250.
C ROOF TRANSFER COEF.
      UOARA=.15*AREA
C DUCT TRANSFER COEF.
      UARAD=30.
C
C
C PUT IN LARGE LOOP TO RUN EACH BIN CASE
C
C
      DO 500 I=1,NDAT
      CADB=OAT(I)
      THOURS=OURS(I)
      TSPLOD=(TSPLO+TSPHI)/2.
C
C GIVE WATER AND HEAT LOADS FOR ZONE
C
      OATP=CADB
      IF(OATP.LE.66)OATP=66.
      QLIZ=(UAZ*(OATP-TSPLOD)+EQUIPZ+ELITE+PEEPS)*ZFAC
      WLIZ=PEOPZ*.24
      QCARA=UOARA*(OADB-TSPLOD)
C
C CALL CONTROL STRATEGY
C
      CALL CONTRO(OADB,OARH,ZCDB,PATM,WLIZ,WLOZ,QLIZ,QLOZ,
     1ZFO,DLHI,DLHO,QOARA,UARAD,CPAIR,WFAN,EFFAN,OAMIN,FCA,FRA,
     2WATEXH,WATINP,WATREM,NLP,RCOMRH,ZTI,ZHI,TOI,HOI,TRAC,HRAO,
     3TINFAN,HINFAN,TAC,HAC,FST,FCHW,HHEAT,HCOOL,TMB,HMB,IFLAG,DWATBL,
     4TSPHI,TSPLO,RHSPHI,TRHSP,TSPDEH,FRESP)
C
C GET RESULTS FOR YEARLY BASIS
C
      HS=OURS(I)*HHEAT
```

```
      HC=CURS(I)*HCOOL
      FS=OURS(I)*FST
      FC=CURS(I)*FCHW
      HSYR=HSYR+HS
      HCYR=HCYR+HC
      FSYR=FSYR+FS
      FCYR=FCYR+FC
C WRITE RESULTS
C
      WRITE(6,ONE)
      WRITE(26,410)OAT(I),OURS(I),QLIZ,WLIZ,ZODB,ROOMRH
 410  FORMAT(1X,6(1PE15.5,2X))
      WRITE(27,420)OAT(I),HHEAT,HCOOL,FST,FCHW,TMB,TAC
 420  FORMAT(1X,7(1PE15.5,2X))
      WRITE(28,430)OAT(I),OURS(I),HS,HC,FS,FC
 430  FORMAT(1X,6(1PE15.5))
      WRITE(6,345)
 345  FORMAT('     '///)
      WRITE(6,346)
 346  FORMAT(' NEXT CASE '//)
 500  CONTINUE
      STOP
      END
C
C
C
C
C               SUBROUTINE CALALL
C
C
C PURPOSE:
C      THE PURPOSE OF THIS SUBROUTINE IS TO CALL ALL COMPONENT
C SUBROUTINES OF THE SINGLE DUCT AIR HANDLER SIMULATION.  THIS
C ROUTINE IS RESPONSIBLE FOR DETERMINING THE ROOM RELATIVE
C HUMIDITY THAT WILL GAURANTEE CONVERGENCE OF THE WATER BALANCE.
C
C INPUT:
C      OADB=OUTSIDE AIR DRY BULB TEMPERATURE              F
C      OARH=OUTSIDE AIR RELATIVE HUMIDITY                 %
C      ZODB=ZONE DRY BULB TEMPERATURE (SET POINT)         F
C      PATM=ATMOSPHERIC PRESSURE                          IN HG
C      WLIZ=WATER LOAD INTO ZONE                          LBS H2O/HR
C      WLOZ=WATER LOAD OUT OF ZONE                        LBS H2O/HR
C      QLIZ=HEAT LOAD INTO ZONE                           BTU/HR
C      QLOZ=HEAT LOAD OUT OF ZONE                         BTU/HR
C      ZFO=AIR FLOW THROUGH ZONE                          LBS/HR
C      DLHI=LATENT HEAT OF WATER                          BTU/LB
C      DLHO=LATENT HEAT OF WATER                          BTU/LB
C      QOARA=HEAT GAIN IN RETURN AIR DUCT FROM OUTSIDE BTU/HR
C      UARAD=HEAT TRANSFER COEFFICIENT BETWEEN SUPPLY
C            DUCT AND RETURN AIR DUCT                     BTU/HR-F
C      CPAIR=HEAT CAPACITY OF AIR                         BTU/LB-F
C      WFAN=HORSEPOWER OF FAN                             HP
C      EFFAN=FAN EFFICIENCY
C      OAMIN=MINIMUM OUTSIDE AIR REQUIREMENT              LBS/HR
C
C
C OUTPUT:
C      FOA=OUTSIDE AIR FLOW                               LBS/HR
C      FRA=RETURN AIR FLOW                                LBS/HR
```

```
C       WATEXH=WATER EXHAUSTED                                  LBS/HR
C       WATINP=WATER INPUT WITH OLTSIDE AIR                     LBS/HR
C       NUM=NUMBER CF ITERATICN CF CALALL
C       ROOMRH=RELATIVE HUMIDITY CF ZONE (SEARCH VARIALBLE)  %
C       ZTI=TEMPERATUE OF AIR FEC TC ZONE                       F
C       ZHI=ENTHALPY OF AIR FED TC ZONE                         BTU/LB
C       TDI=TEMPERATURE OF AIR FEC TO SUPPLY DUCT               F
C       HDI=ENTHALPY OF AIR FED TC SUPPLY DUCT
C       TRAC=TEMPERATURE OF RETLRN AIR                          F
C       HRAC=ENTHALPY OF RETURN AIR                             BTU/LB
C       TINFAN=TEMPERATURE OF AIR INPUT TO FAN                  F
C       HINFAN=ENTHALPY OF AIR INPUT TO FAN                     BTU/LB
C       TAC=TEMPERATLRE OF AIR AFTER COIL CALCULATIONS          F
C       HAC=ENTHALPY OF AIR AFTER CCIL CALCULATIONS             BTU/LB
C       FST=STEAM FLCW                                          LBS/HR
C       FCHW=CHILLED WATER FLOW                                 LBS/HR
C       HHEAT=DUTY CF HEATING CCILS                             BTU/HR
C       HCOOL=DUTY CF COOLING CCILS                             BTU/HR
C       TMB=TEMPERATURE OF MIXING BCX AIR                       F
C       HMB=ENTHALPY CF MIXING ECX AIR                          BTU/LB
C       IFLAG=ZERCUT CONVERGENCE FLAG =1 IF CONVERGED
C       DWATBL=WATER BALANCE CCNVERGENCE CHECK
C
C
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C
C
      SUBROUTINE CALALL(OADB,CARH,ZODB,PATM,WLIZ,WLOZ,CLIZ,CLCZ,
     1ZFC,DLHI,DLHO,QOARA,UAFAC,CPAIR,WFAN,EFFAN,OAMIN,FOA,FRA,
     2WATEXH,WATINP,WATREM,NUM,RCOMRH,ZTI,ZHI,TDI,HDI,TRAC,HRAC,
     3TINFAN,HINFAN,TAC,HAC,FST,FCHW,HHEAT,HCOOL,TMB,HMB,IFLAG,DWATBL)
C
C ZEROUT WILL BE USED TO DETERMINE CONVERGENCE CF THE AIR HANCLER
C WATER BALANCE
C
      REAL WORK(10)
C
C GIVE HIGH AND LOW LIMITS OF THE SEARCHED FOR VARIABLE, WHICH IS
C THE RELATIVE HUMIDITY IN THE ZCNE
C
      RHHI=90.
      RHLO=20.
C
C FIND THE CONDITICNS OF THE OLTSIDE AIR
C
      CALL SSYCH3(OADB,OAWB,PATM,CADP,OAWAT,OAH,CAV,CARH)
C
C SET IFLAG ECUAL TC ZERC AND GIVE TOLERANCE IN WORK(1)
C
      IFLAG=0.
      WCRK(1)=.01
C
C BEGIN ZEROUT
C
      DO 500 I=1,25
      NUM=I
C
C LET ZEROUT SET VALUE OF ROOM RELATIVE HUMIDITY
```

```
C
      ROOMRH=ZEROUT(IFLAG,RHHI,RHLO,DWATBL,WORK)
C
C FIND CCNDITICNS CF AIR LEAVING ZCNE
C
      CALL SSYCH3(ZODB,ZOWB,PATM,ZODP,ZOWAT,ZOH,ZOV,ROOMRH)
C
C FIND CONDITIONS CF AIR INPUT TO ZONE
C
      CALL AIRSM2(WLIZ,WLOZ,QLIZ,CLOZ,0.,0.,0.,0.,PATM,ZFI,ZWI,
     1ZTI,ZHI,ZFC,ZOWAT,ZODB,ZCH,DLHI,DLHO)
C
C DO DUCTEX HEAT EXCHANGER ROUTINE WHICH DETERMINES THE AMCUNT CF
C OF HEAT TRANSFERRED BETWEEN THE AIR SUPPLY DUCT AND THE RETURN AIR
C DUCT, AND THEREFCRE THE CONDITICNS OF THE AIR INPUT TC THE SUPPLY
C DUCT AND LEAVING THE RETURN AIR CUCT
C
      CALL DUCTEX(ZFO,ZODB,ZOH,ZFI,ZTI,ZHI,PATM,QDARA,UARAD,
     1QDUCT,TDI,HDI,TRAO,HRAC)
C
C FIND CONDITIONS OF AIR INPUT TC THE FAN
C
      CALL SFAN(TDI,ZFI,ZHI,WFAN,EFFAN,CPAIR,TINFAN,HINFAN,FINFAN)
C
C DO MIXING BOX CALCULATION WHICH CETERMINES THE NECESSARY FLOWS
C OF RETURN AND OUTSIDE AIR ANC CALCULATES REQUIREMENTS OF STEAM
C OR CHILLED WATER
C
      CALL MIXBCX(TRAO,HRAO,FRA,OADB,OAH,FOA,TINFAN,HINFAN,
     1FINFAN,TMB,HMB,FMB,OAMIN,PATM,TAC,HAC,FST,FCHW,RHAC,HHEAT,
     2HCOOL,WATREM)
C
C DETERMINE THE AMCUNT OF WATER EXHAUSTED AND INPUT
C
      WATEXH=FCA*ZCWAT
      WATINP=FOA*CAWAT
C
C CHECK THE WATER BALANCE
C
      DWATBL=(WLIZ-WLCZ)+WATINP-WATEXH-WATREM
C     WRITE(6,340)NUM,WATEXH,WATINP,WATREM,DWATBL,RCOMRH
C 340 FORMAT(' NUM= 'I3,'  WATEXH= 'F12.6,' WATINP= 'F12.6,
C    1'  WATREM= 'F12.6,'  DWATBL= 'F12.6,'  ROOMRH= 'F12.6)
C     WRITE(6,350)
C 350 FORMAT('    '///)
C
C CHECK FLAG TC SEE IF CONVERGEC
C
      IF(IFLAG.EQ.1.OR.IFLAG.LT.0.)GO TO 600
  500 CCNTINUE
  600 RETURN
      END
C
C
C
C
C               SUBROUTINE CONTFC
C
C
C PURPOSE:
```

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C       THE PURPOSE OF THIS SUBROUTINE IS TO CALL ALL COMPONENT
C  SUBROUTINES OF THE SINGLE DUCT AIR HANDLER SIMULATION.  THIS
C  ROUTINE IS RESPONSIBLE FOR DETERMINING THE ROOM RELATIVE
C  HUMIDITY THAT WILL GAURANTEE CONVERGENCE OF THE WATER BALANCE.
C
C  INPUT:
C       TSPHI=THE ZONE HIGH TEMPERATURE SET POINT              F
C       TSPLO=THE ZONE LOW TEMPERATURE SET POINT               F
C       RHSPHI=THE ZONE HIGH RELATIVE HUMIDITY SET POINT       %
C       TRHSP=THE REHEAT TEMPERATURE SET POINT                 F
C       OADB=OUTSIDE AIR DRY BULB TEMPERATURE                  F
C       OARH=OUTSIDE AIR RELATIVE HUMIDITY                     %
C       ZODB=ZONE DRY BULB TEMPERATURE (SET POINT)             F
C       PATM=ATMOSPHERIC PRESSURE                              IN HG
C       WLIZ=WATER LOAD INTO ZONE                              LBS H2O/HR
C       WLOZ=WATER LOAD OUT OF ZONE                            LBS H2O/HR
C       QLIZ=HEAT LOAD INTO ZONE                               BTU/HR
C       QLOZ=HEAT LOAD OUT OF ZONE                             BTU/HR
C       ZFO=AIR FLOW THROUGH ZONE                              LBS/HR
C       DLHI=LATENT HEAT OF WATER                              BTU/LB
C       DLHC=LATENT HEAT OF WATER                              BTU/LB
C       QOARA=HEAT GAIN IN RETURN AIR DUCT FROM OUTSIDE        BTU/HR
C       UARAD=HEAT TRANSFER COEFFICIENT BETWEEN SUPPLY
C             DUCT AND RETURN AIR DUCT                         BTU/HR-F
C       CPAIR=HEAT CAPACITY OF AIR                             BTU/LB-F
C       WFAN=HORSEPOWER OF FAN                                 HP
C       EFFAN=FAN EFFICIENCY
C       OAMIN=MINIMUM OUTSIDE AIR REQUIREMENT                  LBS/HR
C
C
C  OUTPUT:
C       FOA=OUTSIDE AIR FLOW                                   LBS/HR
C       FRA=RETURN AIR FLOW                                    LBS/HR
C       WATEXH=WATER EXHAUSTED                                 LBS/HR
C       WATINP=WATER INPUT WITH OUTSIDE AIR                    LBS/HR
C       NUM=NUMBER OF ITERATION OF CALALL
C       ROOMRH=RELATIVE HUMIDITY OF ZONE (SEARCH VARIALBLE)    %
C       ZTI=TEMPERATUE OF AIR FED TO ZONE                      F
C       ZHI=ENTHALPY OF AIR FED TO ZONE                        BTU/LB
C       TDI=TEMPERATURE OF AIR FED TO SUPPLY DUCT              F
C       HDI=ENTHALPY OF AIR FED TO SUPPLY DUCT
C       TRAO=TEMPERATURE OF RETURN AIR                         F
C       HRAO=ENTHALPY OF RETURN AIR                            BTU/LB
C       TINFAN=TEMPERATURE OF AIR INPUT TO FAN                 F
C       HINFAN=ENTHALPY OF AIR INPUT TO FAN                    BTU/LB
C       TAC=TEMPERATURE OF AIR AFTER COIL CALCULATIONS         F
C       HAC=ENTHALPY OF AIR AFTER COIL CALCULATIONS            BTU/LB
C       FST=STEAM FLOW                                         LBS/HR
C       FCHW=CHILLED WATER FLOW                                LBS/HR
C       HHEAT=DUTY OF HEATING COILS                            BTU/HR
C       HCOOL=DUTY OF COOLING COILS                            BTU/HR
C       TMB=TEMPERATURE OF MIXING BOX AIR                      F
C       HMB=ENTHALPY OF MIXING BOX AIR                         BTU/LB
C       IFLAG=ZEROUT CONVERGENCE FLAG =1 IF CONVERGED
C       DWATBL=WATER BALANCE CONVERGENCE CHECK
C
C
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
```

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C
C
      SUBROUTINE CONTRO(OADB,OARH,ZODB,PATM,WLIZ,WLOZ,QLIZ,QLOZ,
     1ZFO,DLHI,DLHO,QOARA,UARAD,CPAIR,WFAN,EFFAN,OAMIN,FCA,FRA,
     2WATEXH,WATINP,WATREM,NUM,ROOMRH,ZTI,ZHI,TDI,HDI,TRAC,HRAO,
     3TINFAN,HINFAN,TAC,HAC,FST,FCHW,HHEAT,HCOOL,TMB,HMB,IFLAG,DWATBL,
     4TSPHI,TSPLO,RHSPHI,TRHSP,TSPDEH,FRESP)
C
C
C FIRST CHECK THE HIGH TEMPERATURE SET POINT
C
      COMMON/WHAT/TACCC,HACCC
C
C
      CALL CALALL(OADB,OARH,TSPHI,PATM,WLIZ,WLOZ,QLIZ,QLOZ,ZFO,DLHI,
     1DLHO,QOARA,UARAD,CPAIR,WFAN,EFFAN,OAMIN,FCA,FRA,WATEXH,
     2WATINP,WATREM,NUM,ROOMRH,ZTI,ZHI,TDI,HDI,TRAC,HRAC,TINFAN,
     3HINFAN,TAC,HAC,FST,FCHW,HHEAT,HCOOL,TMB,HMB,IFLAG,DWATBL)
      ZODB=TSPHI
C
C IF STEAM WAS REQUIRED-- GO TO 400 TO CHECK THE LOW SET POINT:TSPLO
C
      IF(FST.GT.0.)GO TO 400
C
C IF NOT CHECK HUMIDITY CONSTRAINT, GO TO 500 FOR COOL AND REHEAT
C DEHUMIDIFICATION
C
      IF(ROOMRH.GT.RHSPHI)GO TO 500
C
C IF NOT CHECK USE OF CHILLED WATER
C
      IF(FCHW.GT.0.)GO TO 900
C
C IF NOT GO FOR FREE USE OF OUTSIDE AND RETURN AIR
C
      GO TO 700
C
C NOW CHECK LOW SET POINT
C
 400  CALL CALALL(OADB,OARH,TSPLO,PATM,WLIZ,WLOZ,QLIZ,QLOZ,ZFO,DLHI,
     1DLHO,QOARA,UARAD,CPAIR,WFAN,EFFAN,OAMIN,FCA,FRA,WATEXH,
     2WATINP,WATREM,NUM,ROOMRH,ZTI,ZHI,TDI,HDI,TRAO,HRAO,TINFAN,
     3HINFAN,TAC,HAC,FST,FCHW,HHEAT,HCOOL,TMB,HMB,IFLAG,DWATBL)
      ZODB=TSPLO
C
C IF CHILLED WATER REQUIRED OR STEAM IS NOT GO FOR FREE USE OF
C OUTSIDE AIR PLUS RETURN AIR
C
      IF(FCHW.GT.0.)GO TO 700
      IF(FST.GT. 0.)GO TO 900
      GO TO 700
C
C NOW DO REHEAT AND DEHUMIDIFICATION
C
 500  CALL REHEA2(OADB,OARH,ZODB,PATM,WLIZ,WLOZ,QLIZ,QLOZ,ZFO,DLHI,
     1DLHO,QOARA,UARAD,CPAIR,WFAN,EFFAN,OAMIN,FCA,FRA,WATEXH,
     2WATINP,WATREM,NUM,ROOMRH,ZTI,ZHI,TDI,HDI,TRAO,HRAO,TINFAN,
     3HINFAN,TAC,HAC,FST,FCHW,HHEAT,HCOOL,TMB,HMB,IFLAG,DWATBL,
     4TSPHI,TSPLO,RHSPHI,TRHSP,TACC,HACC)
      TACCC=TACC
```

