MODEL AND PERFORMANCE CHARACTERISTICS OF A COMMERCIALLY-SIZED HYBRID AIR CONDITIONING SYSTEM WHICH UTILIZES A ROTARY DESICCANT DEHUMIDIFIER

by

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ABSTRACT

Air conditioning operating costs for commercial buildings can be reduced significantly by using a desiccant dehumidifier and an indirect evaporative cooler in conjunction with a vapor compression machine. A hybrid air conditioning system of this sort is presented, and models for the individual components are developed.

Performance maps are developed which enable the operating characteristics of the hybrid system to be analyzed. Effects of ambient conditions and the size and type of building loads on system performance are studied using this concept.

Four methods for supplying the desiccant regeneration thermal energy are presented and examined. Long term system performance of each of these methods are presented for a variety of U.S. locations.

The hybrid system which reclaims condenser thermal energy for regeneration is shown to be the most economical of the systems studied. Its use can provide significant annual fuel savings compared to a conventional vapor compression system.

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NOMENCLATURE

A_c	collector area
AFS	annual fuel savings
AMB	ambient state
C_A	collector area-dependent cost
C_E	cost of equipment independent of collector area
cfm	cubic feet per minute
COND	condenser
COP	coefficient of performance
CP	specific heat of air
CPR	compressor power ratio
D	actual dehumidifier outlet state
D*	ideal dehumidifier outlet state
DEH	dehumidifier
EVAP	evaporator
F_i	characteristic potentials for heat and mass transfer
FLR	fraction of full load air conditioner capacity
F'	collector efficiency factor
GJ	gigajoule
h	enthalpy of moist air
HFG	latent heat of vaporization of water
HXR	indirect evaporative cooler
kwh	kilowatt-hour

NOMENCLATURE (continued)

LCS	life cycle savings
MB	moisture capacitance of building
MBASE	rated air flow rate for fan calculations
MLIQ	amount of water making up the latent load
MREG	regeneration air mass flow rate
MSYS	system air mass flow rate
MVENT	required ventilation air mass flow rate
MWh	megawatt-hour
N_s	number of solar collectors in series
P	pressure, or dehumidifier process inlet state
Q_{AUX}	auxiliary energy
QAUX	auxiliary energy
QCOND	heat rejected by condenser
QEVAP	heat removed from process stream at evaporator
QFAN	heat generated by fan
QLAT	latent cooling load
QLITES	heat generated by lights
QSENS	sensible cooling load
QTOT	total cooling load
R	dehumidifier regeneration stream inlet state
SHR	sensible heat ratio
t	time
T	temperature

NOMENCLATURE (continued)

T^*_{EVAP}	wet bulb temperature of air entering evaporator
T_{FAN}	air temperature leaving fan
T_{REG}	required regeneration temperature
T_{wb}	wet bulb temperature of ambient air
TCOND	air temperature leaving condenser
TIEC	air temperature leaving indirect evaporative cooler
TON	nominal cooling capacity of air conditioner, in tons
U_L	collector loss coefficient
VAV	variable air volume
W_{COMP}	power consumption of vapor compression unit
W_{FAN}	power consumption of fan
W_{SHAFT}	fan shaft horsepower
W_{STATIC}	fan static horsepower
W_{TOTAL}	fan total horsepower (W_{FAN})
w	humidity ratio
w_{MAX}	maximum allowable building humidity ratio
w_{SAT}	saturation humidity ratio at a given temperature
w_{SMAX}	maximum allowable supply humidity ratio
α	absorbtance of collector plate
ε_{F_i}	dehumidifier effectivenesses
ε_{IEC}	indirect evaporative cooler effectiveness
$\eta_{f,s}$	static fan efficiency
$\eta_{d,m}$	fan drive-motor efficiency

NOMENCLATURE (continued)

ρ	density of air
τ	transmittance of collector covers

<u>Subscripts</u>

A	ambient
i	individual components
IN	inlet
OUT	outlet
R	room (building)
RA	return air
RI	initial room (building)
S	supply

CHAPTER 1 INTRODUCTION

Webster's Third New International Dictionary (1971) defines the word 'hybrid' as, "Having characteristics resulting from the blending of two diverse cultures." When applied to commercial air conditioning, the two diverse cultures of a hybrid system refer to the vapor compression cooling systems in use today and the new desiccant cooling systems being proposed for residential use. A hybrid air conditioning system combines a desiccant dehumidifier for latent cooling with a vapor compression machine for sensible cooling. This work will study the features of one such hybrid cooling system for use in conditioning small commercial buildings. As it will study only one basic system configuration it cannot be, and is not intended to be, an all-encompassing assessment of commercial hybrid systems in general. Rather, analysis of the operating characteristics of the system developed here will hopefully be used as a cornerstone for future hybrid studies.

1.1 Air Conditioning of Commercial Buildings

Commercial buildings often have significant heat generation within their structure, due to people, lights and other heat-generating equipment. Due to the tight construction of these buildings, there is not enough infiltration of outside air to remove this internally generated heat, so it has become necessary to supply virtually all commercial buildings with some type of air conditioning system.

Small commercial buildings typically have vapor compression air conditioning systems for their cooling needs. The mechanical refrigeration units in these air conditioning systems use a significant amount of electrical energy, a high grade form of energy, to provide this cooling. In addition to charging for the use of this electricity, the electric utility companies also charge a 'demand' rate to commercial users based on their peak electrical demand (use). In the study of a small commercial building, Spielvogel [1] showed that its peak electrical energy consumption occurs during peak air conditioning hours. In effect, what this means is that commercial electricity users pay twice for their air conditioning electrical energy; they pay for the electricity consumed (as we all do), and they pay higher demand charges because the air conditioning increases their peak electric demand. For this reason, there is great interest in developing a commercial-sized air conditioning system which is less electrical energy intensive.

1.2 Desiccant Air Conditioning of Commercial Buildings

In the mid-1960's, Dunkle [2] proposed a method of airconditioning which was driven principally by solar, rather than electrical, energy. His system uses a desiccant for dehumidification along with evaporative coolers and heat exchangers for sensible cooling. This concept has spawned much research into developing desiccant air conditioning systems for residential use [3, 4, 5, 6]. To date, however, very little has been done to apply the desiccant concept to commercial-sized buildings. Close and Sheridan [7] extended the use of a residential desiccant system to meet commercial-sized cooling loads. Sheridan and Mitchell [8] examined the use of a commercial-sized hybrid solar desiccant cooling system in Australian climates. Their study showed great promise for an air conditioning system which combines the present technology of a vapor compression system with the advanced concept of desiccant dehumidification.

1.3 Objective

The objective of this work is to evaluate the potential of a commercial hybrid air conditioning system similar to the one studied by Sheridan and Mitchell. The evaluation will involve four major steps:

1. Development of a computer model for the hybrid air conditioning system applicable to small commercial buildings. This entails:
    a) development of models for the individual system components, and
    b) development of a reasonable control strategy for system operation.
2. Evaluation of various sources of thermal energy, including solar collectors, for regeneration of the desiccant dehumidifier.

3. Examination of the operating characteristics of the hybrid system. In particular, studies will focus on the system response to the building cooling load, the individual component performances and variations in the ambient conditions.

4. Simulation of the hybrid system long-term performance for various U.S. cities representing different climates and load conditions.

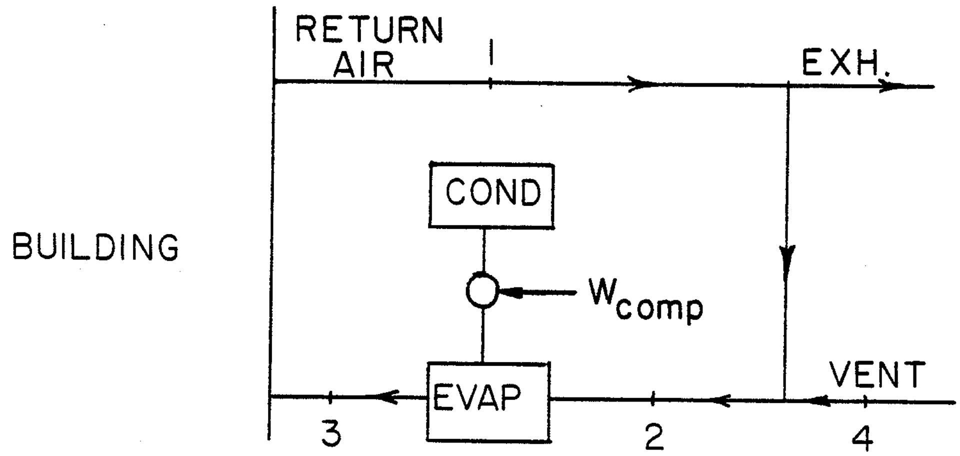
Chapter 2 presents the hybrid system which is to be evaluated. In Chapter 3, the component models which make up the system are given, and the system control strategy is explained. Chapter 4 studies the operating characteristics of the hybrid system in response to various external factors. Chapter 5 presents three alternative methods for regenerating the desiccant in the dehumidifier, and Chapter 6 gives the results of system simulations for the hybrid system in various U.S. locations. The work is concluded in Chapter 7 with an assessment of the hybrid system as a viable air conditioning option.

CHAPTER 2 THE HYBRID AIR CONDITIONING SYSTEM

The development of a hybrid cooling system requires a knowledge of the air conditioning systems whose characteristics it shares. In this chapter, the operation of a vapor compression system and two desiccant air conditioning systems--the ventilation and recirculation cycles--are detailed. A hybrid cooling system is then presented which is a combination of the three systems mentioned above. A brief description of its operation is given to make fully clear the system development in Chapter 3.

2.1 Standard Vapor Compression Air Conditioning System

A schematic drawing and psychrometric diagram of a typical vapor compression air conditioning system is shown in Figure 2.1. The system consists of one primary component, the mechanical refrigeration (vapor compression) unit, which provides both the sensible and the latent cooling. Return air from the building zones at state 1 is mixed with fresh air at state 4, which produces air at state 2 entering the cooling coil (evaporator) of the vapor compression unit. Moisture from this air condenses on the cold evaporator tubes, dehumidifying the air as it is cooled to the supply air temperature of state 3. The air is then delivered back to the conditioned space (building) and follows the process path from state 3 to state 1, called the load line, as it absorbs the building sensible and latent heat gains.



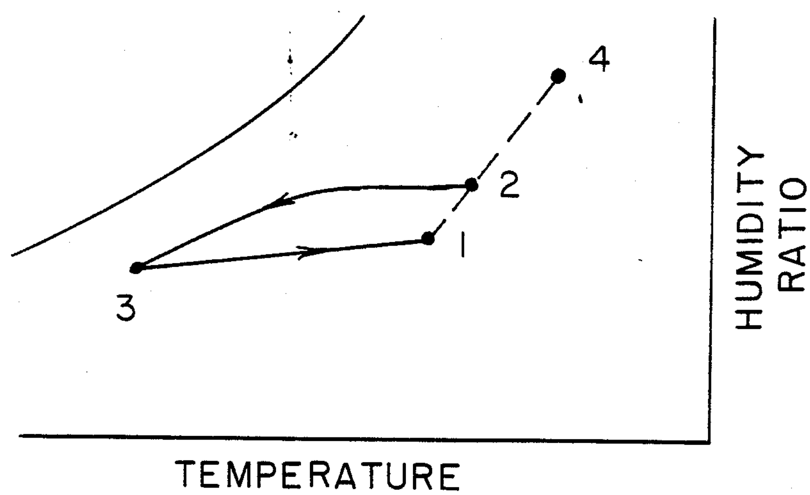


Figure 2.1 Conventional vapor compression air conditioning system

The major disadvantage of this system is that it is electrical energy intensive. The entire cooling load is handled by a single machine which is driven exclusively with electrical energy (W_{COMP} in Figure 2.1). As mentioned in Chapter 1, a premium price is paid for electricity used in the commercial sector.

Another disadvantage of this system is that the building temperature and humidity level cannot both be regulated. Typically, the building temperature is maintained at some acceptable level by turning on the vapor compression unit when the building becomes too hot. Because the cooling and dehumidifying processes occur simultaneously and are not independent, latent cooling is available only when sensible cooling is needed. The amount of latent cooling available is dependent on the cooling coil inlet state and the length of time the refrigeration machine is on, rather than how much dehumidification the building needs. The humidity level floats uncontrolled. Normally this is not a problem, however, because sufficient dehumidification occurs during normal system operation to handle typical latent loads, and the building humidity level floats within an acceptable range.

If strict humidity control is needed, the vapor compression system must incorporate a reheat feature. The coil supply temperature is set at a lower temperature, allowing sufficient condensation from the air. At this point the air is too cold to use in the building, and must be reheated using an auxiliary source of energy. The increased cooling and reheat required for humidity control increases the air conditioning energy consumption considerably.

2.2 Desiccant Cooling Systems

Desiccant cooling systems were designed to reduce the operating expenses of residential air conditioning. The basic concept of these systems is to replace the vapor compression air conditioner with a desiccant dehumidifier for latent cooling and heat exchangers and evaporative coolers for sensible cooling. These components require much less electrical energy than the vapor compression unit. The dehumidifier does, however, require thermal energy for regeneration of the desiccant.

Two of the more well-studied desiccant systems are the ventilation and recirculation cycles. Figure 2.2 contains a schematic and psychrometric diagram of the ventilation cycle; Figure 2.3 contains the same for the recirculation cycle. As their names imply, the ventilation cycle uses outside air for the process air stream, while the recirculation cycle reuses the room air.

Both the ventilation and recirculation cycles combine a dehumidifier with regenerative evaporative cooling. The dehumidification process resembles a constant enthalpy process. The airstream is heated as it is dried out. Actually, the dehumidifier does not reduce the total cooling load; it replaces the latent load with additional sensible load. However, the hot, dry air leaving the dehumidifier can be sensibly cooled in a rotary heat exchanger (regenerator) using evaporatively cooled air as the heat sink. This cooling is essentially free (excluding power to turn the rotary heat exchanger)

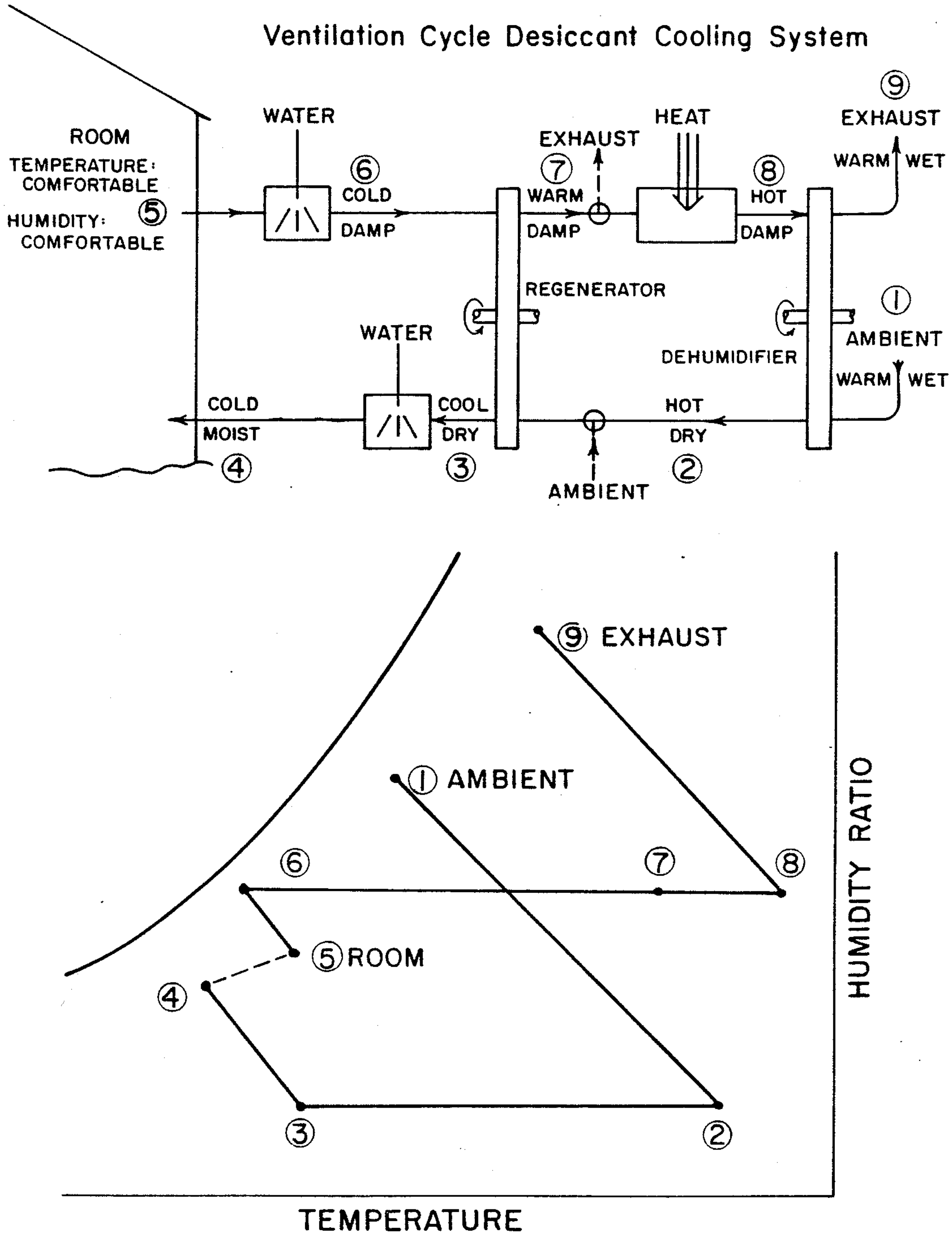


Figure 2.2 Ventilation cycle desiccant cooling system (from ref [9])

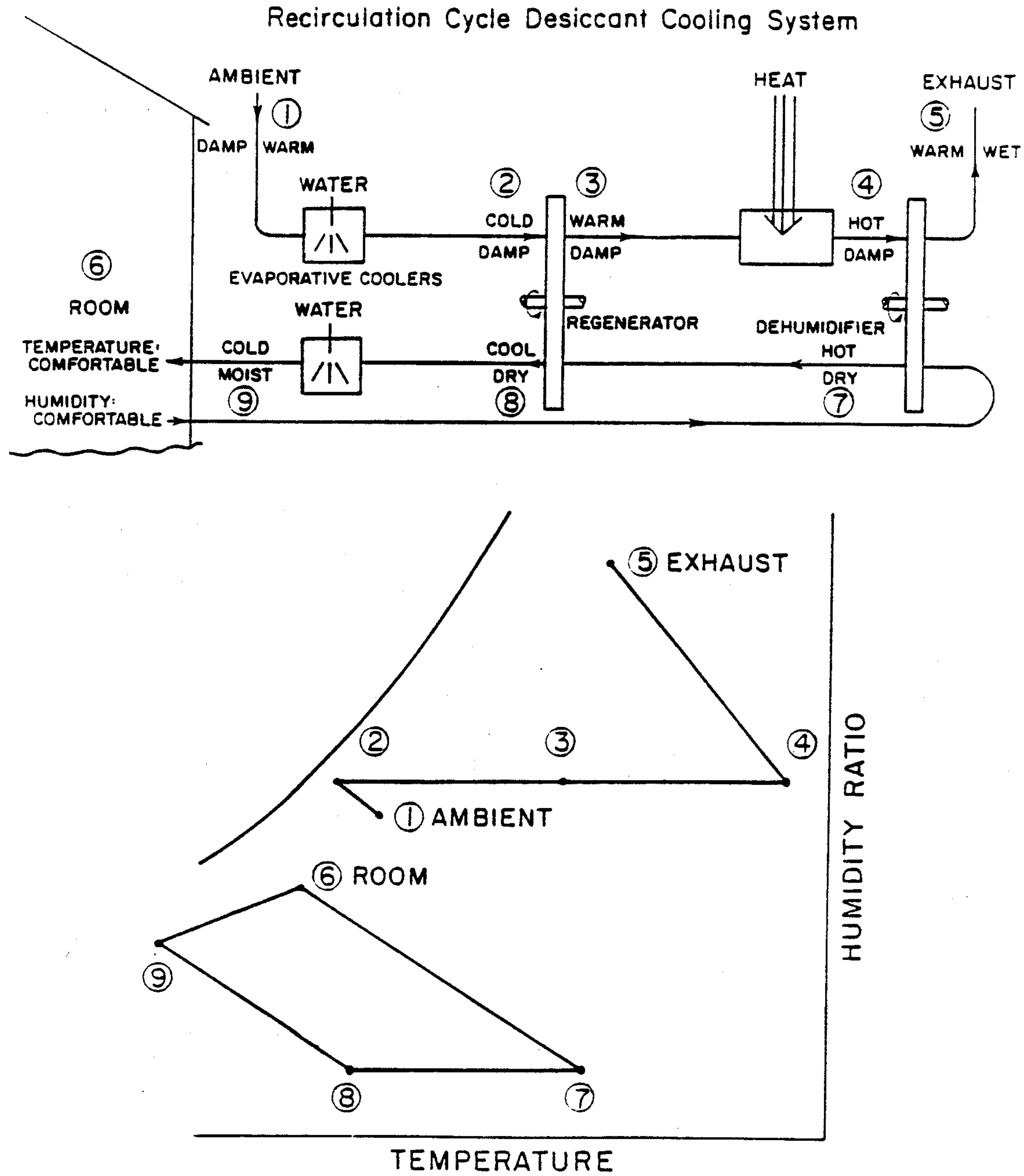


Figure 2.3 Recirculation cycle desiccant cooling system (from ref [9])

and can be further increased by evaporatively cooling this cool air leaving the regenerator. Both cycles use this principle to obtain low cost air conditioning. They are very similar, differing only in their source of the primary and secondary air streams. A more complete discussion of these two cycles is given in [9].

2.3 Hybrid Air Conditioning Systems (General)

Hybrid air conditioning systems have been proposed as a means of reducing the electrical energy consumption of commercial air conditioning systems. This is done by combining the features of a vapor compression system with those of the ventilation and recirculation cycle desiccant systems. As mechanical refrigeration is the primary source of electricity usage in a conventional vapor compression system, hybrid systems are designed to reduce the amount of cooling which is required of the vapor compression unit, thereby reducing the total electrical energy consumption.

Hybrid systems utilize a dehumidifier to handle the latent portion of the cooling load, and one or more heat exchangers and evaporative coolers to provide a portion of the sensible cooling. Except for small parasitic power requirements, these components use no electrical energy. This leaves only a fraction of the sensible cooling load (no latent load) to be handled by the vapor compression unit.

Along with the primary advantage of reduced electrical consumption, hybrid systems also offer both temperature and humidity control. This is possible because the sensible and latent cooling are handled independently. Regulation of the desiccant dehumidifier will permit humidity control, while regulation of the vapor compression machine allows temperature control.

As stated earlier, the dehumidifier needs no electrical energy for operation except for small parasitic power requirements. However, the desiccant cannot adsorb moisture from the air indefinitely without being regenerated. Regeneration is the addition of thermal energy, which is low grade energy, to the desiccant for the purpose of driving off the moisture it holds. The point to be made here is that hybrid systems offer a tradeoff; electrical energy consumption is reduced at the expense of using thermal energy. If a source of thermal energy is not available, it must be purchased.

2.4 A Hybrid Air Conditioning System Presented

In their study, Sheridan and Mitchell [8] examined a hybrid cooling system which contains a desiccant dehumidifier, an indirect evaporative cooler and a vapor compression unit. The work presented here will further develop this particular system, as it has shown tremendous potential in Australian climates.

A simplified schematic of the commercial hybrid air conditioning system is shown in Figure 2.4. Fresh outside air at state 6 is mixed with the building return air at state 1 to produce air at state 2. This mixing process allows the system to operate in either the ventilation or the recirculation mode, depending on the quantity of

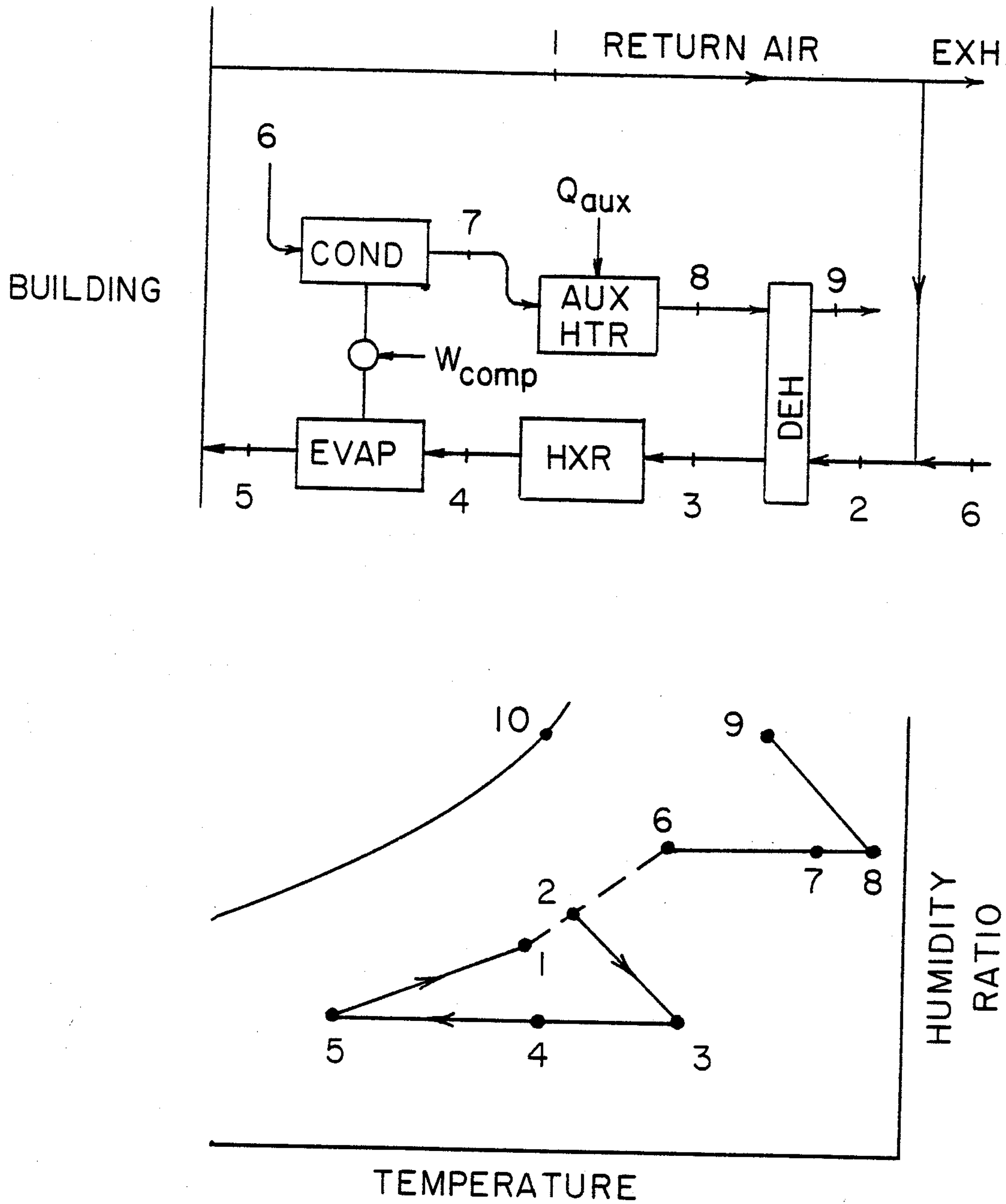


Figure 2.4 Simplified schematic of hybrid air conditioning system

outside air being brought in. The desiccant dehumidifier (DEH) heats and dehumidifies the air to state 3. An indirect evaporative cooler (HXR) uses the wet bulb temperature of the ambient air, state 10, as a heat sink to sensibly cool the process air stream to state 4. Additional sensible cooling to the building supply air condition, state 5, is accomplished with a vapor compression unit. The air is delivered to the conditioned space and follows the process path from state 5 to state 1, called the load line, as it picks up the building sensible and latent heat.

Moisture in the desiccant, adsorbed from the process air stream in going from state 2 to state 3, must be driven off for continued operation. To accomplish this, thermal energy is added to ambient air, reaching a temperature at state 8 which will enable the desiccant to be regenerated. A portion of this thermal energy is supplied by the vapor compression condenser, heating the air to state 7. Additional thermal energy to reach the regeneration temperature at state 8 is supplied by an auxiliary natural gas heater. The regeneration stream is cooled and humidified to state 9 as it passes through the desiccant and picks up moisture.

The location and effect of the system fans have been omitted from the discussion and diagram of the system so as to make the basic operation more clear. A complete detailed diagram of the hybrid system will be presented in the final section of Chapter 3. Chapter 5 will present some alternative methods for supplying regeneration energy to the desiccant which involve solar collectors.