```
      HACCC=HACC
      GC TC 900
C
C DO FREE USE OF OUTSIDE AIR ANC RETURN AIR
C
 700  CALL FREECL(OADB,OARH,ZCDB,PATM,WLIZ,WLOZ,CLIZ,CLCZ,ZFC,CLHI,
     1DLHO,COARA,UARAD,CPAIR,WFAN,EFFAN,OAMIN,FOA,FRA,WATEXH,
     2WATINP,WATREM,NUM,ROCMFH,ZTI,ZHI,TDI,HDI,TRAC,HRAO,TINFAN,
     3HINFAN,TAC,HAC,FST,FCHW,HHEAT,HCOOL,TMB,HMB,IFLAG,CWATBL,
     4FRESP,TSPHI,TSPLC,RHSPHI)
C
C
C EXIT
C
 900  RETURN
      END
C
C
C
C
C
C               SUBRCUTINE FREECL
C PURPOSE:
C     THE PURPOSE CF THIS RCUTINE IS TO SIMULATE THE CONTROL SITUATION
C REQUIRING NO HEATING OR COOLING-- JUST A MIXTURE OF CUTSIDE AIR ANC
C RETURN AIR
C
C
C INPUT:
C     FRESP=BEGINNING GUESS AT FREE TEMPERATURE SET POINT    F
C
C ****    ALL CTHER INPUT AND CUTPUT VARIABLES ARE DEFINED THE SAME
C AS IN ROUTINE    *******   CALALL.FOR    ******
C
      SUBRCUTINE FREECL(OADB,CARH,ZODB,PATM,WLIZ,WLOZ,CLIZ,CLOZ,
     1ZFO,DLHI,DLHO,QOARA,UARAD,CPAIR,WFAN,EFFAN,OAMIN,FCA,FRA,
     2WATEXH,WATINP,WATREM,NUM,RCOMRH,ZTI,ZHI,TDI,HDI,TRAO,HRAO,
     3TINFAN,HINFAN,TAC,HAC,FST,FCHW,HHEAT,HCCOL,TMB,HMB,IFLAG,DWATBL,
     4FRESP,TSPHI,TSPLO,RHSPHI)
      I=0
      FREESP=FRESP
C
C CALL AIR HANDLER CALLING ROUTINE
C
 20   CALL CALALL(CADB,OARH,FREESP,PATM,WLIZ,WLOZ,QLIZ,QLOZ,ZFO,DLHI,
     1DLHO,QOARA,UARAD,CPAIR,WFAN,EFFAN,OAMIN,FCA,FRA,WATEXH,
     2WATINP,WATREM,NUM,ROOMRH,ZTI,ZHI,TDI,HDI,TRAC,HRAC,TINFAN,
     3HINFAN,TAC,HAC,FST,FCHW,HHEAT,HCOOL,TMB,HMB,IFLAG,CWATBL)
C CHECK NUMBER OF PASSES THROUGH LCOP
      I=I+1
      WRITE(6,*)I
      IF(I.EQ.15)GO TO 130
C
C IF STEAM WAS REQUIRED, DECREMENT THE FREE SET PCINT
C
      IF(FST.GT.0.)GO TO 60
C
C IF CHILLED WATER WAS REQUIREC, INCREMENT THE FREE SET POINT
C
      IF(FCHW.GT.C.)GO TO 70
```

```
C
C IF HUMIDITY IS A PROBLEM GIVE ERROR MESSAGE
C
      IF(ROOMRH.GT.RHSPHI)GO TC 80
C
C HAVE WE GCNE ABCVE OR BELOW CUR CRIGINAL SET POINT
C
      IF(FREESP.GT.TSPHI)GO TC 90
      IF(FREESP.LT.TSPLO)GO TC 100
C
C WE HAVE A SOLUTICN
C
      GO TO 150
C
C DECREMENT
C
 60   FREESP=FREESP-1.
      GO TC 20
C
C INCREMENT
C
 70   FREESP=FREESP+1.
      GC TC 20
C
C HUMIDITY MESSAGE
C
 80   WRITE(6,85)
 85   FORMAT(' THERE IS A HUMIDITY PRCBLEM IN FREECL ')
      GO TO 150
C
C SET PCINT VICLATCNS
C
 90   WRITE(6,95)
 95   FORMAT(' HIGH SET POINT VICLATICN IN FREECL ')
      GC TC 150
 100  WRITE(6,1C5)
 105  FORMAT(' LOW SET POINT VICLATION IN FREECL ')
      GC TC 150
C
C TOO MANY INCREMENTS OR DECREMENTS VIOLATIONS
C
 130  WRITE(6,135)
 135  FORMAT(' INCREMENT VIOLATICN IN FREECL ')
C FINISH
 150  ZOD8=FREESP
      RETURN
      END
C
C
C
C
C
C                     SUBROUTINE AIRSM
C
C PURPCSE:
C      CALCULATICN CF EITHER EXIT STREAM OR CNE FEED STREAM
C CONDITIONS FOR A SYSTEM WHICH SUMS TWO AIR STREAMS.  MUST BE
C GIVEN CCNDITICNS CF CTHER TWC STREAMS, WATER LCAD, ANC HEAT LOAD.
C
C INPUT:
```

```
C       DLHI=LATENT HEAT OF INPLT WATER              BTU/LB H2C
C       DLHC=LATENT HEAT OF OUTFUT WATER             BTU/LB H2O
C       WLI=WATER LCAD INTO THE SYSTEM               LB H2O/HR
C       WLO=WATER LCAD OUT OF SYSTEM                 LB H2O/HR
C       QLI=HEAT LCAD INTO SYSTEM                    BTU/HR
C       QLO=HEAT LOAD OUT OF SYSTEM                  BTU/HR
C       FJ=FEED STREAM J TOTAL FLCW                  LBS/HR
C       WJ=FEED STREAM J WATER CCNTENT               LB H2O/LB DRY AIR
C       TJ=FEED STREAM J TEMPERATLRE                 F
C       HJ=FEED STREAM J ENTHALPY                    BTU/LB
C       PATM=SYSTEM PRESSURE                         IN HG
C
C  INPUT/OUTPUT:
C       FI=FEED STREAM I TOTAL FLOW                  LBS/HR
C       WI=FEED STREAM I WATER CCNTENT               LBS H2O/LB DRY AIR
C       TI=FEED STREAM I TEMPERATLRE                 F
C       HI=FEED STREAM I ENTHALPY                    BTU/LB
C       FO=OUTLET FLCW                               LBS/HR
C       WO=CUTLET WATER CONTENT                      LBS H2O/LB DRY AIR
C       TC=OUTLET TEMPERATURE                        F
C       HO=OUTLET ENTHALPY                           BTU/LB
C
C
C  ENTRY AIRSM CLACULATES EXIT STREAM FROM TWO GIVEN FEEDS
C
C
        SUBROUTINE AIRSM(WLI,WLC,QLI,QLO,FJ,WJ,TJ,HJ,PATM,FI,WI,
       1TI,HI,FO,WC,TO,HO,DLHI,CLHC)
C
C  TCTAL MASS BALANCE
C
        FO=FI+FJ
C  WATER MASS BALANCE
        WO=(FI*WI+FJ*WJ+WLI-WLO)/FO
C  ENERGY BALACNE
        HO=(FI*HI+FJ*HJ+QLI+WLI*DLHI-WLC*DLHC-QLC)/FO
C  DRY BULB TEMPERATURE
        CALL SSYCH8(TO,WBO,PATM,CPC,WO,HO,VO,RHO)
        RETURN
C
C                 ENTRY AIRSM2
C
C  CALCULATE ONE FEED STREAM GIVEN CNE FEED AND EXIT STREAM
C
        ENTRY AIRSM2(WLI,WLO,QLI,CLC,FJ,WJ,TJ,HJ,PATM,FI,WI,TI,HI,
       1FO,WO,TO,HC,DLHI,DLHO)
C
C  TOTAL MASS BALANCE
        FI=FC-FJ
C  WATER MASS BALANCE
        WI=(WLC+FC*WO-FJ*WJ-WLI)/FI
C  ENERGY BALANCE
        HI=(WLO*DLHC+QLO+FO*HO-FJ*HJ-QLI-WLI*DLHI)/FI
C  DRY BULB TEMPEPATURE
        CALL SSYCH8(TI,WBI,PATM,DPI,WI,HI,VI,RHI)
        RETURN
        END
C
C
C
```

```
C
C
C               SLBRCUTINE MIXBCX
C
C PURPCSE:
C      (TO BE USED WITH SINGLE DUCT STRATEGY)
C      GIVEN THE CESIRED CONDITICNS FOR THE AIR LEAVING THE CCILS,
C AND THE KNOWN CCNDITIONS OF THE RETURN AIR AND CUTSIDE AIR;
C CALCULATE THE COMBINATION OF RETLRN AIR, CUTSIDE AIR, ANC CCIL
C CPERATION.
C
C
C INPUT:
C      TRA=RETURN AIR TEMPERATLRE                 F
C      HRA=RETURN AIR ENTHALPY                    BTU/LB
C      TOA=OUTSICE AIR TEMPERATURE                F
C      HOA=OUTSIDE AIR ENTHALPY                   BTU/LB
C      TMBD=DESIREC MIXING BCX TEMFERATURE        F
C      HMBD=DESIRED MIXING BOX ENTHALPY           BTU/LB
C      FMBD=DESIREC MIXING BOX FLOW               LBS/HR
C      OAMIN=MINIMUM OUTSIDE AIR FLOW             LBS/HR
C      PATM=ATMOSPHERIC PRESSURE                  IN HG
C
C OUTPUT:
C      FRA=RETURN AIR FLOW                        LBS/HR
C      FOA=CUTSICE AIR FLOW                       LBS/HR
C      FMB=MIXING BOX FLOW                        LBS/HR
C      TMB=MIXING BCX TEMPERATLRE                 F
C      HMB=MIXING BOX ENTHALPY                    BTU/LB
C      TAC=TEMPERATURE AFTER CCILS                F
C      HAC=ENTHALPY AFTER COILS                   BTU/LB
C      RHAC=RELATIVE HUMIDITY AFTER COILS
C      HHEAT=STEAM DUTY                           BTU/HR
C      HCOOL=COOLING COIL DUTY                    BTU/HR
C      FCHW=CHILLEC WATER FLOW                    LBS/HR
C      ALWT=LEAVING CHILL WATER TEMP.             F
C      WATREM=WATER REMOVED BY C.W. COILS         LBS/HR
C
C
C
      SUBRCUTINE MIXBOX(TRA,HRA,FRA,TOA,HOA,FOA,TMBD,HMBD,FMBD,TMB,
     1HMB,FMB,CAMIN,PATM,TAC,HAC,FST,FCHW,RHAC,HHEAT,HCCCL,WATREM)
      REAL WORK(1C)
      COMMON/SMIX/MXSWIT
      EWT=44.                    !!!!!!! NOTE:MAY BE ADJULSTED
      FSMAX=855.                 !!!!!!! NOTE: MAY BE ADJUSTED
      STLH=962.2                 !!!!!!! NOTE: MAY BE ACJUSTEC
C
C DEFINE PRCPERTIES OF GIVEN AIR
C
      CALL SSYCH9(TRA,WBRA,PATM,DPRA,WRA,HRA,VRA,RHRA)
      CALL SSYCH9(TOA,WBOA,PATM,DPOA,WOA,HCA,VOA,RHCA)
      CALL SSYCH9(TMBD,WBMBD,PATM,DPMBD,WMBD,HMBD,VMBD,RHMBD)
C
C CHECK MXSWIT FOR CASE  MXSWIT=1 MEANS MEET SET POINT
C                        MXSWIT=2 MEANS FULL R.A. PLUS MIN O.A.
C
      IF(MXSWIT.EQ.1)GO TO 175
C
C USE MINIMUM CUTSIDE AIR PLUS RETLRN AIR
```

```
C
C       MXSWIT=2
C
C
        FOA=CAMIN
        FRA=FMBD-OAMIN
        FMB=FCA+FRA
        CALL AIRSM(C.,O.,O.,O.,FRA,WRA,TRA,HRA,PATM,FOA,WCA,TCA,HOA,
       1FMB,WMB,TME,HMB,O.,O.)
C
C
C CHECK NEED FCR CCOL OR STEAM
C
        IF(TMB.LT.TMBD)GO TO 10C
C
C USE CHILL WATER COILS WITH MIN. C.A. AND RETURN AIR
C
        CALL CWCCAL(FMB,EWT,FCHW,TME,HME,TAC,HAC,ALWT,HCOOL,WATREM,RATO,
       1TMBD,PATM,HCOO2,RHAC,L1,L2)
        HHEAT=0.
        FST=0.
        GO TO 999
C
C USE STEAM WITH MIN. O.A. AND R.A.
C
 100    HCOCL=0.
        FCHW=0.
        CALL HEACOL(HMB,WMB,TMBC,FMB,HHC,FMBDUM,STLH,FSMAX,HHEAT,
       1FST,PATM)
        CALL SSYCHS(TAC,WBAC,PATM,DPAC,WMB,HHC,VAC,RHAC)
        HAC=HHC
        GC TC 999
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C
C NCW HAVE CPTION CF OUTSIDE AIR FCR FREE COOLING
C
 175    IF(TOA.LT.TRA)GO TO 200
C
C
C RETURN AIR IS LESS THAN RETURN AIR TEMP.  BUT WWE ARE STILL UNDER
C A CONTROL CONDITIC
C
        IF(TCA.LT.TMBD)GO TO 30C
        IF(TRA.GT.TMBD)GO TO 25C
        WRITE(6,126)
 126    FORMAT('  ***** TROUBLE IN EASE CASE MIXBOX ***** ')
        WRITE(6,127)
 127    FORMAT(' TOA.GT.TMBSP.GT.TRA. :::::: IMPOSSIBLE :::::: ')
        WRITE(5,128)
 128    FCRMAT(' ****** MIXBOX IMPOSSIBLE CASE ****** ')
        GO TO 250
C
 200    IF(TRA.LT.TMBD)GO TO 30C
        IF(TOA.LT.TMBD)GO TO 40C
C
C USE FULL CUTSICE AIR PLUS CCCLING
C
 250    FCA=FMBC
```

```
      FRA=0.
      FMB=FOA
      FST=0.
      HHEAT=0.
      TMB=TCA
      HMB=HCA
      CALL CWCCAL(FMB,EWT,FCHW,TMB,HMB,TAC,HAC,ALWT,HCCOL,WATREM,RATO,
     1TMBD,PATM,HCOO2,RHAC,L1,L2)
      GO TO 999
C
C USE MINIMUM OUTSIDE AIR PLUS RETURN AIR PLUS STEAM
C
 300  FOA=OAMIN
      FRA=FMBD-FCA
      FMB=FOA+FRA
      FCHW=0.
      HCOCL=0.
      CALL AIRSM(C.,0.,0.,0.,FRA,WRA,TRA,HRA,PATM,FOA,WCA,TCA,HCA,
     1FMB,WMB,TMB,HMB,0.,0.)
      CALL HEACCL(HMB,WMB,TMBC,FMB,HHC,FMBDUM,STLH,FSMAX,HHEAT,
     1FST,PATM)
      CALL SSYCH5(TAC,WBAC,PATM,DPAC,WMB,HHC,VAC,RHAC)
      HAC=HHC
      GC TC 999
C
C USE COMBINATION OF OUTSIDE AIR PLUS RETURN AIR
C
 400  FCHW=0.
      FST=0.
      HHEAT=0.
      HCOOL=0.
C
C ZEROUT WILL BE USED TO FIND FLOW COMBINATION
C
C CHECK LINEAR GUESS
      FRA1=FMBD*((TMBD-TOA)/(TRA-TOA))
      FOA1=FMBD-FRA1
      FMB=FOA1+FRA1
      CALL AIRSM(0.,0.,0.,0.,FRA1,WRA,TRA,HRA,PATM,FOA1,WCA,TOA,HOA,
     1FMB,WMB,TMB,HMB,0.,0.)
      TAC=TMB
      HAC=HMB
      DELT1=TAC-TMBD
      IF(DELT1.LT.0.)GO TO 420
C
C SET HIGH AND LOW RANGE FOR FLOW GUESS
C
      FRAHI=FRA1
      FRALO=FRA1-CAMIN
      GO TO 440
 420  FRAHI=FRA1+CAMIN
      FRALO=FRA1
C
C INITIALIZE IFLAG AND GIVE TOLERANCE IN WORK(1)
C
 440  IFLAG=0.
      WORK(1)=.01
C
C DC ZEROUT TO FINC FLOW OF R.A. AND O.A.
C
```

```
      DO 480 I=1,25
      FRA=ZERCUT(IFLAG,FRALO,FRAHI,DELT,WORK)
      IF(FRA.EQ.FRA1)GO TO 460
      FOA=FMBD-FRA
      CALL AIRSM(C.,0.,0.,0.,FRA,WRA,TRA,HRA,PATM,FOA,WOA,TOA,HOA,
     1FMB,WMB,TMB,HMB,0.,0.)
      TAC=TMB
      HAC=HMB
      DELT=TAC-TMBD
      GC TC 470
 460  FOA=FOA1
      DELT=DELT1
 470  IF(IFLAG.EQ.1.OR.IFLAG.LT.0.)GO TC 490
 480  CCNTINUE
 490  CALL SSYCH7(TAC,WBAC,PATM,DPAC,WAC,HAC,VAC,RHAC)
      IF(FOA.LT.OAMIN)GO TO 3C0
 999  RETURN
      END
C
C
C
C
C
C                 SUBROUTINE MIXBC2
C
C PURPOSE:
C       ONLY DOES MIXBOX DAMPER CALCULATION.
C       USES ECONOMIZER APPROACH
C       GIVEN THE DESIRED CONDITIONS FOR THE AIR LEAVING THE COILS,
C AND THE KNOWN CONDITIONS OF THE RETURN AIR AND OUTSIDE AIR;
C CALCULATE THE COMBINATION OF RETURN AIR AND OUTSIDE AIR.
C
C
C INPUT:
C       TRA=RETURN AIR TEMPERATURE                      F
C       HRA=RETURN AIR ENTHALPY                         BTU/LB
C       TOA=OUTSIDE AIR TEMPERATURE                     F
C       HOA=OUTSIDE AIR ENTHALPY                        BTU/LB
C       TMBD=DESIRED MIXING BOX TEMPERATURE             F
C       HMBD=DESIRED MIXING BOX ENTHALPY                BTU/LB
C       FMBD=DESIRED MIXING BOX FLOW                    LBS/HR
C       OAMIN=MINIMUM OUTSIDE AIR FLOW                  LBS/HR
C       PATM=ATMOSPHERIC PRESSURE                       IN HG
C
C OUTPUT:
C       FRA=RETURN AIR FLOW                             LBS/HR
C       FOA=OUTSIDE AIR FLOW                            LBS/HR
C       FMB=MIXING BOX FLOW                             LBS/HR
C       TMB=MIXING BOX TEMPERATURE                      F
C       HMB=MIXING BOX ENTHALPY                         BTU/LB
C
C
C
      SUBROUTINE MIXBO2(TRA,HRA,FRA,TOA,HOA,FOA,TMBD,HMBD,FMBD,TMB,
     1HMB,FMB,CAMIN,PATM)
      REAL WORK(10)
      COMMON/SMIX/MXSWIT
C
C DEFINE PROPERTIES OF GIVEN AIR
C
```

```
      CALL SSYCH9(TRA,WBRA,PATM,DPRA,WRA,HRA,VRA,RHRA)
      CALL SSYCH9(TOA,WBOA,PATM,DPOA,WOA,HOA,VOA,RHOA)
      CALL SSYCH9(TMBD,WBMBD,PATM,DPMBD,WMBD,HMBD,VMBD,RHMBD)
C
C CHECK MXSWIT FOR CASE MXSWIT=1 MEANS MEET SET POINT
C                       MXSWIT=2 MEANS FULL R.A. PLUS MIN. C.A.
C
      IF(MXSWIT.EQ.1)GO TO 175
C
C
C USE MINIMUM OUTSIDE AIR PLUS RETURN AIR
C
C
      FOA=OAMIN
      FRA=FMBD-OAMIN
      FMB=FOA+FRA
      CALL AIRSM(0.,0.,0.,0.,FRA,WRA,TRA,HRA,PATM,FOA,WOA,TOA,HOA,
     1FMB,WMB,TMB,HMB,0.,0.)
      GO TO 999
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C
C NOW HAVE OPTION OF OUTSIDE AIR FOR FREE COOLING
C
 175  IF(TOA.LT.TRA)GO TO 200
C
C  RETURNA AIR IS LESS THAN RETURN AIR TEMP.  BUT WE ARE STILL UNDER
C A CONTROL CONDITION
C
      IF(TOA.LT.TMBD)GO TO 300
      IF(TRA.GT.TMBD)GO TO 250
      WRITE(6,126)
 126  FORMAT('  **** TROUBLE IN BASE CASE MIXBOX **** ')
      WRITE(6,127)
 127  FORMAT('  TOA.GT.TMBSP.GT.TRA. ::: IMPOSSIBLE CASE::::: ')
      WRITE(5,128)
 128  FORMAT(' **** MIXBOX IMPOSSIBLE CASE  2  ***** ')
      GO TO 250
 200  IF(TRA.LT.TMBD)GO TO 300
      IF(TOA.LT.TMBD)GO TO 400
C
C USE FULL OUTSIDE AIR PLUS COOLING
C
 250  FOA=FMBD
      FRA=0.
      FMB=FOA
      TMB=TOA
      HMB=HOA
      GO TO 999
C
C USE MINIMUM OUTSIDE AIR PLUS RETURN AIR PLUS STEAM
C
 300  FOA=OAMIN
      FRA=FMBD-FOA
      FMB=FOA+FRA
      CALL AIRSM(0.,0.,0.,0.,FRA,WRA,TRA,HRA,PATM,FOA,WOA,TOA,HOA,
     1FMB,WMB,TMB,HMB,0.,0.)
      GO TO 999
C
```

```
C USE COMBINATION CF OUTSIDE AIR PLUS RETURN AIR
C
C
C ZEROUT WILL BE USED TO FIND FLCW COMBINATION
C
C CHECK LINEAR GUESS
 400    FRA1=FMBD*((TMBD-TOA)/(TRA-TOA))
        FOA1=FMBD-FRA1
        FMB=FOA1+FRA1
        CALL AIRSM(C.,0.,0.,0.,FRA1,WRA,TRA,HRA,PATM,FOA1,WOA,TOA,HOA,
        1FMB,WMB,TMB,HMB,0.,0.)
        TAC=TMB
        HAC=HMB
        DELT1=TAC-TMBD
        IF(DELT1.LT.0.)GO TO 42C
C
C SET HIGH AND LCW RANGE FOR FLCW GUESS
C
        FRAHI=FRA1
        FRALO=FRA1-CAMIN
        GC TC 440
 420    FRAHI=FRA1+CAMIN
        FRALO=FRA1
C
C INITIALIZE IFLAG AND GIVE TOLERANCE IN WORK(1)
C
 440    IFLAG=0.
        WORK(1)=.C1
C
C DO ZERCUT TO FIND FLOW OF R.A. AND O.A.
C
        DC 480 I=1,25
        FRA=ZERCUT(IFLAG,FRALO,FRAHI,DELT,WORK)
        IF(FRA.EQ.FRA1)GO TO 46C
        FOA=FMBC-FRA
        CALL AIRSM(C.,0.,0.,0.,FRA,WRA,TRA,HRA,PATM,FOA,WCA,TCA,HCA,
        1FMB,WMB,TMB,HMB,0.,0.)
        TAC=TMB
        HAC=HMB
        DELT=TAC-TMBC
        GO TO 470
 460    FCA=FOA1
        DELT=DELT1
 470    IF(IFLAG.EQ.1.OR.IFLAG.LT.0.)GO TO 490
 480    CCNTINUE
 490    CALL SSYCH7(TAC,WBAC,PATM,DPAC,WAC,HAC,VAC,RHAC)
        IF(FCA.LT.CAMIN)GO TO 3CC
 999    RETURN
        END
C
C
C
C
C
C
C                SUBROUTINE DUCTEX
C
C PURPOSE
C     THIS SUBROUTINE CALCULATES THE AMOUNT OF HEAT TRANSFERRED
C FROM THE RETURN AIR TO THE DUCT AIR AND PROVIDES THE RESULTING
```

```
C PSYCHRCMETRIC CONCITIONS OF THE EXIT RETURN AIR AND THE INLET
C DUCT AIR.
C
C INPUTS:
C       FR=RETURN AIR FLOW                          LBS/HR
C       TRI=RETURN AIR TEMPERATURE IN               F
C       HRI=RETURN AIR ENTHALPY IN                  BTU/LB
C       FD=DUCT AIR FLOW                            LBS/HR
C       TDO=DUCT AIR TEMPERATURE CUT                F
C       HDO=DUCT AIR ENTHALPY CUT                   BTU/LB
C       PATM=SYSTEM PRESSURE                        PSIA
C       QOA=HEAT GAIN BY RETURN AIR                 BTU/LB
C       UA=DUCT LUMPED TRANSFER CCEFFICIENT         BTU/HR-F
C
C OUTPUTS:
C       QD=HEAT GAIN BY DUCT-LOST BY RETURN AIR BTU/HR
C       TDI=UCT TEMPERATURE IN                      F
C       HDI=DUCT ENTHALPY IN                        BTU/LB
C       TRO=RETURN AIR TEMPERATURE CUT              F
C       HRO=RETURN AIR ENTHALPY CUT                 BTU/LB
C
C       THIS ROUUTINE USES THE ZERCUT ROUTINE TO SEARCH FOR THE
C APPROPRIATE VALUE OF QD.  A VALUE OF QD WILL BE GUESSED.  WHEN
C THE VALUE CF UA GIVEN IS MATCHED, THE SEARCH WILL TERMINATE.
C
      SUBROUTINE DUCTEX(FR,TRI,HRI,FD,TDC,HDO,PATM,QOA,UA,CD,
     1TDI,HDI,TRC,HRO)
C     NAMELIST/TWC/AQD,DUA,DTLNMN,QD1,QD2,DLHRI,DLHRO,WRI,WRO,
C    1VRI,VDC,DPRI,DPRO
      REAL WORK(1C)
C
C SET UP RANGE OF GUESSES FOR CD
C
      QD1=0.
      CD2=UA*30.
C INITIALIZE IFLAG AND GIVE TOLERANCE IN WORK(1)
      IFLAG=0.
      WCRK(1)=0.1
C FIND NEEDED CONDITIONS FOR GIVEN AIR
      CALL SSYCH9(TRI,WBRI,PATM,DPRI,WRI,HRI,VRI,RHRI)
      CALL SSYCH9(TDO,WBDO,PATM,DPDO,WDC,HDO,VDC,RHDO)
C CC ZERCUT TO FINE A QD THAT WILL MATCH UA
      DC 30 I=1,25
      AQD=ZEROUT(IFLAG,QD1,QD2,DUA,WORK)
      CALL AIRSM2(0.,0.,AQD,0.,0.,0.,0.,0.,PATM,FDI,WDI,TDI,HDI,
     1FD,WDO,TDC,HDO,DLHDI,DLHCC)
      CALL AIRSM(C.,0.,QOA,AQC,FR,WRI,TRI,HRI,PATM,0.,C.,0.,0.,
     2FRO,WRC,TRC,HRO,DLHRI,DLHRC)
      DELA=TRO-TDI
      DELB=TRI-TCC
      IF(DELA.EC.C.AND.DELB.EQ.0.)GO TO 50
      TRMLOG=DELA/DELB
      IF(TRMLCG.LE.0.)GC TO 20
      IF(TRMLOG.EQ.1.)GO TO 5C
      DTLNMN=(DELA-DELB)/TRMLCG
      GO TO 25
20    DTLNMN=(DELA+DELB)/2.
25    GUA=AQD/DTLNMN
      DUA=UA-GUA
      IF(IFLAG.EC.1.OR.IFLAG.LT.0)GO TO 40
```

```
 30     CONTINUE
 40     CD=ACD
        GO TO 60
 50     QD=0.
        AQD=0.
 60     RETURN
        END
C
C
C
C
C
C
C               SUBRCUTINE SFAN
C
C PURPOSE:
C       GIVEN THE EXIT CONDITION FOR THE FAN, THE VOLUME OF AIR
C MOVED, AND THE HP RATING, CLACULATE THE ENTRANCE AIR CCNDITIONS.
C
C INPUT:
C       TEX=AIR TEMPERATURE EXIT THE FAN                        F
C       FEX=AIR FLOW EXIT THE FAN                               LBS/HR
C       HEX=AIR ENTHALPY EXIT THE FAN                           BTU/LB
C       WF=BRAKE HORSEPOWER OF FAN                              HP
C       EFF=FAN EFFICIENCY                                      UNITLESS
C       CPAIR=SPECIFIC HEAT OF AIR                              BTU/LB-F
C
C OUTPUT:
C       TIN=AIR TEMPERATURE INTO FAN                            F
C       HIN=AIR ENTHALPY INTO FAN                               BTU/LB
C       FIN=AIR FLOW INTO THE FAN                               LBS/HR
C
        SUBROUTINE SFAN(TEX,FEX,HEX,WF,EFF,CPAIR,TIN,HIN,FIN)
C
C USE EQUATION FOR DELTA H ACROSS FAN
        DELH=(2546.*WF)/(EFF*FE))
C DELTA T ACROSS FAN
        DELT=DELH/CPAIR
C CALCULATE FAN ENTRANCE CONDITIONS
        TIN=TEX-DELT
        HIN=HEX-DELH
        FIN=FEX
        RETURN
C
C
C               ENTRY SFAN2
C
C PURPCSE:
C       SFAN2 CLCULATES THE OUPUT CONDITIONS OF THE SUPPLY FAN GIVEN
C THE INPUT CONDITIONS, THE WORK OF THE FAN AND THE EFFICIENCY
C
C INPUT:
C       TIN,FIN,HIN,WF,EFF,CPAIR
C CUTPUT:
C       TEX,FEX,HEX
C
        ENTRY SFAN2(TEX,FEX,HEX,WF,EFF,CPAIR,TIN,HIN,FIN)
C USE EQUATION FOR DELTA H ACROSS FAN
        DELH=(2546.*WF)/(EFF*FE))
C DELTA T ACROSS FAN
```

```
      DELT=DELH/CPAIR
C CALCULATE FAN EXIT CONDITIONS
      TEX=TIN+DELT
      HEX=HIN+DELH
      FEX=FIN
      RETURN
      END
C
C
C
C
C
C
C                      SUBROUTINE HEACCL
C
C PURPOSE:
C      GIVEN THE AIR CONDITIONS ACROSS THE HEATING CCIL, CALCULATE
C THE LBS/HR OF STEAM REQUIRED TC MEET THE CESIRED TEMPERATURE.
C
C INPUT:
C      PATM=ATMOSPHERIC PRESSURE                          IN HG
C      TMBD=TEMPERATURE DESIREC AFTER COIL                F
C      WMB=AIR WATER CCNTENT                              LBS H2O/LB DRY AIR
C      HMB=AIR ENTHALPY BEFORE CCILS                      BTU/LB
C      FMB=AIR FLOW BEFORE COILS                          LBS/HR
C      HHC=AIR ENTHALPY AFTER HEATING COILS               BTU/LB
C      FHC=AIR FLOW AFTER HEATING COILS                   LBS/HR
C      STLH=STEAM  LATENT HEAT                            BTU/LB
C      FSMAX=MAXIMUM STEAM FLOW                           LBS/HR
C CUTPUT:
C      QHC=HEAT TRANSFER ACROSS COILS                     BTU/HR
C      FST=STEAM FLOW                                     LBS/HR
C      HHCD=ENTHALPY REQUIRED TC MATCH TEMP. AFTER COIL  BTU/LB
C
C
      SUBROUTINE HEACOL(HMB,WMB,TMBD,FMB,HHC,FHC,STLH,FSMAX,QHC,FST,
     1PATM)
C
C CALCULATE DESIREC CONDITIONS AFTER CCIL
C
      CALL SSYCH6(TMBD,WBMBD,PATM,DPMBD,WMB,HHCD,VMBD,RHMBD)
      HHC=HHCD
C
C CALCULATE COIL DUTY
C
      FHC=FMB
      QHC=FHC*(HHCD-HMB)
C
C CALCULATE STEAM FLOW
C
      FST=QHC/STLH
C
C CHECK FLCW LIMITS
C
      IF(FST .LT. 0.)GO TO 10
      IF(FST .GT. FSMAX)GO TO 20
      GC TC 30
C WRITE ERRCR MESSAGES IF LIMITS VIOLATED
 10   WRITE(6,15)
 15   FORMAT(' STEAM FLOW ON HEATING COIL WAS NEGATIVE ')
```

```
      GO TO 30
 20   WRITE(6,25)
 25   FORMAT(' STEAM FLOW EXCEECS MAXIMUM ALLOWED ')
 30   RETURN
      END
C
C
C
C
C
C
C
C               SUBROUTINE CWCCAL
C
C PURPOSE:
C     THE PURPOSE OF THIS SUBROUTINE IS TO FIND A CHILLED WATER FLOW RATE
C THAT WILL PROVIDE THE DESIREC EXIT AIR TEMPERATURE.
C THE ROUTINE USES THE ZEROUT SEARCH TECHNIQUE AND CALLS RCUTINE CWCOIL
C
C
C INPUTS:
C     TACC=DESIREC TEMPERATURE AFTER COOLING CCILS                F
C     FAIR=FLOW OF AIR                                            LBS/HR
C     EWT=ENTERING WATER TEMPERATURE                              F
C     EDB=ENTERING AIR DRY BULB TEMPERATURE                       F
C     EH=ENTERING AIR ENTHALPY                                    BTU/LB
C     PATM=SYSTEM PRESSURE                                        IN HG
C
C CUTPUTS:
C     CWFLO=CHILL WATER FLOW                                      LBS/HR
C     ALDB=LEAVING AIR DRY BULB TEMPERATURE                       F
C     ALH=LEAVING AIR ENTHALPY                                           BTU/LB
C     ALWT=LEAVING CHILL WATER TEMPERATURE                        F
C     QWAT=HEAT REMOVED                                           BTU/HR
C     WATER=WATER REMOVED                                         LBS/HR
C     RATO=RATIC CF SENSIBLE TO TCTAL HEAT REMOVED                DIMENSIONLESS
C
C
      SUBRCUTINE CWCCAL(FAIR,EWT,CWFLO,EDB,EH,ALDB,ALH,ALWT,CWAT,WATER,
     1RATO,TACC,PATM,QEXC,ALFH,NCNE,NTWO)
      REAL WORK(1C)
      CCMMCN/AFLAG/IFCWCL,IFCCIL
C
C     FIND CONDITION OF INPUT AIR
C
      CALL PSYCH7(EDB,EWB,PATM,EDP,EWAT,EH,EV,ERH)
C
C
C USE          ********  ZEROUT  ********
C
C SET UP RANGE CF GUESSES FOR FLCW RATE OF CHILLED WATER
C
      EWT=44.          |||||||||  NOTE: MAY BE ADJUSTED
      CWMAX=110124              ||||||| NCTE:MAY BE ADJUSTEC
      CWMIN=0.0C1
C INITIALIZE IFLAG AND GIVE TOLERANCE IN WORK(1)
      IFLAG=0.
      WCRK(1)=.1
C DO ZERCUT TO FIND FLOW RATE CF CHILLED WATER
      DC 30 I=1,25
```

```
      NONE=I
      CWFLC=ZERCUT(IFLAG,CWMIN,CWMAX,DTEMP,WORK)
 20   CALL CWCOIL(FAIR,EWT,CWFLC,EDB,EH,ALDB,ALH,ALWT,CWAT,WATER,
     1RATO,PATM,QEXC,ALRH,NTWC,EV,EDP,EWAT)
      DTEMP=ALDB-TACC
      IF(IFLAG.EQ.1.OR.IFLAG.LT.0.)GO TC 40
 30   CCNTINUE
 40   IF(IFLAG.EC.1.)GO TO 60
      WRITE(6,55)
 55   FCRMAT(' WARNING:::::::: IFLAG=0. :::::  '/)
 60   IFCWCL=IFLAG
      RETURN
      END
C
C
C
C
C
C
C               SUBRCUTINE CWCOIL
C
C PURPCSE:
C      THE PURPOSE OF THIS SUBRCUTINE IS TO DESCRIBE THE CPERATICN OF THE
C CHILLED WATER COILS.  GIVEN THE CONDITIONS OF THE INPUT AIR, THE CHILLED
C WATER TEMPERATURE, AND THE CHILLED WATER FLOW; THIS ROUTINE WILL GIVE
C THE OUTLET WATER TEMPERATURE AND AIR CONDITIONS.
C
C INPUT:
C      FAIR=AIR FLCW (DRY)                                LBS/HR
C      EWT=TEMPERATURE OF INPUT CHILLEC WATER             F
C      CWFLC=FLOW CF CHILLED WATER                        LBS/HR
C      EDB=DRY BULB TEMPERATURE CF INPUT AIR              F
C      EH=ENTHALPY CF INPUT AIR                           BTU/LB
C
C INTERICR DEFINED INPUTS:
C      BRC#=CONSTANTS FOR BASE RATING                     DIMENSIONLESS
C      WSC#=CONSTANTS FOR WETTEC SURFACE FACTOR           DIMENSIONLESS
C      WLOFLO=LOW ACCEPTED WATER FLOW                     FT/SEC
C      CPW=HEAT CAPACITY OF WATER                         BTU/LB-F
C      ANR=NUMBER CF ROWS DEEP CF COILS                   DIMENSIONLESS
C      FA=FACE AREA OF COILS                              FT**2
C      AID=INSIDE DIAMETER OF CCILS                       FT
C      ANC=NUMBER CF PARALLEL CIRCUITS                    DIMENSIONLESS
C
C OUTPUTS:
C      ALDB=LEAVING AIR DRY BULB TEMPERATURE              F
C      ALH=LEAVING AIR ENTHALPY                                BTU/LB
C      ALWT=LEAVING WATER TEMPERATURE                     F
C      QWAT=HEAT REMOVED                                  BTU/HR
C      WATER=WATER REMCVED                                LBS/HR
C      RATO=RATIO OF SENSIBLE TC TCTAL HEAT REMOVED
C
C
C ****************************************************************************
C
C
      SUBROUTINE CWCOIL(FAIR,EWT,CWFLC,EDB,EH,ALDB,ALH,ALWT,CWAT,WATER,
     1RATC,PATM,CEXC,ALRH,NTWC,EV,EDP,EWAT)
      CCMMON/THREE/ANR,FA,AID,ANC,CPW,PATMR,BRCW,WSF
      REAL WORK(1C)
```