CHAPTER 3 HYBRID SYSTEM COMPONENT MODELS

In this chapter, the operation and control of the hybrid system is explained fully. Each of the system components is examined in detail. Simple, easily-calculated models representing the actual complex heat and mass transfer processes involved are presented for each of the components. The chapter concludes with a schematic and psychrometric diagram of the complete hybrid system.

3.1 Building Controller

The primary function of the building controller is to determine the desired supply air temperature and humidity ratio (Fig. 2.4, state 5) to best meet the building load while minimizing the system energy required. It is assumed that the controller 'knows' the building sensible and latent loads during any time period.

3.1.1 Supply Temperature and Mass Flow Rate

A general rule of thumb states that air flow in a building should not exceed 400 cfm per ton of cooling required. This establishes a minimum enthalpy difference beyween the supply and room air of ~ 15 kJ/kg. That is, $h_R - h_S \geq 15$ kJ/kg. With this minimum enthalpy criterion, the supply air temperature, T_S, and the system flow rate, MSYS, are determined from energy balances as,

$$T_S = T_R - \frac{(h_R - h_S)\ \text{QSENS}}{\text{CP}\ (\text{QSENS} + \text{QLAT})} \tag{3.1}$$

$$MSYS = \frac{QSENS}{CP\ (T_R - T_S)} \tag{3.2}$$

where

T = air temperature, °C

h = air enthalpy, kJ/kg

QSENS = building sensible load, kJ/hr

QLAT = building latent load, kJ/hr

CP = air specific heat, kJ/kg-°C

and subscripts S = supply

R = room

If the calculated value of MSYS is less than the fresh air ventilation requirement, MVENT, then MSYS is set equal to MVENT and the supply air temperature is recalculated[1] as,

$$T_S = T_R - \frac{QSENS}{MSYS \times CP} \tag{3.3}$$

3.1.2 Supply Humidity Ratio

A mass balance on the building yields the following differential equation for the time rate of change of the room (building) humidity ratio:

$$\frac{dw_R}{dt} + \frac{MSYS}{MB} w_R = \frac{MLIQ + w_S \times MSYS}{MB} \tag{3.4}$$

---

[1]The 400 cfm per ton criterion is no longer in effect.

where

w_R = room humidity ratio, kg/kg

MSYS = system flow rate, kg/hr

MB = building moisture capacitance, kg

w_S = supply humidity ratio, kg/kg

MLIQ = QLAT/HFG

QLAT = building latent load, kJ/hr

HFG = latent heat of vaporization of water, kJ/kg

For an initial room humidity ratio, w_{RI}, the solution to the differential equation is

$$w_R = \frac{\text{MLIQ} + w_S \times \text{MSYS}}{\text{MSYS}} + \left[w_{RI} - \frac{\text{MLIQ} + w_S \times \text{MSYS}}{\text{MSYS}} \right] e^{-\frac{\text{MSYS} \times t}{\text{MB}}} \tag{3.5}$$

There is a maximum permissible room humidity level, w_{MAX}, specified by ASHRAE [10] for maintaining comfort. Substituting w_{MAX} for w_R, equation 3.5 can be solved explicitly for the maximum allowable supply humidity ratio, w_{SMAX}. As this is the humidity ratio the controller will 'tell' the dehumidifier to supply, this ensures that the air conditioning system does not dehumidity any more than is absolutely necessary to maintain occupant comfort (per ASHRAE [10]).

If the ambient humidity ratio, w_A, is less than w_{SMAX}, the controller instructs the economizer to supply 100% outside air. This means the supply humidity ratio is w_A instead of w_{SMAX}, and the corresponding room humidity ratio (Eq. 3.5) will be less than w_{MAX}. Under these conditions, no dehumidification of the supply air is needed.

It is possible under high sensible loads for the supply condition (T_S, w_S) determined by the controller to lie to the left of the saturation line on a psychrometric chart. These air states do not physically exist. A curve fit for w_S (kg/kg) as a function of T_S (°C) at the saturation line, in the region with which we are concerned, was developed as:

$$w_{SAT} = 8.087 \times 10^{-5} \, T_S^{1.6005} + 4.377 \times 10^{-3} \tag{3.6}$$

If w_S is greater than w_{SAT}, we have a condition as described above where (T_S, w_S) lies to the left of the saturation line. In this situation, the controller sets w_S equal to w_{SAT}, which is less than w_S.

3.2 Return Air Vents

Air which leaves the building zones is passed across the room lighting fixtures, removing a portion of the lighting load before it is felt by the occupants. Anywhere from 30% to 80% of the lighting load is typically removed by using return air vents along the lighting fixtures [11]. The building modeled here is assumed to remove 40% of the lighting load. The return air temperature, T_{RA}, is:

$$T_{RA} = T_R + \frac{.40 \text{ QLITES}}{\text{MSYS} \times \text{CP}} \tag{3.7}$$

where

T_R = room temperature, °C

QLITES = total lighting load, kJ/hr

MSYS = system flow rate, kg/hr

CP = air specific heat, kJ/kg-°C

3.3 Economizer

The economizer governs the ventilation and recirculation modes of operation. That is, it regulates the amount of fresh outside air to be introduced into the building. Figure 3.1 illustrates the economizer control scheme. If the outside enthalpy, h_A, is greater than the return air enthalpy, h_{RA}, the system uses only the minimum required outside air and recirculates the return air (recirculation mode). If the outside enthalpy is less than that of the return air, 100% of the building air requirement is brought in from the outside and the return air is exhausted (ventilation mode). Outdoor air below the supply enthalpy, h_S, is mixed in the proper quantity with return air to deliver supply air at h_S. This is a free cooling mode. Beyond some point, the outside air is too cold, and mixing with return air will require additional heating. Our system ignores this condition.

3.4 Rotary Desiccant Dehumidifier

The dehumidifier used in the hybrid system is a rotary type heat and mass exchanger containing nominal silica gel as the desic-

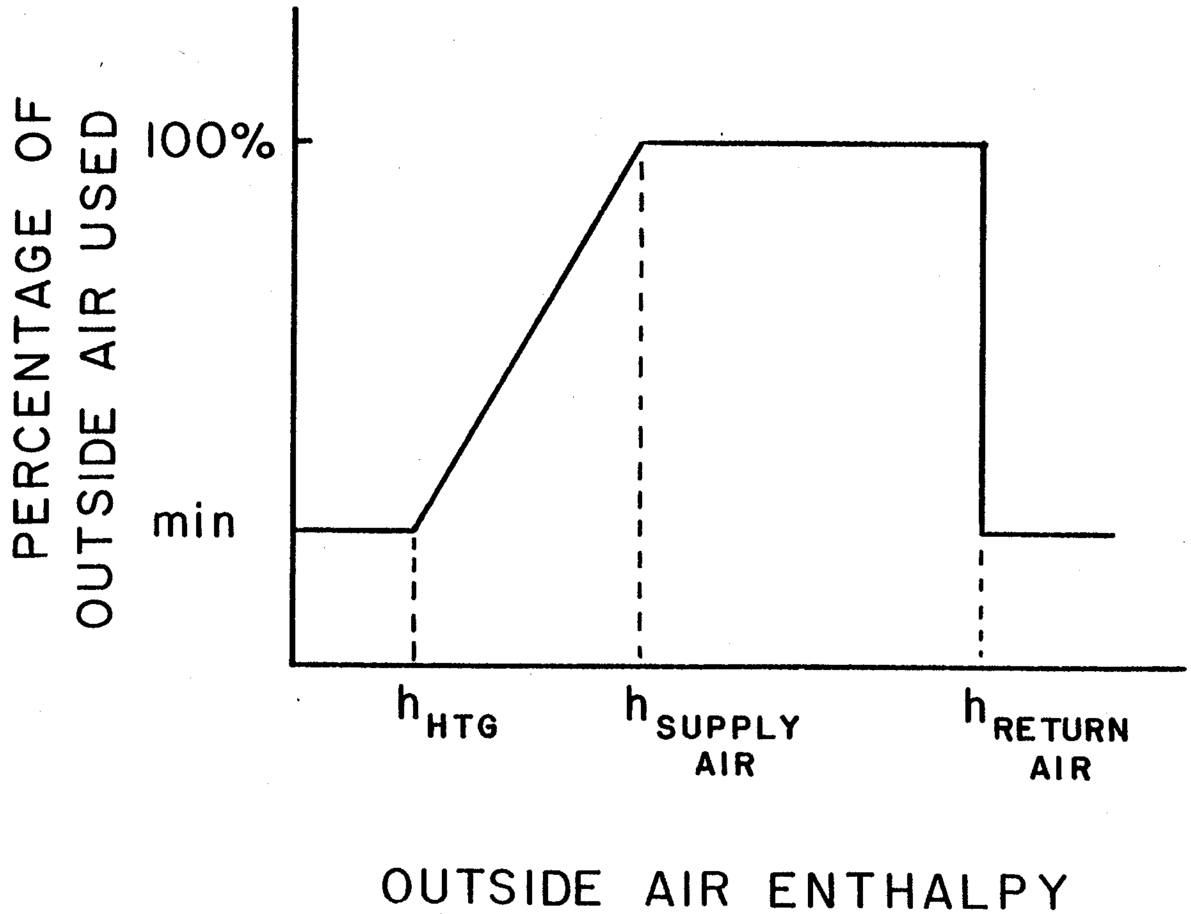


Figure 3.1 Economizer control scheme

cant. Its performance is modeled using the non-linear analogy theory proposed by Banks [12]. This method assumes that the mathematical description of the heat and mass transfer taking place within the rotary dehumidifier is analogous to that of heat transfer only in a rotary heat exchanger.

Potential lines, denoted F_1 and F_2, describe the ideal process path for the air stream. Referring to Figure 3.2a, where P is the process stream inlet state and R is the regeneration stream inlet state, the ideal process stream outlet state is at the intersection of the process stream F_1 line and the regeneration stream F_2 line and is denoted D*. Jurinak [9] has developed three-parameter curve fits to the F_1 and F_2 potentials for nominal silica gel. They are functions of the air stream temperature (°K) and humidity ratio (kg/kg) given by:

$$F_1 = \frac{-2865}{T^{1.490}} + 4.244w^{.8624} \tag{3.8}$$

$$F_2 = \frac{T^{1.490}}{6360} - 1.127w^{.07969} \tag{3.9}$$

Two effectivenesses of the F_1 and F_2 potentials, ε_{F_1} and ε_{F_2}, determine the actual dehumidifier outlet state. Banks [12] defines these effectivenesses as:

$$\varepsilon_{F_i} = \frac{F_{iD} - F_{iP}}{F_{iR} - F_{iP}}, \quad i = 1,2 \tag{3.10}$$

where D is the actual outlet state. Figure 3.2b illustrates this effectiveness concept. Note that for the ideal dehumidifier,

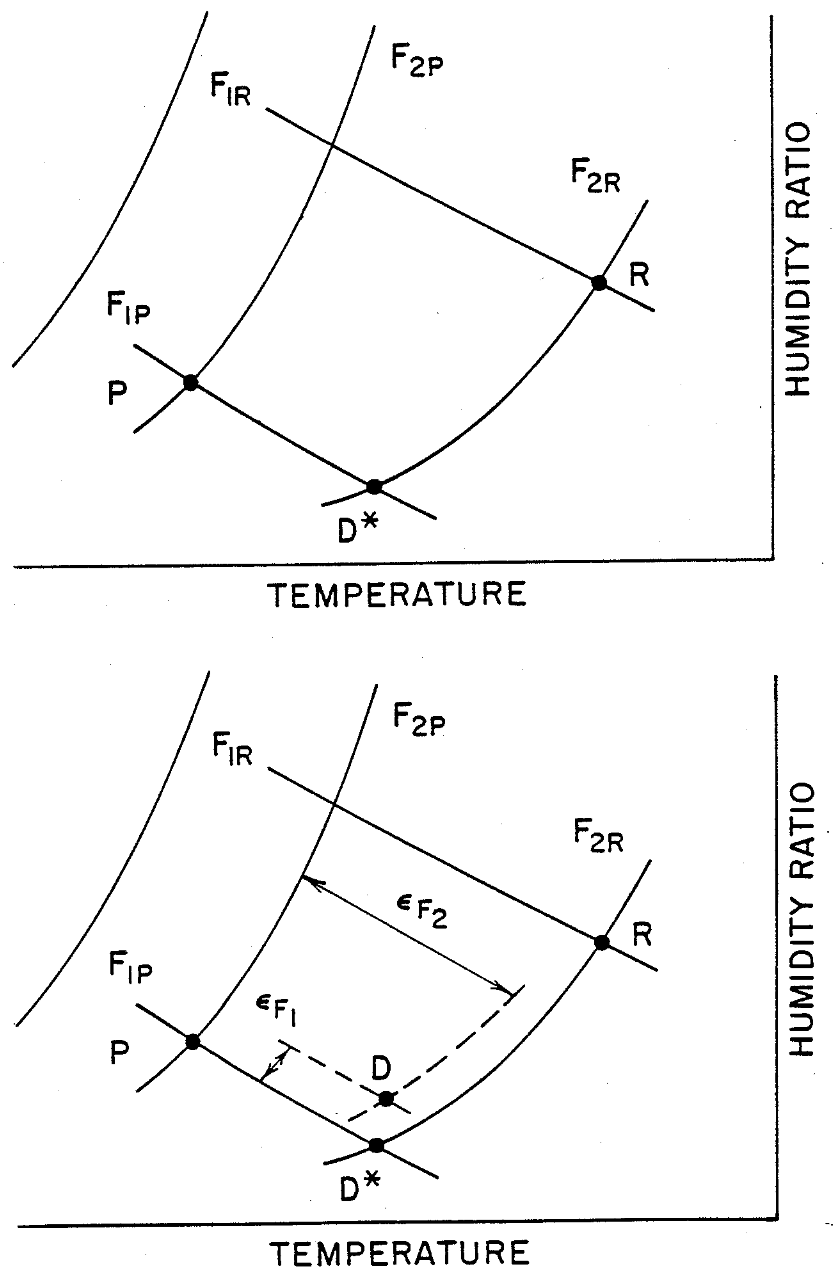


Figure 3.2 a) Ideal dehumidifier process paths
b) F_i effectivenesses for actual dehumidifier process (adapted from ref [20])

$\varepsilon_{F_1} = 0.0$ and $\varepsilon_{F_2} = 1.0$.

The dehumidifier is modeled as an infinite capacity machine. This implies that the dehumidifier outlet state D is not affected by the mass flow rate of the process or regenerating streams (provided, of course, that reasonable flows are chosen). Unlike the residential desiccant systems, the hybrid system varies the regeneration temperature to provide only the minimum amount of latent cooling required, rather than operate at a fixed regeneration temperature. The advantage of this is that a minimum of auxiliary thermal energy for regeneration is used. The disadvantage is that at any time, the regeneration temperature is not known. Equations 3.8, 3.9 and 3.10 must be solved implicitly using Newton's method to determine both the process outlet temperature at state D and the regeneration temperature. The process outlet humidity ratio is known from the building controller (w_S) and the regeneration humidity ratio is the same as the ambient.

3.5 Fan

The fan model calculates the power consumed by each fan in the system and the amount of heat which is added to the building because of the fan motors. It is assumed that the entire fan heating load enters the air handling ducts directly, for ease of calculation.

The total system static pressure drop is needed for computations, and is calculated as,

$$\Delta P_{STATIC} = \Sigma(\Delta P_i) * \left(\frac{MSYS}{MBASE}\right)^2 \tag{3.11}$$

where

$\Sigma(\Delta P_i)$ = sum of the duct and component pressure drop (Pa) at a reference flowrate, MBASE

MSYS = system air flow rate, kg/hr

The ideal static fan power is

$$W_{STATIC} = MSYS * \Delta P_{STATIC}/\rho \tag{3.12}$$

where

ρ = air density, kg/m^3

The shaft power is defined as,

$$W_{SHAFT} = W_{STATIC}/\eta_{f,s} \tag{3.13}$$

where

$\eta_{f,s}$ = static fan efficiency

Tht total power required is then,

$$W_{TOTAL} = W_{SHAFT}/\eta_{d,m} \tag{3.14}$$

where

$\eta_{d,m}$ = drive-motor efficiency

Finally, using a first law energy balance, the heat added to the airstream is,

$$QFAN = W_{TOTAL} - W_{STATIC} \tag{3.15}$$

$$= (\frac{1}{\eta_{f,s}\ \eta_{d,m}} - 1) * W_{STATIC} \tag{3.16}$$

The outlet temperature of the fan is then calculated as,

$$T_{FAN} = T_{IN} + \frac{QFAN}{MSYS \times CP} \tag{3.17}$$

3.6 Indirect Evaporative Cooler

The indirect evaporative cooler is modeled after a plate type crossflow sensible heat exchanger described by Pescod [13]. Water is sprayed into the secondary circuit passages to evaporatively cool an ambient air stream. Sensible heat transfer occurs between the warm process air from the fan and this wet bulb temperature heat sink. No moisture transfer occurs between the two air streams. Pescod defines an overall effectiveness for rating the cooler performance as

$$\varepsilon_{IEC} = \frac{T_{IN} - T_{OUT}}{T_{IN} - T_{wb}} \tag{3.18}$$

where T_{IN} is the process stream inlet temperature, T_{OUT} is the process stream outlet temperature and T_{wb} is the wet bulb temperature of the secondary stream.

3.7 Vapor Compression Unit

The vapor compression performance is based on published data from the Trane Company and Carrier Corporation. A general model for the coefficient of performance (COP) resulted from a non-linear regression analysis of Trane 20-70 ton rooftop air conditioner data at standard evaporator and condenser inlet conditions [14]. The resulting model was modified to account for non-standard operating conditions using data for the Carrier 38HQ Weathermaster III heat pump [15]. The model is given by equation 3.19, found below.

$$COP = 3.68 + .162\ FLR * TON * e^{-.183FLR \times TON} - .753FLR - .0073\ TON - .03998\ (T_A - T^*_{EVAP} - 15) \tag{3.19}$$

$$FLR = \frac{QEVAP}{TON \times 12000 \times 1.0548} \tag{3.20}$$

$$QEVAP = MSYS * CP\ (TIEC - T_S) \tag{3.21}$$

where

FLR = ratio of the cooling load to nominal cooling capacity

QEVAP = cooling load, kJ/hr

TON = nominal cooling capacity, tons

T_A = air temperature entering condenser (ambient), °C

T^*_{EVAP} = wet bulb temperature of air entering evaporator, °C

MSYS = system flow rate, kg/hr

CP = air specific heat, kJ/kg-°C

TIEC = air temperature entering evaporator (from IEC), °C

T_S = supply air temperature, °C

Equation 3.19 is valid for machines between 20 and 70 tons in size.

Also, it is only valid for a fractional load ratio (FLR) between 25% and 100%. However, errors in the COP prediction below FLR = .25 probably will not result in a significant energy use error, as the loads under these conditions are small. Under typical operating conditions, equation 3.19 predicts values for the COP in the range of 3.0 to 3.5.

Using the definition of COP, the compressor work is computed as,

$$W_{COMP} = \frac{QEVAP}{COP} \tag{3.22}$$

The heat rejected at the condenser is the sum of the cooling load and the compressor work,

$$QCOND = QEVAP + W_{COMP} \tag{3.23}$$

$$= QEVAP \left(1 + \frac{1}{COP}\right) \tag{3.24}$$

The temperature of the air leaving the condenser is,

$$TCOND = T_A + \frac{QCOND}{MREG \times CP} \tag{3.25}$$

where

MREG = condenser airflow rate, kg/hr

3.8 Auxiliary Heater

The auxiliary heater is modeled as a natural gas burner with infinite capacity for supplying thermal energy to the regeneration air stream. The required regeneration temperature, T_{REG}, is determined by the dehumidifier (sec. 3.4) and relayed to the auxiliary heater. The heater controls an intake/exhaust vent to the outside. In this way, it can maintain T_{REG} exactly. If the air entering the heater is lower than T_{REG}, it is supplied with the necessary amount of energy. If the air temperature exceeds T_{REG} entering the heater, ambient air is mixed with it to obtain T_{REG}.

The auxiliary thermal energy required to maintain T_{REG} is computed as,

$$QAUX = MREG \times CP \, (TREG - T_{IN})^{+} \tag{3.26}$$

where

MREG = regeneration flow rate, kg/hr

CP = air specific heat, kJ/kg-°C

T_{IN} = inlet air temperature to heater, °C

and "+" indicates that only positive values are considered (otherwise, QAUX = 0).

The purchased natural gas must have an energy content greater than QAUX due to the inefficiency of the combustion process. For this analysis, the heater is assumed to have an efficiency of 70%.

3.9 Complete Hybrid System

The simplified schematic and psychrometric diagram presented in Chapter 2 can now be replaced with a detailed schematic of the system which will be used in this work. Figure 3.3 shows a detailed schematic of the hybrid system, while Figure 3.4 is its correspond-

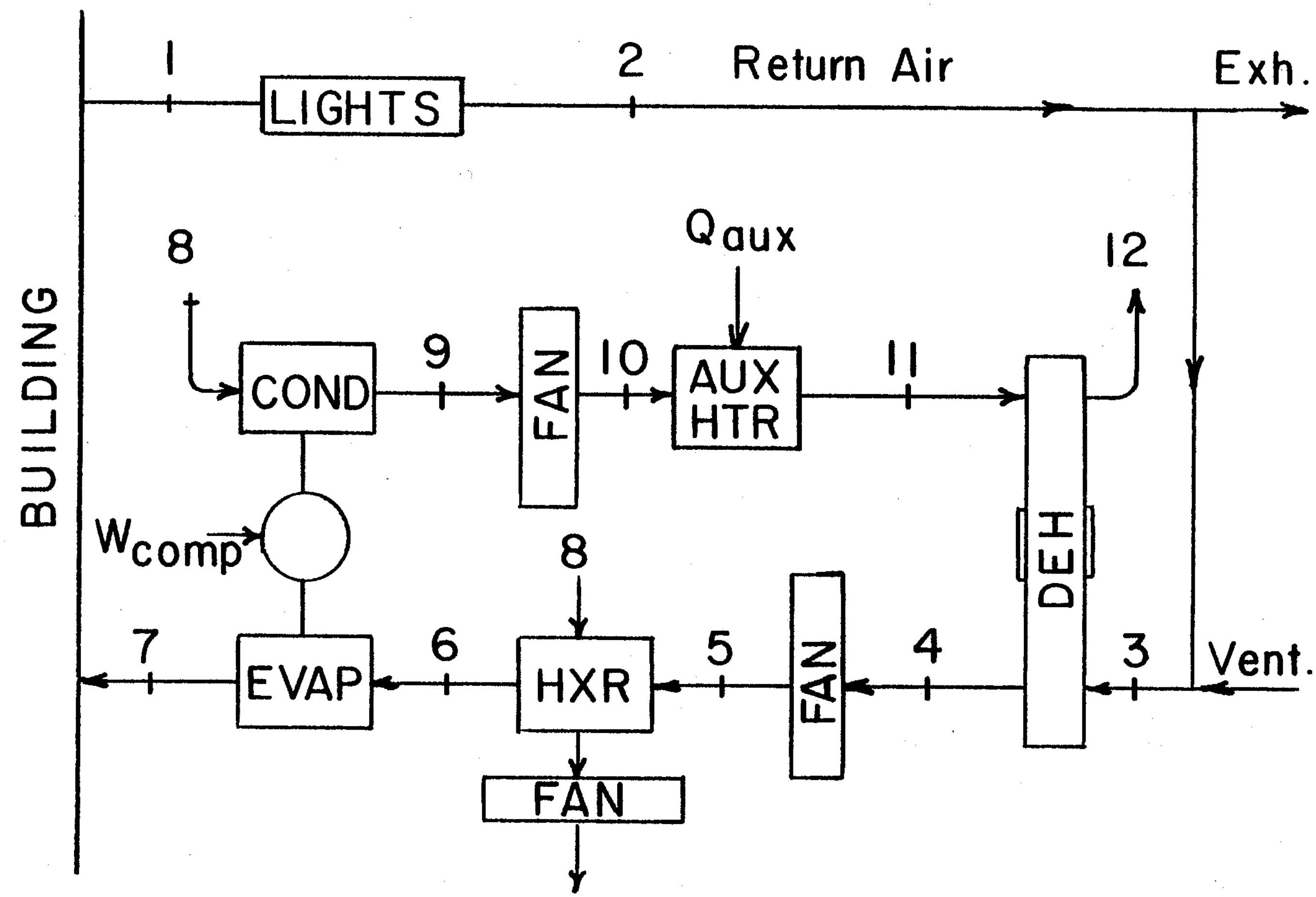


Figure 3.3 Schematic of hybrid air conditioning system

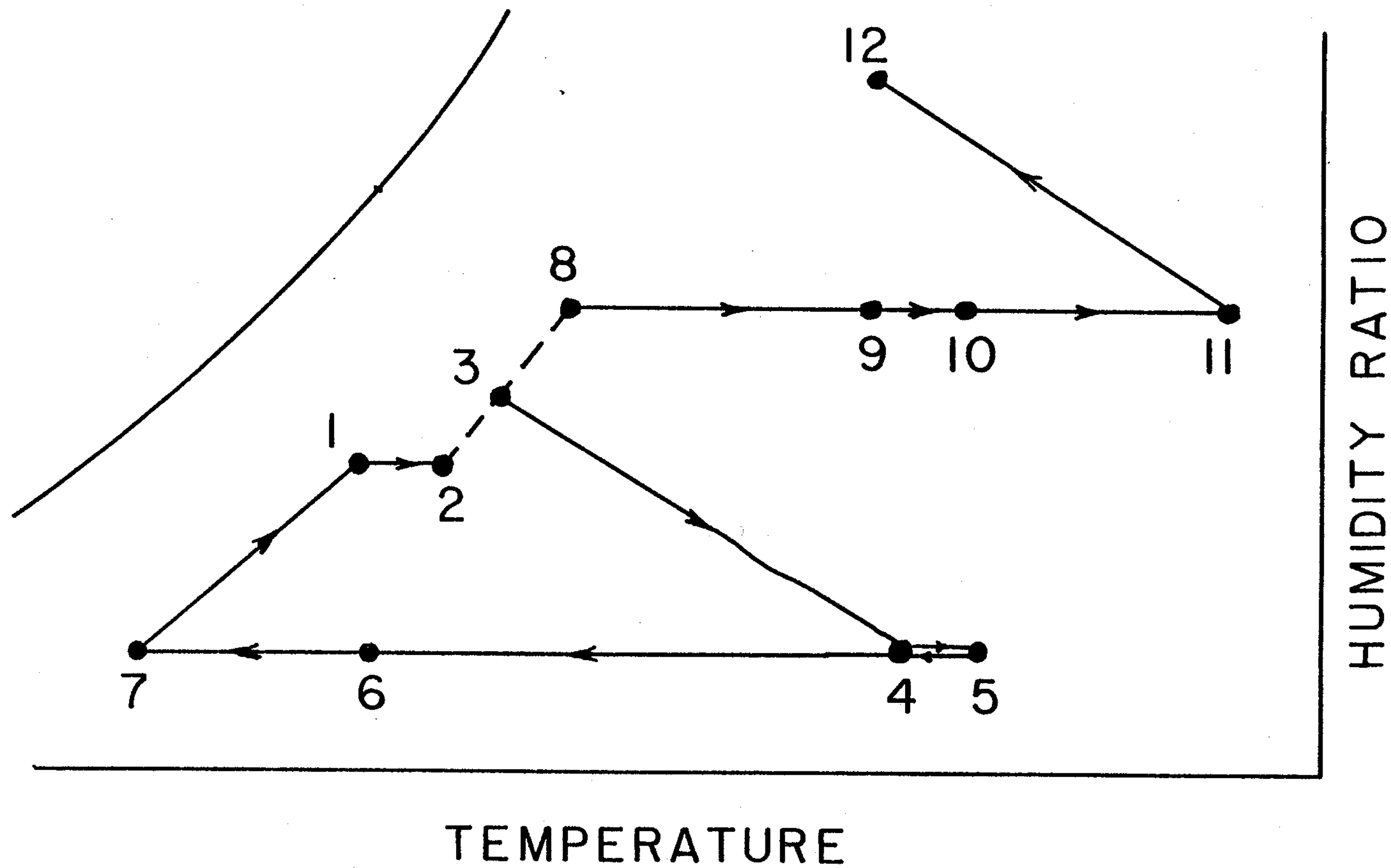


Figure 3.4 Psychrometric diagram of hybrid cooling process

ing psychrometric process. They are quite similar to those presented in Chapter 2 with a couple of exceptions. Three system fans have been added, as well as the building lights. This is the basic system which will be analyzed in the following chapters.

CHAPTER 4 OPERATING CHARACTERISTICS OF HYBRID COOLING SYSTEMS

The quantities of importance which characterize air conditioning systems are the amounts of energy which they use to achieve satisfactory cooling. In this study, we are concerned with three general energy uses: compressor power (W_{COMP}) for the vapor compression unit, fan power (W_{FAN}) to move the air, and auxiliary thermal energy (QAUX) for regeneration of the desiccant. Other parasitic energy uses are ignored. They are small compared with the three mentioned above (W_{COMP}, W_{FAN} and QAUX) which will be used to characterize system performance.

System performance is affected by the interactions between the ambient conditions, the system control strategy, the building load, individual component performance, and others. This chapter studies these interactions, ultimately to determine under what conditions, if any, the hybrid system performance is superior to that of a conventional vapor compression system.

4.1 Effect of Ambient Conditions on System Components

Each of the major components in the hybrid system interacts, sometimes subtly, with the ambient. One would expect their performance to change as the ambient conditions change. This section examines all of the interactions between the outdoor conditions and each of the system components.

4.1.1 Economizer

The ambient affects the economizer in two ways. In the first, it determines which mode, ventilation or recirculation, the economizer will select for operation. The second effect is more subtle. Because the economizer mixes outside air directly with return air in the recirculation mode or supplies outside air alone in the ventilation mode, any change in the ambient conditions will change the air state which it delivers to the dehumidifier. This in turn would have an effect throughout the entire system.

4.1.2 Desiccant Dehumidifier

Figure 4.1 best illustrates the change which occurs in the dehumidifier operation when the ambient conditions change. For a fixed outlet state, the ambient humidity ratio determines the regeneration temperature, while the ambient temperature has no effect at all. Figure 4.1 shows that an increase in the ambient humidity ratio increases the regeneration temperature necessary to maintain a constant outlet humidity ratio. In an ideal dehumidifier, as in Figure 4.1, the outlet temperature would remain the same. But for a real dehumidifier, the outlet temperature would increase also.

4.1.3 Indirect Evaporative Cooler

The indirect evaporative cooler uses evaporatively cooled ambient air as the secondary stream of the heat exchanger process.

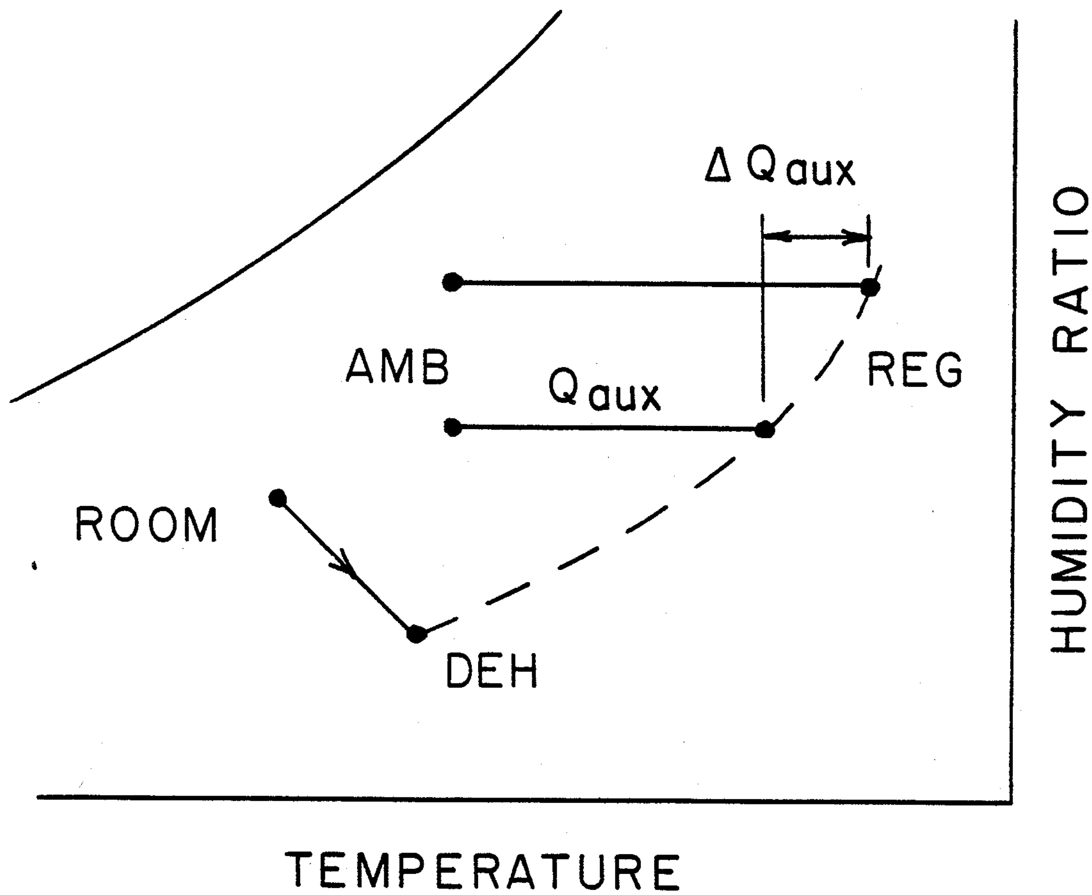


Figure 4.1 Effect of ambient humidity ratio on the regeneration temperature of a desiccant dehumidifier for a fixed outlet state

As shown in Figure 4.2, an increase in the ambient wet bulb temperature[1] will reduce the indirect evaporative cooler's cooling potential. This will mean more cooling must be done by the vapor compression unit, and more power consumed.

4.1.4 Vapor Compression Unit

The vapor compression condenser is cooled by the ambient air. As shown in Equation 3.18, the unit COP decreases as the ambient temperature increases. More compressor power is needed under these conditions. The impact of this interaction is far greater for the vapor compression system than for the hybrid system because the vapor compression system relies exclusively on mechanical refrigeration for total cooling. Any adverse effect to the COP hurts the vapor compression system the most.

4.2 Effect of Ambient Conditions on Control Strategy

In addition to affecting the system components, the ambient conditions affect the way the system is operated. A psychrometric chart is a convenient way to represent the hybrid system control strategy for any ambient condition. Figure 4.3 divides a chart into regions of ambient conditions in which the system will operate in the ventilation or recirculation modes. The figure is further

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[1] Technically, the thermodynamic wet bulb, or adiabatic saturation, temperature. The evaporative cooler model is based on the adiabatic saturation process.

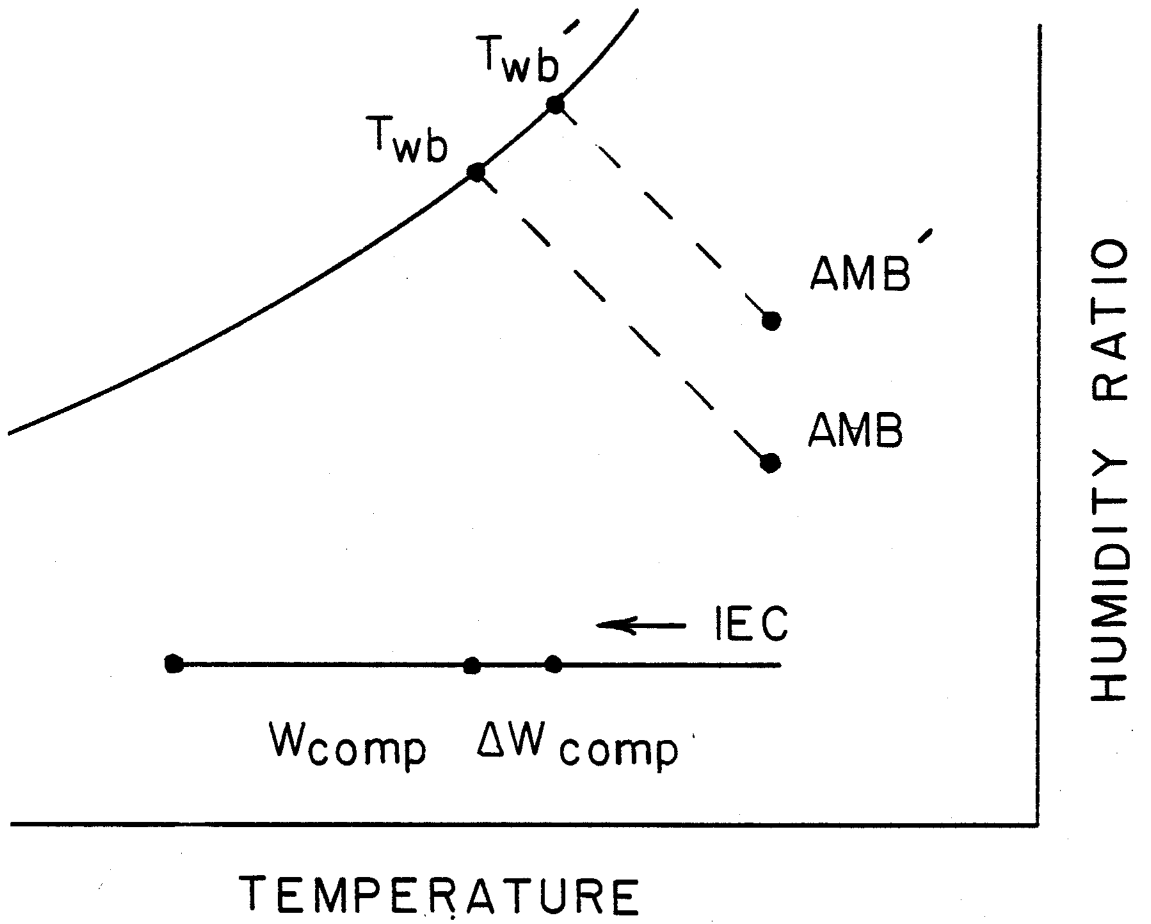

Figure 4.2 Effect of ambient wet bulb temperature on the cooling capacity of an indirect evaporative cooler

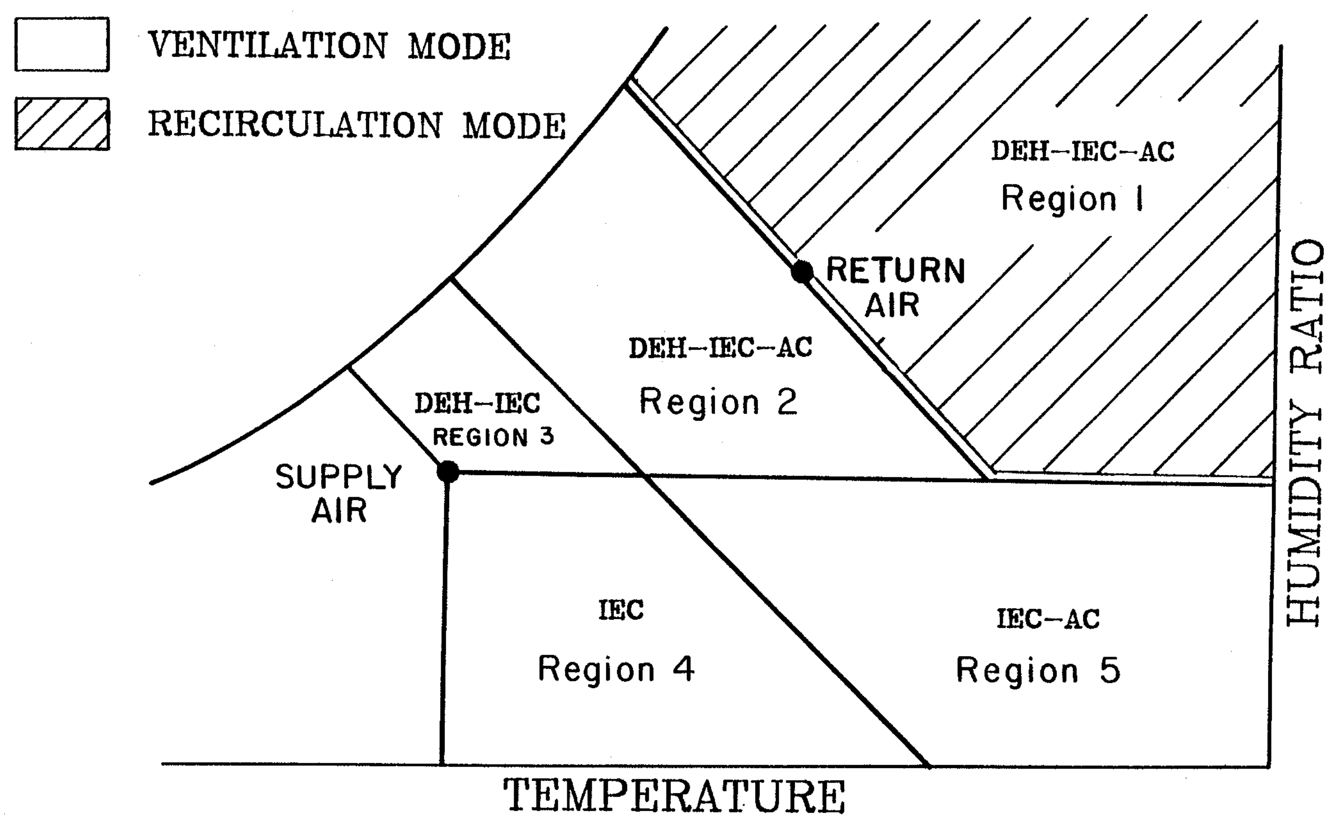

Figure 4.3 Variance of hybrid control strategy with ambient conditions

divided to show which hybrid components are to be used for any climatic condition to deliver the required supply air. Regions 4 and 5 make up those ambient conditions whose humidity ratio is less than or equal to the supply humidity ratio as determined by the building controller (sec. 3.1.2). In these two regions, the desiccant dehumidifier is not used (auxiliary thermal energy for regeneration is zero in these regions). Above the supply humidity ratio, regions 1, 2 and 3 all require dehumidification to meet the supply condition.

Note that regions 3 and 4 do not require the use of a vapor compression unit. In these regions, the supply temperature is sufficiently above the wet bulb temperature of the ambient air that the indirect evaporative cooler can supply the full sensible cooling.

A point to be made of all this is that, while the system components performance are affected by the ambient conditions, the control strategy is also a function of the weather and this determines which of the system components are used at any given time.

4.3 Performance Maps

In section 4.2, a psychrometric chart was used to illustrate the hybrid control strategy for any ambient condition. We can extend the use of the psychrometric chart even further to show the level of system performance at any ambient condition. Using electrical and thermal energy consumption as a measure of system performance, contour plots of energy consumption as a function of the ambient temperature and humidity ratio can be overlaid on a psychrometric chart. These performance maps, as they shall be called, have the benefits listed below.

1. Convenient way to summarize system performance for any climate.
2. System analysis is simple using Figure 4.3 and Section 4.1.
3. Trends are easily seen.
4. Convenient for parametric studies.

Performance maps will be used throughout this chapter in the manners mentioned above.

To generate the performance maps, the building cooling system is subjected to a dummy load, and the steady-state operation of the system is simulated over a wide range of ambient conditions. The purpose of using a dummy load is to eliminate changing loads as a reason for any changes in system performance. Because the load is artificial, the performance maps are generic, rather than for a specific building size and construction. A large well-constructed building may well have similar loads to a smaller, poorly-constructed one in the same climate.

4.3.1 Base Case Performance Maps

To allow later comparison, a base case system was established using the parameters given in Table 4.1. The component efficiencies and effectivenesses are reasonable values, achievable with present

TABLE 4.1

Base Case System Parameters

System Parameters

Total load	= 209 MJ/hr
Sensible load	= 124 MJ/hr
Regeneration flow (MREG)	= .60 x system flow (MSYS)

Component Parameters

Dehumidifier,	ε_{F_1}	= .08
	ε_{F_2}	= .95
Indirect Evaporative Cooler	ε_{IEC}	= .9025
Fan	$\eta_{f,s}$	= .70
	$\eta_{d,m}$	= .70

technology.

The regeneration air flow rate is set at 60% of the process air flow. This is consistent with what Jurinak [9] has shown to be the optimum flow for cycles operating in the recirculation mode.[1] While the hybrid system does not always operate in the recirculation mode, the ventilation mode is used only when no dehumidification of the outside air is required (and thus no regeneration stream is required) or when the outside air enthalpy is less than the return air (see Figure 4.3). With this arrangement, air entering the dehumidifier is always in the range of that found under recirculation mode conditions.

Figure 4.4a is the base case performance map of the total electrical energy use (W_{COMP} + W_{FAN}) for the hybrid system; Figure 4.4b is the same for a vapor compression system. As the ambient temperature or humdity ratio increase, both systems increase their electrical energy use. This is due to the effects discussed in sections 4.1 and 4.2. The constant electrical energy lines are steeper for the vapor compression system, showing that it is less affected by changes in the ambient humidity ratio than the hybrid system. Of its two systems components,[2] only the economizer is affected by the ambient humidity ratio. The hybrid system de-

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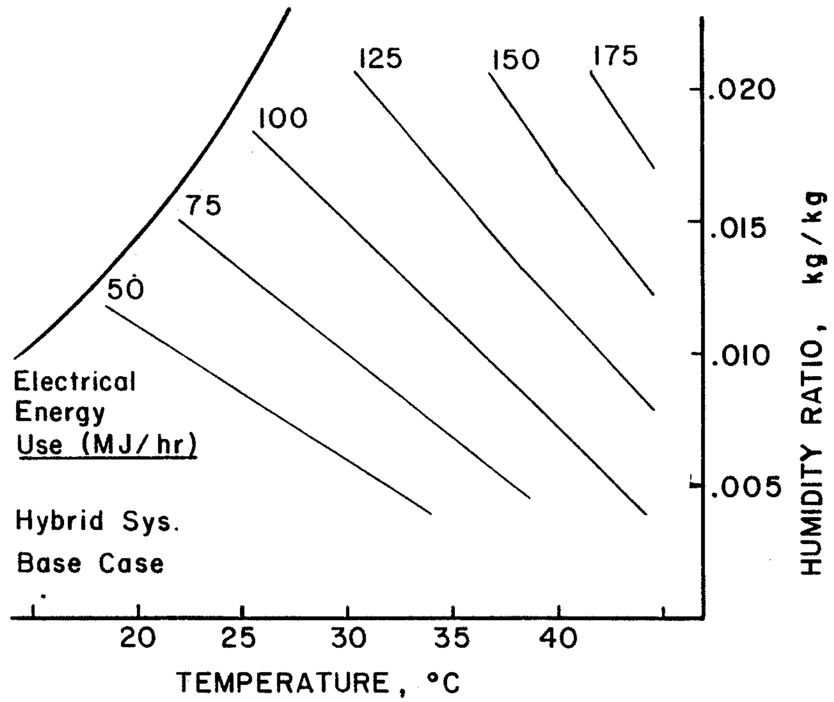
[1]Jurinak's conclusion is based on the operation of a finite capacity dehumidifier. Our assumption of an infinite capacity dehumidifier is realistic only if a reasonable regeneration flow rate is chosen.

[2]Vapor compression unit and an economizer.



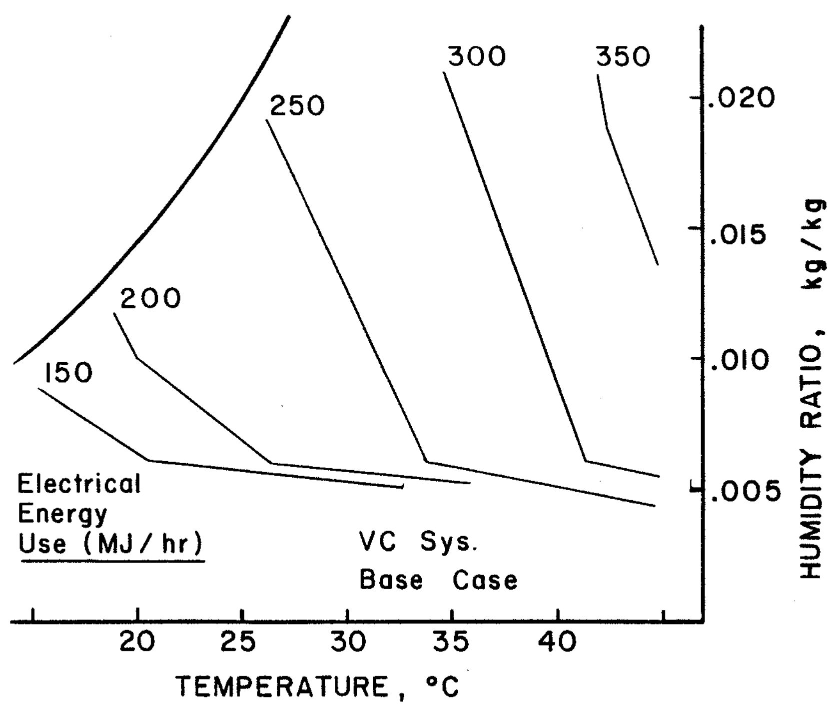


Figure 4.4 Base case performance maps

a) Hybrid system electrical energy use (W_{FAN} = 35.5 MJ/hr)

b) Vapor compression system electrical energy use (W_{FAN} = 22.7 MJ/hr) humidifier, indirect evaporative cooler and economizer are all affected by the ambient humidity ratio.

The break in the contour lines for the vapor compression system occur at the supply air humidity ratio. Below this point, 100% outside air is used for cooling and does not require dehumidification. Above the break, additional electrical energy is used to dehumidify the air. There is no discontinuity in the contour lines for the hybrid system, Figure 4.4a, because all dehumidification is done by the desiccant dehumidifier which needs only thermal, not electrical, energy for operation.

Note that over the entire range of climatic conditions studied, the hybrid system uses less electrical energy than the vapor compression system.

Figure 4.5a is the base case performance map of the ratio of hybrid compressor work to vapor compression system compressor work (compressor power ratio, CPR). The CPR tends toward zero at low temperatures and humidity ratios. The control strategy plot of Figure 4.3 helps explain this: the system is operating near regions 3 and 4 where the hybrid system does not need the vapor compression unit. However, in this region the vapor compression system uses outside air to help reduce its load, and needs only a small amount of mechanical refrigeration, so there is very little difference in the compressor power used. At high temperatures and humidities, the CPR increases but the difference in compressor power is significant.

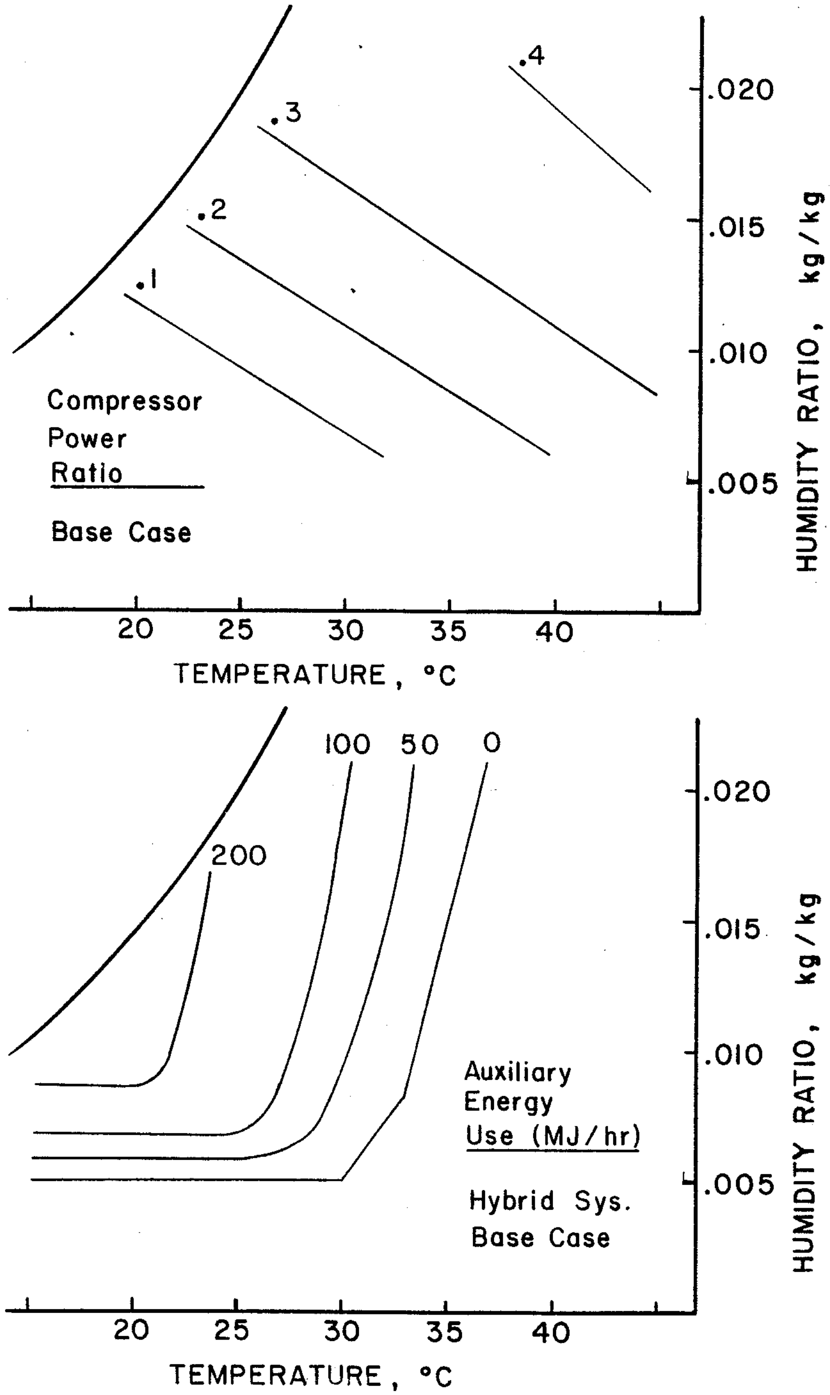


Figure 4.5 Base case performance maps
a) Ratio of hybrid system compressor work to vapor compression system compressor work (CPR)
b) Hybrid system auxiliary energy use

The CPR lines resemble lines of constant enthalpy or constant adiabatic saturation. The vapor compression system cooling load is based on the enthalpy difference between the return and supply air states. Along ambient constant enthalpy lines, this cooling load remains constant. The hybrid system vapor compression unit cooling load is based on the temperature difference between the indirect evaporative cooler outlet and the supply air states. But along constant ambient adiabatic saturation lines, the outlet of the indirect evaporative cooler stays nearly unchanged, and therefore the cooling load on the vapor compression unit remains nearly unchanged. As the vapor compression cooling loads stay constant along these lines, one would expect the CPR lines to be nearly constant also.

Figure 4.5b plots the hybrid system auxiliary energy use under the base case conditions. The slope of the curves changes dramatically along the line which separates the ventilation mode region from the recirculation mode region. In the ventilation mode, the auxiliary energy is essentially independent of the ambient temperature, while in the recirculation mode it is nearly independent of the ambient humidity ratio.

4.3.2 Effect of Sensible Heat Ratio on System Performance

For the same total load of 200 MJ/hr, the hybrid system was simulated with sensible heat ratios (SHR) of 1.0 and 0.0. The sensible heat ratio is defined as,

$$SHR = \frac{QSENS}{QTOT} \tag{4.1}$$

where

QSENS = sensible load, kJ/hr

QTOT = total load, kJ/hr

For the base case, the SHR = 0.60.

Figures 4.6a and 4.6b show the effects of decreasing the sensible heat ratio. As the SHR decreases, the supply air state--as determined by the building controller--moves along a constant enthalpy line[1] to a warmer, drier state. The dehumidifier must dry the process air to this lower humidity ratio, which increases the regeneration temperature. Additional auxiliary thermal energy, $\Delta QAUX$, is required to meet this higher regeneration temperature.

Even with the increased dehumidifier outlet temperature (see Fig. 4.6a), the indirect evaporative cooler is able to cool the process stream to, or nearly to, the ambient wet bulb temperature. Since the supply air temperature is increased, less sensible cooling and less compressor work (ΔW_{COND}) is required of the vapor compression unit.

It is interesting to note that any process which reduces the amount of compressor work also reduces the amount of heat added to the regeneration stream at the condenser. The implication is that

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[1]The controller maintains a 15 kJ/kg enthalpy difference between the room and supply air states.