```
      COMMON/AFLAG/IFCWCL,IFCCIL
      PATMR=PATM
C
C PROVIDE THE CONSTANT VALUES FOR THE WETTED SURFACE FACTCR ECUATICN
C
      WSC1=1.0OC7596
      WSC2=0.42652727E-01
      WSC3=-.22742528E-02
      WSC4=0.16232278E-02
      WSC5=0.41384645E-04
      WSC6=-.24564063E-04
      WSC7=-.24092761E-06
      WSC8=-0.11066765E-08
      WSC9=0.543C2870E-10
C
C
C PROVIDE MAXIMUM CHILLED WATER FLCW RATE IN LBS/HR
C
      CWMAX=110145.
C
C DETERMINE CONSTANTS FOR EQUATICN PROVIDING THE RATIO OF SENSIBLE
C TO TOTAL HEAT REMOVAL
C
      A=EDP/EWT
      IF(A.LT.1.)A=1.
      IF(A.GT.2.2)A=2.2
      IF(CWFLC.GT.CWMAX)GOTO 18
      B=CWFLO/CWMAX
      GC TC 19
 18   B=1.0
 19   C=.6                                    !ADJUSTABLE PARAMETER ****
C DETERMINE RATIC
      RATO=1-C*B*((A-1.0)**2.)
C DEFINE CCIL PARAMETER
      ANR=4.0
      FA=75.8
      AID=0.052C83
      ANC=64.
C DEFINE HEAT CAPACITY OF WATER
      CPW=0.24
C FIND AIR FLCW IN FT/MIN (FPM) ANC WATER FLOW IN FT/SEC (FPS)
      FPM=EV*FAIR/(60.*FA)
      FPS=CWFLO/(62.42*3600*.7854*AID*AID*ANC)
C
C DETERMINE BASE RATING
      CALL BRCWCL(FPM,FPS,BRCW)
C NOW DETERMINE WETTED SURFACE FACTOR
 100  PW=EDP-EWT
      BW=EDB-EWT
      WSF=WSC1+WSC2*PW+WSC3*PW*BW+WSC4*PW*PW+WSC5*BW*BW*PW+WSC6*BW*PW*PW
     1+WSC7*PW*(BW**3.)+WSC8*PW*PW*(BW**3.)+WSC9*(BW**3.)*(PW**3.)
C
C NOW BEGIN TRIAL AND ERROR PRCCESS WHCIH MATCHES THE WATER SIDE Q WITH
C THE AIR SIDE Q
C
C THE PROCEDURE REQUIRES GUESSING AN EXIT WATER TEMPERATURE.  WHEN Q=MC(TIN-TOUT)
C FOR THE WATER IS EQUAL TO Q=ERCW*NR*FA*WSF*MLTD1 FOR THE TOTAL EXCHANGER
C THE ROUTINE IS CCNVERGED
C
C ZEROUT WILL BE USED
```

```
C
C       CHECK 1 DEGREE FROM DRY BULB FOR LEAVING WATER TEMPERATURE
C
        ALWT1D=EDB-1.
        CALL CCWCAL(CWFLO,ALWT1D,EWT,EH,EDB,FAIR,ALH,ALRH,ALWAT,
        1ALDB,EWAT,RATO,QWAT,QEXC,AMLTD)
        DELQ1D=QWAT-QEXC
        IF(DELQ1D.GT.0.)GO TO 150
C
C SET HIGH AND LOW RANGE OF WATER TEMPERATURE OUT FOR ZEROUT
C
        ALWTHI=EDB-.000001
        ALWTHO=EDB-1.
        GO TO 160
 150    ALWTHI=EDB-1.
        ALWTHO=EWT+.5
C
C INITIALIZE IFLAG AND GIVE TOLERANCE IN WORK(1)
C
 160    IFLAG=0.
        WORK(1)=.00005
C
C
C DO ZEROUT TO FIND LEAVING WATER TEMPERATURE
C
C
        DO 300 I=1,25
        NTWO=I
        ALWT=ZEROUT(IFLAG,ALWTHO,ALWTHI,DELQ,WORK)
        IF(ALWT.EQ.ALWT1D)GO TO 265
        IF(ALWT.LT.EWT+.5)GO TO 350
 240    CALL CCWCAL(CWFLO,ALWT,EWT,EH,EDB,FAIR,ALH,ALRH,ALWAT,
        1ALDB,EWAT,RATO,QWAT,QEXC,AMLTD)
        DELQ=QWAT-QEXC
        GO TO 270
 265    DELQ=DELQ1D
 270    IF(IFLAG.EQ.1.OR.IFLAG.LT.0.)GO TO 500
 300    CONTINUE
        GO TO 500
 350    ALWT=EWT+.001+.499*EXP((ALWT-EWT+.5)/.499)
 360    GO TO 240
 500    IFCOIL=IFLAG
        WATER=FAIR*(EWAT-ALWAT)
        RETURN
        END
C
C
C
C
C
C       SUBROUTINE CCWCAL
C
C
C       PURPOSE:
C               TO PERFORM COMBINATION ENERGY BALANCES ON AIR AND
C       CHILLED WATER
C
C
        SUBROUTINE CCWCAL(CWFLO,ALWT,EWT,EH,EDB,FAIR,ALH,ALRH,
        1ALWAT,ALDB,EWAT,RATO,QWAT,QEXC,AMLTD)
```

```
      COMMON/THREE/ANR,FA,AID,ANC,CPW,PATMR,BRCW,WSF
C FIND Q FOR WATER
      QWAT=CWFLC*CPW*(ALWT-EWT)
C FIN H FOR LEAVING AIR
      ALH=EH-QWAT/FAIR
C FIND DRY BULB BASED ON SENSIBLE HEAT REMOVED
      SENS=EH-RATC*QWAT/FAIR
      CALL SSYCH8(ALDB,DWB,PATMR,CDP,EWAT,SENS,DV,DRH)
C FIND ACTUAL CONDITIONS BASED ON LEAVING DRY BULB AND ENTHALPY
      CALL SSYCH7(ALDB,ALWB,PATMR,ALDP,ALWAT,ALH,ALV,ALRH)
      IF(ALRH.GT.90.)GO TO 400
C IF RELATIVE HUMIDITY WAS GREATER THAN 90%, IT IS ADJUSTED TO 90%
C OTHERWISE FIND Q-EXCHANGER
 250  D1=EDB-ALWT
      D2=ALDB-EWT
      IF(D1.LE.0..OR.D2.LE.0.)GO TO 377
      AMLTD=(D1-D2)/(ALOG(D1/D2))
      QEXC=BRCW*ANR*FA*WSF*AMLTD
      GO TO 380
 377  QEXC=0.
 380  GO TO 500
C ADJUST TO 90% RELATIVE HUMIDITY AND LEAVING AIR ENTHALPY
 400  ALRH=90.
      CALL SSYCH4(ALDB,ALWB,PATMR,ALDP,ALWAT,ALH,ALV,ALRH)
      GO TO 250
 500  WATER=FAIR*(EWAT-ALWAT)
      RETURN
      END
C
C
C
C
C
C     SUBROUTINE BRCWCL
C
C PURPOSE:
C
C     THIS ROUTINE IS USED TO CALCULATE THE BASE RATING OF THE
C CHILLED WATER COIL.  IT IS CALLED BY CWCOIL.
C THE ROUTINE DOES NOT ALLOW THE LAST THREE TERMS OF THE BASE
C RATING EQUATION TO DOMINATE AND CAUSE INSTABILITY.
C
C
      SUBROUTINE BRCWCL(FPM,FFS,BRCW)
C
C PROVIDE THE CONSTANT VALUES FOR THE BASE RATING EQUATION
C
      BRC1=2.2074304
      BRC2=1417.8455
      BRC3=3.3925543
      BRC4=-.55238537
      BRC5=-11071271.
      BRC6=-392885790.
C
C PROVIDE THE MAXIMUM RATIO CONSTANT FOR BRCW2/BRCW1
C
      ACON=-.171944556
C
C CALCULATE BRCW1 AND BRCW2
C
```

```
 5      BRCW1=BRC1+ERC2/FPM+BRC3/FPS
        IF(FPM.LT..0005)GO TO 25
 10     BRCW2=BRC4/(FPS*FPS)+BRC5/(FPM**3)+BRC6/(FPS*FPS*FPM**4)
C
C CHECK LIMITS OF RATIO
C
        IF((BRCW2/BRCW1).LE.ACON)GO TO 25
        GC TC 40
C
C SET BRCW2 AT MAXIMUM RATIO CF BRCW1
C
 25     BRCW2=BRCW1*ACON
C
C CALCULATE BRCW
C
 40     BRCW=1000./(BRCW1+BRCW2)
        RETURN
        END
C
C
C
C
C       SUBROUTINE PSYCHO
C
C  BORROWED FROM RCUTINES OF CLIFTCN MARCEV
C
        SUBROUTINE PSYCH(DB,WB,PATM,DP,W,H,V,RH)
C
C   PURPCSE:
C
C       THIS SUBRCUTINE CALCULATES THE PSYCHOMETRIC
C       PROPERTIES CF MOIST AIR.
C
C       THIS ROUTINE HAS 7 ENTRY PCINTS:
C                 PSYCH - WHEN DRY BULB AND WET BULB ARE KNOWN
C                 PSYCH2- WHEN DRY BULB AND DEW POINT ARE KNOWN
C                 PSYCH3- WHEN DRY BULB AND RELATIVE HUMIDITY ARE KNOWN
C                 PSYCH4- WHEN ENTHALPY AND RELATIVE HUMIDITY ARE KNOWN
C                 PSYCH5- WHEN WATER CONTENT AND ENTHALPY ARE KNOWN
C                 PSYCH6- WHEN DRY BULB TEMP. AND WATER CCNTENT ARE KNCWN
C                 PSYCH7- WHEN DRY BULB TEMP. AND ENTHALPY ARE KNOWN
C
C
C   INPUTS:
C       DB=DRY BULB TEMPERATURE
C       PATM-ATMOSPHERIC PRESSURE, IN. HG.
C
C   CONDITIONAL INPUTS/OUTPUT:
C       WB - WET BULB TEMPERATURE, F
C       RH - RELATIVE HUMIDITY, PERCENT
C       DP - DEW POINT, F
C       H = ENTHALPY CF MOIST AIR, BTU/LB DRY AIR
C
C   OUTPUTS:
C       W = HUMIDITY RATIO, LB WATER/LB AIR
C       V = SPECIFIC VOLUME MOIST AIR, CU FT/LB
C
C   SUBROUTINES CALLED:
C       PVSF
C       WBF
```

```
C       DPF
C
C    ERROR CODES RETURNED BY ROUTINE:
C       IF THE DRY BULB SPECIFIED AND THE WET BULB SPECIFIED
C       ARE PHYSICALLY IMPOSSIBLE, AN ERROR CODE IS GENERATED.
C
C    USAGE OF ROUTINE:
C       PROVIDE DRY BULB TEMP. ATMOSPHERIC PRESSURE AND ONE OF THE BELOW:
C               WET BULB, DEWPOINT, RELATIVE HUMIDY OR ENTHALPY.
C
        REAL WORK(10)
C
C    ENTRY PSYCH
C       INPUTS:  DB, WB, PATM
C
        PVP = PVSF(WB)
        IF(DB.LE.WB) GO TO 4
        WSTAR = 0.622*PVP/(PATM-PVP)
        IF(WB.GT.32.) GO TO 2
        PV = PVP-5.704E-4*PATM*(DB-WB)/1.8
        GO TO 3
4       PV = PVP
        GO TO 3
2       CDB = (DB-32.)/1.8
        CWB = (WB-32.)/1.8
        HL = 597.31+0.4409*CDB-CWB
        CH = 0.2402+0.4409*WSTAR
        EX = (WSTAR-CH*(CDB-CWB)/HL)/0.622
        PV = PATM*EX/(1.+EX)
3       W = 0.622*PV/(PATM-PV)
        V = 0.754*(DB+459.7)*(1.+7000.*W/4360.)/PATM
        H = 0.24*DB+(1061.+0.444*DB)*W
*       IF(PV.LE.0.0) CALL ERREFA(109,700,0,0.0,0.0)
        DP = DPF(PV)
        RH = 100.*(PV/PVSF(DB))
        RETURN
C
C
C    ENTRY PSYCH2
C    INPUTS:  DB,DP,PATM
C
        ENTRY PSYCH2(DB,WB,PATM,DP,W,H,V,RH)
        PV = PVSF(DP)
        PVS = PVSF(DB)
        RH = 100.*PV/PVS
        W = 0.622*PV/(PATM-PV)
        V = 0.754*(DB+459.7)*(1.+7000.*W/4360.)/PATM
        H = 0.24*DB+(1061.0+0.444*DB)*W
        WB = WBF(H,PATM)
        RETURN
C
C
C    ENTRY PSYCH3
C    INPUTS:  DB,RH,PATM
C
        ENTRY PSYCH3(DB,WB,PATM,DP,W,H,V,RH)
        PVS = PVSF(DB)
        PV = RH/100.*PVS
        W = 0.622*PV/(PATM-PV)
        V = 0.754*(DB+459.7)*(1.+7000*W/4360.)/PATM
```

```
      H = 0.24*CB+(1061.+0.444*CB) *W
      DP = DPF(PV)
      WB = WBF(H,PATM)
      RETURN
C
C     INPUTS: RH, ENTHALPY (H) ANC PATM.
C     NCTE: THIS METHCD REQUIRES AN ITERATIVE SEARCH FOR THE VARIABLES
C           SC CARE SHOULD BE EXERCISED IN THE USE OF THIS ENTRY
C
      ENTRY PSYCH4(DB,WB,PATM,DP,W,H,V,RH)
      TWB=WBF(H,PATM)
      TDB1=TWB-500.*H/(RH*RH)
      TDB2=TDB1-0.2*(100.-RH)
      IFLAG=0
      WORK(1)=0.0C1
      DO 120 I=1,25
        TDB=ZERCUT(IFLAG,TDB1,TCB2,DH,WORK)
        PVS=PVSF(TDB)
        PV=0.01*RH*PVS
        W=C.622*PV/(PATM-PV)
        DH=H-(0.24*TDB+W*(1061.C+C.444*TDB))
        IF(IFLAG.EC.1.OR.IFLAG.LT.0) GOTO 140
  120   CONTINUE
*     CALL ERRERA(110,700,0,TCB,DH)
  140 DB=TDB
      WB=TWB
      DP=DPF(PV)
      V=0.754*(DP+459.7)*(1.0+7C00.0*W/4360.0)/PATM
      RETURN
C
C     INPUTS: WATER CONTENT (W),EATHALPY(H) AND PATM
C
      ENTRY PSYCH5(DB,WB,PATM,CP,W,H,V,RH)
      PV=(W*PATM)/(.622+W)
      DB=(H-1061.0*W)/(.24+.444*W)
      V=.754*(DB+459.7)*(1.+7C0C.*W/4360.)/PATM
      PVS=PVSF(CB)
      RH=100.*PV/PVS
      DP=DPF(PV)
      WB=WBF(H,PATM)
      RETURN
C
C     INPUTS:DB,W,PATM
C
      ENTRY PSYCH6(DB,WB,PATM,DP,W,H,V,RH)
      H=0.24*DB+(1061.+.444*CB)*W
      V=.754*(DB+459.7)*(1.+7C0C.*W/4360.)/PATM
      PV=(W*PATM)/(.622+W)
      PVS=PVSF(DB)
      RH=100.*PV/PVS
      DP=DPF(PV)
      WB=WBF(H,PATM)
      RETURN
C
C     INPUTS:DB,H,PATM
C
      ENTRY PSYCH7(DB,WB,PATM,CF,W,H,V,RH)
      PVS=PVSF(CB)
      WB=WBF(H,PATM)
      W=(H-.24*DB)/(1061.+0.444*CB)
```

```
      PV=(W*PATM)/(.622+W)
      RH=100.*PV/PVS
      DP=DPF(PV)
      V=.754*(DB+459.7)*(1.+7000.*W/4360.)/PATM
      RETURN
      END
C
C
C
C
C     FUNCTION WBF
C
C  CALLED FROM PSYCROMETRIC CALCULATIONS
C
      FUNCTION WBF(H,PB)
C     THIS PROGRAM APPROXIMATES THE WET BULB TEMPERATURE WHEN
C     ENTHALPY AND BAROMETRIC PRESSURE ARE GIVEN.
      IF(PB.NE.29.92) GO TO 2
      Y=ALOG(H)
      IF(H.GT.11.758)GO TO 3
C     WBF=0.6041+3.4841*Y+1.3601*Y*Y+0.97307*Y*Y*Y
      WBF=0.6041+(3.4841+(1.3601+0.97307*Y)*Y)*Y
      GO TO 4
    3 WBF=30.9185+(-39.68200+(20.5841-1.758*Y)*Y)*Y
      GO TO 4
    2 WB1=150.
      PV1=PVSF(WB1)
      W1=0.622*PV1/(PB-PV1)
      X1=0.24*WB1+(1061.+0.444*WB1)*W1
      Y1=H-X1
    9 WB2=WB1-1.
      PV2=PVSF(WB2)
      W2=0.622*PV2/(PB-PV2)
      X2=0.24*WB2+(1061.+0.444*WB2)*W2
      Y2=H-X2
      IF(Y1*Y2)6,7,8
    8 WB1=WB2
      Y1=Y2
      GO TO 9
    7 IF(Y1.NE.0.0)GO TO 10
   11 WBF=WB1
      GO TO 4
   10 WBF=WB2
      GO TO 4
    6 Z=ABS(Y1/Y2)
      WBF=(WB2*Z+WB1)/(1.+Z)
    4 RETURN
      END
C
C
C     FUNCTION PVSF
C
C CALLED FROM PSYCHROMETRIC CALCULATIONS
C
      FUNCTION PVSF(X)
      DIMENSION A(6),B(4),P(4)
      DATA A/-7.9028,5.02808,-1.3816E-7,11.344,
     1        8.1328E-3,-3.49149/
      DATA B/-9.09718,-3.56654,0.876793,0.0060273/
      T=(X+459.688)/1.8
```

```
      IF(T.LT.273.16)GO TO 3
      Z=373.16/T
      P(1)=A(1)*(Z-1.)
      P(2)=A(2)*ALCG10(Z)
      Z1=A(4)*(1.-1./Z)
      P(3)=A(3)*(10.**Z1-1.)
      Z1=A(6)*(Z-1.)
      P(4)=A(5)*(10.**Z1-1.)
      GO TO 4
    3 Z=273.16/T
      P(1)=B(1)*(Z-1.)
      P(2)=B(2)*ALOG10(Z)
      P(3)=B(3)*(1.-1./Z)
      P(4)=ALCG10(B(4))
    4 SUM = 0.0
      DC 5 I=1,4
    5 SUM=SUM+P(I)
      PVSF=29.921*10.**SUM
      RETURN
      END
C
C
C     FUNCTION DPF
C
C CALLED FROM PSYCHRCMETRIC CALCULATIONS
C
      FUNCTION CPF(PV)
C     THIS FUNCTICN CALCULATES CEW POINT TEMPERATURE FCR
C     A GIVEN VAPCR PRESSURE.
C
      Y=ALOG(PV)
      IF(PV.GT.0.1836) GO TO 1
      DPF=71.98+24.873*Y+0.8927*Y*Y
      GO TO 2
    1 DPF=79.047+30.579*Y+1.8893*Y*Y
    2 RETURN
      END
      FUNCTION ENTHAL(DB,W)
      ENTHAL=0.24*DB+(1061.0+C.444*DB)*W
      RETURN
      END
C
C
C
C
C
C     SUBROUTINE SSYCHO
C
C PURPCSE:
C     SIMPLIFIED VERSION OF "FSYCHO"
C     CNLY CALCULATES SELECTEC PSYCHROMETRIC PROPERTIES AS
C     GIVEN AT THE FIRST OF EACH ENTRY
C
C     IF PCSSIBLE THIS ROUTINE IS ADVISED OVER PSYCHO BECAUE
C     IT ELIMINATES MANY UNNECESSARY TIME CONSUMING CALCULATIONS
C     MADE BY "PSYCHO".
C
C::::::NOTATICN IS THE SAME AS FOR PSYCHO ::::::::::::
C
C
```

```
      SUBROUTINE SSYCH(DB,WB,PATM,DP,W,H,V,RH)
C
      REAL WORK(10)
C
C   ENTRY SSYCH
C      INPUTS:  DB, WB, PATM
C
      PVP = PVSF(WB)
      IF(DB.LE.WB) GO TO 4
      WSTAR = 0.622*PVP/(PATM-PVP)
      IF(WB.GT.32.) GO TO 2
      PV = PVP-5.704E-4*PATM*(DB-WB)/1.8
      GO TO 3
4     PV = PVP
      GO TO 3
2     CDB = (DB-32.)/1.8
      CWB = (WB-32.)/1.8
      HL = 597.31+0.4409*CDB-CWB
      CH = 0.2402+0.4409*WSTAR
      EX = (WSTAR-CH*(CDB-CWB)/HL)/0.622
      PV = PATM*EX/(1.+EX)
3     W = 0.622*PV/(PATM-PV)
      V = 0.754*(DB+459.7)*(1.+7000.*W/4360.)/PATM
      H = 0.24*DB+(1061.+0.444*DB)*W
*     IF(PV.LE.0.0) CALL ERREFA(109,700,0,0.0,0.0)
      DP = DPF(PV)
      RH = 100.*(PV/PVSF(DB))
      RETURN
C
C   ENTRY SSYCH3
C   INPUTS:  DB,RH,PATM
C      OUTPUTS:W,H
C
      ENTRY SSYCH3(DB,WB,PATM,DP,W,H,V,RH)
      PVS = PVSF(DB)
      PV = RH/100.*PVS
      W = 0.622*PV/(PATM-PV)
      H = 0.24*DB+(1061.+0.444*DB) *W
      RETURN
C
C
C
C     INPUTS: RH, ENTHALPY (H) AND PATM.
C     OUTPUTS: W AND DB
C     NOTE: THIS METHOD REQUIRES AN ITERATIVE SEARCH FOR THE VARIABLES
C           SO CARE SHOULD BE EXERCISED IN THE USE OF THIS ENTRY
C
      ENTRY SSYCH4(DB,WB,PATM,DP,W,H,V,RH)
      TDB1=1.5*H+400./RH
      IF(TDB1.LE.-100.)TDB1=-100.
      TDB2=TDB1+400./RH+30.
      IFLAG=0
      WORK(1)=0.001
      DO 120 I=1,25
        TDB=ZEROUT(IFLAG,TDB1,TDB2,DH,WORK)
        PVS=PVSF(TDB)
        PV=0.01*RH*PVS
        W=0.622*PV/(PATM-PV)
        DH=H-(0.24*TDB+W*(1061.0+0.444*TDB))
        IF(IFLAG.EQ.1.OR.IFLAG.LT.0) GOTO 140
```

```
  120    CONTINUE
  140   DB=TDB
        RETURN
C
C       INPUTS: WATER CONTENT (W),ENTHALPY(H) AND PATM
C       OUTPUTS: DB,RH
C
        ENTRY SSYCH5(DB,WB,PATM,DP,W,H,V,RH)
        PV=(W*PATM)/(.622+W)
        DB=(H-1061.0*W)/(.24+.444*W)
        PVS=PVSF(DB)
        RH=100.*PV/PVS
        RETURN
C
C       INPUTS:DB,W,PATM
C       OUTPUTS: H,RH
C
        ENTRY SSYCH6(DB,WB,PATM,DP,W,H,V,RH)
        H=0.24*DB+(1061.+.444*DB)*W
        PV=(W*PATM)/(.622+W)
        PVS=PVSF(DB)
        RH=100.*PV/PVS
        RETURN
C
C       INPUTS:DB,H,PATM
C       OUTPUTS:W,RH
C
        ENTRY SSYCH7(DB,WB,PATM,DP,W,H,V,RH)
        PVS=PVSF(DB)
        W=(H-.24*DB)/(1061.+0.444*DB)
        PV=(W*PATM)/(.622+W)
        RH=100.*PV/PVS
        RETURN
C
C       INPUTS:WATER CONTENT(W),ENTHALPY(H),AND PATM
C       OUTPUTS:DB
C
        ENTRY SSYCH8(DB,WB,PATM,DP,W,H,V,RH)
        DB=(H-1061.0*W)/(.24+.444*W)
        RETURN
C
C       INPUTS:DB,H,PATM
C       OUTPUTS:W
C
        ENTRY SSYCH9(DB,WB,PATM,DP,W,H,V,RH)
        W=(H-.24*DB)/(1061.+.444*DB)
        RETURN
        END
C
C
C
C
```

```
C
C      SUBROUTINE ZEROUT
C
C      WRITTEN BY C. RUSSELL SMITH
C      UNIVERSITY OF TENNESSEE APPLIED CONTROL LABS
C
       REAL FUNCTION ZEROUT(IFLAG,AX,BX,F,W)
C
C   PURPOSE:
C      ROUTINE TO PROVIDE A SYSTEMATIC SEARCH FOR THE ZEROS OF
C        A FUNCTION CALCULATED BY THE CALLER.
C        THUS, THE ROUTINE RETURNS A VALUE OF THE INDEPENDENT VARIABLE,
C          AND USES THE FUNCTION VALUE USING PROVIDED ON THE NEXT CALL
C          TO DETERMINE THE SUBSEQUENT STEP.
C
C   USE OF THE ROUTINE:
C      PROVIDE THE INITIAL RANGE OF THE SEARCH, VIA 'AX' AND 'BX'
C      ALLOCATE A WORKING ARRAY IN THE CALLING ROUTINE, WORK(10)
C       THE ARRAY MUST BE DIMENSIONED AT LEAST TEN IN LENGTH.
C      INITIALLY SET THE VARIABLE 'IFLAG' TO 0.
C       THE USER MUST MONITOR THE VALUE OF IFLAG TO DETERMINE WHEN
C        THE SOLUTION HAS BEEN FOUND.
C      AFTER THE INITIAL CALL, PROVIDE THE FUNCTIONAL EVALUATION
C       FOR THE VALUE RETURN BY THE FUNCTION FROM THE PREVIOUS CALL.
C      THE TOLERANCE FOR THE ACCURACY OF THE SOLUTION IS PROVIDED BY
C        THE USER IN WORK(1).  IT MAY BE EQUAL TO ZERO AND THE ROUTINE
C        WILL FIND AN ANSWER TO THE ACCURACY OF THE MACHINE EPSILON.
C      ON COMPLETION OF THE ROUTINE, IFLAG IS SET TO 1.
C
C   INTERMEDIATE DETAILS:
C      WORK(1) - USER SPECIFIED ERROR TOLERANCE. IT IS UNALTERED ACROSS
C                    CALLS.
C      WORK(2),WORK(3) - POINT A,F(A).
C      WORK(4),WORK(5) - POINT B,F(B)
C      WORK(6),WORK(7) - POINT C,F(C)
C      WORK(8) - VARIABLE 'D'
C      WORK(9) - VARIABLE 'E'
C
C              MEANING OF THE VARIOUS FLAG NUMBERS.
C       -1 - THE SEARCH INTERVAL HAS SHRUNK TO BELOW THE MACHINE
C              PRECISION WITHOUT FINDING AN ANSWER.
C       -2 - MORE THAN 150 ITERATIONS ARE REQUIRED.
C        0 - USER SET FLAG TO ZERO FOR THE INITIAL CALL.
C        1 - THE ROUTINE SET IFLAG TO ONE WHEN A ZERO IS FOUND.
C              THE VALUE OF THE ZERO IS 'ZEROUT'.
C        2 - THE SECOND CALL TO THE ROUTINE IS BEING MADE.  THUS,
C              ONLY ONE POINT IS KNOWN FULLY AND THE FUNCTIONAL
C              VALUE OF THE SECOND POINT WILL BE REQUESTED UPON
C              COMPLETION OF THIS CALL TO THE ROUTINE.
C        3 - THERE IS NO ZERO IN THE INCREMENT SPECIFIED BY THE CALLER.
C              THE ROUTINE WILL EXTRAPOLATE TO FIND A POTENTIAL INTERVAL.
C              A LIMITED TWO POINT SECANT METHOD IS USED.  THE LIMITS
C              RANGE FROM 0.2 TO 2.0 TIMES THE CURRENT SEARCH INTERVAL.
C        4 - AN INTERVAL IS BEING USED THAT CONTAINS A ZERO.  THIS IS
C              THE USUAL RETURNED VALUE UNTIL A ZERO IS ACTALLY FOUND.
C          ****IMPORTANT**** THE USER MUST NOT MODIFY 'IFLAG' AFTER THE
C              INITIAL CALL. THIS IMPLIES THAT IFLAG MUST BE A VARIABLE.
C
C
C      SEE FORSYTHE, MALCOLM AND MOLER FOR THE BASIC PROCEDURE.
```

```
C
C
      REAL AX,BX,F,TOL,W(10)
      REAL A,B,C,D,E,EPS,FA,FB,TCL1,XM,P,Q,R,S
C
C     PROVIDE THE MACHINE PRECISICN, EXPANSICN FACTOR ANC CCNTRACTION
C       FACTORS FCR EXTRAPOLATICN OUTSIDE CF THE PROVICEC REGICN.
C
      DATA EPS/3.752529032E-09/,EF/2.0/,CF/0.2/
C
      A=W(2)
      FA=W(3)
      B=W(4)
      FB=W(5)
      C=W(6)
      FC=W(7)
C
C     CETERMINE THE ACTICN TO BE TAKEN WITH THE CURRENT VALUE OF IFLAG.
C       IF:
C             IFLAG .LT. 0 - STCP THE ROUTINE COLD. THE USER HAS
C                            IGNCRED THE ERROR RETURNED PREVIOUSLY.
C             IFLAG .EQ. 0 - INITIALIZE THE RCUTINE.
C             IFLAG .EQ. 1 - SIMPLY RETURN, THE ROUTINE HAS PREVIOUSLY
C                            FCUND A ZERO AND THE USER HAS NOT HEEDED
C                            THIS RESULT.  IT IS ASSUMED THE USER HAS
C                            THE ROUTINE W/I A DO LCOP SO IT WILL NOT
C                            BE LOOPING FCR EVER.
C             IFLAG .GT. 1 - SEE ABOVE.
C
      IF(IFLAG.GE.4) GOTO 190
      IF(IFLAG) 100,110,120
  100 STOP 'IFLAG.LT.0'
  110 A=AX
      B=A
      IFLAG=2
      GOTC 90
  120 IF(IFLAG-2) 99,125,130
  125 B=BX
      FA=F
      IFLAG=3
      GOTC 90
  130 FB=F
      IF(ABS(FA-FB).LT.(EPS*(ABS(FA)+ABS(FB)))) GOTC 95
      P=B-FB*(A-B)/(FA-FB)
      IF(A.LT.B) GOTO 140
        C=A
        A=B
        B=C
        FC=FA
        FA=FB
        FB=FC
  140 IF(P.GT.A) GOTO 150
      IF((A-P).LT.EF*(B-A)) P=A-EF*(B-A)
      IF((A-P).GT.CF*(B-A)) P=A-CF*(B-A)
      B=P
      GOTC 90
  150 IF(P.LT.B) GOTO 160
      IF((P-B).GT.EF*(B-A)) P=B+EF*(B-A)
      IF((P-B).LT.(CF*(B-A))) P=B+CF*(B-A)
      A=B
```

```
      FA=FB
      B=P
      GOTO 90
 160  IFLAG=4
      GCTC 20
 190  IFLAG=IFLAG+1
      FB=F
      IF((FB*(FC/ABS(FC))).LE.0.0) GOTO 30
C
  20  C=A
      FC=FA
      W(8)=B-A
      W(9)=W(8)
  30  IF(ABS(FC).GE.ABS(FB)) GCTO 40
      A=B
      B=C
      C=A
      FA=FB
      FB=FC
      FC=FA
  40  TCL1=2.0*EPS*ABS(B)+0.5*W(1)
      XM=0.5*(C-B)
      IF(ABS(XM).LE.TOL1) GOTC 94
      IF(FB.EC.0.0) GCTC 94
C
      IF(ABS(W(9)).LT.TOL1) GCTC 70
      IF(ABS(FA).LE.ABS(FB)) GCTC 70
C
      IF(A.NE.C) GCTO 50
      S=FB/FA
      P=2.0*XM*S
      Q=1.0-S
      GOTO 60
  50  Q=FA/FC
      R=FB/FC
      S=FB/FA
      P=S*(2.0*XM*Q*(Q-R)-(B-A)*(R-1.0))
      Q=(Q-1.0)*(R-1.0)*(S-1.0)
C
  60  IF(P.GT.0.0) Q=-Q
      P=ABS(P)
C
      IF((2.0*P).GE.(3.0*XM*Q-ABS(TOL1*Q))) GOTO 70
      IF(P.GE.ABS(0.5*W(9)*Q)) GCTO 70
      W(9)=W(8)
      W(8)=P/Q
      GCTC 80
C
  70  W(8)=XM
      W(9)=W(8)
  80  A=B
      FA=FB
      IF(ABS(W(8)).GT.TOL1) B=B+W(8)
      IF(ABS(W(8)).LE.TOL1) B=B+SIGN(TOL1,XM)
C
C     THIS IS THE COMMON ROUTINE EXIT.  THE WORKING ARRAY IS RESTORED
C       FOR THE NEXT CALL.  NOTE: THIS WORKING ARRAY IS NOT NECESSARY
C       IF ONLY ONE SEARCH IS BEING DONE AT A TIME.
C
  90  IF(IFLAG.GT.150) IFLAG=-2
```

```
      ZEROUT=B
      W(2)=A
      W(3)=FA
      W(4)=B
      W(5)=FB
      W(6)=C
      W(7)=FC
      RETURN
C
C     A SOLUTION HAS BEEN FOUND. SET THE FLAG AND GO TC THE CCMMON EXIT.
C
   94 IFLAG=1
      GOTO 90
C
C     INTERVAL OF SEARCH HAS SHRUNK TCO SMALL TC BE USED WITHCUT
C       FINDING A CRCSS-OVER FCINT.  THIS USUALLY OCCURS WHEN A LOCAL
C       MINIMUM HAS BEEN FOUNE WHILE TRYING TC FIND AN INTERVAL
C       THAT HAS A CROSSING OF THE X-AXIS.  THIS CANNCT CCCUR IF
C       THE INITIAL INTERVAL HAS FUNCTICNAL EVALUATIONS WITH
C       DIFFERENT SIGNS.  FOR THE CASE CF AND INITIAL ECUNCS ON THE
C       ZERO, THE RCUTINE WILL CCNVERGE IN A FINITE NUMBER OF STEPS.
C
   95 IFLAG=-1
   99 RETURN
      END
```

```
C
C
C
C
C
C
C
C               DUAL-DUCT SIMULATION
C
C
C
C
C
C
C       NOTE:  THIS SIMULATION USES SEVERAL ROUTINES JUST PRESENTED
C FOR THE SINGLE DUCT AIR HANDLER.
C
C       THE COMMON ROUTINES ARE:
C               SFAN.FOR, AIRSM.FOR, PSYCHC.FOR, SSYCHC.FOR,
C               MIXBC2.FOR, CWCCAL.FOR, CWCOIL.FOR,
C               QCWCAL.FOR, BRCYCL.FOR, AND ZEROUT.FOR
C
C
CC
C       ANOTHER ROUTINE USED FOR THIS PROGRAM THAT IS NOT LISTED
C       HERE IS THE ROUTINE  "ZSCNT" BORROWED FROM THE IMSL LIBRARIES
C
C
C
C
C
C               DUAL DUCT BASE CASE
C
C
C       MAIN ROUTINE FOR DUAL DUCT SIMULATION
C       THIS ROUTINE PROVIDES THE SYSTEM LOADS AND CALLS "ZSCNT"
C       WHICH CALLS THE ROUTINE "DDLOOP" TO SIMULATE THE SYSTEM LOOP
C
C       NOTATION CAN BE DETERMINED BY CHECKING DIRECT EQUALITIES IN
C       ROUTINE "DDLOOP"
C
C
C
        EXTERNAL FCN
        DIMENSION X(2),WK(60),OAT(100),OURS(100),OAH(100)
        NAMELIST/ONEI/CDSPT,HDSPT,Z1SPT,Z2SPT,Z3SPT,Z1Q,Z2Q,Z3Q,Z1W,
       1Z2W,Z3W,SPMBT,ATMP,ADLH,ASTLH,STFMAX,CWEWT,WFAN,EFFAN,AIRCP,
       2Z1F,Z2F,Z3F,OAIRMN,OATEMP,CAENTH,TFLOW,X
        NAMELIST/ONEO/CDTOTF,HDTOTF,OHFNI,OTFNI,GUESCT,GUESHT,GUESCH,
       1GUESHH,Z1T,Z1H,Z1FCD,Z1FHD,Z2T,Z2H,Z2FCD,Z2FHD,Z3T,Z3H,
       2Z3FCD,Z3FHD,QHEAT,QCOOL,STFLOW,CWFLO,TEMACC,ENTACC,RHACC,
       3TEMAHC,ENTAHC,DELHC,DELCC,RAWAT,RATEM,RAENT,FINTOF,FINHOF,
       4FNRHOF,FNORM,IER,WTLTEP,J,K,JMAX,FRAMBX,FOAMBX,TMXBX,HMXBX
        COMMON/DDUCTI/CDSPT,HDSPT,Z1SPT,Z2SPT,Z3SPT,Z1Q,Z2Q,Z3Q,Z1W,
       1Z2W,Z3W,SPMBT,ATMP,ADLH,ASTLH,STFMAX,CWEWT,WFAN,EFFAN,AIRCP,
       2Z1F,Z2F,Z3F,OAIRMN,OATEMP,CAENTH,TFLOW
        COMMON/DDUCTO/CDTOTF,HDTOTF,OHFNI,OTFNI,GUESCT,GUESHT,GUESCH,
       1GUESHH,Z1T,Z1H,Z1FCD,Z1FHD,Z2T,Z2H,Z2FCD,Z2FHD,Z3T,Z3H,
       2Z3FCD,Z3FHD,QHEAT,QCOOL,STFLOW,CWFLO,TEMACC,ENTACC,RHACC,
       3TEMAHC,ENTAHC,DELHC,DELCC,RAWAT,RATEM,RAENT,FINTOF,FINHOF,
```

```
      4FNRHOF,WTLTEM,J,K,JMAX,FRAMBX,FOAMBX,TMXBX,HMXBX
      CCMMCN/DDCUT2/Z1RH,Z2RH,Z3RH
      COMMON/ACMAX/CWMAX
      COMMCN/CMIX/MXSWIT
      K=0
C
C DUAL DUCT PARAMETERS
C
      SWIT=2.
      TFLOW=70000.
      OAIRMN=.15*TFLOW
      ATMP=29.921
      ADLH=1037.9
      ASTLH=962.2
      STFMAX=855.
      CWEWT=44.
      WFAN=18.5
      EFFAN=.7
      AIRCP=.24
      CWMAX=110124.
      Z1F=30000.
      Z2F=24000.
      Z3F=16000.
      OARH=50.
C
C READ IN BINS DATA
C
      DC 50 J=1,100
      READ(20,*,END=60)OAT(J),OURS(J)
      NDAT=NDAT+1
 50   CCNTINUE
C
C DETERMINE VALUES NEEDED FOR LOAD CALCULATIONS
C
C
C ZCNE FLOCR AREA
C
 60   AREA1=8350                    !APPROX. 1.35*CFM ZONE FLOW
      AREA2=6700
      AREA3=4450
C ZCNE WALL OUTSIDE EXPOSURE AREA  WITH 12 FOOT CEILING SQUARE ROOM
      FAREA1=(AREA1**.5)*12.
      FAREA2=(AREA2**.5)*12.
      FAREA3=(AREA3**.5)*12.
C TRANSFER COEFF TC OUTSIDE-UA
      UAZ1=.1*FAREA1
      UAZ2=.1*FAREA2
      UAZ3=.1*FAREA3
C PEOPLE IN ZONE     1 PER 250 FT**2
      PEOP1=33.
      PEOP2=27.
      PEOP3=18.
C EQUIPMENT GAINS ASSUME 100 WATTS/PERSON
      EQUIP1=PECP1*100.*3.4122
      EQUIP2=PECP2*100.*3.4122
      EQUIP3=PECP3*100.*3.4122
C ELECTRIC LIGHT GAINS  2 WATTS/FT**2  FLOOR AREA
      ELITE1=AREA1*2.*3.4122
      ELITE2=AREA2*2.*3.4122
      ELITE3=AREA3*2.*3.4122
```