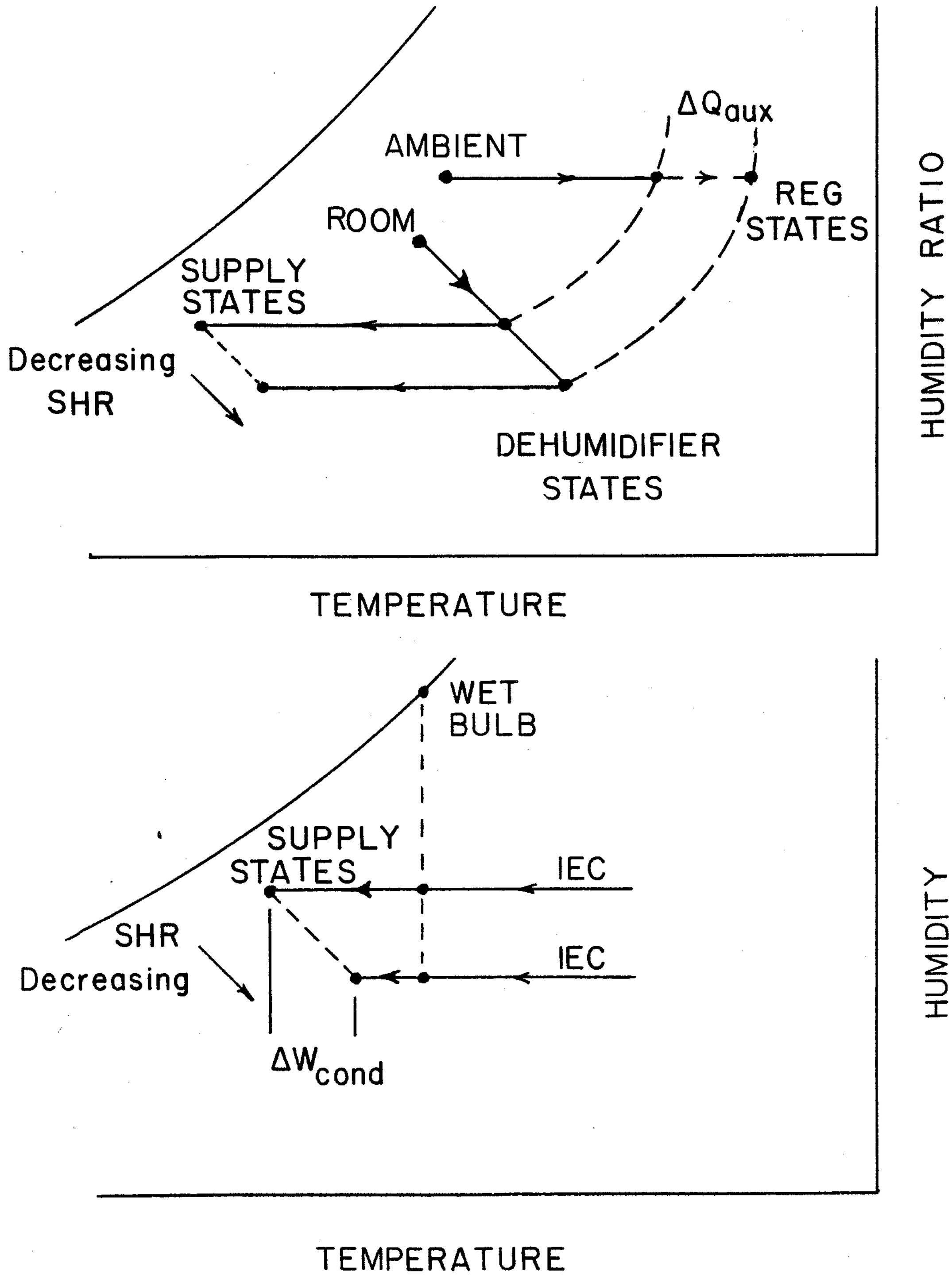


Figure 4.6 Effect of varying sensible heat ratio (SHR)
a) on dehumidifier outlet and regeneration states
b) on vapor compression work

it may not be optimal to reduce the vapor compression work to zero, as the auxiliary energy requirements will increase dramatically.

Performance maps of the hybrid electrical energy use at different sensible heat ratios are found in Figures 4.7a and 4.7b. As predicted, there is a significant decrease in electrical energy use as the latent fraction of the load increases (i.e., the SHR decreases). The opposite is true as the SHR increases.

The hybrid auxiliary energy use is found in the performance maps in Figures 4.8a and 4.8b. As the sensible heat ratio increases the auxiliary energy use drops off dramatically, until at a fully sensible load, auxiliary energy is required only at near-saturated ambient conditions. An outline of the control strategy map (Fig. 4.3) has been overlaid on Figure 4.8b to help distinguish the different regions of operation. Notice that the size of the regions change, as do their locations, as the supply air state changes (due to the changing sensible heat ratio). As the sensible fraction of the load decreases, the line separating the regions which require vapor compression cooling (regions 3 and 4, Fig. 4.3) from those which do not (regions 2 and 5, Fig. 4.3), moves to the right on a psychrometric chart. At the high latent load in Figure 4.8b, this line lies within the recirculation mode region (region 1). This will never occur in practical cooling situations. That is, operation in the recirculation mode will always require some vapor compression cooling.

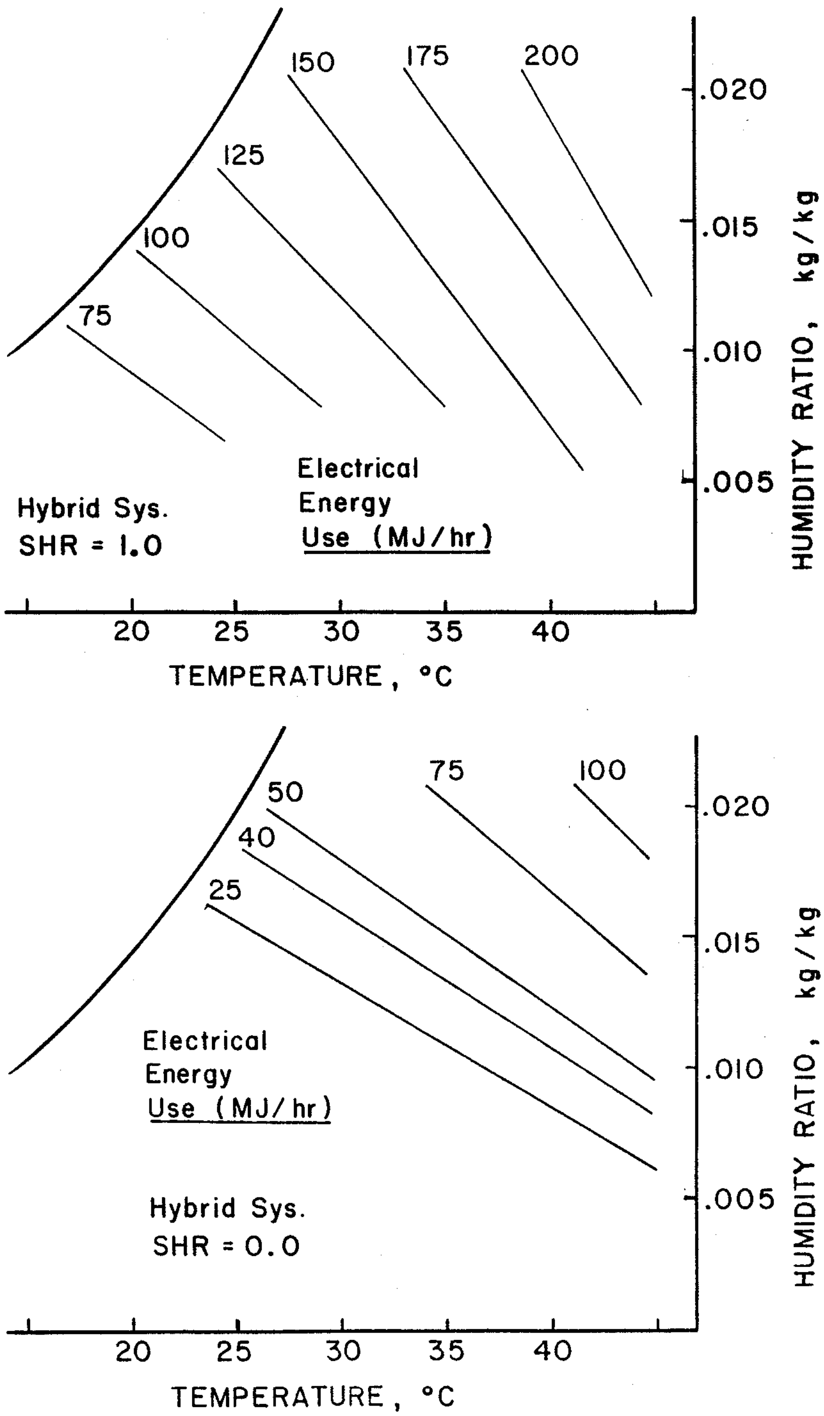


Figure 4.7 Performance maps of hybrid system electrical energy use--variable SHR
a) SHR = 1.0
b) SHR = 0.0

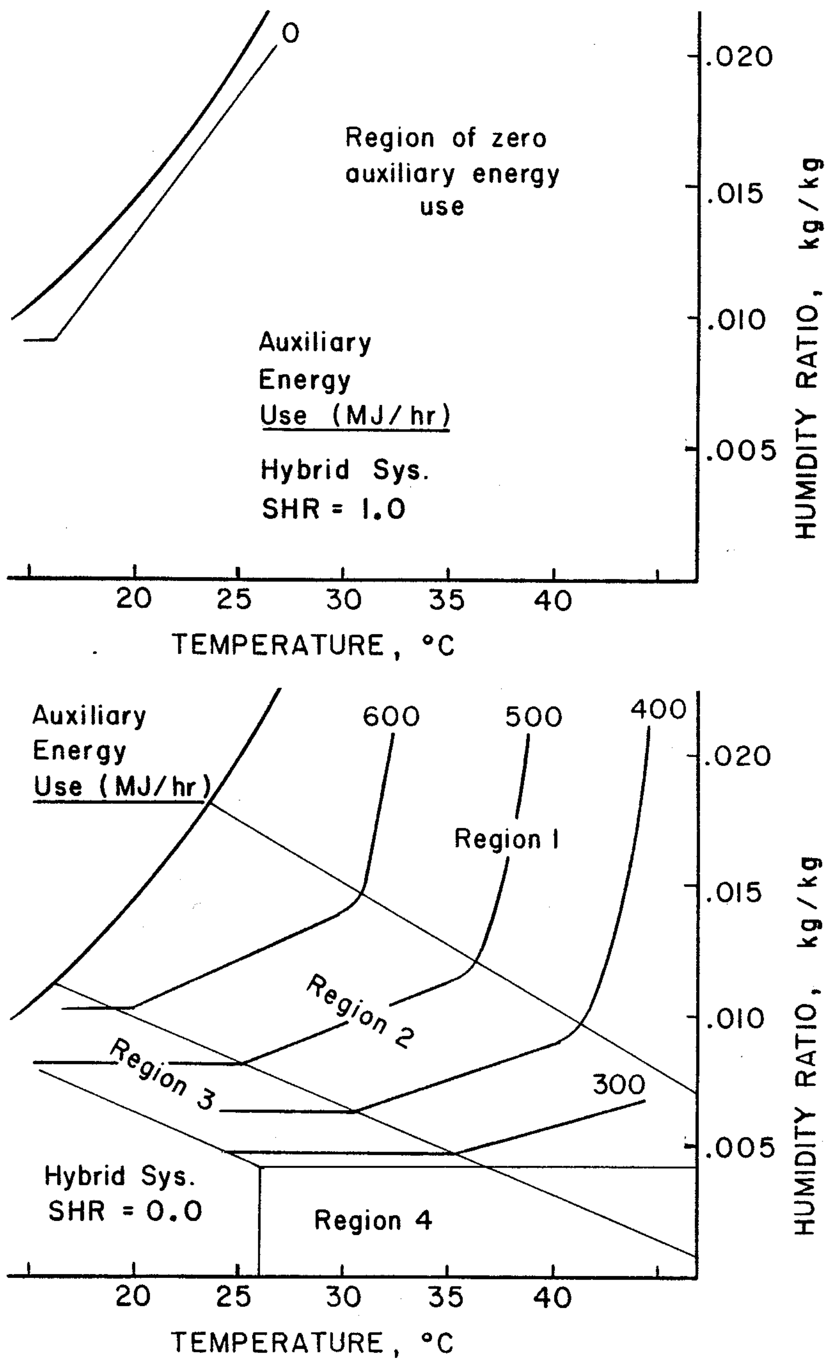


Figure 4.8 Performance maps of hybrid system auxiliary energy use--variable SHR
a) SHR = 1.0
b) SHR = 0.0

In the recirculation mode (region 1), the constant auxiliary energy lines resemble the dehumidifier F_2 lines at its regeneration state. This is because the air state entering the dehumdifier is relatively constant in the recirculation mode and therefore the regeneration temperature is a function of the ambient humidity ratio only. The ideal dehumidifier regeneration state will lie on the F_2 line which passes through the dehumidifier outlet state.

At high regeneration temperatures, the F_2 lines are quite steep, so the regeneration temperature does not increase substantially with an increased ambient humidity ratio. In this situation, auxiliary energy use does not increase appreciably with the ambient humidity ratio.

4.3.3 Effect of Load on System Performance

To determine whether the magnitude of the building cooling load affects the system performance in any unusual ways, the dummy load was increased from 209 MJ/hr to 313.5 MJ/hr. Performance maps in Figures 4.9a and 4.9b show the hybrid and vapor compression systems electrical energy use. Comparing these to the base case results in Figures 4.4a and 4.4b, both systems have increased their electrical energy use significantly, which is an obvious result. The only effect to the process air stream is to increase its mass flow rate, which increases the fan power and compressor work. The system state points remain the same.

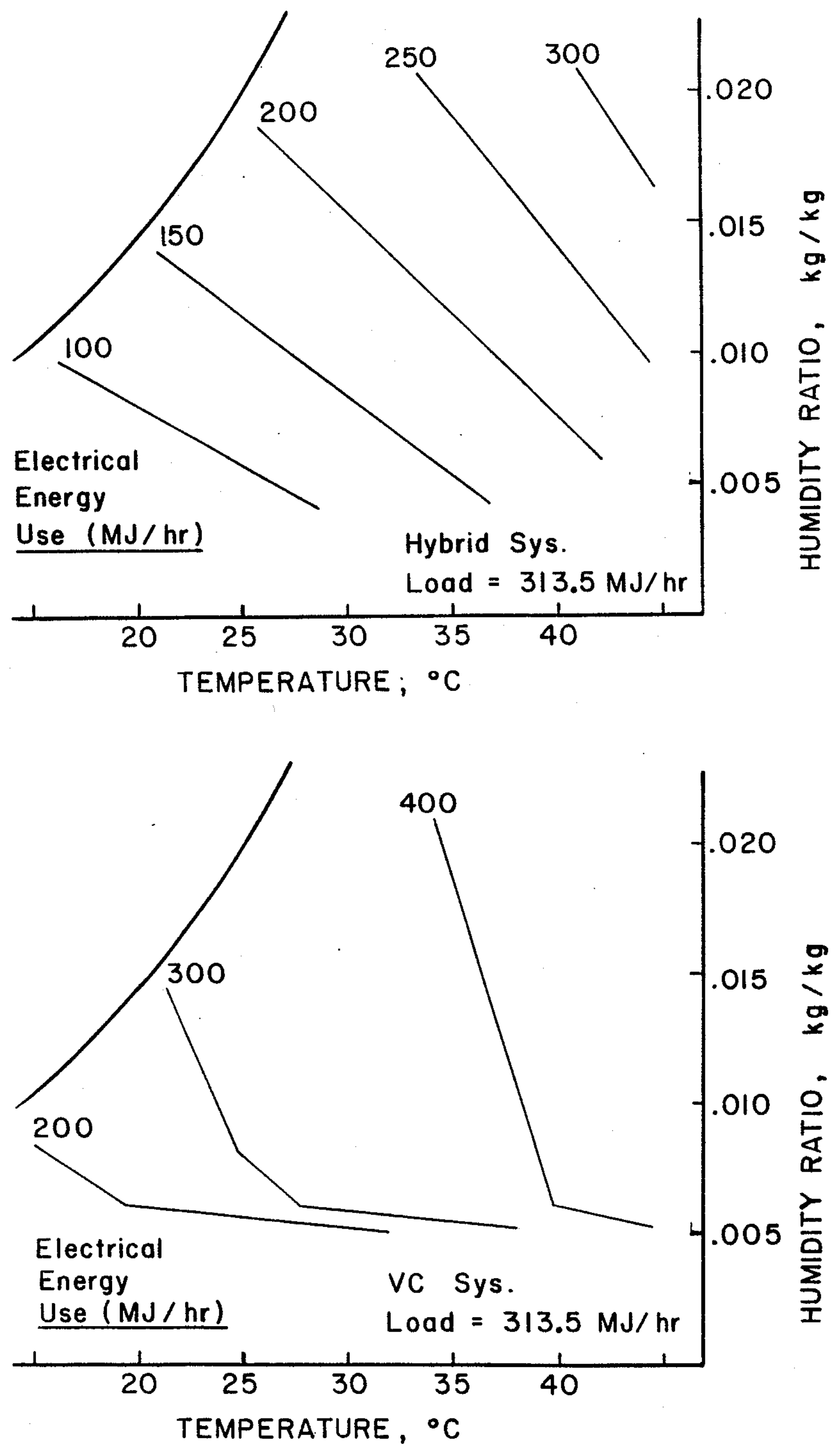


Figure 4.9 Performance maps--variable load
a) Hybrid system electrical energy use
b) Vapor compression system electrical energy use

Due to the increased compressor work, the heat rejected to the regeneration stream at the condenser increases. The percentage increase in condenser heat is slightly greater than the percentage increase in the mass flow rate. Thus, the regeneration air stream enthalpy out of the condenser increases with increased loads so that slightly less auxiliary energy is needed.

4.3.4 Effect of Dehumidifier Performance on System Performance

The base case dehumidifier parameters of $\varepsilon_{F_1} = .08$ and $\varepsilon_{F_2} = .95$ are achievable with present technology and describe a relatively good dehumidifier. The performance map approach is a convenient method of determining the system effect if poor dehumidifier parameters are selected. For this study, the parameters chosen for system analysis are:

$$\varepsilon_{F_1} = .10$$

$$\varepsilon_{F_2} = .80$$

By increasing the F_1 effectiveness and decreasing the F_2 effectiveness, the equilibrium regeneration and outlet temperatures are both increased. This is illustrated in Figure 4.10. The effect of increasing the regeneration temperature is a significant increase in the auxiliary energy used, as is shown in the performance map of Figure 4.11. This increase occurs for all sensible heat ratios.

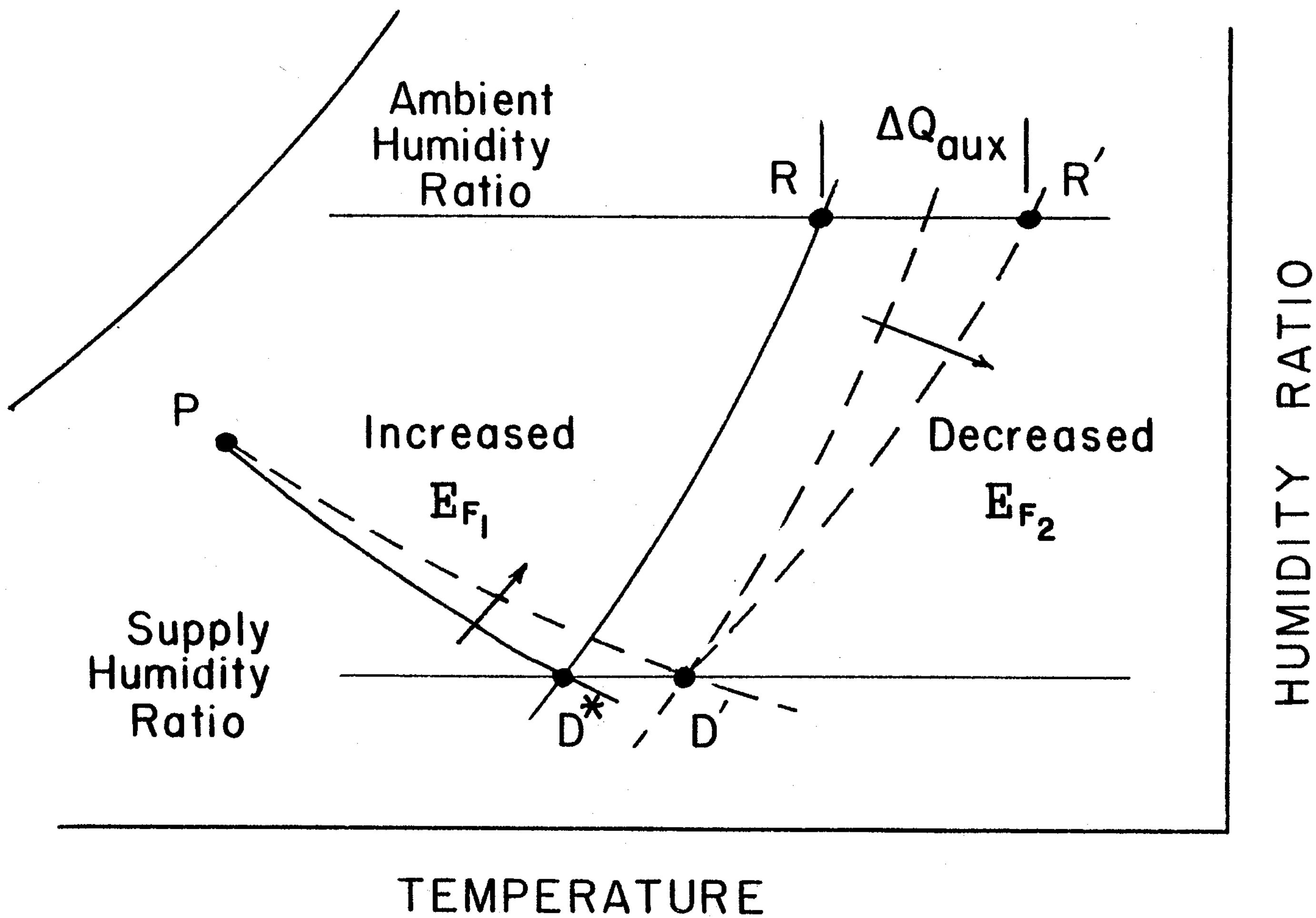


Figure 4.10 Effect of varying F_i effectivenesses on the dehumidifier outlet and regeneration temperatures

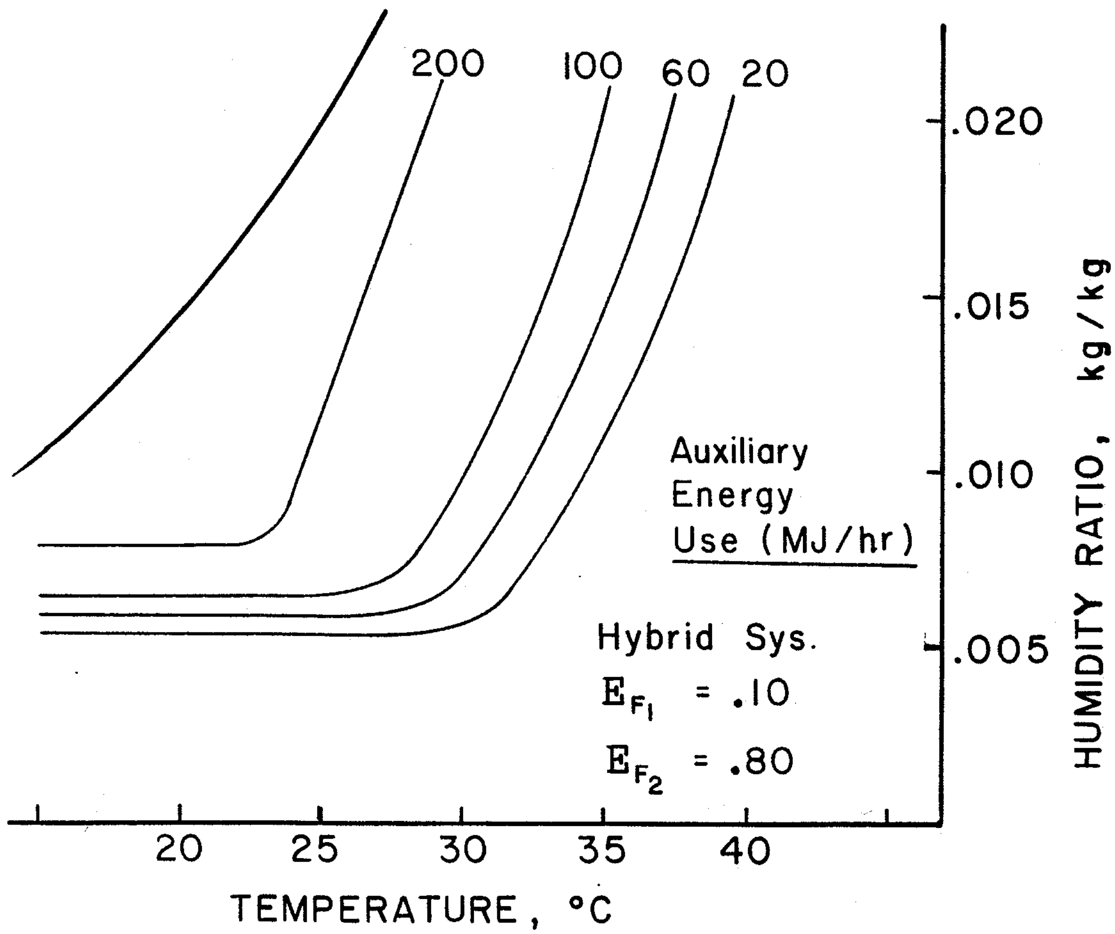


Figure 4.11 Performance map of hybrid system auxiliary energy--variable dehumidifier effectivenesses

The higher dehumidifier outlet temperature has very little effect on the process stream electrical energy use. The indirect evaporative cooler is still able to cool the air nearly to the ambient wet bulb temperature. The inlet condition to the vapor compression unit is thus unchanged, so its electrical energy consumption remains constant.

It is expected that improved dehumidifier performance would reduce the amount of auxiliary energy required, but again have little impact on the electrical energy use.

4.3.5 Effect of Indirect Evaporative Cooler Performance on System Performance

An ideal indirect evaporative cooler reduces the inlet air stream to the ambient wet bulb temperature. While this is not realistic, this would minimize the vapor compression electrical energy used. Conversely, the poorer the indirect evaporative cooler performance, the higher will be its outlet temperature, and compressor work will increase. Figure 4.12a is a performance map of the electrical energy use for the hybrid system with an indirect evaporative cooler effectiveness of $\varepsilon_{IEC} = .80$. Compared with the base case system ($\varepsilon_{IEC} = .9025$) of Figure 4.4a, the electrical energy is shown to increase by about 20 MJ/hr.

One would expect that, as a result of the increased compressor work, the system auxiliary energy use would be reduced. Figure 4.12b shows about a 60 MJ/hr reduction (when compared to the base case system,

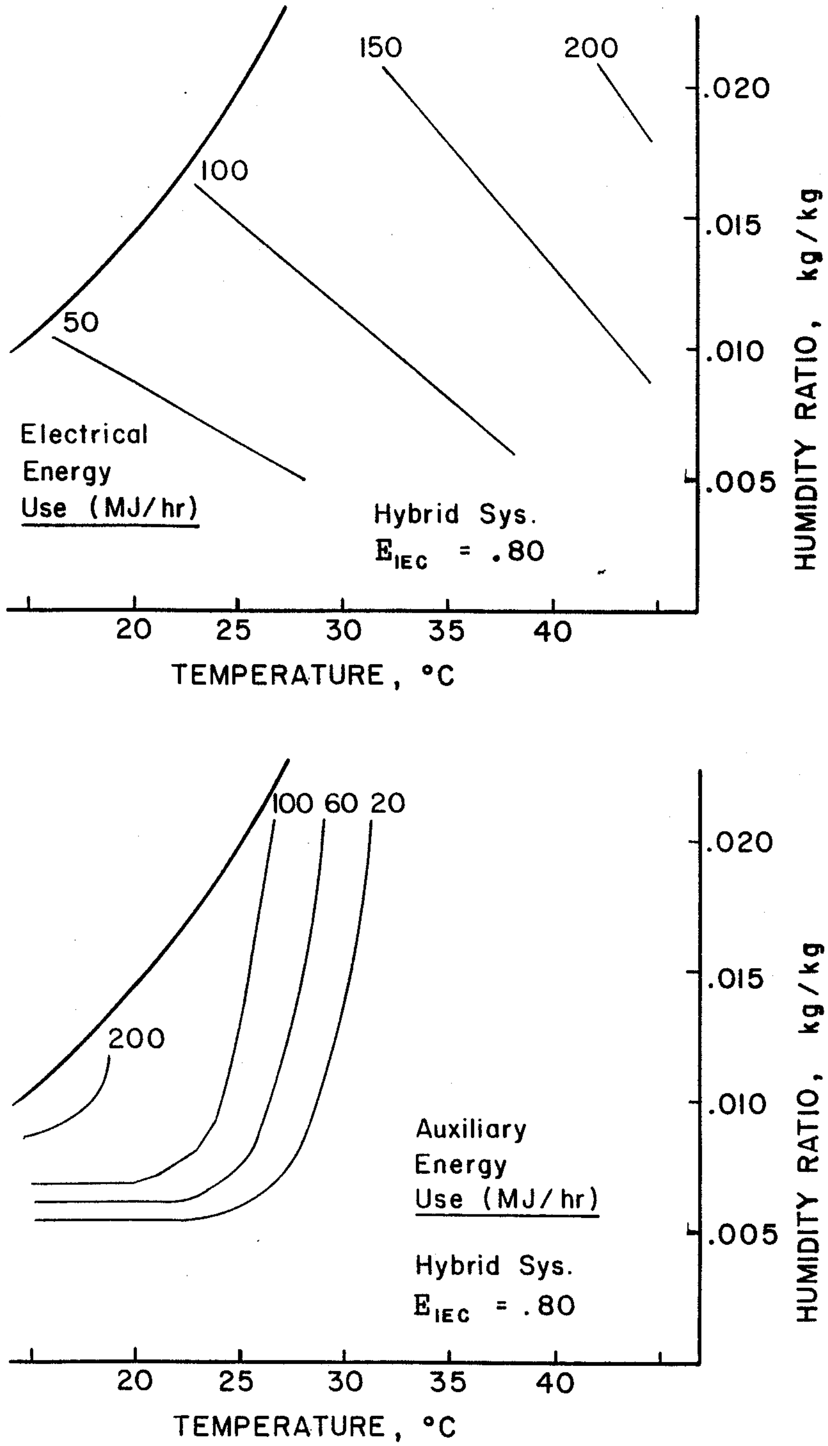


Figure 4.12 Performance maps--variable ε_{IEC}
a) Hybrid system electrical energy use
b) Hybrid system auxiliary energy use

Fig. 4.5b).

To best see the effect of reduced indirect evaporative cooler performance on the vapor compression unit, Figure 4.13 is a map of the compressor power ratio (CPR). Comparing this with the base case CPR map in Figure 4.5a, the contour lines have been shifted down, meaning the gap between the vapor compressing and hybrid systems mechanical refrigeration power use is closing but it is still significant.

4.3.6 Effect of Vapor Compression Performance on System Performance

If the COP of the vapor compression unit is increased, the electrical energy consumption for both the hybrid and the vapor compression systems will decrease. For an equal increase in COP for each system, the compressor power ratio (CPR) between the two will remain identical to that of the base case (Fig. 4.5a). But this means that the absolute reduction (not fractional) in the electrical energy use for the vapor compression system is much greater, because it consumes much more. The point to be made is that, as the performance of the vapor compression unit improves, the hybrid system begins to lose some of its advantage over the vapor compression system. Figures 4.14a and 4.14b show the electrical energy use of the two systems when the COP is raised by 50%.

Again, the result of reducing the compressor power for the hybrid system is to increase the amount of auxiliary energy needed.

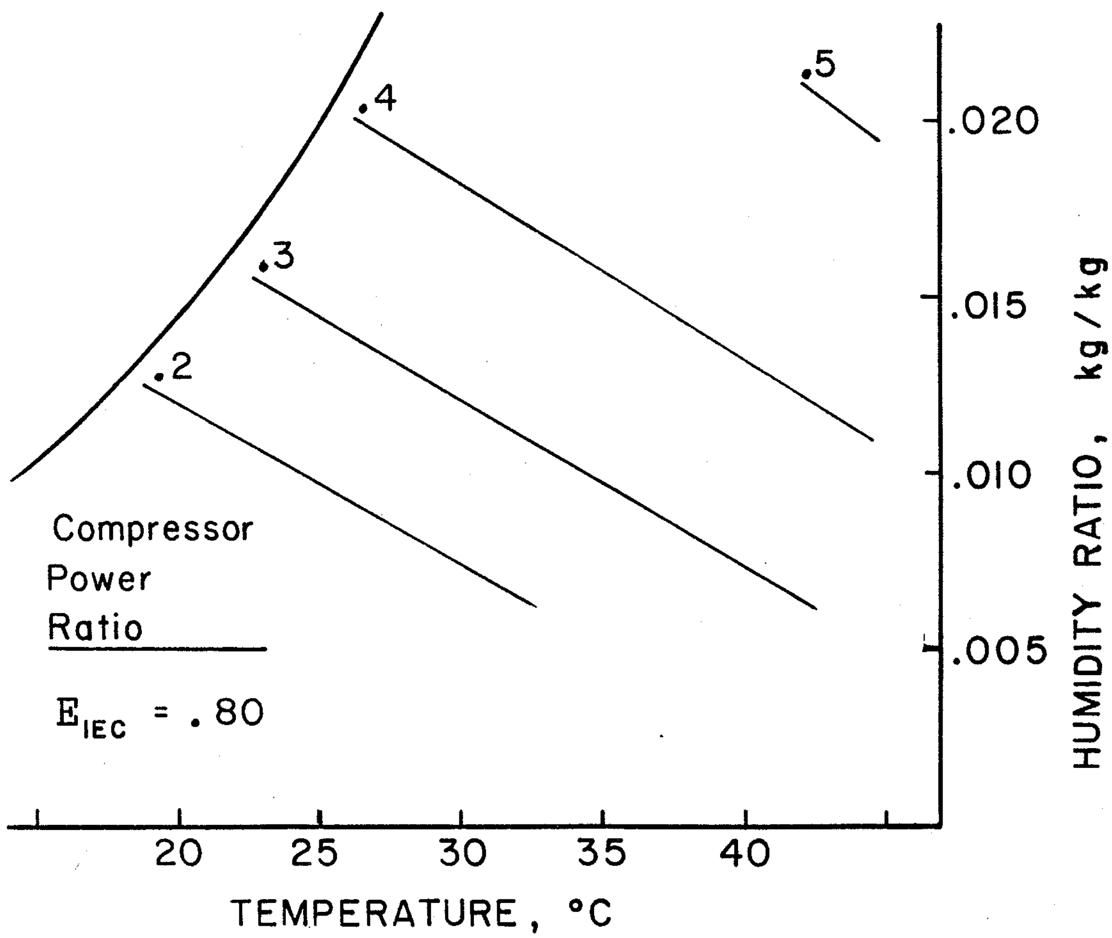
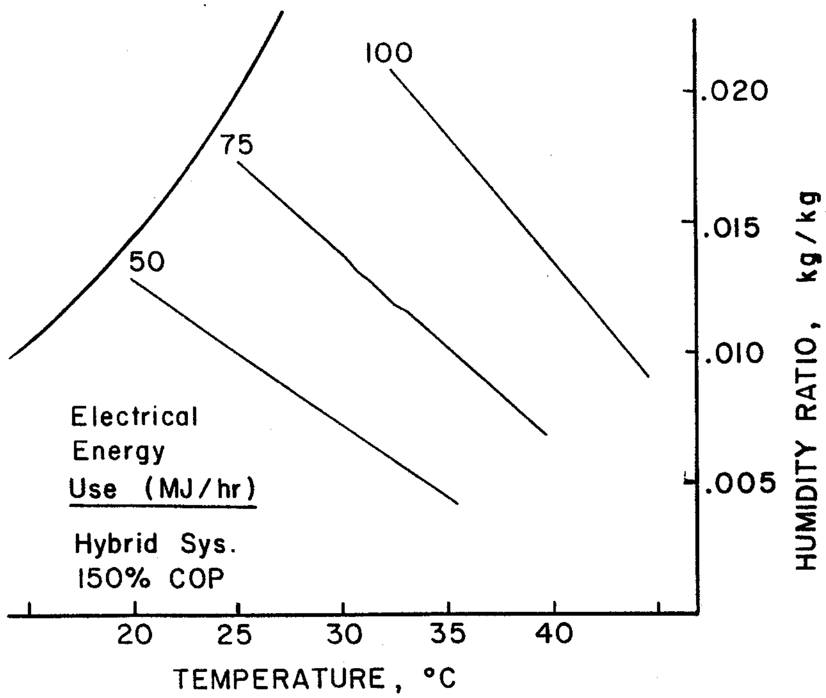


Figure 4.13 Performance map of compressor power ratio (CPR)--variable ε_{IEC}



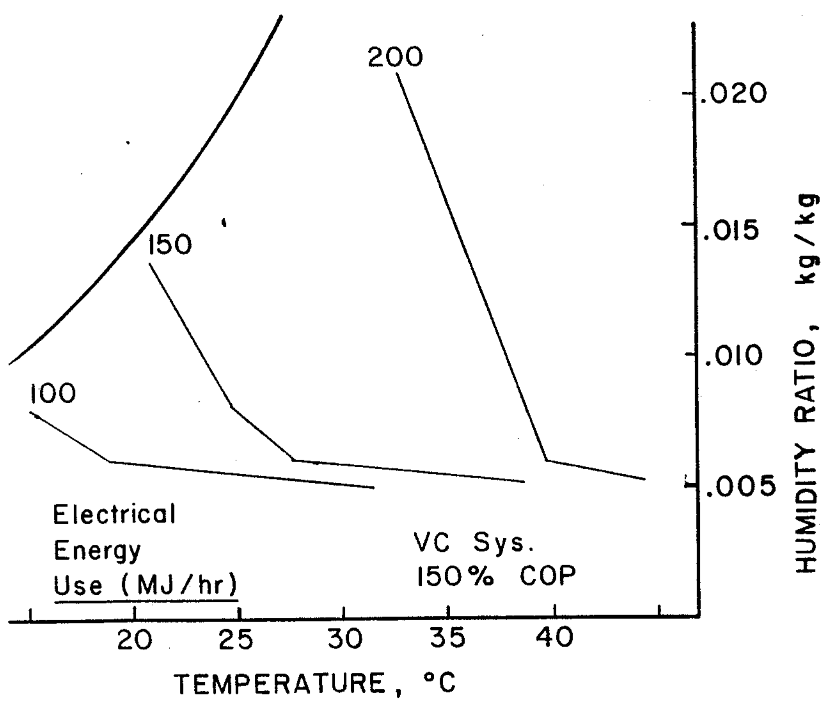


Figure 4.14 Performance maps--variable COP
a) Hybrid system electrical energy use
b) Vapor compression system electrical energy use

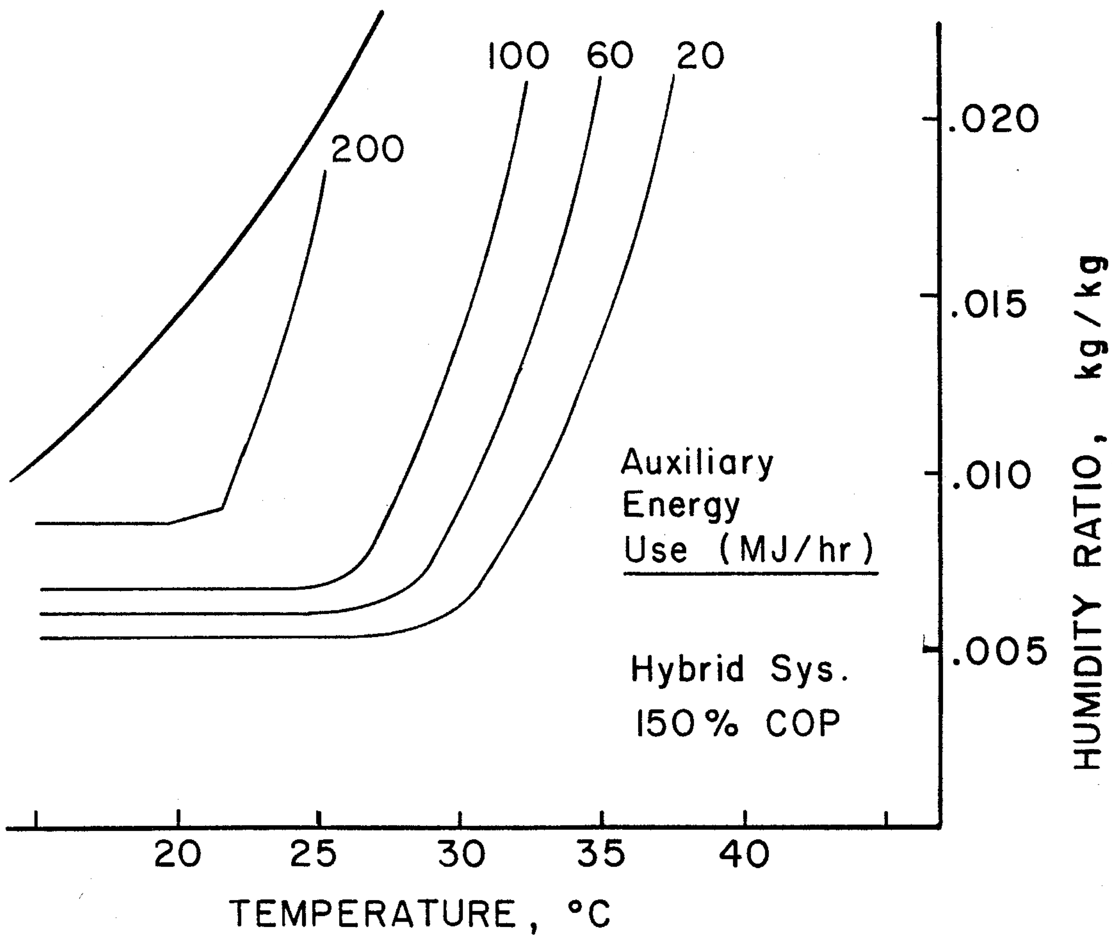


Figure 4.14 (continued) Performance maps--variable COP
c) Hybrid system auxiliary energy use

This is shown in the auxiliary energy use plot of Figure 4.14c.

4.3.7 Concluding Remarks Regarding Performance Maps

The utility of performance map analysis is to spot trends in system performance, to compare various systems, and to aid in the understanding of the system operation. However, the performance map data was generated using a fixed load, at a fixed room state, and with a hybrid system that is slightly modified from the one presented in Chapter 3. Because of these restrictive conditions, using data from the performance maps as representative of the actual hybrid system under normal operationg conditions may not be quite accurate. Results of simulations for the 'real' systems under actual conditions are presented in Chapter 6, and these reflect a more accurate portrayal of system performance.

CHAPTER 5 METHODS OF SUPPLYING REGENERATION ENERGY

Hybrid systems offer a tradeoff in air conditioning operating costs. They provide reduced electrical energy use over a conventional vapor compression system, but at the same time they require thermal energy for regeneration of the desiccant. The hybrid system described in Chapter 3 reclaims the vapor compression condenser heat to supply a portion of the thermal energy needs, while using an auxiliary heater to supply the remainder. But is this the most economical way to provide the regeneration energy?

In this chapter, the benefits and detriments of supplying the regeneration energy using the condenser and auxiliary heater are discussed. Three alternatives are presented involving the use of a flat plate solar collector system. Each system will be simulated, with the results being presented in Chapter 6 and an evaluation made.

5.1 Strategies for Supplying Regeneration Energy

Four different methods for supplying thermal energy to regenerate the desiccant will be considered here. They are designated S1-S4. Schematic drawings of the components involved in the regeneration process are given. To eliminate confusion, fans have been omitted from the drawings because, while they do contribute to the thermal energy input to the regeneration stream, each option (S1-S4) contains a fan, so they do not help distinguish one option from another.

5.1.1 Condenser and Auxiliary (S1)

Figure 5.1 shows diagrammatically the arrangement of components to supply thermal energy to the desiccant using the vapor compression condenser and an auxiliary heater. This is the standard method, as presented in Chapter 2. The condenser is cooled with ambient air. If the air temperature at the condenser exit is below the regeneration temperature, the auxiliary heater is turned on, supplying the additional thermal energy needed.

Because this option does not entail use of a solar system, the initial equipment cost of providing for regeneration is low. Another advantage is that, generally, as the building load increases, the condenser supplies a greater fraction of the required regeneration energy (as explained in Sec. 4.3.3). This results in a reduced percentage of auxiliary energy being supplied.

5.1.2 Solar and Auxiliary (S2)

Equation 3.19 shows that the vapor compression unit COP increases as the air temperature entering the condenser decreases. This reduces the electrical energy consumption. Option S2 takes advantage of this characteristic by using a flat plate solar collector array and an auxiliary heater to supply all of the regeneration energy. A cooling tower can then be used to provide air near its wet bulb temperature to cool the condenser. The heat from the

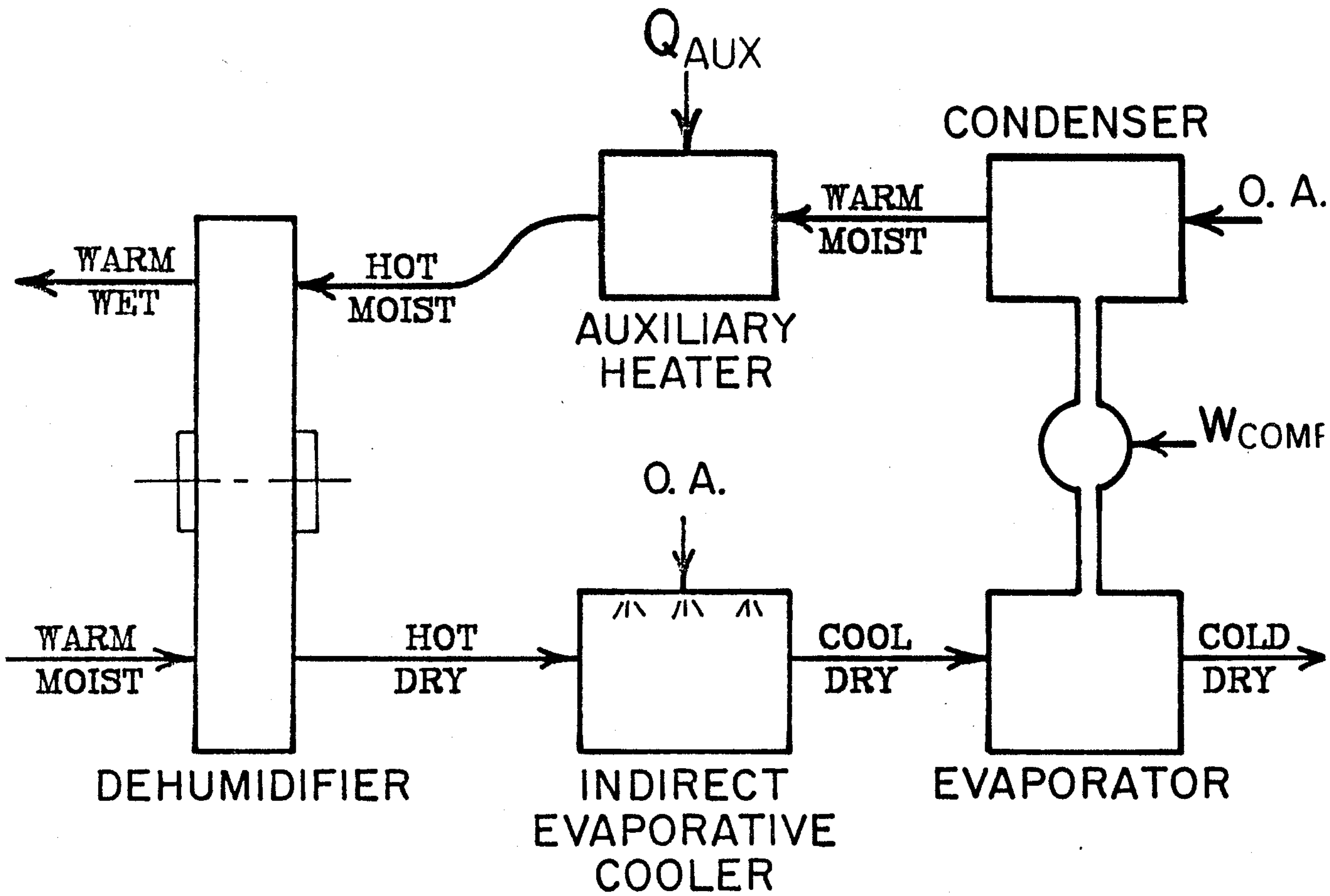


Figure 5.1 Regeneration option S1--Condenser and auxiliary (standard option)

condenser is not used for regeneration. Figure 5.2 illustrates this option.

The advantage of this arrangement, as mentioned, is that electrical energy consumption is reduced due to the cooler air used in the condenser. This advantage becomes more significant as the difference between the ambient wet and dry bulb temperatures increase, because the cooling tower can achieve a greater wet bulb depression with the ambient air.

Because it relies on a solar collector system, the regeneration stream relies entirely on the auxiliary heater during the night and other non-solar times unless an energy storage system is provided. The collectors will have a high first cost, above the S1 option, and inclusion of a storage system will increase that cost.

5.1.3 Solar, Condenser and Auxiliary (S3)

Option S3 consists of an array of solar collectors, the vapor compression condenser and an auxiliary heater operating in series to provide the regeneration stream with thermal energy. If the air leaving the solar collectors is at or above the regeneration temperature, the air is bypassed around the condenser. The condenser is then cooled with air from a cooling tower for improved COP. This arrangement is shown in Figure 5.3.

With the bypass feature, the condenser is cooled with solar-heated air a portion of the time, with cooling tower air a portion of the time, and with ambient air (during non-solar hours) a portion

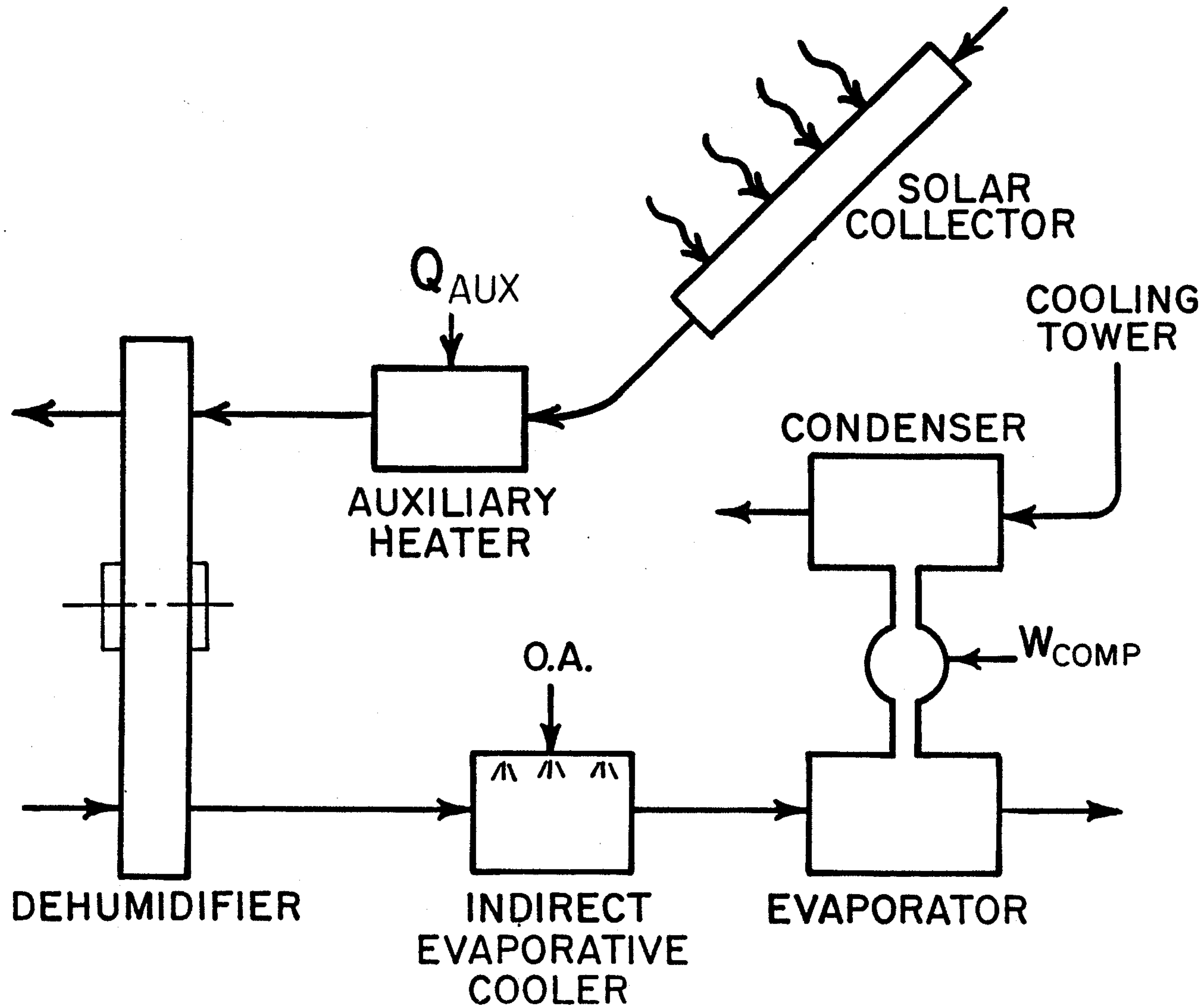


Figure 5.2 Regeneration option S2--Solar and auxiliary

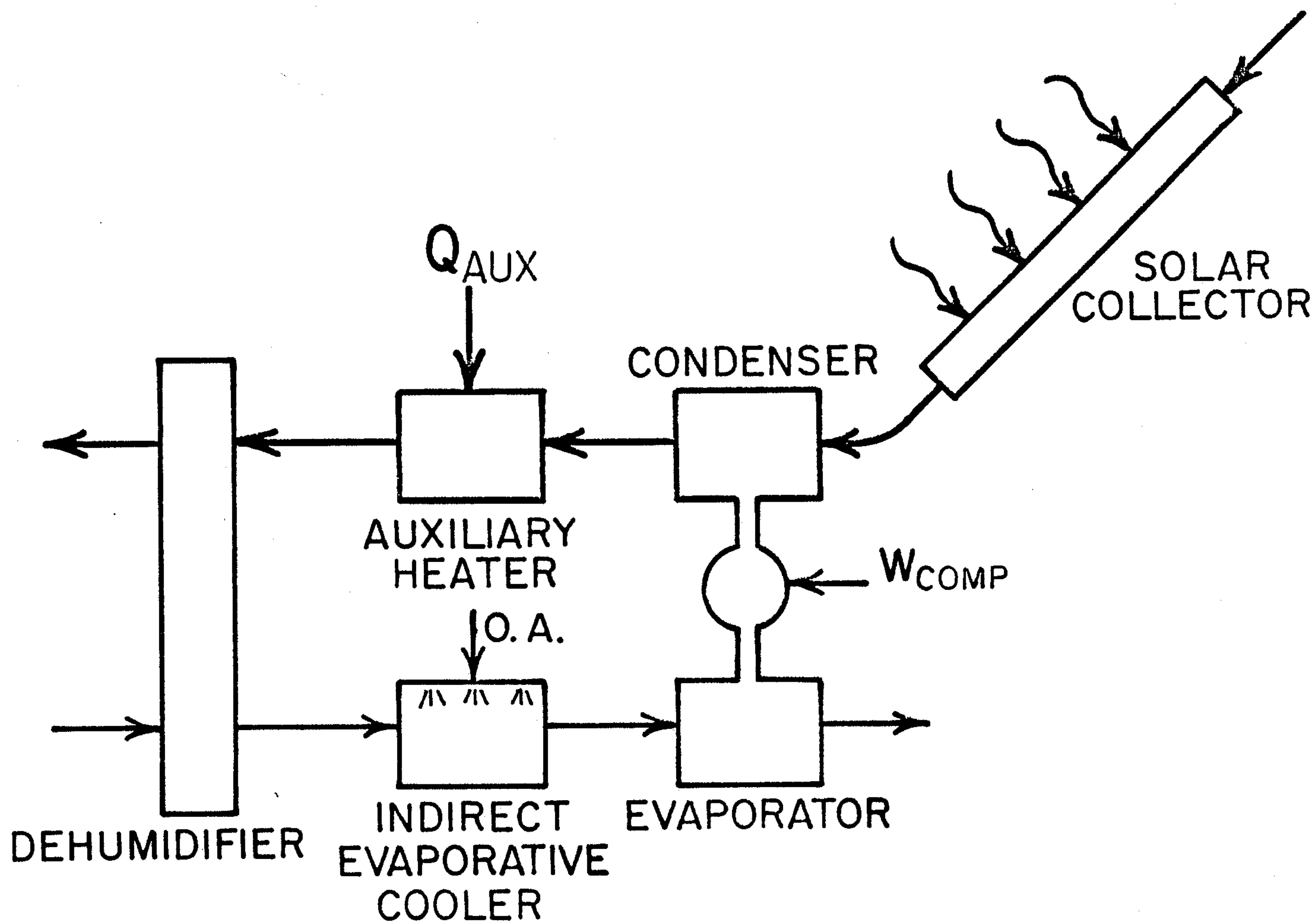


Figure 5.3 Regeneration option S3--Solar, condenser and auxiliary

of the time. The fractions of these times will depend on the climate, the load and the operating point of the system. Because of all this, the COP may or may not be higher than that of the S1 option, and the electrical energy consumption may increase or decrease.