```
C PEOPLE SENSIBLE LOAD  250 BTL/FEFSON
      PEEPS1=PECP1*250
      PEEPS2=PECP2*250.
      PEEPS3=PEOP3*250.
C
C PUT IN LARGE LCP DIFFERENT CCNDITIONS
C
      DC 500 I=1,NCAT
      OATEMP=OAT(I)
      THOURS=OURS(I)
      CALL SSYCH3(CATEMP,WB,ATMP,CP,OAW,OAENTH,OAV,OARH)
      OAH(I)=OAENTH
C
C  BASE CASE DUAL CUCT SET PCINTS
C
C  COLD DUCT
      CDSPT=53.
C HOT DUCT
      IF(OATEMP.LE.0.)GO TO 111
      IF(OATEMP.GE.60.)GO TO 113
      HDSPT=90.+OATEMP*(-.25)
      GO TO 115
 111  HDSPT=90.
      GO TC 115
 113  HDSPT=75.
C MIXING BCX
C
C MIXING BOX SWITCH DESIGINATICNS
C
C  MXSWIT=1 MEANS MIXING BOX CCNTRCLLED ABOUT SPMBT
C  MXSWIT=2 MEANS CNLY RETURN AIR AND MINIMUM OUTSIDE AIR
C MUST USE ROUTINE MXDDBC.FOR TC UTILIZE THIS SWITCH
C
 115  SPMBT=55.
      IF(OATEMP.GE.60.)GO TO 120
      MXSWIT=1
      GC TC 121
 120  MXSWIT=2
C
C CHECK SUMMER/WINTER SWITCH TC DETERMINE OPERATICN
C SWIT=1 MEANS SUMMER
C SWIT=2 MEANS CCNSTANT ROOM SET PCINT ALL YEAR
C
 121  IF(SWIT.EC.1.)GO TO 140
      IF(SWIT.EC.2.)GO TO 135
      Z1SPT=66.
      Z2SPT=66.
      Z3SPT=66.
      GO TO 150
 135  Z1SPT=72.
      Z2SPT=72.
      Z3SPT=72.
      GO TO 150
 140  Z1SPT=78.
      Z2SPT=78.
      Z3SPT=78.
C
C GIVE LOADS WATER AND HEAT FOF ZCNE
C
 150  CAIRTP=CATEMP
```

```
      IF(OAIRTP.LE.66.)OAIRTP=66.
C
C GIVE FACTOR FOR ENERGY MULTIFLIERS
C
      FZF1=0.5
      FZF2=1.0
      FZF3=1.5
C
C CALCULATE LOADS
C
      Z1Q=(UAZ1*(OAIRTP-Z1SPT)+EQUIP1+ELITE1+PEEPS1)*FZF1
      Z2Q=(UAZ2*(OAIRTP-Z2SPT)+EQUIP2+ELITE2+PEEPS2)*FZF2
      Z3Q=(UAZ3*(OAIRTP-Z3SPT)+EQUIP3+ELITE3+PEEPS3)*FZF3
      Z1W=PEOP1*.241
      Z2W=PEOP2*.241
      Z3W=PEOP3*.241
C
C SOLVE SYSTEM
C
      X(1)=60.
      X(2)=50.
      NSIG=5
      N=2
      ITMAX=20.
      CALL ZSCNT(FCN,NSIG,N,ITMAX,PAR,X,FNORM,WK,IER)
      WRITE(6,ONEI)
      WRITE(6,ONEC)
C
C
C SUM TERMS TO DESCRIBE TOTAL USAGE WITH WEIGHT FOR HOURS
C
      QH=OURS(I)*QHEAT
      QC=OURS(I)*QCOOL
      FS=OURS(I)*STFLOW
      FC=OURS(I)*CWFLO
      QSTMYR=QSTMYR+QH
      FSTMYR=FSTMYR+FS
      QCWYR=QCWYR+QC
      FCWYR=FCWYR+FC
C
C WRITE VALUES IN FILES FOR PLOTTING PURPOSES
C
      WRITE(25,281)OATEMP,THOURS,QHEAT,QCOOL,STFLOW,CWFLO
 281  FORMAT(1X,6(1PE15.5,2X))
      WRITE(26,282)OATEMP,THOURS,QH,QC,FS,FC
 282  FORMAT(1X,6(1PE15.5,2X))
      WRITE(27,283)OATEMP,Z1Q,Z2Q,Z3Q,Z1W,Z2W,Z3W
 283  FORMAT(1X,7(1PE15.5,2X))
      WRITE(28,284)OATEMP,Z1T,Z2T,Z3T,Z1RH,Z2RH,Z3RH
 284  FORMAT(1X,7(1PE15.5,2X))
C WRITE
C
      WRITE(6,332)QH,FS,QC,FC
 332  FORMAT(' AT GIVEN TEMP.  QH= '1PE15.5,3X,' FS= '1PE15.5,3X,'
     1 QC= '1PE15.5,3X,' FC= '1PE15.5/)
      WRITE(6,334)QSTMYR,FSTMYR,QCWYR,FCWYR
 334  FORMAT(' TOTAL TO DATE  QSTMYR= '1PE15.5,3X,' FSTMYR= '1PE15.5,
     13X,' QCWYR= '1PE15.5,3X,' FCWYR= '1PE15.5/)
C
C
```

```
      WRITE(6,345)
 345  FORMAT('    '///)
      WRITE(6,346)
 346  FORMAT(' NEXT CASE '//)
      WRITE(5,347)CATEMP
 347  FORMAT(' CASE FINISHED WITH OUTSIDE TEMP. = 'F12.5)
 500  CONTINUE
      STOP
      END
C
C
C
C
C
C
C     SUBROUTINE DDLFCN
C
C PURPOSE:
C     USED BY RCUUTINE ZSCNT CF IMSL TO CALCULATE FUNCTIONS IN
C DETERMINATION CF TEMPERATURE AND RELATIVE HUMIDITY CUT CF THE SUPPLY
C FAN.
C
      SUBROUTINE FCN(X,F,N,PAR)
      DIMENSICN X(2),F(2),PAR(1)
C
C     X(1)   = TEMPERATURE OUT OF THE SUPPLY FAN
C     X(2)   = RELATIVE HUMIDITY CUT CF THE SUPPLY FAN
C
C  X(1) AND X(2) ARE THE INITIAL GUESSES  UPON INPUT AND THE FINAL
C RESULTS UPON OUPUT.  THROUGH ZSCNT OF IMSL VALUES OF X(1) AND
C X(2) ARE FCUND SUCH THAT F(1) AND F(2) ARE EQUAL TC ZERC.
C
      CALL DDLOOP(X(1),X(2),TC,RHC)
      F(1)=X(1)-TC
      F(2)=X(2)-RHO
      RETURN
      END
C
C
C
C
C
C
C     SUBROUTINE DDLOOP
C
C PURPCSE:
C     GIVEN SET PCINTS CF ZCNE TEMPERATURES FCR EACH ZCNE(1,2,3), THE
C HCT ANC COLD DUCT SET POINTS, A MIXING BOX OPERATION AND THE TEMPERATURE
C AND RELATIVE HUMIDITY OF THE AIR LEAVING THE FAN, THE ROUTINE CALCULATES:
C STEAM ANC CHILLED WATER REQUIREMENTS, THE HOT AND CCLC CUCT FLCWS, THE
C RETURN AIR CCNDITCNS AND GIVES A RESULTING VALUE OF AIR LEAVING THE
C FAN.  THE ROUTINE IS CALLED BY A HIGHER LEVEL SEARCH ROUTINE WHICH
C ATTEMPTS TO MATCH THE STARTNG ANC RESULTING CONDITIONS OF AIR LEAVING THE
C FAN.
C
C INPUT:
C     FTOT=TOTAL AIR FLCW                    LBS/HR
C     TMBSP=MIXING BOX SET POINT             F
C     TCDSP=CCLC CUCT TEMPERATURE SET POINT  F
C     THDSP=HOT DLCT TEMPERATLRE SET POINT   F
```

```
C       TZ1SP=ZONE 1 TEMPERATURE SET POINT      F
C       TZ2SP=ZONE 2 TEMPERATURE SET POINT      F
C       TZ3SP=ZONE 3 TEMPERATURE SET POINT      F
C       FZ1=ZONE 1 AIR FLOW                     LBS/HR
C       FZ2=ZONE 2 AIR FLOW                     LBS/HR
C       FZ3=ZONE 3 AIR FLOW                     LBS/HR
C       TEXFAN=INPUT TEMPERATURE EXIT FAN       F
C       RHEXFN=RELATIVE HUMIDITY EXIT FAN       %
C       QZ1=HEAT LOAD ZONE 1                    BTU/HR
C       QZ2=HEAT LOAD ZONE 2                    BTU/HR
C       QZ3=HEAT LOAD ZONE 3                    BTU/HR
C       WZ1=WATER LOAD ZONE 1                   LBS/HR
C       WZ2=WATER LOAD ZONE 2                   LBS/HR
C       WZ3=WATER LOAD ZONE 3                   LBS/HR
C       PATM=ATMOSPHERIC PRESSURE               IN HG
C       DLH=LATENT HEAT OF WATER                BTU/LB
C       STLH=STEAM LATENT HEAT                  BTU/LB
C       TOA=OUTSIDE AIR TEMPERATURE             F
C       HOA=OUTSIDE AIR ENTHALPY                BTU/LB
C       CAMIN=MINIMUM OUTSIDE AIR REQUIREMENT   LBS/HR
C       FSMAX=MAXIMUM STEAM FLOW                LBS/HR
C       EWT=ENTERING CHILLED WATER TEMPERATURE  F
C       WF=HORSEPOWER RATING OF SUPPLY FAN      HP
C       EFF=SUPPLY FAN EFFICIENCY               %
C       CPAIR=HEAT CAPACITY OF AIR              BTU/LB-F
C
C
C
C OUTPUTS:
C       FCDTOT=TOTAL COLD DUCT AIR FLOW         LBS/HR
C       FHDTOT=TOTAL HOT DUCT AIR FLOW          LBS/HR
C       HOFNI=INITIAL GUESS ENTHALPY OUT OF FAN BTU/LB
C       TCGUES=GUESS AT COLD DUCT TEMPERATURE   F
C       THGUES=GUESS AT HOT DUCT TEMPERATURE    F
C       HCGUES=GUESS AT COLD DUCT ENTHALPY      BTU/LB
C       HHGUES=GUESS AT HOT DUCT ENTHALPY       BTU/LB
C       TZ1=ZONE 1 TEMPERATURE                  F
C       TZ2=ZONE 2 TEMPERATURE                  F
C       TZ3=ZONE 3 TEMPERATURE                  F
C       HZ1=ZONE 1 ENTHALPY                     BTU/LB
C       HZ2=ZONE 2 ENTHALPY                     BTU/LB
C       HZ3=ZONE 3 ENTHALPY                     BTU/LB
C       FCDZ1=COLD DUCT FLOW INTO ZONE 1        LBS/HR
C       FCDZ2=COLD DUCT FLOW INTO ZONE 2        LBS/HR
C       FCDZ3=COLD DUCT FLOW INTO ZONE 3        LBS/HR
C       FHDZ1=HOT DUCT FLOW INTO ZONE 1         LBS/HR
C       FHDZ2=HOT DUCT FLOW INTO ZONE 2         LBS/HR
C       FHDZ3=HOT DUCT FLOW INTO ZONE 3         LBS/HR
C       HHEAT=DUTY ON HEATING COILS             BTU/HR
C       HCOOL=DUTY ON COOLING COILS             BTU/HR
C       FST=STEAM FLOW                          LBS/HR
C       FCHW=CHILLED WATER FLOW                 LBS/HR
C       TACC=TEMPERATURE AIR AFTER COOLING COILS  F
C       HACC=ENTHALPY AIR AFTER COOLING COILS   BTU/LB
C       ALRH=RELATIVE HUMIDITY AFTER COOLING COILS  %
C       TAHC=AIR TEMPERATURE AFTER HEATING COILS  F
C       HAHC=AIR ENTHALPY AFTER HEATING COILS   BTU/LB
C       DELT=DIFFERENCE IN HEATING COIL TEMP. AND GUESS
C       DELT2=DIFFERENCE IN COOLING COIL TEMP. AND GUESS
C       WRA=WATER CONTENT OF RETURN AIR         LBS H2O/LB DRY AIR
```

```
C       TRA=TEMPERATURE OF RETURN AIR              F
C       HRA=ENTHALPY OF RETURN AIR                 BTU/LB
C       TCFANF=END OF LOOP TEMP. OUT OF FAN        F
C       HCFANF=END OF LOOP ENTHALPY OUT OF FAN     BTU/LB
C       RHOFNF=RELATIVE HUMIDITY OUT OF FAN ON FINAL LOOP
C       FRAMBX=RETURN AIR FLOW INTO MIXING BOX               LBS/HR
C       FOAMBX=OUTSIDE AIR FLOW INTO MIXING BOX              LBS/HR
C       TMXBX=MIXING BOX TEMPERATURE                         F
C       HMXBX=MIXING BOX ENTHALPY                            BTU/LB
C
C
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C
C
        SUBROUTINE DDLOOP(TOFANI,RHCFNI,TOFANF,RHOFNF)
C
C PROVIDE COMMONS
C
        COMMON/DDUCTI/CDSPT,HDSPT,Z1SPT,Z2SPT,Z3SPT,Z1Q,Z2Q,Z3Q,Z1W,
       1Z2W,Z3W,SPMBT,ATMP,ADLH,ASTLH,STFMAX,CWEWT,WFAN,EFFAN,AIRCP,
       2Z1F,Z2F,Z3F,OAIRMN,OATEMP,OAENTH,TFLOW
        COMMON/DDUCTO/CDTOTF,HDTOTF,OHFNI,OTFNI,GUESCT,GUESHT,GUESCH,
       1GUESHH,Z1T,Z1H,Z1FCD,Z1FHD,Z2T,Z2H,Z2FCD,Z2FHD,Z3T,Z3H,
       2Z3FCD,Z3FHD,QHEAT,QCOOL,STFLOW,CWFLO,TEMACC,ENTACC,RHACC,
       3TEMAHC,ENTAHC,DELHC,DELCC,RAWAT,RATEM,RAENT,FINTOF,FINHOF,
       4FNRHOF,WTLTEM,J,K,JMAX,FRAMBX,FOAMBX,TMXBX,HMXBX
        COMMON/DDCUT2/Z1RH,Z2RH,Z3RH
        COMMON/ACMAX/CWMAX
C
C RENAME COMMON ELEMENTS SO SUBROUTINE CALLS CAN BE MADE
C
C INPUTS:
        TCDSP=CDSPT
        THDSP=HDSPT
        TZ1SP=Z1SPT
        TZ2SP=Z2SPT
        TZ3SP=Z3SPT
        TEXFAN=TOFANI
        RHEXFN=RHOFNI
        QZ1=Z1Q
        QZ2=Z2Q
        QZ3=Z3Q
        WZ1=Z1W
        WZ2=Z2W
        WZ3=Z3W
        TMBSP=SPMBT
        PATM=ATMP
        DLH=ADLH
        STLH=ASTLH
        FSMAX=STFMAX
        FTOT=TFLOW
        OAMIN=OAIRMN
        TOA=OATEMP
        HOA=OAENTH
        FZ1=Z1F
        FZ2=Z2F
        FZ3=Z3F
        EWT=CWEWT
        WF=WFAN
        EFF=EFFAN
```

```
      CPAIR=AIRCP
C
C
C SET J=0 FOR SUCC. SUBST. COUNTER, AND SET UP K COUNTER FOR NUMBER
C OF CHANGES OF TEMP. AND RH LEAVING FAN
C
      J=0
      K=K+1
C
C
C GIVE CHECKS TO INSURE KEEPING TEMPERATURE AND RELATIVE HUMIDITY
C GUESSES IN REASONALBLE RANGES
C
C
C CHECK TEMPERATURES
C
      IF(TEXFAN.GT.100.)GO TO 20
      IF(TEXFAN.LT.0.)GO TO 22
      GO TO 30
 20   TEXFAN=110.-10.*EXP((100.-TEXFAN)/10.)
      TOFANI=TEXFAN
      GO TO 30
 22   TEXFAN=-10.+10.*EXP(TEXFAN/10.)
      TOFANI=TEXFAN
C
C NOW CHECK RELATIVE HUMIDITY EXIT THE FAN
C
 30   IF(RHEXFN.GT.99.0)GO TO 32
      IF(RHEXFN.LT.1.)GO TO 34
      GO TO 40
 32   RHEXFN=99.+(RHEXFN-99.)/201
      RHOFNI=RHEXFN
      GO TO 40
 34   RHEXFN=1.+(RHEXFN-1.)/100.
      RHOFNI=RHEXFN
C
C
C DETERMINE CONDITIONS OF INPUT GUESS FOR AIR EXIT FAN
C
 40   CALL SSYCH3(TEXFAN,WZOFNI,PATM,DPOFNI,WOFNI,HOFNI,VEXFN,
     1RHEXFN)
C
C DETERMINE GUESS CONDITION FOR WATER CONTENT OUT OF COOLING COIL
C
      IF(TCDSP.LT.TOFANI)GO TO 50
      WGACC=WOFNI
      HGACC=HOFNI
      GO TO 70
 50   RHGACC=90.
      CALL SSYCH3(TCDSP,WBG,PATM,DPG,WGACC,HGACC,RHGACC)
C
C DETERMINE FIRST PASS FLOW AFTER COOLIING COIL BY DETERMINING DUCT
C REQUIREMENTS FOR EACH ZONE
C
C FIRST NEED AIR CONDITIONS AFTER HEATING COIL
C
 70   WHD=WOFNI
      CALL SSYCH6(THDSP,WBHD,PATM,DPHD,WHD,HHD,VHD,RHHD)
C
C FIRST PASS TEMPERATURES=SET POINTS
```

```
C
      TCGUES=TCCSP
      THGUES=THDSP
      HCGUES=HGACC
      HHGUES=HHD
C
C
C CALL ZONE MIXING BOXES
C
 80   CALL ZTMXBX(QZ1,WZ1,THGUES,TCGUES,HHGUES,HCGUES,TZ1SP,
     1DLH,PATM,FZ1,TZ1,FHDZ1,FCDZ1,HZ1)
      CALL ZTMXBX(QZ2,WZ2,THGUES,TCGUES,HHGUES,HCGUES,TZ2SP,
     1DLH,PATM,FZ2,TZ2,FHDZ2,FCDZ2,HZ2)
      CALL ZTMXBX(QZ3,WZ3,THGUES,TCGUES,HHGUES,HCGUES,TZ3SP,
     1DLH,PATM,FZ3,TZ3,FHDZ3,FCDZ3,HZ3)
C
C CHECK FLOWS
C
      FCDTOT=FCDZ1+FCDZ2+FCDZ3
      FHDTOT=FHDZ1+FHDZ2+FHDZ3
C
C CHECK NEMD FOR HEATING COIL CALCULATION
C
      IF(THDSP.LE.TOFANI)GO TO 300
      CALL HEACOL(HOFNI,WOFNI,THDSP,FHDTOT,HAHC,FAHC,STLH,FSMAX,
     1HHEAT,FST,PATM,TAHC)
      GO TO 310
 300  HHEAT=0.
      FST=0.
      TAHC=TOFANI
      HAHC=HOFNI
C
C WAS HOT DUCT TEMPERATURE MET
C
 310  DELT=ABS(TAHC-THGUES)
      IF(DELT.LT..001)GO TO 320
      THGUES=TAHC
      HHGUES=HAHC
      GO TO 80
C
C CHECK NEED FOR COOLING COIL
C
 320  IF(TCDSP.LT.TOFANI)GO TO 330
C
C NO COOLING REQUIRED
C
      TACC=TOFANI
      WACC=WOFNI
      HACC=HOFNI
      HCOCL=0.
      FCHW=0.
      GO TO 350
C
C
C DO COOLING COIL CALCULATION
C
 330  IF(JCWFL.EQ.1)GO TO 340
      CALL CWCCAL(FCDTOT,EWT,FCHW,TOFANI,HOFNI,ALDBL,ALH,ALWT,
     1QWAT,WATER,RATC,TCDSP,PATM,QEXC,ALRH,NONE,NTWC)
      IF(FCHW.GT.CWMAX)GO TO 340
```