This option reduces the amount of auxiliary energy required because both the solar and the condenser heat are used. However, like option S2, it has a high initial cost due to the solar collectors and any accompanying storage system.

5.1.4 Condenser, Solar and Auxiliary (S4)

Option S4 is similar in appearance to option S3, except that the positions of the condenser and the solar collectors have been reversed. Condenser heat is added to ambient air before being passed to the collector array. The regeneration stream bypasses the collector array when the collector losses exceed the gains, as during nighttime operation. A schematic of option S4 is shown in Figure 5.4.

The condenser is always cooled with ambient air, as in option S1, and thus has the same vapor compression COP as option S1 (and the same electrical energy usage). But with the addition of the collector array, the amount of auxiliary energy required is reduced. The economic question is whether this reduced auxiliary energy use is worth the cost of the solar collectors.

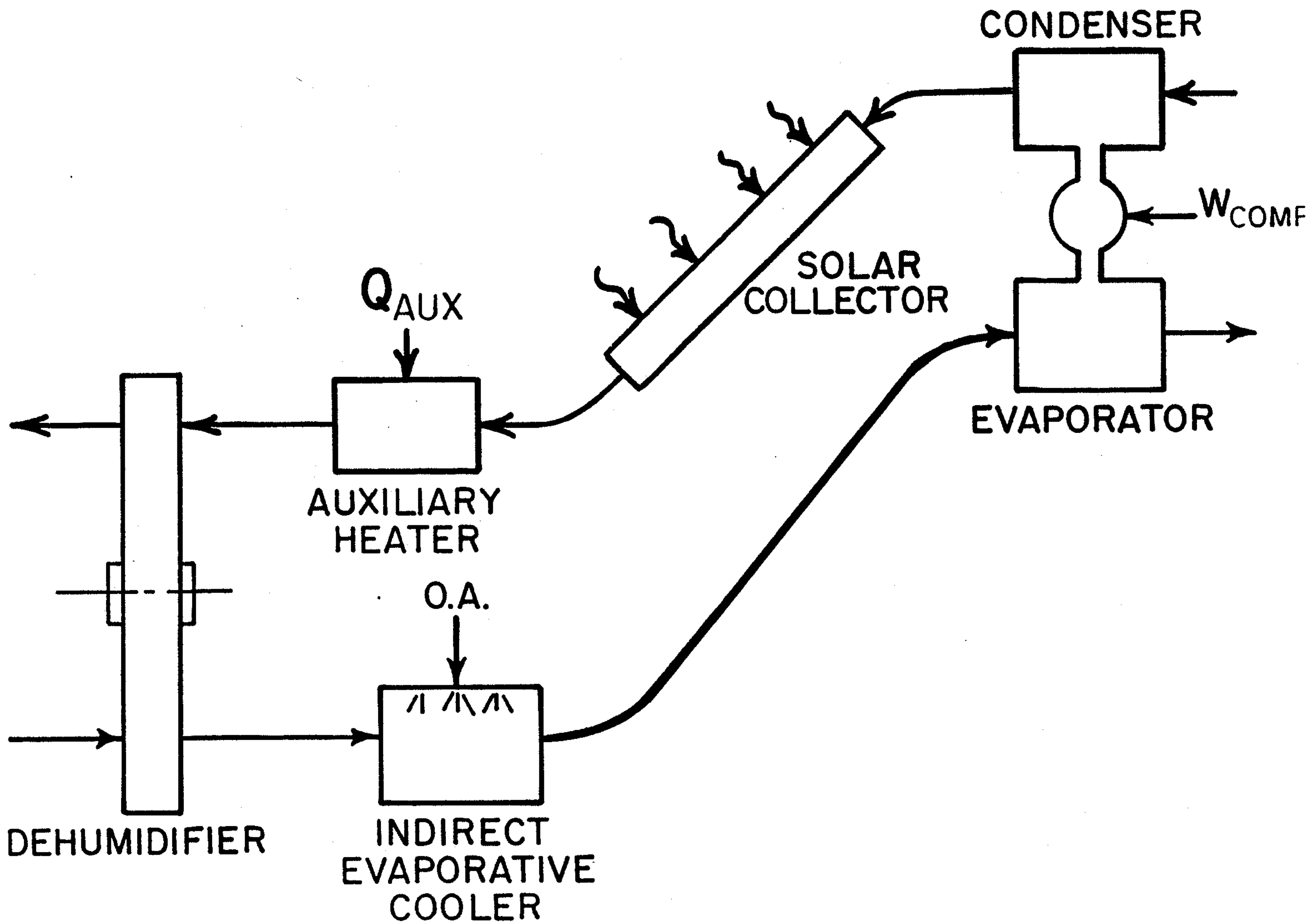


Figure 5.4 Regeneration option S4--Condenser, solar and auxiliary

CHAPTER 6 ESTIMATES OF ANNUAL PERFORMANCE OF HYBRID AIR CONDITIONING SYSTEMS

In Chapter 4, the characteristics of the hybrid system were examined while operating under restricted load and building conditions. While this is helpful in understanding the system operation, the full potential of hybrid systems cannot be determined without observing its performance under actual operating conditions. In this chapter, the results of annual simulations for the four hybrid system options (S1-S4), as described in Chapter 5, are presented for a variety of U.S. locations. These results will be compared with the annual simulation of a conventional vapor compression air conditioning system operating in the same location.

6.1 Building Model

The building selected for simulation purposes is the same structure used by Ottenstein [16] in his work with multizone buildings. The building is three stories, measuring 30m x 30m x 11m, for a total floor area of 2700 m^2. It is a flat-roofed, multizone, heavily-constructed structure comprised of six zones as shown in Figure 6.1. The floors and interior walls are considered adiabatic, so that the lower interior zone only has internally generated heat gains. The outer walls have the properties of ASHRAE Wall No. 1, with 20% window area on each face. The roof is described by ASHRAE Roof No. 8.

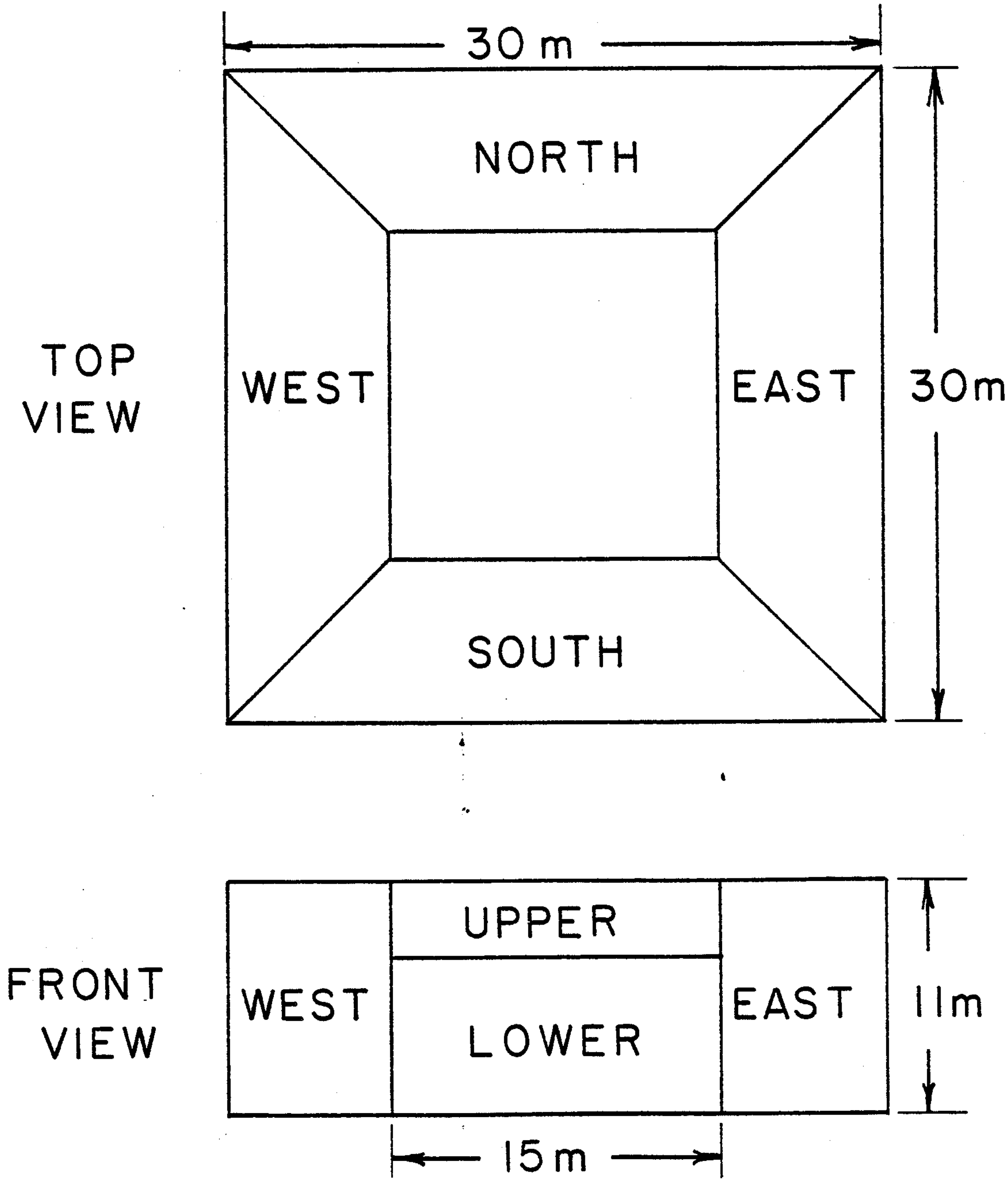


Figure 6.1 Schematic of multizone building

The building has a variable air volume (VAV) air handling system, and uses a 30-ton vapor compression unit as part of its air conditioning system.

6.2 Building Load Model

The system simulation results were obtained by using the transient simulation program TRNSYS [17]. Standard wall and room components (Type 17 and Type 19) were used to generate the building loads. In calculating these loads, the TRNSYS energy rate control option was specified. This control assumes that the computed load at each timestep is fully met by the air conditioning system. This allows the load to be calculated independent of the cooling system. The annual hourly cooling loads for a building need only be calculated once and stored in a file. This load data file can be used repeatedly to examine many air conditioning systems.

The alternative to energy rate control is temperature level control. Temperature level control provides a more realistic simulation of building load-cooling system interaction because it accounts for the dynamics of system operation and the finite capacity of cooling systems. However, energy rate control results in greater numerical stability, reduces the computational time and allows for pre-processing of the loads, making it much cheaper and easier to use. It was chosen for these reasons.

The building cooling load is calculated based on the Room Transfer Function Method described in ASHRAE [18]. This provides a time-dependent sum of solar, conduction and internally generated heat gains within a given zone. The room temperature floats within a specified deadband in response to the building load. The load is zero while the room temperature is within the deadband setting. If the room temperature exceeds the maximum setting, the cooling load is calculated which, when met, will maintain the room temperature at the maximum setting. The ASHRAE comfort zone [10] was used to determine the deadband temperature range. The latent cooling load is calculated based on a constant room humidity ratio of .012 kg/kg. However, the air conditioning system models allow the humidity level to float within the ASHRAE comfort zone boundaries.

Figure 6.2 shows the lighting and equipment, people, and ventilation schedules supplied to TRNSYS for calculation of internal loads. The maximum lighting and electrical load of 4 W/ft^2 is within the range of typical building loads as stated by ASHRAE [11]. The people and ventilation schedules also follow ASHRAE guides. It was assumed that the building is occupied for 365 days a year. Weather data for the locations studied was supplied by Typical Meteorological Year (TMY) weather tapes [19].

6.3 Component Parameters for System Simulations

The solar collector model used to predict the behavior of the

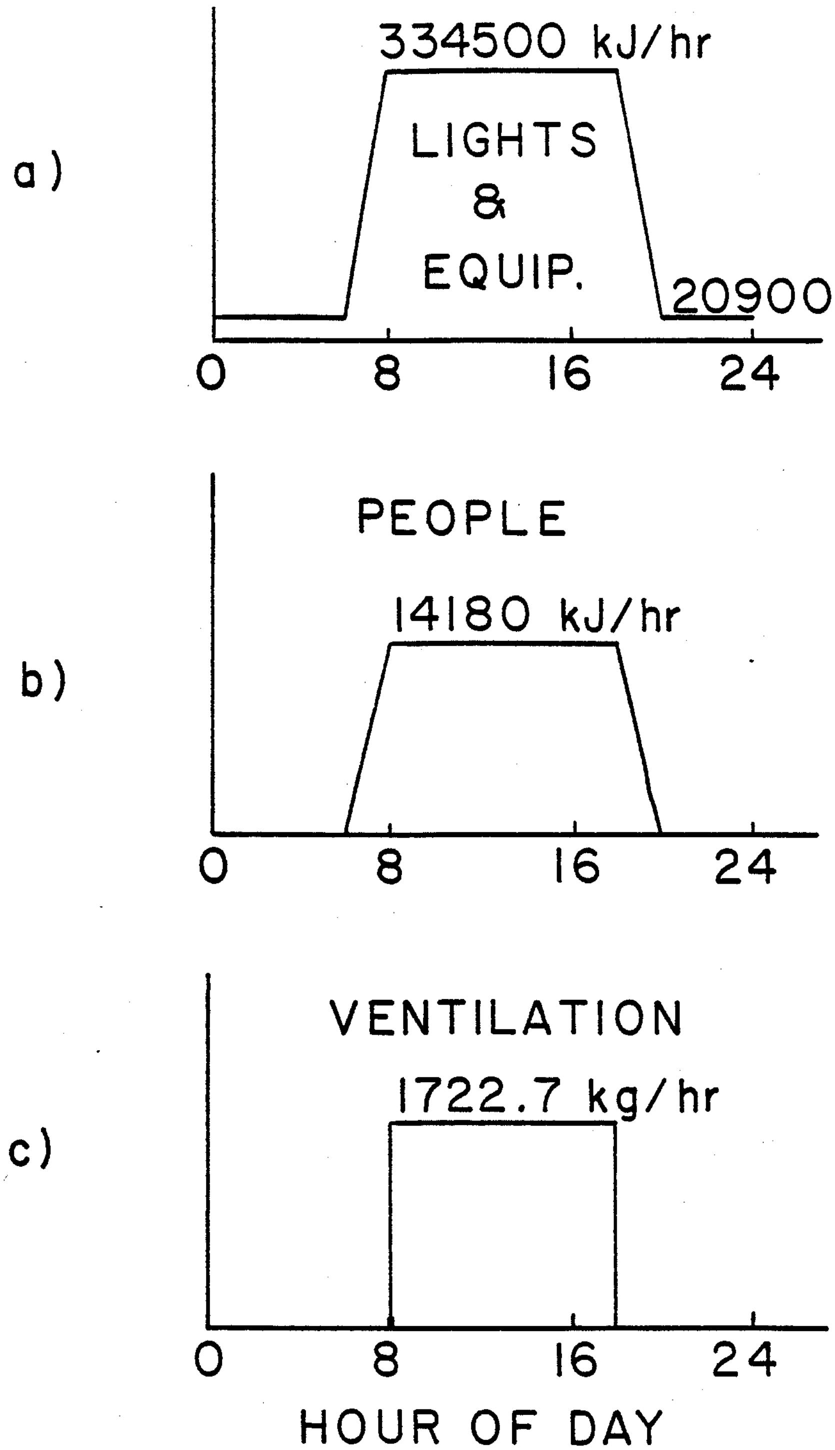


Figure 6.2 a) Simulation lighting and equipment schedule
b) Simulation building occupancy schedule
c) Simulation ventilation schedule

flat plate collector array is a modification[1] of the TRNSYS Type 1 flat plate collector model. Using mode 1, five collector parameters (aside from the collector area) help determine the array performance. N_S is the number of collectors in series, F' is the collector efficiency factor, U_L is the collector loss coefficient, τ is the transmittance of the collector covers, and α is the absorbtance of the collector plate. These parameters, as well as those chosen for the building and other components, are given in Table 6.1.

The collector parameters were selected from Ottenstein's work with solar collectors in multizone buildings [16]. All of the components parameters reflect current technological feasibility.

6.4 Method of Presenting Annual Performance Results

Performance of the hybrid and conventional vapor compression air conditioning systems are measured in terms of the amount of electrical and thermal energy they consume. Comparing these systems on the basis of their energy consumption is misleading because thermal and electrical energies have different values, both economically and thermodynamically. In studying hybrid systems, Sheridan and Mitchell [8] and Howe, et al. [21, 22] used the concept of re-

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[1] The modification is for control purposes only and does not change the performance of the collectors.

TABLE 6.1

Building and Component Parameters for Simulation Studies

Dehumidifier,	ε_{F_1}	= .08
	ε_{F_2}	= .95
Indirect Evaporative Cooler,	ε_{IEC}	= .90
Solar Collector,	F'	= .95
	U_L	= 4.0 W/m^2-°C
	τ	= .82
	α	= .94
	N_S	= 4
Fan,	$\eta_{f,s}$	= .70
	$\eta_{d,m}$	= .70
Component Pressure Drops,	ΔP_{DEH}	= 250 Pa
	ΔP_{IEC}	= 150 Pa
	ΔP_{EVAP}	= 60 Pa
	ΔP_{COND}	= 60 Pa @ Rated flow rate of MBASE = 23300 kg/hr
	$\Delta P_{AUX\ HTR}$	= 100 Pa
	ΔP_{SOLAR}	= 50 Pa
	ΔP_{DUCT}	= 500 Pa
	ΔP_{FILTER}	= 32 Pa
	ΔP_{INTAKE}	= 50 Pa

MREG = .60 x MSYS

Building thermal capacitance = 6.24 x 10^5 kJ/°C

Building moisture capacitance = 59450 kg (5 times mass of bldg. air) source energy to account for the fact that there is a conversion efficiency of primary fossil fuel into electricity. While this approach can be helpful, it is still not entirely meaningful as this does not reflect the true cost of these energies, which is the critical result.

In presenting the results of the hybrid and vapor compression systems, comparison will be done on an economic basis. The term annual fuel savings will be used as the annual operating cost of a vapor compression system minus the annual cost of operating the hybrid system. In equation form,

$$\begin{aligned}\text{Annual Fuel Savings (AFS)} = &\ (\text{Reduction in electrical consumption}) \times \$/\text{kWh} \\ &+ (\text{Reduction in peak demand usage}) \times \$/\text{kW} \\ &- (\text{Hybrid gas consumption}) \times \$/\text{GJ}\end{aligned} \tag{6.1}$$

The reduction in peak electrical demand usage can be a significant fraction of the annual savings, but it cannot be tied directly to the air conditioning operation because it is dependent on the building demand profile without the air conditioning system. To circumvent this problem, an effective price of electrical energy can be defined as,

$$\$/\text{kWh})_{\text{eff}} = \frac{(\text{Reduction in kWh}) \times \$/\text{kWh} + (\text{Reduction in demand}) \times \$/\text{kW}}{\text{Reduction in kWh}} \tag{6.2}$$

This essentially increases the base electrical energy cost per kilowatt-hour by incorporating the demand cost into the kilowatt-hour consumption cost. This assumes that the demand charge is directly proportional to the total electrical energy consumption. Equation 6.1 can now be rewritten as,

$$\text{Annual Fuel Savings} = (\text{Reduction in kWh consumption}) \times \$/\text{kWh})_{\text{eff}} - (\text{Hybrid gas consumption}) \times \$/\text{GJ} \qquad (6.3)$$

In the following annual simulation results, the energy price for the economic analyses are:

$\$/\text{kWh}_{\text{eff}} = \$.06/\text{kWh}$

$\$/\text{GJ} = \$6.00/\text{GJ}$

6.5 Results for Phoenix

The results of the annual simulations in Phoenix for the four hybrid systems S1-S4 (discussed in Chapter 5) and the vapor compression systems are listed in Table 6.2. Those hybrid systems which use solar energy for regeneration were each run with four collector areas. The table gives the compressor and fan electrical energy consumption for each system. Also listed is the auxiliary natural gas consumption for the hybrid systems with no storage and with an infinite storage system. These provide lower and upper bounds for the amount of auxiliary energy the system must

Table 6.2 Phoenix Annual Simulation Results

Hybrid System	Collector Area, m^2	Electricity (MWh) Compressor	Electricity (MWh) Fans	Natural Gas (GJ) No Storage	Natural Gas (GJ) Infinite Storage	Annual Fuel Savings ($) No Storage	Annual Fuel Savings ($) Infinite Storage
S1	--	98.0	75.1	3.9	0	9703	9726
S2-	0	81.6	75.1	452.3	452.3	7996	7996
	200	81.6	75.1	272.1	0	9077	10710
	400	81.6	75.1	245.3	0	9238	10710
	600	81.6	75.1	236.9	0	9289	10710
S3-	0	98.0	75.1	3.9	0	9703	9726
	200	86.9	75.7	3.3	0	10336	10356
	400	85.3	75.7	3.3	0	10432	10452
	600	84.8	75.7	3.3	0	10462	10482
S4-	0	98.0	75.1	3.9	0	9703	9726
	200	98.0	75.7	3.7	0	9668	9690
	400	98.0	75.7	3.6	0	9668	9690
	600	98.0	75.7	3.6	0	9668	9690
Conventional Vapor Compression	-	286.4	48.8	-	-	--	--

supply. The concept of infinite storage means that all energy from the condenser and solar collectors that is not needed immediately for regeneration can be stored for future use with no losses. The infinite storage model did not monitor the temperature at which the thermal energy was stored or re-used. Instead, it was assumed that the energy was available at whatever temperature was required. Because of possible second law repercussions, this infinite storage concept may exceed its theoretical limits. However, its practicality as a pseudo-upper bound on energy storage is still of value.

The annual fuel savings, discussed in the previous section, are given in Table 6.2 for both the no-storage and the infinite storage cases. In the following sections, each of the hybrid systems will be analyzed individually. Concluding remarks about hybrid systems in Phoenix will then be made.

6.5.1 Condenser and Auxiliary (S1)

From Table 6.2, the hybrid S1 compressor work is reduced to 34% of that of a vapor compression system. The hot, dry Phoenix climate allows the maximum benefit from the indirect evaporative cooler, which uses the low ambient wet bulb temperature as a heat sink to provide a significant fraction of the sensible cooling. The fan power consumed is 54% greater than the vapor compression system due to the added system pressure drop of the hybrid components and an additional fan required. The 3.9 GJ of natural gas used for regeneration (with no storage) represents a very small additional cost.

This indicates that the condenser is able to supply almost all of the thermal energy needed for regeneration. By storing the excess condenser thermal energy, no auxiliary energy would be needed year-round in Phoenix.

The hybrid system has an annual fuel savings of $9703 without benefit of a storage system. If an infinite storage system could be purchased, it would only save an additional $23 annually. Obviously, a storage system of any size is not practical economically for Phoenix.

The annual fuel savings of $9703 must offset the cost of the additional equipment needed for the hybrid S1 system, if it is to be an economically sound alternative to a vapor compression system. In the commercial sector, an economic analysis based on a 5-year payback is not unreasonable. To determine the value of the $9703 annual savings, a life cycle savings analysis using the P_1, P_2 method [23] is employed. The life cycle savings is defined as,

$$LCS = P_1 \times (C_{F_1} L \mathcal{F}) - P_2 (C_A A_C + C_E) \qquad (6.4)$$

where

$C_{F_1} L \mathcal{F}$ = annual fuel savings, \$ (1st year)

C_A = solar collector area-dependent costs, $\$/m^2$

A_C = collector area,[1] m^2

C_E = cost of equipment independent of collector area, \$

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[1]For the hybrid S1 option, the collector area is zero.

P_1 and P_2 are functions of many variables, including the market and fuel inflation rates. These variables must be projected into the future, and in general are not equal for all sectors of the U.S. However, the ratio of P_1 to P_2 is almost always nearly equal to the payback analysis time. By assuming P_1/P_2 = 5 and setting LCS = 0, the break-even cost of the additional equipment for the hybrid S1 system is:

$$\text{Cost of Equipment } (C_E) = 5 \times \$9703 = \$48515$$

In other words, an additional outlay of $48515 to install the hybrid S1 system will pay itself off in five years. For a nominal 30-ton cooling system, spending an additional $1617/ton of cooling is economically justifiable. To put this in perspective, vapor compression units cost on the order of $200-$500/ton. Figures such as these should provide incentive for air conditioning manufacturers to develop and produce the hybrid S1 system, as increased sales dollars would result.

6.5.2 Solar and Auxiliary (S2)

The compressor work for the hybrid S2 system is only 28% of that of the vapor compression system. This 6% improvement over the hybrid S1 option is due to the use of a cooling tower to cool the condenser for improved COP. Since the condenser heat is not used for regeneration, a large amount of natural gas is consumed because solar energy cannot replace the condenser heat. Natural gas consumption decreases from 452.3 GJ with 0 m^2 of collector area, to 236.9 GJ with 600 m^2. Table 6.2 contains these figures.

Using a similar analysis to that of the previous section, the break-even cost of additional equipment for the S2 option with no solar system (0 m^2) is,

$$C_E = 5 \times \$7996 = \$39980$$

This is about \$8500 less than the break-even equipment cost of the S1 option, even though the S2 option also requires a cooling tower. On a per ton basis, the cost of equipment is \$1333/ton, which is still attractive, but is not as economically promising as the hybrid S1 system.

With 200 m^2 of collectors added to the system, the annual fuel savings of \$10710 with infinite storage exceeds the S1 option. Using equation 6.4, the maximum cost which would make the addition of 200 m^2 of collectors economically feasible is

$$C_A = \frac{5 \times \$10710 - \$39980}{200\ m^2} = \$68/m^2$$

Unfortunately, the actual cost of installing collectors is roughly 5-7 times greater than this, making the addition uneconomical.

As shown in Table 6.2, adding additional collector area does not increase the annual fuel savings.

6.5.3 Solar, Condenser and Auxiliary (S3)

In the limit of a 0 m^2 solar collector system, the S3 option is identical to the S1 option, and the energy use for both is the same. At increased collector areas, solar energy is able to regenerate the desiccant more and more, allowing the condenser to be cooled by a cooling tower. The result is that the compressor work decreases but the fan power increases slightly due to the collector pressure drop. The annual fuel savings increases, but the allowable cost per area of collector is much too low to be economical. Table 6.3 summarizes the economic analysis for the S3 option.

6.5.4 Condenser, Solar and Auxiliary (S4)

As with the S3 option, at 0 m^2 of solar collectors, the hybrid S4 system is identical to the S1 option. As the collector area is increased, the compressor work remains constant because the condenser is always cooled with ambient air. The auxiliary natural gas consumption is reduced slightly, but not enough to increase the annual fuel savings. Actually, the annual fuel savings decrease at increased collector areas because the cost of the increased fan power (due to the added collector pressure drop) is greater than the savings in auxiliary energy.

6.5.5 Concluding Remarks About Hybrid Systems in Phoenix

The low ambient humidity ratios and high building sensible loads

TABLE 6.3

Phoenix S3 Economic Analysis Results

(No Storage)

Collector area, m^2	Break-even Equipment Cost (C_E), \$	$\frac{\text{Cost}}{\text{ton}}$	Collector Cost Per Area (C_A), \$/m^2
0 m^2	\$ 48515	\$1617/ton	---
200	48515	1617	\$16/m^2
400	48515	1617	9
600	48515	1617	6

characteristic of the Phoenix climate allow the hybrid systems to operate at relatively low regeneration temperatures, and in fact the dehumidifier is needed only about 20% of the operating time. Because of this, the condenser heat is sufficient to meet a large fraction of the required regeneration energy, so auxiliary energy requirements are small. This is why the hybrid S1 system works well in Phoenix. Solar energy is not needed. The S1 option is clearly the most economical choice in Phoenix, because of its high annual fuel savings and its low equipment costs relative to the other hybrid options.

The economic analyses for the hybrid systems was based on an effective electrical energy cost of \$.06/kWh and a gas price of \$6/GJ. Since gas and electricity prices vary widely, and because the effective price of electricity will vary from building to building, and from region to region, it is helpful and interesting to plot the annual fuel savings of the S1 option as a function of the two energy prices. Figure 6.3 shows this plot. The annual fuel savings is shown to be a much stronger function of the electrical energy price than the gas price. One can conclude that even if the trend toward rapidly increasing gas prices continue, the hybrid S1 system will be cost-effective.