```
      GO TO 345
C
C FIND CPERATICN AT MAXIMUM COCLING RATE
C
 340  FCHW=CWMAX
      CALL PSYCH7(TOFANI,EWB,PATM,EDP,EWAT,HOFNI,EV,ERH)
      CALL CWCOIL(FCDTOT,EWT,FCHW,TOFANI,HOFNI,ALDBL,ALH,ALWT,
     1QWAT,WATER,RATO,PATM,QEXC,ALRH,NTWO,EV,EDP,EWAT)
      JCWFL=1
 345  TACC=ALCBL
      HACC=ALH
      CALL SSYCH9(ALDB,ALWB,PATM,ALDP,WACC,ALH,ALV,ALRH)
      IF(ALDBL.LT.TCDSP)JCWFL=0
C
C SEE IF COLD DUCT TEMPERATURE GUESS OK
C
 350  DELT2=ABS(TACC-TCGUES)
      IF(DELT2.LT..001)GO TO 400
      TCGUES=TACC
      HCGUES=HACC
      GC TC 80
C
C SEE IF WATER BALANCE IS OK
C
C SET COUNTER J TO KICK OUT IF GREATER THAN FIFTY ITERATICNS
C
 400  J=J+1
      IF(J.GT.JMAX)JMAX=J
      IF(J.EQ.50)GC TC 500
C IF HUMIDITY EQUALS 90% THE GLESS WAS OK
      IF(ALRH.GE.90.)GO TO 60C
C IF ENTHALPY AFTER COOLING COIL AGREES WITH GUESS THAN OK
      CHK=ABS(HCGUES-HACC)
      IF(CHK.LT..001)GO TO 60C
C OTHERWISRETURN WITH NEW GUESSES
      THGUES=TAHC
      TCGUES=TACC
      HHGUES=HAHC
      HCGUES=HACC
      GO TO 80
C
C WRITE ERROR MESSAGE FOR ITERATION VIOLATION
C
 500  WRITE(6,510)
 510  FORMAT(' GREATER THAN FIFTY ITERATIONS IN DDLOOP ')
      GO TO 999
C
C NOW SUM RETURN AIR   (AFTER RETURNING CW FLAG TO 0)
C
 600  JCWFL=0
      CALL SSYCH7(TZ1,WB1,PATM,DP1,WZ1,HZ1,V1,RH1)
      CALL SSYCH7(TZ2,WB2,PATM,DP2,WZ2,HZ2,V2,RH2)
      CALL SSYCH7(TZ3,WB3,PATM,DP3,WZ3,HZ3,V3,RH3)
      CALL AIRSM(C.,0.,0.,0.,FZ1,WZ1,TZ1,HZ1,PATM,FZ2,WZ2,TZ2,
     1HZ2,FZ12,WZ12,TZ12,HZ12,0.,0.)
      CALL AIRSM(C.,0.,0.,0.,FZ12,WZ12,TZ12,HZ12,PATM,FZ3,WZ3,
     1TZ3,HZ3,FRA,WRA,TRA,HRA,C.,0.)
C
C DO MIXING BOX CALCULATION
C
```

```
      CALL MIXBC2(TRA,HRA,FRAMBX,TOA,HOA,FCAMBX,TMBSP,HMBC,FTOT,
     1TMXBX,HMXBX,FMB,OAMIN,PATM)
C
C DO FAN CALCULATION
C
      CALL SFAN2(TOFANF,FTOT,HCFANF,WF,EFF,CPAIR,TMXBX,HMXBX,FTOT)
      CALL SSYCH7(TOFANF,WBOF,PATM,DPCF,WOFANF,HOFANF,VOF,RHCFNF)
C
C REDEFINE ALL COMMON OUTPUTS
C
      COTCTF=FCCTCT
      HDTOTF=FHDTCT
      CHFNI=HCFNI
      OTFNI=TOFANI
      GUESCT=TCGUES
      GUESHT=THGUES
      GUESCH=HCGUES
      GUESHH=HHGUES
      Z1RH=RH1
      Z2RH=RH2
      Z3RH=RH3
      Z1T=TZ1
      Z1H=HZ1
      Z1FCD=FCDZ1
      Z1FHD=FHDZ1
      Z2T=TZ2
      Z2H=HZ2
      Z2FCD=FCDZ2
      Z2FHD=FHDZ2
      Z3T=TZ3
      Z3H=HZ3
      Z3FCD=FCDZ3
      Z3FHD=FHDZ3
      QHEAT=HHEAT
      QCOOL=QWAT
      STFLOW=FST
      CWFLO=FCHW
      TEMACC=TACC
      ENTACC=HACC
      RHACC=ALRH
      TEMAHC=TAHC
      ENTAHC=HAHC
      DELHC=DELT
      DELCC=DELT2
      RAWAT=WRA
      RATEM=TRA
      RAENT=HRA
      FINTOF=TOFANF
      FINHCF=HOFANF
      FNRHOF=RHCFNF
      WTLTEM=ALWT
* 999 WRITE(5,1000)TOFANI,RHCFNI,FINTCF,FNRHCF
* 1000 FORMAT(4F15.5)
*     RETURN
 999  RETURN
      END
C
C
C
C
```

```
C
C
C               SUBROUTINE ZTMXEX
C
C
C PURPOSE:
C      THIS ROUTINE CALCULATES THE NECESSARY MIXTURE OF HOT AND COLD AIR
C TO MEET THE ZONE SET POINTS
C
C INPUT:
C      QZ=HEAT LOAD ON ZONE                        BTU/HR
C      WZ=WATER LOAD ON ZONE                       LBS/HR
C      THD=HOT DUCT TEMPERATURE                    F
C      TCD=COLD DUCT TEMPERATURE                   F
C      HHD=HOT DUCT ENTHALPY                       BTU/LB
C      HCD=COLD DUCT ENTHALPY                      BTU/LB
C      TZSP=ZONE TEMPERATURE SET POINT             F
C      DLH=WATER LATENT HEAT                       BTU/LB
C      PATM=ATMOSPHERIC PRESSURE                   IN HG
C      FZTOT=TOTAL ZONE AIR FLOW                   LBS/HR
C
C OUTPUT:
C      TZOUT=ZONE TEMPERATURE OUT                  F
C      FHD=HOT DUCT AIR FLOW                       LBS/HR
C      FCD=COLD DUCT AIR FLOW                      LBS/HR
C      HZOUT=ZONE ENTHALPY OUT                     BTU/LB
C
C
C
      SUBROUTINE ZTMXBX(QZ,WZ,THD,TCD,HHD,HCD,TZSP,DLH,PATM,FZTOT,
     1TZOUT,FHD,FCD,HZOUT)
C
C USE ZEROUT TO DETERMINE PROPER FLOW OF EACH AIR STREAM
C
      REAL WORK(10)
C
C FIND AIR CONDITONS IN DUCT
C
      CALL SSYCH9(THD,WBHD,PATM,DPHD,WHD,HHD,VHD,RHHD)
      CALL SSYCH9(TCD,WBCD,PATM,DPCD,WCD,HCD,VCD,RHCD)
C
C CHECK TO SEE IF HOT AND COLD DUCT ARE THE SAME
C
      IF(ABS(THD-TCD).GE..001)GO TO 5
      IF(TCD.GE.TZSP)GO TO 7
      FHD=FZTOT
      FCD=0.
      CALL AIRSM(WZ,0.,QZ,0.,FHD,WHD,THD,HHD,PATM,FCD,WCD,TCD,HCD,
     1FZTOTA,WZT,TZOUT,HZOUT,DLH,DLH)
      IF(TZOUT.LE.TZSP)GO TO 900
 7    FCD=FZTOT
      FHD=0.
      CALL AIRSM(WZ,0.,QZ,0.,FHD,WHD,THD,HHD,PATM,FCD,WCD,TCD,HCD,
     1FZTOTA,WZT,TZOUT,HZOUT,DLH,DLH)
      GO TO 900
C
C CHECK TOTAL FLOW OF HOT DUCT
C
 5    FHD=FZTOT
      FCD=0.
```

```
      CALL AIRSM(WZ,0.,QZ,0.,FHC,WHD,THD,HHD,PATM,FCD,WCD,TCD,HCD,
     1FZTOTA,WZT,TZOUT,HZOUT,DLH,DLH)
 10   DELT1=TZOLT-TZSP
      IF(TZCUT.LE.TZSP) GO TO 900
C
C
C NOW CHECK FULL CCLD DUCT FLCW
C
      FCD=FZTOT
      FHD=0.
      CALL AIRSM(WZ,0.,QZ,0.,FHC,WHD,THD,HHD,PATM,FCD,WCD,TCD,HCD,
     1FZTOTA,WZT,TZOUT,HZOUT,DLH,DLH)
      DELT2=TZOLT-TZSP
      IF(TZOUT.GE.TZSP)GO TO 90C
C
C OTHERWISE USE ZEROUT TO FIND INTERMEDIATE SOLUTION
C
C
C INITIALIZE IFLAG AND GIVE TOLERANCE IN WORK(1)
C
 100  IFLAG=0.
      WORK(1)=.C1
      FHDHI=FZTCT
      FHDLO=0.
C
C DO ZEROUT TC FIND FLOW OF HCT ANC COLD DUCT
C
      DC 500 I=1,25
      FHD=ZEROUT(IFLAG,FHDHI,FHCLC,DELT,WORK)
      IF(FHD.EQ.FZTOT)GO TO 450
      IF(FHD.EQ.0.)GO TC 470
      FCD=FZTOT-FHD
      CALL AIRSM(WZ,0.,QZ,0.,FHD,WHD,THD,HHD,PATM,FCD,WCD,TCD,HCD,
     1FZTOTA,WZT,TZOUT,HZOUT,DLH,DLH)
      DELT=TZCUT-TZSP
      GC TC 480
 450  DELT=DELT1
      GC TC 480
 470  DELT=DELT2
 480  IF(IFLAG.EQ.1.OR.IFLAG.LT.0.)GO TO 600
 500  CCNTINUE
C
C CHECK LIMITS  (IF OUTSIDE SET CUCT AT MAXIMUM)
C
 600  IF(FCD.GT.FZTOT)GO TO 7C0
      IF(FHD.CT.FZTOT)GO TO 8C0
      GO TO 9C0
 700  FCD=FZTOT
      FHD=0.
      WRITE(5,710)
 710  FCRMAT(' ERROR IN COLC CUCT OF ZTMXBX ')
      GO TO 850
 800  FCD=0.
      FHD=FZTCT
      WRITE(5,810)
 810  FCRMAT(' ERROR IN HOT DLCT CF ZTMXBX ')
 850  CALL AIRSM(WZ,0.,QZ,0.,FHC,WHD,THD,HHD,PATM,FCD,WCD,TCC,HCD,
     1FZTOTA,WZT,TZOUT,HZOUT,DLH,DLH)
 900  RETURN
      END
```

```
C
C
C
C
C
C
C                SUBROUTINE MAXMIN
C
C
C  PURPOSE:
C
C       THIS ROUTINE WILL RETURN THE HIGHEST AND LOWEST NUMBER
C FROM A THREE NUMBER SET
C
      SUBROUTINE MAXMIN(A,B,C,AMAX,AMIN)
      IF(A.GE.B.AND.A.GE.C)GO TO 10
      IF(B.GE.A.AND.B.GE.C)GO TO 20
      AMAX=C
      IF(B.GT.A)GO TO 5
      AMIN=B
      GO TO 100
 5    AMIN=A
      GO TO 100
 10   AMAX=A
      IF(B.GT.C)GO TO 15
      AMIN=B
      GO TO 100
 15   AMIN=C
      GO TO 100
 20   AMAX=B
      IF(A.GT.C)GO TO 25
      AMIN=A
      GO TO 100
 25   AMIN=C
 100  RETURN
      END
```

```
C
C
C
C
C
C
C
C               PERIMETER SYSTEM SIMULATION
C
C
C   MAIN ROUTINE FOR ORIGINAL FEEDFORWARD CONTROL STRATEGY
C
C       THIS ROUTINE PROVIDES SYSTEM CONDITIONS AND CALLS
C       ROUTINE "PSOAT" TO SIMULATE THE ORIGINAL CONTROL STRATEGY.
C
C
C       NOTATIONS ARE THE SAME AS GIVEN IN "PSOAT"
C
C
C
C
      DIMENSION ABIAS(4),ATOUT(4),AUN(4),ARTEM(4),APT(4),AQP(4),
     1AQW(4),AQR(4),ATWO(4),APH(4),ATWI(4),
     2ARECYS(4),ARETFL(4)
      NAMELIST/ONE/RL,RW,RH,ATWI,FWI,D,AUN,RK,ATOUT,ARTEM,APT,
     1AQP,AQW,AQR,ATWO,APH,STLH,TWOALL,QSTEAM,STFLOW,RWATFL,
     2ABIAS,ARETFL,ARECYS
C
C DEFINE PARAMETERS EQUIVALENT FOR EACH FACE
C
      WRITE(5,10)
 10   FORMAT('  INPUT MULTIPLIER FOR CONDUC.  ')
      READ(5,*)AMUL
      RH=12.0
      RL=150.
      RW=25.
      D=1.0
      FWI=25000.*((D/2.0)**2)  !    5 FT/SEC IN PIPE
      RK=.01488*AMUL           !LINEAR 68F ASSUMED
      CP=1.0
      STLH=962.2
C
C SET MODE OF RECYCLE OR NO RECYCLE  MODE   NO-RECYCLE=1, RECYCLE=2
C
      MODE=2
C
C
C PARAMETERS FOR EACH FACE
C
      ABIAS(1)=0.
      ABIAS(2)=0.
      ABIAS(3)=0.
      ABIAS(4)=0.
      ARTEM(1)=66.
      ARTEM(2)=66.
      ARTEM(3)=66.
      ARTEM(4)=66.
C
C LARGE DO LOOP WITH VARYING VALUES OF AUN
C
```

```
      DO 200 J=1,6
      C=J-1
      UVAL=.05+.04*C
      WRITE(6,21)
 21   FORMAT(' '///)
      WRITE(6,22)J,UVAL
 22   FORMAT(' CASE= 'I2,5X,' U VALUE= 'F12.6///)
      AUN(1)=.05+.04*C
      AUN(2)=.05+.04*C
      AUN(3)=.05+.04*C
      AUN(4)=.05+.04*C
C
C DO LOOP WITH VARYING TEMPERATURES
C
      DO 100 I=1,13
      B=I-1
      ATOUT(1)=.001+B*5.
      ATOUT(2)=.001+B*5.
      ATOUT(3)=.001+B*5.
      ATOUT(4)=.001+B*5.
      CALL PSOAT(RL,RW,FWI,D,RK,CP,ATOUT,AUN,ARTEM,STLH,APT,
     1AQP,AQW,AQR,ATWO,APH,ATWI,ARECYS,ARETFL,ABIAS,TWOALL,QSTEAM,
     2STFLOW,RWATFL,RH,MODE)
C
C WRITE U VALUES AND OUUTSIDE AIR TEMPS.
C
      WRITE(6,43)AUN(1),AUN(2),AUN(3),AUN(4)
 43   FORMAT(' U1= 'F12.6,' U2= 'F12.6,' U3= 'F12.6,' U4= 'F12.6/)
      WRITE(6,44)ATOUT(1),ATOUT(2),ATOUT(3),ATOUT(4)
 44   FORMAT(' T1= 'F12.6,' T2= 'F12.6,' T3= 'F12.6,' T4= 'F12.6/)
      WRITE(6,ONE)
      WRITE(6,50)
 50   FORMAT(' '///)
      WRITE(6,55)
 55   FORMAT(' NEXT CASE ')
      WRITE(6,58)
 58   FORMAT(' '//)
C WRITE OUT VALUES FORM PLOTTING
      WRITE(25,97)ATOUT(1),QSTEAM,APT(1),APT(2),APT(3),APT(4),AUN(1)
 97   FORMAT(1X,7F14.4)
 100  CONTINUE
      WRITE(25,98)
 98   FORMAT(' ****** ')
 200  CONTINUE
      STOP
      END
C
C
C
C
C
C
C
C
C     SUBROUTINE PSOAT
C
C PURPOSE:
C     TO SIMULATE THE PERIMETER STRATEGY BASED ON OUTSIDE AIR TEMPERATURE
C TO DETERMINE WATER INPUT TEMFEFATURE
C     1=N,2=S,3=E,4=W
```

```
C INPUT:
C       RL,RW,FWI,D,RK,CP: SAME FCR ALL FACES AS DEFINED BY PERIMZ
C       ATOUT(1-4)         OUUTSIDE AIR TEMPERATURE           F
C       AUN(1-4)           WALL TRANSFER CCEFFICIENT          BTU/FT-HR-F
C       ARTEM(1-4)         BULK ROCM TEMPERATURE              F
C       ABIAS(1-4)         CONTRCLLER BIAS FOR DECREASING
C                          DESIRED WATER TEMP.                F
C       STLH               LATENT HEAT OF STEAM               BTU/LB
C       MODE               1=NO RECYCLE ,  2= RECYCLE
C
C OUTPUT:
C       APT(1-4)           WALL PLANE TEMPERATURE             F
C       AQP(1-4)           HEAT GAINED FROM PIPE              BTU/HR
C       AQW(1-4)           HEAT LOST THROUGH WALL             BTU/HR
C       AQR(1-4)           HEAT LOST TC ROOM                  BTU/HR
C       ATWC(1-4)          RETURN WATER TEMP.                 F
C       APH(1-4)           PIPE TRANSFER CCEF.                BTU/FT**2-HR-F
C       ATWI(1-4)          WATER TEMP. INTO ZONE              F
C       ARECYS(1-4)        RECYCLE WATER FLOW                 LBS/HR
C       ARETFL(1-4)        INDIVIDUAL RETURN WATER FLOW       LBS/HR
C       TWOALL             COMBINEC WATER RETURN TEMP         F
C       QSTEAM             STEAM LCAD                         BTU/HR
C       STFLCW             STEAM FLOW                         LBS/HR
C       RWATFL             RETURN WATER FLCW                  LBS/HR
C
C
        SUBROUTINE PSOAT(RL,RW,FWI,D,RK,CP,ATOUT,AUN,ARTEM,STLH,
       1APT,AQP,AQW,AQR,ATWO,APH,ATWI,ARECYS,ARETFL,ABIAS,TWCALL,
       2QSTEAM,STFLCW,RWATFL,RH,MODE)
C
C DIMENSION
C
        REAL WORK(10)
C
        DIMENSICN APTD(4),ATOUT(4),AUN(4),ARTEM(4),APT(4),AQP(4),
       1AQW(4),AQR(4),ATWO(4),APH(4),ATWI(4),ARECYS(4),ARETFL(4),
       2ABIAS(4)
C
C DETERMINE WATER TEMPERATURE IN
C
        DO 20 I=1,4
        IF(ATOUT(I).LT.0.)GO TO 5
        IF(ATOUT(I).GE.60.)GO TC 10
        ATWI(I)=200.0-1.667*ATOUT(I)+ABIAS(I)
        IF(MODE.EC.1) GC TO 30
        GO TO 20
 5      ATWI(I)=200.
        GO TO 20
 10     ATWI(I)=100.
 20     CCNTINUE
        GO TO 40
C
C SET CASE WITH NO RECYCLE
 30     DO 35 J=2,4
        ATWI(J)=ATWI(1)
 35     CONTINUE
C
C
C NOW CALL PERIMZ
C
```