6.6 Results for Miami

The climate in Miami is warm and humid, in contrast to Phoenix, which is hot and dry. Due to the longer cooling season in Miami,

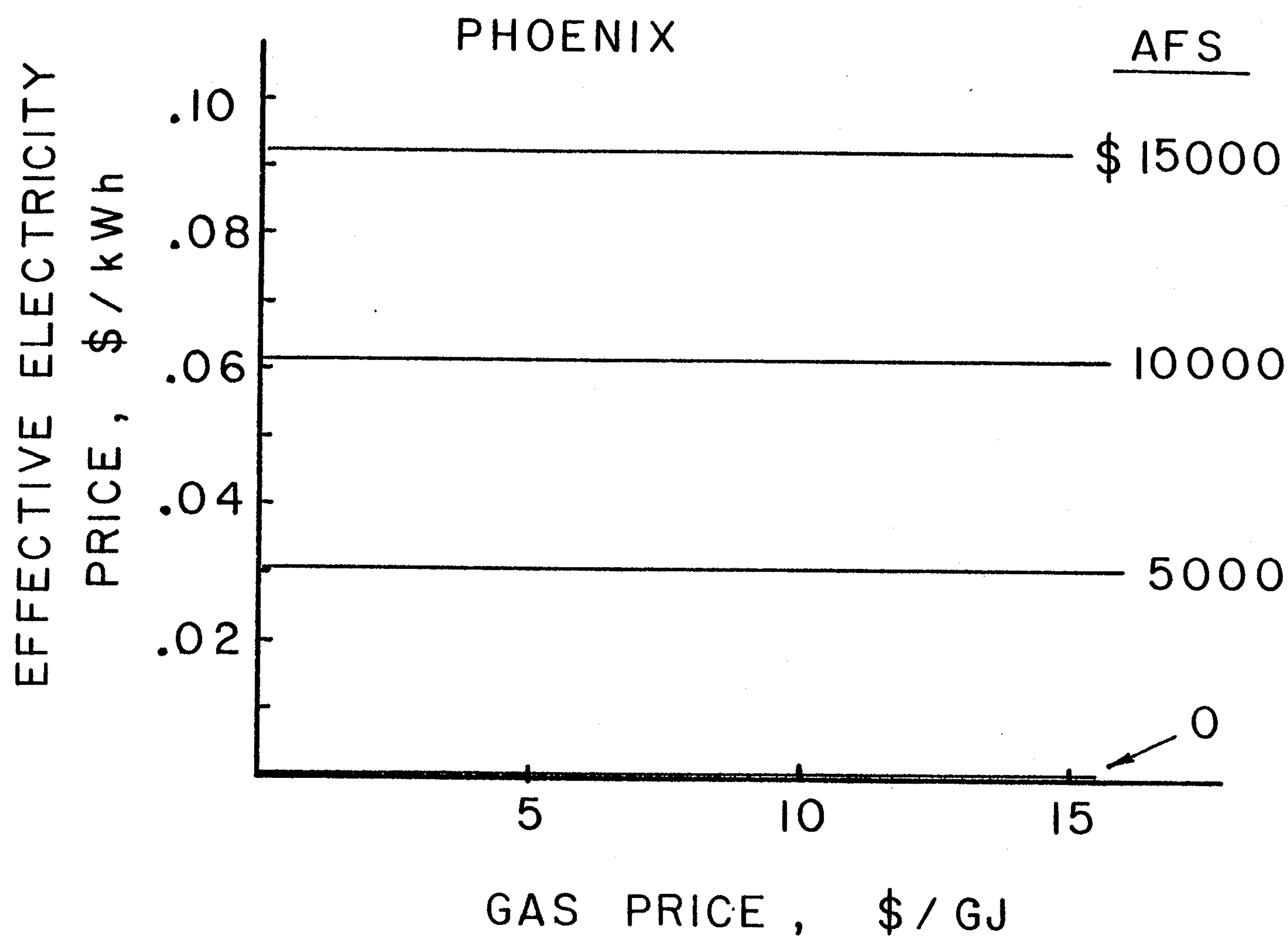

Figure 6.3 Annual fuel savings (AFS) for the hybrid S1 system in Phoenix

the sensible load in our building is nearly identical in the two cities. However, because the ambient air for ventilation is much moister in Miami than in Phoenix, the latent cooling load is significantly greater in Miami. Because of this, the energy consumption of the hybrid and vapor compression air conditioning systems in Miami is greater than in Phoenix. The results for Miami are summarized in Table 6.4. The economic analysis results for the hybrid systems are found in Table 6.5. These results will be discussed in the following sections.

6.6.1 Condenser and Auxiliary (S1)

From Table 6.4, the hybrid S1 system shows large annual fuel savings for both the storage and the non-storage options. Due to the higher ambient humidity ratios and the large latent loads, the desiccant dehumidifier is required to operate nearly 100% of the time, and at relatively high regeneration temperatures. Without a storage system, the condenser cannot provide all of the regeneration energy, and requires a large amount of auxiliary energy. However, with an infinite storage system, the auxiliary heater is not needed. Table 6.5 indicates that one can purchase the hybrid equipment at a price of up to $1536 per ton of cooling capacity (with no storage) and still have a positive life cycle savings. That is, even if the cost of the hybrid equipment is 3-7 times greater per ton than a vapor compression unit, it would still be economical to purchase it. A storage system at an additional cost

Table 6.4 Miami Annual Simulation Results

System	Collector Area, m^2	Electricity (MWh) Compressor	Electricity (MWh) Fans	Natural Gas (GJ) No Storage	Natural Gas (GJ) Infinite Storage	Annual Fuel Savings ($) No Storage	Annual Fuel Savings ($) Infinite Storage
S1	-	172.3	65.8	231.3	0	9214	10602
S2-	0	163.8	65.8	3161.4	3161.4	-7856	-7856
	200	163.8	65.8	2247.1	1886.6	-2371	-220
	400	163.8	65.8	1960.0	884.6	-648	5804
	600	163.8	65.8	1854.3	66.9	-14	10711
S3-	0	172.3	65.8	231.3	0	9214	10602
	200	182.9	66.4	195.6	0	8756	9930
	400	176.0	66.4	191.6	0	9194	10344
	600	173.8	66.4	190.3	0	9334	10476
S4-	0	172.3	65.8	231.3	0	9214	10602
	200	172.3	66.4	215.9	0	9271	10566
	400	172.3	66.4	212.0	0	9294	10566
	600	172.3	66.4	210.6	0	9302	10566
Conventional Vapor Compression	-	372.5	42.3	-	-	-	-

Table 6.5 Hybrid System Economic Analysis Results for Miami

System	Collector Area, m^2	No Storage: Equipment Cost (C_E), \$	No Storage: Cost per ton, \$/ton	No Storage: Collector Cost (C_A), \$/m^2	Infinite Storage: Equipment Cost (C_E) \$	Infinite Storage: Cost per ton, \$/ton	Infinite Storage: Collector Cost (C_A), \$/m^2
S1	-	46070	1536	-	53010	1767	-
S2	0	***	***	-	***	***	-
	200	***	***	137	***	***	191
	400	***	***	90	***	***	171
	600	***	***	65	***	***	155
S3	0	46070	1536	-	53010	1767	-
	200	46070	1536	***	53010	1767	***
	400	46070	1536	***	53010	1767	***
	600	46070	1536	1	53010	1767	***
S4	0	46070	1536	-	53010	1767	-
	200	46070	1536	1	53010	1767	***
	400	46070	1536	1	53010	1767	***
	600	46070	1536	1	53010	1767	***

***Negative cost.

of \$6940, or \$231/ton, would also be justified.

6.6.2 Solar and Auxiliary (S2)

The hybrid S2 system reduces the electrical energy consumption of the compressor by 8.5 MWh over the S1 option, but uses an additional 2930.1 GJ of natural gas at 0 m^2 of solar collectors and no storage. This system increases the annual fuel cost (as shown by the negative annual fuel savings in Table 6.4) over the vapor compression system, and the negative equipment cost (***) shown in Table 6.5 says that even if the hybrid S2 equipment were free, it would not be economical. The addition of solar collectors significantly reduces the amount of natural gas consumed, but not enough to show a positive annual fuel savings. If an infinite storage system is added, an expected annual fuel savings of \$10711 results from 600 m^2 of collectors. However, this is not a large enough savings to justify the cost of equipment, collectors and storage.

The large difference between the storage and the no-storage results indicates that the solar energy system may be able to meet the regeneration requirements during the daytime, but that much auxiliary energy is used at night when there are no solar gains. A building operating schedule which shuts down the air conditioning system at night will enhance the prospects for the hybrid S2 system in Miami, but it is unlikely that it will prove more economical than the hybrid S1 system.

6.6.3 Solar, Condenser and Auxiliary (S3)

The results for the hybrid S3 system are found in Tables 6.4 and 6.5. The compressor electrical energy consumption found in Table 6.4 is plotted as a function of collector area in Figure 6.4 to illustrate an interesting characteristic of the S3 option in humid climates. As the solar system collector area is increased from 0 m^2, the compressor work begins to increase! It reaches a maximum somewhere between 0 m^2 and 200 m^2 of collector. Beyond this point, the work decreases as expected. The reason for the initial increase is as follows: at small collector areas, increasing the area will increase the air temperature leaving the array, but it is not sufficiently high to meet the large regeneration temperatures seen in humid climiates. The condenser must be used to supply thermal energy, then. The air temperature entering the condenser increases with collector area, increasing the work required. Above some threshold collector area, the solar system can supply the required regeneration energy often enough for the condenser to use the bypass option of cooling tower air for improved performance.

Because of this characteristic of the S3 option, it is not economical to use this system in humid climates.

6.6.4 Condenser, Solar and Auxiliary

The results for the hybrid S4 system are found in Tables 6.4

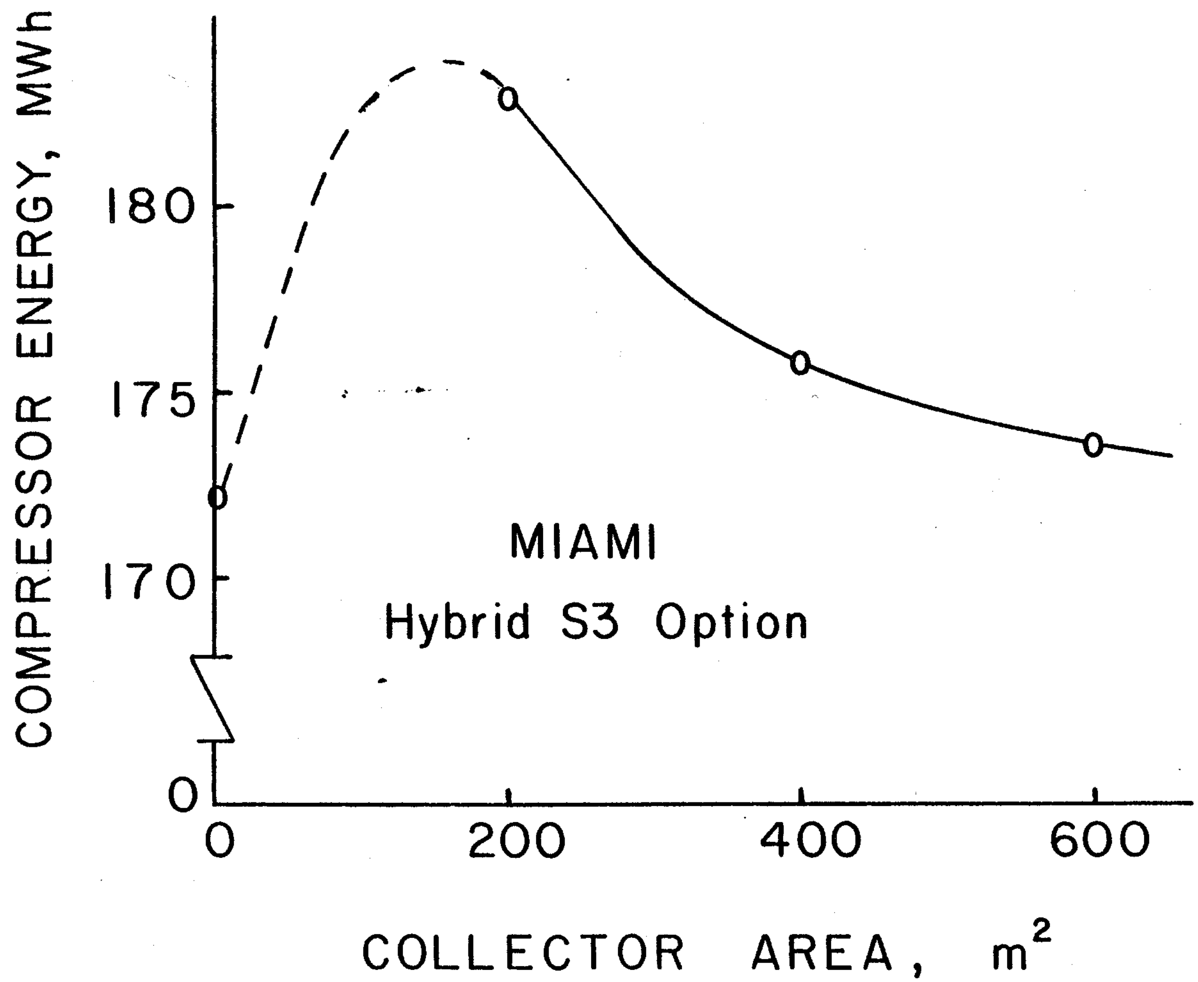

Figure 6.4 Hybrid S3 compressor electrical energy consumption for Miami

and 6.5. As the solar collector area is increased, the amount of auxiliary energy required is decreased, but only slightly. The condenser alone is able to supply most of the regeneration energy during the daylight hours, and therefore the solar system is of little use unless a storage system is also used.

As with the S3 option, the hybrid S4 system can afford a high initial equipment cost (C_E), but it is uneconomical to add any collector area. Unlike the S3 option, there is a slight increase in annual fuel savings with increased collector area (and no storage), but this is too small to justify the collector expense.

6.6.5 Concluding Remarks About Hybrid Systems in Miami

While the auxiliary natural gas requirements are much greater in Miami than in Phoenix, they are not large enough to warrant the purchase of a solar collector system to reduce these requirements. Hybrid options S2-S4 use solar energy, and are all shown to be uneconomical.

The hybrid S1 system, which uses only condenser heat with a backup auxiliary heater for regeneration, is seen to be a very attractive system in Miami. Along with high annual fuel savings, it has a low first cost of equipment relative to the other hybrid systems.

A plot of the Miami annual fuel savings for the S1 no-storage hybrid system is shown in Figure 6.5 as a function of the electric and gas prices. Compared to the same plot for Phoenix (Fig. 6.3),

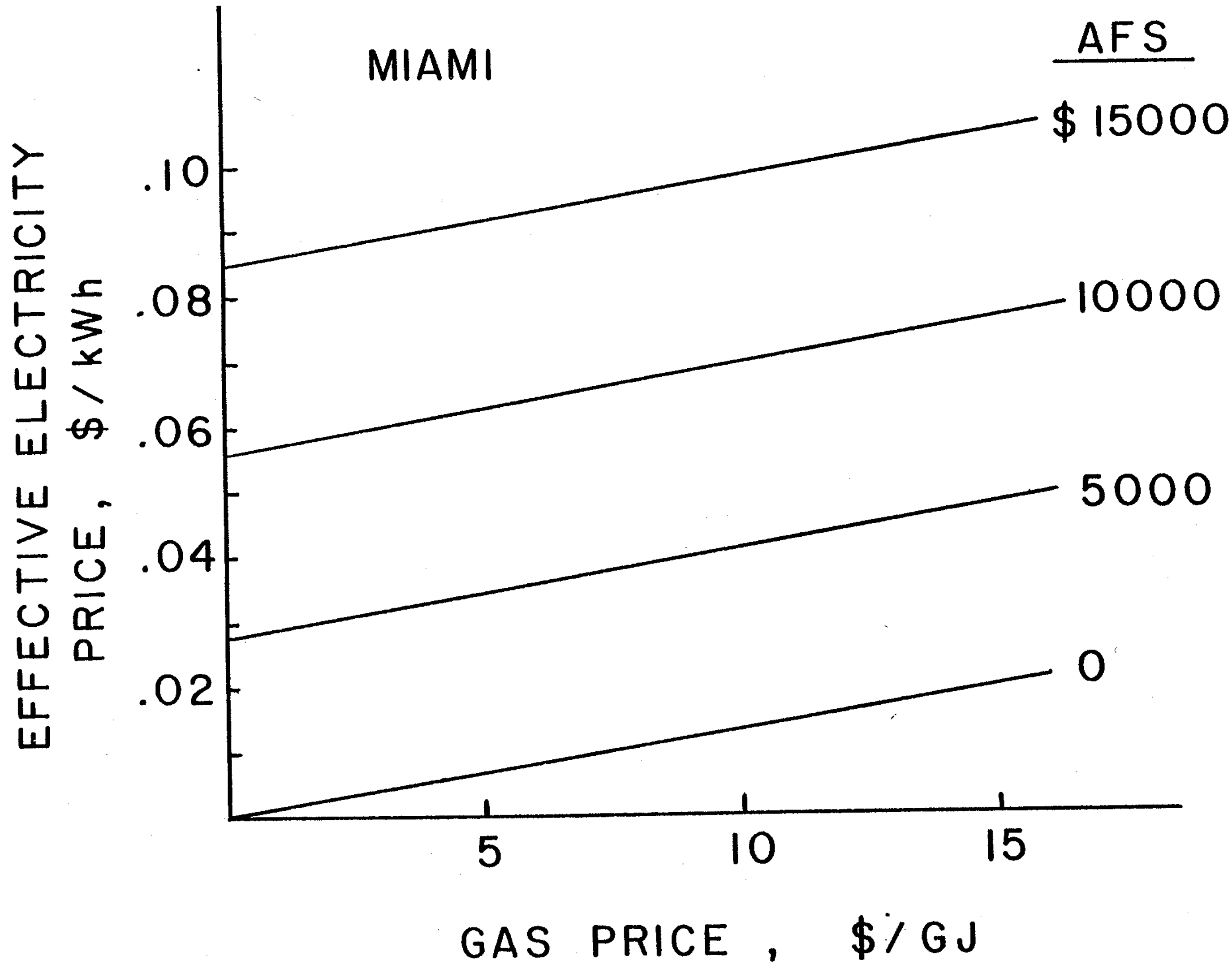

Figure 6.5 Annual fuel savings (AFS) for the hybrid S1 system in Miami

the Miami annual fuel savings is a much stronger function of the gas price. This is because it uses a large amount of natural gas. A large increase in the price of gas would reduce the attractiveness of this system, while any increase in the effective electrical energy price would make it even more favorable compared to a vapor compression system.

6.7 Results for Other Cities

Simulations were run for five other U.S. cities varying in climate and location. The cities are: Charleston, SC, Ft. Worth, TX, Fresno, CA, Madison, WI and Sterling, VA. Tables A.1 through A.10 in Appendix A contain the performance and economic analysis results for each of these cities. The pattern of their results closely match those of Phoenix and Miami. In all cases, the hybrid S1 system is the best of the hybrid options. The solar energy systems of options S2-S4 cannot 'pay' for themselves in fuel savings.

Table 6.6 shows the annual fuel savings for the no-storage S1 option in the five locations. Even in Madison, Wisconsin, which has a relatively short cooling season, the hybrid S1 system yields a $2330 annual savings over a conventional vapor compression air conditioning system. As expected, the annual fuel savings increases for cities with larger cooling loads. The more humid climates tend to benefit more from the hybrid S1 system than do the drier climates.

TABLE 6.6

Annual Fuel Savings of No-Storage Hybrid S1 System for Various Locations

Location	Annual Fuel Savings, $
Charleston, SC	5104
Ft. Worth, TX	6737
Fresno, CA	5094
Madison, WI	2330
Sterling, VA	3432

CHAPTER 7 CONCLUSIONS AND RECOMMENDATIONS

The following is a discussion of the conclusions and recommendations for the commercial hybrid air conditioning systems.

7.1 Conclusions

Compared to a conventional vapor compression air conditioning system, hybrid systems reduce the compressor work of the mechanical refrigeration unit from 55-70%. The fan power will increase by 50-55% normally, due to the added components pressure drop and an additional system fan. The net effect is an electrical energy reduction of from 40% to 50%.

Of the four hybrid systems studied, the three which use solar energy for regeneration (S2-S4) are not economically attractive. In many cases, their use can result in high annual savings, but not sufficiently high to justify the cost of the solar collectors. That is, the concept behind these systems may be good, but the cost of solar collectors makes their use impractical in these applications.

The hybrid S1 system, the standard hybrid system which uses the condenser thermal energy along with an auxiliary heater for regeneration, is superior to a conventional vapor compression system in terms of energy consumption, operating cost and climate control. Larger building loads will yield larger savings. Humid climates are better suited to the hybrid system, although the savings in dry climates are also quite high.

System performance is significantly affected by the performance of each of the individual components. Poor dehumidifier performance will increase the need for auxiliary energy. Poor indirect evaporative cooler performance will increase the need for vapor compression cooling, which will in turn increase the electrical energy consumption but reduce the auxiliary energy requirements. Poor vapor compression performance will have the same final effect as poor indirect evaporative cooler performance.

Performance maps are found to be an effective way of analyzing the behavior of hybrid systems under a whole range of ambient conditions. They are inexpensive too, relative to running annual simulations.

The hybrid system has advantages for all concerned parties. For the commercial building owner, the air conditioning operational expenses will be reduced significantly.

For the electric utilities, the hybrid system will help reduce peak electricity demand on their generators, as air conditioning systems operate primarily during peak demand times.

For the gas utilities, the hybrid system is a way to level out the gas demand over the entire year. Typically, gas consumption is drastically reduced during the summer months. Hybrid systems will create a whole new summer market for natural gas.

And finally, air conditioning equipment manufacturers stand to gain from hybrid systems because the initial cost of these systems is greater than that of a conventional vapor compression system. With greater dollars being spent on equipment, greater sales profits can be realized.

7.2 Recommendations

The hybrid system developed in this work has some simplifications built into its components that can be improved with further work.

Presently the desiccant dehumidifier is modeled as an infinite capacity machine. A model which bases the capacity on the heat and mass transfer rates, as well as the capacitance and hysterisis effects, is more realistic.

The feasibility of manufacturing an indirect evaporative cooler of the size required for commercial applications should be studied. The cooler is such an important part or the hybrid system. Therefore it must be certain that an efficient cooler of this size can be manufactured with no problems.

The system building controller is based on the principle of energy rate control. Simulations should be repeated using temperature level control, which is more realistic, to see the effect on system performance.

Peak electrical demand for both the hybrid and conventional systems should be examined. It is important to determine the magnitude of the electrical energy reduction at peak demand to more accurately assess the value of hybrid systems in reducing demand charges and reducing electric utility peaks.

The hybrid system uses much less mechanical refrigeration than a conventional vapor compression system. Because of this, the hybrid system can use a smaller vapor compression unit, which will reduce the initial equipment cost. This appropriate vapor compression size should be determined for its first cost and performance consequences.

Finally, it is necessary to determine the probable cost of all the hybrid system equipment in order to see if their annual fuel savings will justify the initial expense.

APPENDIX A

Annual Performance and Economic Analysis Results For:

1. Charleston, SC
2. Ft. Worth, TX
3. Fresno, CA
4. Madison, WI
5. Sterling, VA

Table A.1 Charleston Annual Simulation Results

System	Collector Area, m^2	Electricity (MWh) Compressor	Fans	Natural Gas (GJ) No Storage	Infinite Storage	Annual Fuel Savings ($) No Storage	Infinite Storage
S1	-	95.6	46.5	144.3	0	5104	5970
S2-	0	91.1	46.5	1755.7	1755.7	-4294	-4294
	200	91.1	46.5	1214.9	720.7	-1049	1916
	400	91.1	46.5	1062.9	0	-137	6240
	600	91.1	46.5	1004	0	216	6240
S3-	0	95.6	46.5	144.3	0	5104	5970
	200	100.1	46.9	118.4	0	4966	5676
	400	96.8	46.9	116.6	0	5176	5874
	600	95.7	46.9	116.0	0	5244	4940
S4-	0	95.6	46.5	144.3	0	5104	4970
	200	95.6	46.9	129.7	0	5168	5946
	400	95.6	46.9	127.7	0	5180	5946
	600	95.6	46.9	127.3	0	5182	5946
Conventional Vapor Compression	-	211.4	30.2	-	-		

Table A.2 Hybrid System Economic Analysis Results for Charleston, SC

System	Collector Area, m^2	No Storage: Equipment Cost (C_E), $	No Storage: Cost per Ton, $/ton	No Storage: Collector Cost (C_A), $\$/m^2$	Infinite Storage: Equipment Cost (C_E), $	Infinite Storage: Cost per Ton, $/ton	Infinite Storage: Collector Cost (C_A), $\$/m^2$
S1	-	25520	851	-	29850	995	-
S2	0	***	***	-	***	***	-
	200	***	***	81	***	***	155
	400	***	***	52	***	***	132
	600	***	***	38	***	***	88
S3	0	25520	851	-	29850	995	-
	200	25520	851	***	29850	995	***
	400	25520	851	1	29850	995	***
	600	25520	851	1	29850	995	***
S4	0	25520	851	-	29850	995	-
	200	25520	851	2	29850	995	***
	400	25520	851	1	29850	995	***
	600	25520	851	1	29850	995	***

***Negative cost.

Table A.3 Ft. Worth Annual Simulation Results

System	Collector Area, m^2	Electricity (MWh) Compressor	Electricity (MWh) Fans	Natural Gas (GJ) No Storage	Natural Gas (GJ) Infinite Storage	Annual Fuel Savings ($) No Storage	Annual Fuel Savings ($) Infinite Storage
S1	-	101.8	53.3	69.1	0.1	6737	7151
S2-	0	93.2	53.3	1562.9	1562.9	-1709	-1709
	200	93.2	53.3	1012.1	426.4	1595	5110
	400	93.2	53.3	883.4	0	2368	7668
	600	93.2	53.3	836.4	0	2650	7668
S3-	0	101.8	53.3	69.1	.1	6737	7151
	200	105.4	53.7	53.4	0	6592	6912
	400	100.5	53.7	51.9	0	6895	7206
	600	99.0	53.7	51.1	0	6989	7296
S4-	0	101.8	53.3	69.1	.1	6737	7151
	200	101.8	53.7	61.3	0	6760	7128
	400	101.8	53.7	60.4	0	6766	7128
	600	101.8	53.7	60.1	0	6767	7128
Conventional Vapor Compression	-	239.7	34.6	-	-		

Table A.4 Hybrid System Economic Analysis Results for Ft. Worth, TX

		No Storage			Infinite Storage		
System	Collector Area, m^2	Equipment Cost (C_E), $	Cost per Ton, $/ton	Collector Cost (C_A), $\$/m^2$	Equipment Cost (C_E), $	Cost per Ton, $/ton	Collector Cost (C_A), $\$/m^2$
S1	-	33685	1123	-	35755	1192	-
S2	0	***	***	-	***	***	-
	200	***	***	83	***	***	170
	400	***	***	51	***	***	117
	600	***	***	36	***	***	78
S3	0	33685	1123	-	35755	1192	-
	200	33685	1123	***	35755	1192	***
	400	33685	1123	2	35755	1192	1
	600	33685	1123	2	35755	1192	1
S4	0	33685	1123	-	35755	1192	-
	200	33685	1123	1	35755	1192	***
	400	33685	1123	0	35755	1192	***
	600	33685	1123	0	35755	1192	***

***Negative cost.

Table A.5 Fresno Annual Simulation Results

System	Collector Area, m^2	Electricity (MWh) Compressor	Electricity (MWh) Fans	Natural Gas (GJ) No Storage	Natural Gas (GJ) Infinite Storage	Annual Fuel Savings ($) No Storage	Annual Fuel Savings ($) Infinite Storage
S1	-	53.2	55.2	0	0	5094	5094
S2-	0	45.8	55.2	155.1	155.1	4607	4607
	200	45.8	55.2	8.39	0	5035	5538
	400	45.8	55.2	80.6	0	5054	5538
	600	45.8	55.2	79.0	0	5064	5538
S3-	0	53.2	55.2	0	0	5094	5094
	200	46.9	55.6	0	0	5448	5448
	400	46.7	55.6	0	0	5460	5460
	600	46.6	55.6	0	0	5466	5466
S4-	0	53.2	55.2	0	0	5094	5094
	200	53.2	55.2	0	0	5070	5070
	400	53.2	55.6	0	0	5070	5070
	600	53.2	55.6	0	0	5070	5070
Conventional Vapor Compression	-	157.4	35.9	-	-	-	-