```
 40     DO 500 K=1,4
        CALL PERIMZ(RL,RW,RH,ATWI(K),FWI,D,AUN(K),RK,ATCUT(K),ARTEM(K),
       1APT(K),ACP(K),AQW(K),ACR(K),ATWO(K),APH(K),CP)
        IF(ATWI(K).EQ.ATWI(1)) GO TO 200
C
C COMPUTE NECESSARY RECYCLE
C
        ARETFL(K)=AQP(K)/(CP*(ATWI(1)-ATWO(K)))
        ARECYS(K)=FWI-ARETFL(K)
        TCCHK=(ARETFL(K)*ATWI(1)+ARECYS(K)*ATWO(K))/FWI
        IF(ABS(TCCHK-ATWI(K)).LE..001)GO TO 500
        WRITE(6,120)TCCHK,ATWI(K)
 120    FORMAT(' CHECK TEMP: ' 2F15.6)
        GO TO 500
C
C NO RECYCLE
C
 200    ARETFL(K)=FWI
        ARECYS(K)=0.
 500    CONTINUE
C
C CONTINUUE TOTAL RETURN WATER FLOW AND ITS TEMPERATURE
C
        RWATFL=ARETFL(1)+ARETFL(2)+ARETFL(3)+ARETFL(4)
        SUM=0.
        DO 600 L=1,4
        SUM=SUM+ARETFL(L)*ATWO(L)
 600    CONTINUE
        TWOALL=SUM/RWATFL
C
C COMPUTE STEAM FLOW AND LOAD
C
        QSTEAM=RWATFL*CP*(ATWI(1)-TWOALL)
        STFLOW=QSTEAM/STLH
        RETURN
        END
C
C
C
C
C
C
C               SUBROUTINE PERIMZ
C
C       PERIMETER ZONE SIMULATOR
C
C FOR ANY ZONE GIVEN:
C
C INPUT:
C       RL=ROOM AND PIPE LENGTH                          FT
C       RW=ROOM WIDTH (PERPENDICULAR TO PIPE)            FT
C       RH=ROOM HEIGHT
C       TWI=WATER TEMPERATURE IN                         F
C       FWI=WATER FLOW                                   LBS/HR
C       D=PIPE DIAMETER                                  INCHES
C       U=WALL TRANSFER COEFFICIENT                      BTU/FT**2-HR-F
C       RK=ROOM CONDUCTIVITY                             BTU/FT-HR-F
C       TCUT=OUTSIDE AIR TEMPERATURE                     F
C       RTEM=BULK ROOM TEMPERATURE                       F
C       CP=HEAT CAPACITY OF WATER                        BTU/LB-F
```

```
C OUTPUT:
C       PT=ROOM TEMERATURE IN PIPE PLANE          F
C       CP=HEAT GAINED FROM PIPE                  BTU/HR
C       QW=HEAT LCST THROUGH WALL                 BTU/HR
C       QR=HEAT LCST INTO ROOM                    BTU/HR
C       TWO=WATER TEMPERATURE OLT                 F
C       PH=CALCULATED PIPE TRANSFER COEFF.        BTU/FT**2-HR-F
C
C
        SUBRCUTINE PERIMZ(RL,RW,RH,TWI,FWI,D,U,RK,TOUT,RTEM,PT,QP,
       1QW,QR,TWC,PH,CP)
C
C
C
C
C THE FCLLCWING EQUATIONS ARE THE ONES WHICH ARE  SOLVED
C
C THE UNKNOWNS IN THESE EQUATICNS ARE:
C
C::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::
C                PT, QP, PH, AND TWO           ::::::::::::::::::
C::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::::
C
C THESE EQUATIONS HAVE BEEN SOLVED TO A QUADRATIC EQUATION IN
C TERMS OF:     ------  PT  -------
C
C
C CALCULATE WATER TEMPERATURE CUT
C       TWO=((FWI*CP*TWI)-QP)/(FWI*CP)
C AVERAGE TEMPERATURE
C       TWG=(TWI+TWC)/2.
C WALL PLANE TEMPERATURE
C       PT=(U*TCUT*RW+RK*RTEM+((CP*RW)/(RL*RH)))/(U*RW+RK)
C
C  PIPE TRANSFER CCEF
C
C       1.90+  FOR A 2" PIPE
C       2.08+  FOR A 1" PIPE
C       1.70+  FOR A 4" PIPE
C
C       8*       MULTIPLIER FOR A FINNED SURFACE
C       (BASED ON KERN HEAT TRANSFER BOOK)
C
C       PH=(2.08+.0C5*(TWG-PT))*.25
C       PH=CP/(3.14159*(D/12.)*RL*(TWG-PT))
C
C
C CALCULATE Q THROUGH WALL
C
 40     QW=-U*(TOUT-PT)*RL*RH
C
C CALCULATE Q INTO ROOM
C
C
C  EVALUATION CONSTANTS IN TERMS OF GIVEN INFORMATION USED TC
C  SOLVE QUACRATIC FOR   ....  PT  .....
C
        FINFAC=1.
C FINFAC IS THE MULTIPLYING FACTCR FOR THE KERN HEAT EQUATION TO
C CORRECT FOR FINS AND/OR INSULATICN
```

```
      A=2.08*FINFAC
      B=.005 *FINFAC
      C=FWI*CP
      E=TWI
      F=U*TOUT*RW
      G=RK*RTEM
      O=RW/(RL*RH)
      P=U*RW
      S=RK
      R=3.14159*(D/12.)*RL
      T=(F+G)/(P+S)
      V=O/(P+S)
      W=1./C
      X=E/2.
      Y=0.5
      Z=A+B*X
      ALP=Z+B*Y*E
      BET=B*Y*W
      GAM=R*X+R*Y*E
      ZET=R*Y*W
      ETA=T/V
      EPS=1./V
      THET=ALP+BET*ETA
      PHI=BET*EPS+B
      ALAM=GAM+ZET*ETA
      SIG=ZET*EPS+R
      ASQ=SIG*PHI
      BSQ=PHI*ALAM+THET*SIG+EPS
      CSQ=THET*ALAM+ETA
C
C NOW SOLVE QUADRATIC FOR PT
C
C FIND SIGN OF SQUARE ROOT TERM
C
      TRMSQR=BSQ**2-4.*ASQ*CSQ
      IF(TRMSQR.GE.0.)GO TO 100
      WRITE(6,47)TRMSQR
 47   FORMAT(' IMAGINARY ROOTS,  TRMSQR=  ' F12.6//)
      GO TO 400
C FIND BOTH ROOTS
 100  PT1=(BSQ+TRMSQR**.5)/(2.*ASQ)
      PT2=(BSQ-TRMSQR**.5)/(2.*ASQ)
*     WRITE(6,101)PT1,PT2
* 101 FORMAT(' PT1= 'F12.6,' ,          PT2= 'F12.6/)
CC
C CHECK TO SEE WHICH ROOT IS PHYSICALLY POSSIBLE
C
      IF(PT1.LT.O.AND.PT2.LT.O.)GO TO 120
      IF(PT1.GT.O.AND.PT2.LT.O.)GO TO 140
      IF(PT1.GT.O.AND.PT2.GT.O.)GO TO 160
C  WARNING FOR ODD SOLUTION
      WRITE(6,112)PT1,PT2
 112  FORMAT(' WARNING, PT1= 'F12.6, ' PT2= 'F12.6//)
      GO TO 400
C WARNING FOR BOTH NEGATIVE
 120  WRITE(6,122)PT1,PT2
 122  FORMAT(' BOTH, PT1= 'F12.6,' AND PT2= 'F12.6, ' ARE NEGATIVE '
     1///)
      GO TO 400
C PT1 LOOK LIKE THE CORRECT SOLUTION
```

```
      CP=1.0
      STLH=962.2
C
C
C PARAMETERS FOR EACH FACE
C
      APTD(1)=66.
      APTD(2)=66.
      APTD(3)=66.
      APTD(4)=66.
      ARTEM(1)=66.
      ARTEM(2)=66.
      ARTEM(3)=66.
      ARTEM(4)=66.
C
C LARGE DO LOOP WITH VARYING VALUES OF "AUN"
C
      DC 200 J=1,6
      C=J-1
      UVAL=.05+.04*C
      WRITE(6,21)
 21   FCRMAT('  '///)
      WRITE(6,22) J,UVAL
 22   FORMAT(' CASE= 'I2,5X,' U VALUE= 'F12.6///)
      AUN(1)=.05+.04*C
      AUN(2)=.05+.04*C
      AUN(3)=.05+.04*C
      AUN(4)=.05+.04*C
C
C DO LOCP WITH VARYING TEMPERATURES
C
      DO 100 I=1,13
      B=I-1
      ATOUT(1)=.001+B*5.
      ATOUT(2)=.001+B*5.
      ATOUT(3)=.0C1+B*5.
      ATOUT(4)=.001+B*5.
      CALL PSWT(RL,RW,FWI,D,RK,CP,APTC,ATOUT,AUN,ARTEM,STLH,APT,
     1AQP,AQW,AQR,ATWO,APH,ATWI,ARECYS,ARETFL,TWOALL,CSTEAM,
     2STFLOW,RWATFL,RH,IFLAG,NUM)
      WRITE(6,43)AUN(1),AUN(2),AUN(3),AUN(4)
 43   FCRMAT(' U1= 'F12.6,' U2= 'F12.6,' U3= 'F12.6,' U4= 'F12.6/)
      WRITE(6,44)ATOUT(1),ATOLT(2),ATCUT(3),ATOUT(4)
 44   FORMAT(' T1= 'F12.6,' T2= 'F12.6,' T3= 'F12.6,' T4= 'F12.6/)
      WRITE(6,ONE)
      WRITE(6,50)
 50   FORMAT('    '///)
      WRITE(6,55)
 55   FORMAT(' NEXT CASE ')
      WRITE(6,58)
 58   FORMAT('  '//)
      WRITE(25,97)ATOUT(1),QSTEAM,APT(1),APT(2),APT(3),APT(4),AUN(1)
 97   FORMAT(1X,7F14.4)
 100  CONTINUE
      WRITE(25,98)
 98   FORMAT(' ****** ')
C
C
C WRITE CASE VALUES
C
```

```
 140    PT=PT1
        GC TC 175
C EITHER PT1 OR PT2 MAY BE CORPECT, BUT PT2 LOCKS BETTEP
 160    PT=PT2
C CALCULATE OTHER LNKNOWNS---QP,TWC,AND PH
 175    QP=EPS*PT-ETA
        TWO=E-W*QP
        TWG=(TWI+TWO)/2.
        PH=Z+B*Y*TWC-B*PT
C IF PT2 USED---OK
        IF(PT.EQ.PT2)GO TO 300
C IF PT1 USED CHECK FOR PHYSICAL VIOLATIONS
        IF(TWG.GT.PT.AND.PH.GT.C.)GC TO 300
        WRITE(6,180)
 180    FORMAT('  '//)
        WRITE(6,190)PT,QP,TWO,PH
 190    FORMAT(' VICLATTION|||||  PT= 'F12.6,' QP= 'F12.6,' TWO= '
        1F12.6,' PH= 'F12.6//)
        GC TC 400
C NOW CALCULATE C THROUGH THE WALL, AND Q INTO THE ROOM
 300    QW=-U*(TOUT-PT)*RL*RH
        QR=((-RK*(PT-RTEM))/RW)*PL*RH
 400    RETURN
        END
C
C
C
C
C
C
C
C       PERIMETER SYSTEM SIMULATICN  WITH NEW SUGGESTED STRATEGY
C
C       USES ROUTINE "PERIMZ" THAT IS GIVEN IN OLD STRATEGY LISTING
C
C PERIMETER DRIVE ROUTINE NEW STRATEGY
C
C       THIS IS THE MAIN ROUTINE FOR THE PERIMETER SIMULATION WITH
C THE SUGGESTED CCNTROL STRATEGY
C       NOTATION IS THE SAME AS GIVEN IN ROUTINE "PSWT"
C
C
        DIMENSION WPTD(4),ATOUT(4),AUN(4),ARTEM(4),APT(4),AQP(4),
        1AQW(4),AQR(4),ATWO(4),APH(4),ATWI(4),
        2ARECYS(4),ARETFL(4)
        NAMELIST/CNE/RL,RW,RH,ATWI,FWI,D,AUN,RK,ATOUT,ARTEM,APT,
        1AQP,AQW,AQR,ATWO,APH,STLH,TWOALL,QSTEAM,STFLOW,RWATFL,IFLAG,
        2NUM,ARECYS,ARETFL
C
C DEFINE PARAMETERS EQUIVALENT FCR EACH FACE
C
        WRITE(5,10)
 10     FORMAT('  INPUT MULTIPLIEF FOR CONDUC. TERM ')
        READ(5,*)AMUL
        RH=12.0
        RL=150.
        RW=25.
        D=1.0
        FWI=25000.*((D/2.0)**2)  |  5 FT/SEC IN PIPE
        RK=.01488*AMUL           |LINEAR 68F ASSUMED
```

```
      WRITE(6,180)
 180  FCRMAT('    '///)
      WRITE(6,190)
 190  FORMAT(' CASE= 'I2,5X,' U VALUE= 'F12.6///)
 200  CONTINUE
      STOP
      END
C
C
C
C
C
C
C
C     SUBROUTINE PSWT
C
C
C
C PURPOSE:
C     TO SIMULATE PERIMETER STRATEGY BASED ON WALL PLANE
C TEMPERATURE FEEDBACK
C DIMENSIONED  1=N,2=S,3=E,4=W
C     WITH 'A' AS A PREFIX
C
C INPUTS:
C     RL,RW,FWI,D,RK,CP:SAME FCR ALL FACES AS DEFINED BY PERIMZ
C     APTD(1-4)         DESIRED WALL PLANE TEMP.        F
C     ATOUT(1-4)        OUTSIDE AIR TEMPERATURE         F
C     AUN(1-4)          WALL TRANSFER CCEFF.            BTU/FT**2-HR-F
C     ARTEM(1-4)        BULK ROCM TEMPS.                F
C     STLH              LATENT HEAT OF STEAM            BTU/LB
C OUTPUT:
C     APT(1-4)          WALL PLANE TEMPERATURE          F
C     AQP(1-4)          HEAT GAINED FROM PIPE           BTU/HR
C     AQW(1-4)          HEAT LOST THROUGH WALL          BTU/HR
C     AQR(1-4)          HEAT LOST TC ROCM               BTU/HR
C     ATWO(1-4)         RETURN WATER TEMP.              F
C     APH(1-4)          PIPE TRANSFER COEF.             BTU/FT**2-HR-F
C     ATWI(1-4)         WATER TEMP. INTO ZONE           F
C     ARECYS(1-4)       RECYCLE WATER FLOW              LBS/HR
C     ARETFL(1-4)       INDIVIDUAL RETURN WATER FLOW    LBS/HR
C     TWOALL            COMBINEC WATER RETURN TEMP      F
C     QSTEAM            STEAM LCAC                      BTU/HR
C     STFLOW            STEAM FLOW                      LBS/HR
C     RWATFL            RETURN WATER FLOW               LBS/HR
C
C
      SUBROUTINE PSWT(RL,RW,FWI,D,RK,CP,APTD,ATCUT,AUN,ARTEM,
     1STLH,APT,AQP,AQW,AQR,ATWC,APH,ATWI,ARECYS,ARETFL,TWCALL,
     2QSTEAM,STFLCW,RWATFL,RH,IFLAG,NUM)
C
C DIMENSION
C
      REAL WORK(10)
      DIMENSICN APTD(4),ATOUT(4),AUN(4),ARTEM(4),APT(4),AQP(4),
     1AQW(4),ACR(4),ATWO(4),APH(4),ATWI(4),ARECYS(4),ARETFL(4)
C
C DETERMINE WATER TEMPERATURE NEEDED FOR EACH FACE
C
      DO 100 J=1,4
```

```
C
C USE ZEROUT
C
      TWATHI=210.
      TWATLO=80.
C
C INITIALIZE IFLAG AND GIVE TOLERANCE IN WORK(1)
C
      IFLAG=0.
      WORK(1)=.1
C
C DO ZEROUT TO FIND APPROP. WATER TEMP.
C
      DO 30 I=1,25
      NUM=I
      ATWI(J)=ZEROUT(IFLAG,TWATHI,TWATLO,DTEMP,WORK)
      CALL PERIMZ(RL,RW,RH,ATWI(J),FWI,D,AUN(J),RK,ATOUT(J),ARTEM(J),
     1APT(J),AQP(J),AQW(J),ACR(J),ATWO(J),APH(J),CP)
      DTEMP=APT(J)-APTD(J)
      IF(IFLAG.EQ.1.OR.IFLAG.LT.0.)GO TO 90
 30   CONTINUE
C
C IF WATER TEMP. IN IS GREATER THAN ALLOWED, SET AT MAXIMUUM OF 200
C
 90   IF(ATWI(J).LE.200.)GO TO 100
      ATWI(J)=200.
      CALL PERIMZ(RL,RW,RH,ATWI(J),FWI,D,AUN(J),RK,ATOUT(J),ARTEM(J),
     1APT(J),AQP(J),AQW(J),ACR(J),ATWO(J),APH(J),CP)
 100  CONTINUE
C
C
      ARETFL(1)=FWI
      ARECYS(1)=0.
C
C NOW DETERMINE RECYCLE FLOWS AND RETURN FLOWS
C
      DO 200 K=2,4
      ARETFL(K)=ACP(K)/(CP*(ATWI(1)-ATWO(K)))
      ARECYS(K)=FWI-ARETFL(K)
      TCCHK=(ARETFL(K)*ATWI(1)+ARECYS(K)*ATWO(K))/FWI
      IF(ABS(TCCHK-ATWI(K)).LE..001)GO TO 200
      WRITE(6,120)TCCHK,ATWI(K)
 120  FORMAT(' CHECK TEMPS: '2F15.6)
 200  CONTINUE
C
C COMPUTE TOTAL RETURN WATER AND ITS TEMPERATURE
C
      RWATFL=ARETFL(1)+ARETFL(2)+ARETFL(3)+ARETFL(4)
      SUM=0.
      DO 250 L=1,4
      SUM=SUM+ARETFL(L)*ATWO(L)
 250  CONTINUE
      TWOALL=SUM/RWATFL
C
C
C COMPUTE STEAM FLOW AND LOAD
C
      QSTEAM=RWATFL*CP*(ATWI(1)-TWOALL)
      STFLOW=QSTEAM/STLH
      RETURN
      END
```

VITA

Winston Bernard Rawlston was born in Bristol, Tennessee on June 17, 1956. He attended elementary schools in that area and graduated from Sullivan Central High School in June 1974. The following September he entered The University of Tennessee, Knoxville, and in June 1978 he received a Bachelor of Science Degree in Chemical Engineering.

The author married Miss Susan Elizabeth Smith of Knoxville, Tennessee on August 9, 1978. That Fall he began study toward a Master's degree at The University of Tennessee, Knoxville. He received the Master of Science with a major in Chemical Engineering in June 1981.

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