Table A.6 Hybrid System Economic Analysis Results for Fresno, CA

System	Collector Area, m^2	No Storage: Equipment Cost (C_E), \$	No Storage: Cost per Ton, \$/ton	No Storage: Collector Cost (C_A), \$/m^2	Infinite Storage: Equipment Cost (C_E), \$	Infinite Storage: Cost per Ton, \$/ton	Infinite Storage: Collector Cost (C_A), \$/m^2
S1	-	25470	849	-	25470	849	-
S2	0	23035	768	-	23035	768	-
	200	23035	768	11	23035	768	23
	400	23035	768	6	23035	768	12
	600	23035	768	4	23035	768	8
S3	0	25470	849	-	25470	849	-
	200	25470	849	9	25470	849	9
	400	25470	849	5	25470	849	5
	600	25470	849	3	25470	849	3
S4	0	25470	849	-	25470	849	-
	200	25470	849	***	25470	849	***
	400	25470	849	***	25470	849	***
	600	25470	849	***	25470	849	***

***Negative cost.

Table A.7 Madison Annual Simulation Results

System	Collector Area, m^2	Electricity (MWh) Compressor	Electricity (MWh) Fans	Natural Gas (GJ) No Storage	Natural Gas (GJ) Infinite Storage	Annual Fuel Savings ($) No Storage	Annual Fuel Savings ($) Infinite Storage
S1	-	32.7	28.0	19.7	0	2330	2448
S2-	0	30.8	28.0	487.7	487.7	-364	-364
	200	30.8	28.0	296.6	0	782	2562
	400	30.8	28.0	255.6	0	1028	2562
	600	30.8	28.0	240.3	0	1120	2562
S3-	0	32.7	28.0	19.7	0	2330	2448
	200	33.0	28.2	14.0	0	2334	2418
	400	32.1	28.2	13.7	0	2390	2472
	600	32.0	28.2	13.6	0	2396	2478
S4-	0	32.7	28.0	19.7	0	2330	2448
	200	32.7	28.2	16.1	0	2339	2436
	400	32.7	28.2	15.9	0	2341	2436
	600	32.7	28.2	15.9	0	2341	2436
Conventional Vapor Compression	-	83.2	18.3	-	-	-	-

Table A.8 Hybrid System Economic Analysis Results for Madison, WI

System	Collector Area, m^2	No Storage: Equipment Cost (C_E), \$	No Storage: Cost per Ton, \$/ton	No Storage: Collector Cost (C_A), \$/m^2	Infinite Storage: Equipment Cost (C_E), \$	Infinite Storage: Cost per Ton, \$/ton	Infinite Storage: Collector Cost (C_A), \$/m^2
S1	-	11650	388	-	12240	408	-
S2	0	***	***	-	***	***	-
	200	***	***	29	***	***	73
	400	***	***	17	***	***	37
	600	***	***	12	***	***	24
S3	0	11650	388	-	12240	408	-
	200	11650	388	0	12240	408	***
	400	11650	388	1	12240	408	0
	600	11650	388	1	12240	408	0
S4	0	11650	388	-	12240	408	-
	200	11650	388	0	12240	408	***
	400	11650	388	0	12240	408	***
	600	11650	388	0	12240	408	***

***Negative cost.

Table A.9 Sterling Annual Simulation Results

System	Collector Area, m^2	Electricity (MWh) Compressor	Electricity (MWh) Fan	Natural Gas (GJ) No Storage	Natural Gas (GJ) Infinite Storage	Annual Fuel Savings ($) No Storage	Annual Fuel Savings ($) Infinite Storage
S1	-	54.7	35.1	50.0	0	3432	3732
S2-	0	51.5	35.1	899.7	899.7	-1414	-1414
	200	51.5	35.1	584.3	2.7	418	3908
	400	51.5	35.1	505.1	0	893	3924
	600	51.5	35.1	474.6	0	1074	3924
S3-	0	54.7	35.1	50.0	0	3432	3732
	200	56.1	35.3	38.7	0	3404	3636
	400	54.6	35.3	37.9	0	3499	3726
	600	53.9	35.3	37.4	0	3544	3768
S4-	0	54.7	35.1	50.0	0	3432	3732
	200	54.7	35.3	43.6	0	3458	3720
	400	54.7	35.3	43.0	0	3462	3720
	600	54.7	35.3	42.9	0	3462	3720
Conventional Vapor Compression	-	129.1	22.9	-	-	-	-

Table A.10 Hybrid System Economic Analysis Results for Sterling, VA

System	Collector Area, m^2	No Storage: Equipment Cost (C_E), \$	No Storage: Cost per Ton, \$/ton	No Storage: Collector Cost (C_A), \$/m^2	Infinite Storage: Equipment Cost (C_E), \$	Infinite Storage: Cost per Ton, \$/ton	Infinite Storage: Collector Cost (C_A), \$/m^2
S1	1	17160	572	-	18660	622	-
S2	0	***	***	-	***	***	-
	200	***	***	46	***	***	133
	400	***	***	29	***	***	67
	600	***	***	21	***	***	44
S3	0	17160	572	-	18660	622	-
	200	17160	572	***	18660	622	***
	400	17160	572	1	18660	622	***
	600	17160	572	1	18660	622	0
S4	0	17160	572	-	18660	622	-
	200	17160	572	1	18660	622	***
	400	17160	572	0	18660	622	***
	600	17160	572	0	18660	622	***

***Negative cost.

APPENDIX B

Program Listings for Hybrid System

1) HOWE.SYSTEM8--Generation of performance map data

2) HOWE*TRNSYS.LOADMODEL--TRNSYS deck for generating building loads

3) HOWE*TRNSYS.STRATEGY1--Typical TRNSYS deck for simulation of hybrid system

4) Individual Component Models

HOWE.SYSTEM8

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C
C   *****************************
C   *                           *
C   *  PROGRAM DETERMINES THE   *
C   *  STATIC PERFORMANCE OF A  *
C   *  HYBRID AIR CONDITIONING  *
C   *  SYSTEM WHICH USES A DEHU- *
C   *  MIDIFIER, AN INDIRECT    *
C   *  EVAPORATIVE COOLER, AND  *
C   *  A VAPOR COMPRESSION UNIT *
C   *                           *
C   *****************************
C
C
C   *****************************
C   *                           *
C   *  THIS PROGRAM WAS SET UP  *
C   *  TO RUN A FIXED LOAD FOR  *
C   *  VARYING AMBIENT CONDIT-  *
C   *  IONS TO GENERATE SYSTEM  *
C   *  CHARACTERISTICS FOR USE  *
C   *  ON PERFORMANCE MAPS      *
C   *                           *
C   *****************************
C
C
      DIMENSION QS(6) ,QL(6)
      REAL MZONE(6),MVENT(6),MOUT(6),MEX(6),MVENTT,MSYS,MRA
      REAL MAS,MRAS,IBAL
C
C
      DATA CP,HFG,D1/1.006,2501.,1.805/
      DATA PA,RHO/101600.,1.201/
C
C
      EEC  = .95
      EIEC = .95
      EFAN = .50
      MBASE= 16917.6
C
      READ(*,*) ICT,IDUMP,IBAL
40    READ(*,*,END=41) TA,WA,TWB
C
C
      HA     = TA + WA * (HFG + D1 * TA)
```

```
      QSENS = 0.
      QLAT  = 0.
      DO 60 J = 1, 4
         QL(J) = 0.
         QS(J) =  0.
60    CONTINUE
      QL(5) = 310500.
      QS(5) =  0.
      QL(6) = 0.
      QS(6) =  0.
      DO 65 K = 1,6
         MVENT(K) = 519.6
         MEX(K)   =   0.0
         QSENS = QSENS + QS(K)
         QLAT  = QLAT + QL(K)
65    CONTINUE
C
C
C  ********************
C  *                  *
C  *  INITIALIZATION  *
C  *                  *
C  ********************
C
C
      TDEH   = 0.
      WDEH   = 0.
      TREG   = 0.
      WREG   = 0.
      TAC    = 0.
      WAC    = 0.
      TDEC   = 0.
      WDEC   = 0.
      MSYS   = 0.0
      MRA    = 0.0
      MVENTT = 0.0
      TROOM = 26.0
      WROOM = .0074
      HROOM = TROOM + WROOM * (HFG + D1 * TROOM)
      TROOMS= TROOM
      WROOMS= WROOM
      HROOMS= HROOM
      SHR   = QSENS / (QSENS + QLAT)
      TFAN  = TROOM - 15.* SHR / CP
      HFAN  = HROOM - 15.
      WFAN  = (HFAN - CP * TFAN)/(HFG + D1 * TFAN)
      TFANS = TFAN
      WFANS = WFAN
      DO 100 I=1,6
         MZONE(I) = (QS(I) + QL(I)) / 15. + MVENT(I)
         MOUT(I)  = MZONE(I) - MEX(I)
```

```
          IF(MOUT(I) .LT. 0.) MOUT(I) = 0.
          MRA = MRA + MOUT(I)
          MSYS = MSYS + MZONE(I)
          MVENTT = MVENTT + MVENT(I)
100    CONTINUE
C
C   ***************************************
C   *                                     *
C   *            FAN CALCULATIONS         *
C   *                                     *
C   ***************************************
C
       DPDEH = 250.
       DPEC  = 100.
       DPIEC = 125.
       DPAC  = 125.
       DPCOND= 125.
       DPFR  = 550.
       DPP   = (DPDEH+DPEC+DPIEC+DPAC+DPFR) * (MSYS/MBASE) ** 2
       QSTATP= MSYS * DPP / (RHO * 1000.)
       QFANP = QSTATP * (1.- EFAN)/EFAN
       DPR   = (DPDEH+DPCOND) * (IBAL*MSYS/MBASE) ** 2
       QSTATR= IBAL * MSYS * DPR / (RHO * 1000.)
       QFANR = QSTATR * (1.- EFAN)/EFAN
       TSET  = TFAN - QFANP / (MSYS * CP)
       WSET  = WFAN
       HSET  = TSET + WSET * (HFG + B1 * TSET)
C
C   ***************************************
C   *                                     *
C   *              COOLING TOWER          *
C   *                                     *
C   ***************************************
C
       WWB = .62198 * PVS(TWB) / (PA - PVS(TWB))
       TEC = TA - EEC * (TA - TWB)
       WEC = WA + EEC * (WWB - WA)
       IF(ICT .EQ. 1) THEN
          TREJ = TEC
          WREJ = WEC
          ELSE
          TREJ = TA
          WREJ = WA
       ENDIF
       IF(IDUMP .EQ. 1) THEN
          WPURGE = WEC
          ELSE
          WPURGE = WA
       ENDIF
C
C   ***************************************
```

```
C  *                                        *
C  *          MODES OF OPERATION            *
C  *                                        *
C  ****************************************
C
      IF(WA .LE. WFAN) THEN
         WDIFF = WFAN - WA
         TFANO = TFAN
         B     = WFAN + .00459 * TFAN
         WFAN  = WA
         TFAN  = (B - WFAN) / .00459
         HFAN  = TFAN + WFAN * (HFG + D1 * TFAN)
         TDIFF = TFAN - TFANO
         TROOM = TROOM + TDIFF
         WROOM = WROOM - WDIFF
         HROOM = TROOM + WROOM * (HFG + D1 * TROOM)
         TSET  = TFAN - QFANP / (MSYS * CP)
         WSET  = WFAN
         HSET  = TSET + WSET * (HFG + D1 * TSET)
         GO TO 600
         ELSE
         IF(HA .GT. HROOM) THEN
            GO TO 500
            ELSE
            GO TO 700
         ENDIF
      ENDIF
C
C
500   MODE = 1
      MA = MVENTT
      MRA = MSYS - MA
      TMIX = (MA * TA + MRA * TROOM) / MSYS
      WMIX = (MA * WA + MRA * WROOM) / MSYS
      WDEH = WFAN
      WREG = WPURGE
      CALL DEHUM(TMIX,WMIX,WDEH,WREG,TDEH,TREG)
      TIEC = TDEH - EIEC * (TDEH - TEC)
      WIEC = WFAN
      IF(TIEC .LE. TSET) THEN
         TIEC = TSET
         ELSE
         TAC  = TSET
         WAC  = WSET
      ENDIF
      GO TO 900
C
C
600   MODE = 2
      TMIX = TA
      WMIX = WA
```

```
      TIEC = TMIX - EIEC * (TMIX - TEC)
      WIEC = WA
      IF(TIEC .LE. TSET) THEN
         TIEC = TSET
         ELSE
         TAC  = TSET
         WAC  = WSET
      ENDIF
      GO TO 900
C
C
700   MODE = 3
      TMIX = TA
      WMIX = WA
      WDEH = WFAN
      WREG = WPURGE
      CALL DEHUM(TMIX,WMIX,WDEH,WREG,TDEH,TREG)
      TIEC = TDEH - EIEC * (TDEH - TEC)
      WIEC = WFAN
      IF(TIEC .LE. TSET) THEN
         TIEC = TSET
         ELSE
         TAC  = TSET
         WAC  = WSET
      ENDIF
C
C
900   IF(TAC .LE. 0.) THEN
         K = 0
         ELSE
         K = 1
      ENDIF
C
C  ****************************************
C  *                                      *
C  *   DEHUMIDIFIER SYSTEM CALCULATIONS   *
C  *                                      *
C  ****************************************
C
      QEVAP  = MSYS * CP * ( TIEC - TAC ) * K
      COPAC = (2.842 - .03998*(TREJ - WETBLB(TIEC,WIEC)))*K
      QCOND  = QEVAP * (1. + 1./ COPAC )
      TCOND  = ( TREJ + QCOND / (IBAL * MSYS * CP))
      IF( IDUMP .EQ. 0  .AND.  ICT .EQ. 1 )  TCOND = TA
      TPURGE = ( TCOND + QFANR / (IBAL * MSYS * CP))
      QAUX   = IBAL * MSYS * CP * ( TREG - TPURGE )
      IF( QAUX .LT. 0. ) QAUX = 0.
      QVC    = QEVAP / COPAC
      QFAN   = QSTATP + QSTATR + QFANP + QFANR
      QELEC  = QVC + QFAN
      COPTH  = MSYS * ( HROOM - HSET ) / QAUX
```

```
      IF( QAUX .LE. 0. ) COPTH = 1000.
      COPEL  = MSYS * ( HROOM - HSET ) / ( QVC + QFAN )
C
C  ************************************
C  *                                  *
C  *    STD VAPOR COMPRESSION SYSTEM  *
C  *                                  *
C  ************************************
C
      PVFANS = PA * WFANS / (.62198 + WFANS)
      ALPHA  = ALOG(PVFANS)
      TDP    = -35.957 - 1.8726 * ALPHA + 1.1689 * ALPHA ** 2
      WDP    = WFANS
      HDP    = TDP + WDP * (HFG + D1 * TDP)
      DPTOTS = DPFR + DPAC + DPCOND
      QSTATS= MSYS * DPTOTS / (RHO * 1000.)
      QFANS = QSTATS * (1.- EFAN)/EFAN
      TSETS  = TFANS - QFANS / (MSYS * CP)
      WSETS  = WFANS
      IF(WA .GT. WFANS) THEN
         IF(HA .GT. HROOMS) THEN
            MAS    = MVENTT
            MRAS   = MSYS - MAS
            TMIXS  = (MAS * TA + MRAS * TROOMS) / MSYS
            WMIXS  = (MAS * WA + MRAS * WROOMS) / MSYS
            ELSE
            TMIXS  = TA
            WMIXS  = WA
         ENDIF
         HMIXS  = TMIXS + WMIXS * (HFG + D1 * TMIXS)
         QEVAPS = MSYS * (HMIXS - HDP)
         QAUXS  = MSYS * CP * (TSETS - TDP)
         ELSE
         IF(HA .GT. HROOMS) THEN
            WMIXS  = WFANS
            TMIXS  = (WMIXS-(TA*WROOMS-TROOMS*WA)/(TA-TROOMS))/
     -               ((WA-WROOMS)/(TA-TROOMS))
            ELSE
            TMIXS  = TA
            WMIXS  = WA
            BS     = WFANS + .00459 * TFANS
            WFANS  = WMIXS
            TFANS  = (BS - WFANS) / .00459
            TSETS  = TFANS - QFANS / (MSYS * CP)
            WSETS  = WFANS
         ENDIF
         QEVAPS = MSYS * CP * (TMIXS - TSETS)
         QAUXS  = 0.0
      ENDIF
      IF(QAUXS .LT. 0.0) THEN
         WRITE(*,11) TA,WA
```

```
11        FORMAT(' QAUXS NEGATIVE AT TA=',F6.3,'   WA=',F6.5)
          ELSE
          CONTINUE
       ENDIF
       COPACS = (2.842 - .03998*(TA - WETBLB(TMIXS,WMIXS)))
       OVCS   = QEVAPS / COPACS
       OFANS  = OSTATS + QFANS
       OELECS = OVCS + OFANS
C
C   ***************************************
C   *                                     *
C   * PERFORMANCE RATIO OF THE TWO SYSTEMS *
C   *                                     *
C   ***************************************
C
       EFFCOP = COPAC / COPACS
       EFFAC  = QEVAP / QEVAPS
       EFFWRK = EFFAC / EFFCOP
       EFFRES = (OELEC/.2895 + QAUX) / (OELECS/.2895 + QAUXS)
       EFRSNP = (OVC/.2895 + QAUX)/(OVCS/.2895 + QAUXS)
       EFFFAN = OFAN / OFANS
       EFFEL  = OELEC / OELECS
       RESDIF = (OELECS/.2895+QAUXS-(OELEC/.2895+QAUX))/1000.
       RSNPDF = (OVCS/.2895+QAUXS-(OVC/.2895+QAUX))/1000.
       QAUX   = QAUX/1000..
       QAUXS  = QAUXS/1000.
       OELEC  = OELEC/1000.
       OELECS = OELECS/1000.
C
C
       WRITE(8,5)TA,WA,EFFWRK,EFFRES,EFRSNP,OELEC,OELECS,QAUX,
      -          RESDIF,RSNPDF
5      FORMAT(' ',F4.1,1X,F5.4,3(1X,F5.2),3(1X,F8.1),2(1X,F7.0))
C
C
       GO TO 40
C
C
41     STOP
       END
C
C
C   ***************************************
C   *                                     *
C   *  SUBROUTINE TO DETERMINE THE VAPOR  *
C   *  PRESSURE OF WATER IN AIR FOR USE   *
C   *  WITH EVAPORATIVE COOLER MODEL      *
C   *                                     *
C   ***************************************
C
C
```

```
      FUNCTION PVS(TT)
      C8  = -5800.2206
      C9  = 1.3914993
      C10 = -.04860239
      C11 = .41764768E-04
      C12 = -.14452093E-07
      C13 = 6.5459673
      TW  = TT + 273.15
      PPPVS = C8/TW + C9 + C10*TW + C11*TW**2 + C12*TW**3 + C13*ALOG(TW)
      PVS = EXP(PPPVS)
      RETURN
      END
C
C
C  ****************************************
C  *                                      *
C  *  SUBROUTINE OF DEHUMIDIFIER ITERATIVE *
C  *  SCHEME TO DETERMINE OUTLET STATES   *
C  *                                      *
C  ****************************************
C
C
      SUBROUTINE DEHUM(TEMP1,W1,W2,W3,TEMP2,TEMP3)
C
C
      DIMENSION TEMP(3), W(3)
C
      DATA A1,A2,A3/2865.,4.344,.8624/
      DATA B1,B2,B3/6360.,1.127,.07969/
      DATA ZZ,ETAF1,ETAF2/1.490,.08,.95/
C
C
      F1(I)= -A1 /  ((TEMP(I) + 273.15) ** ZZ) + A2 * (W(I) ** A3)
      F2(I)= ((TEMP(I) + 273.15) ** ZZ) / B1 - B2 * (W(I) ** B3)
C
C
      TEMP(1) = TEMP1
      W(1)    = W1
      W(2)    = W2
      W(3)    = W3
C
C
      F11= F1(1)
      F21= F2(1)
      F12I= F11
      TEMP(2)= (A1 / (A2 * W(2) ** A3 - F12I)) ** (1. / ZZ) - 273.15
      F22I= F2(2)
      F23I= F22I
      TEMP(3)= (B1 * (F23I + B2 * W(3) ** B3)) ** (1. / ZZ) - 273.15
      GO TO 40
30    TEMP(3)= TEMP(3) + 0.10
```

```
      IF(TEMP(3) .LT. 150.) GO TO 40
      WRITE(8,35)
35    FORMAT('0 REGENERATION TEMPERATURE DID NOT CONVERGE')
      GO TO 99
40    F13= F1(3)
      F23= F2(3)
      F12= F11 + ETAF1 * (F13 - F11)
      F22= F21 + ETAF2 * (F23 - F21)
      WOLD= .005
      DO 45 I=1,25
         FNW= B1*(F22 + B2*(WOLD ** B3)) + A1/(F12 - A2*(WOLD ** A3))
         DFNW= B1 * B2 * B3 * (WOLD ** (B3 - 1.)) + A1 * A2 * A3 *
     -        (WOLD ** (A3 - 1.)) / ((F12 - A2 * (WOLD ** A3)) ** 2)
         WNEW= WOLD - FNW / DFNW
         IF(WNEW.LE.0.0) WNEW= WOLD * .5
         IF(ABS((WNEW - WOLD) / WOLD).LT. .01) GO TO 50
         WOLD= WNEW
45    CONTINUE
      WRITE(8,47)
47    FORMAT('0 HUMIDITY RATIO DID NOT CONVERGE')
99    STOP
50    IF(ABS((WNEW - W(2)) / W(2)) .GT. .01) GO TO 30
      TEMP(2)= (A1 / (A2 * W(2) ** A3 - F12)) ** (1. / ZZ) - 273.15
      TEMP2 = TEMP(2)
      TEMP3 = TEMP(3)
C
C
      RETURN
      END
C
C
C   ****************************************
C   *                                      *
C   *  SUBROUTINE TO DETERMINE WET BULB    *
C   *  TEMPERATURE OF A GIVEN AIR STATE    *
C   *                                      *
C   ****************************************
C
C
      FUNCTION WETBLB(TEMP,HUM)
C
C
      DATA C1,C2,C3/-5800.2206,1.3914993,-.04860239/
      DATA C4,C5,C6/.41764768E-04,-.14452093E-07,6.5459673/
      DATA PA/101600./
C
C
      K(T) = T + 273.15
      PVV(T) = EXP( C1/K(T) + C2 + C3*K(T) + C4*K(T)**2
     -              + C5*K(T)**3 + C6*ALOG(K(T)) )
      W(T) = .62198 * PVV(T)/( PA - PVV(T) )
```

```
      F(T) = ((2501. - 2.381*T)*W(T)-(TEMP-T))/(2501. +
     -         1.805*TEMP-4.186*T) - HUM
C
      EPS = .00000001
      TOL = .000001
C
      A = TEMP/2. - 3.
      B = TEMP
      FA= F(A)
      FB= F(B)
20    C = A
      FC= FA
      D = B-A
      E = D
30    IF(ABS(FC) .GE. ABS(FB)) GO TO 40
      A = B
      B = C
      C = A
      FA= FB
      FB= FC
      FC= FA
40    TOL1 = 2.0 * EPS * ABS(B) + 0.5 * TOL
      XM= 0.5 * (C-B)
      IF(ABS(XM) .LE. TOL1) GO TO 90
      IF(FB .EQ. 0.0) GO TO 90
      IF(ABS(E) .LT. TOL1) GO TO 70
      IF(ABS(FA) .LE. ABS(FB)) GO TO 70
      IF(A .NE. C) GO TO 50
      S = FB/FA
      P = 2.0 * XM * S
      Q = 1.0 - S
      GO TO 60
50    Q = FA/FC
      R = FB/FC
      S = FB/FA
      P = S*(2.*XM*Q*(Q-R) - (B-A)*(R-1.))
      Q = (Q-1.)*(R-1.)*(S-1.)
60    IF(P .GT. 0.0) Q=-Q
      P = ABS(P)
      IF((2.*P) .GE. (3.*XM*Q-ABS(TOL1*Q))) GO TO 70
      IF(P .GE. ABS(.5*E*Q)) GO TO 70
      E = D
      D = P/Q
      GO TO 80
70    D = XM
      E = D
80    A = B
      FA= FB
      IF(ABS(D) .GT. TOL1) B = B+D
      IF(ABS(D) .LE. TOL1) B = B+ SIGN(TOL1,XM)
      FB= F(B)
```

```
      IF((FB * (FC/ABS(FC))) .GT. 0.0) GO TO 20
      GO TO 30
90    WETBLB= B
      RETURN
      END
```

HOWE*TRNSYS.LOADMODEL

```
  SIMULATION 1.0  8760.0 1.0
****************************
*                          *
*      LOAD MODEL FOR      *
*  MULTIZONE VAV BUILDING  *
*                          *
****************************
*
NOLIST
*
UNIT 5 TYPE 9  DATA READER
PARAMETERS 19
9  1  -1  1  0  -2  1  0  -3  1  0  -4  1  0  -5  1  0  -1  1
(10X,F2.0,1X,F2.0,1X,F2.0,1X,F4.0,1X,F4.0,1X,F4.1,12X,F4.1,1X,F6.4,1X,F2.0)
*
UNIT 6 TYPE 14  FORCING FUNCTION...MVENT (15 cfm/PERSON)
PARAMETERS 12
0  0  8  0  8  1722.7  18  1722.7  18  0  24  0
*
UNIT 7 TYPE 14  FORCING FUNCTION...QLITES (2W/FT2) AND QEQUIP (2W/FT2)
PARAMETERS 12
0 3920 6 3920 8 62745 18 62745 20 3920 24 3920
*
UNIT 8 TYPE 14  FORCING FUNCTION...QPEPL (70W/PERSON)
```

```
PARAMETERS 12
0 0 6 0 8 14180 18 14180 20 0 24 0
*
UNIT 9 TYPE 14  FORCING FUNCTION...WI (65W/PERSON)
PARAMETERS 12
0 0 6 0 8 5.4  18  5.4 20 0 24 0
*
UNIT 10 TYPE 16  SOLAR RADIATION PROCESSOR
PARAMETERS 7
5 1 1  33.43 4871. 0 -1
INPUTS 13
5,5  5,4  5,19  5,20  0,0  0,0  0,0  0,0  0,0  0,0  0,0  0,0  0,0
.0   .0   .0    .0    .2   90   0    90   90   90   180  90   -90
*
UNIT 11 TYPE 17  WALL #1
PARAMETERS 21
1 .5 .8 5 5 330 .8 2 .2 .3 -1 .0 .00046 .00225 .00150 .00016 -1.73771
.90936 -.13373 .00496 -.00001
INPUTS 4
5,6  10,6  5,9  25,2
.0   .0   2.24  23.25
*
UNIT 12 TYPE 17  WALL #2
PARAMETERS 21
1 .5 .8 5 5 330 .8 2 .2 .3 -1 .0 .00046 .00225 .00150 .00016 -1.73771
.90936 -.13373 .00496 -.00001
INPUTS 4
5,6  10,11  0,0  26,2
.0   .0     .0   23.25
*
UNIT 13 TYPE 17  WALL #3
PARAMETERS 21
1 .5 .8 5 5 330 .8 2 .2 .3 -1 .0 .00046 .00225 .00150 .00016 -1.73771
.90936 -.13373 .00496 -.00001
INPUTS 4
5,6  10,13  5,9  27,2
.0   .0    2.24  23.25
*
UNIT 14 TYPE 17  WALL #4
PARAMETERS 21
1 .5 .8 5 5 330 .8 2 .2 .3 -1 .0 .00046 .00225 .00150 .00016 -1.73771
.90936 -.13373 .00496 -.00001
INPUTS 4
5,6  10,15  0,0  28,2
.0   .0     .0   23.25
*
UNIT 16 TYPE 17  ROOF #2
PARAMETERS 23
1 .5 .8 6 6 168.75 0 1 0 0 0 .0 .00002 .00046 .00133 .00079 .00011
-1.91091 1.22135 -.31019 .03001 -.00095 .00001
INPUTS 4
```

```
5,6  5,5  5,9  26,2
.0   .0   2.24   23.25
*
UNIT 17 TYPE 17  ROOF #5
PARAMETERS 23
1 .5 .8 6 6 225 0 1 0 0 0 .0 .00002 .00046 .00133 .00079 .00011
-1.91091 1.22135 -.31019 .03001 -.00095 .00001
INPUTS 4
5,6  5,5  5,9  29,2
.0   .0   2.24  23.25
*
UNIT 20 TYPE 15  ALGEBRAIC OPERATOR
INPUTS 5
16,2  11,2  12,2  13,2  14,2
.0    .0    .0    .0    .0
PARAMETERS 16
0 0 3 -4 -11 0 3 -4 -11 0 3 -4 -11 0 3 -4
*
UNIT 21 TYPE 15  ALGEBRAIC OPERATOR FOR #5
INPUTS 5
0,0  6,1  7,1  8,1  9,1
.450  .0   .0   .0   .0
PARAMETERS 16
0  0  1  -4  -11  0  1  -4  -11  0  1  -4  -11  0  1  -4
*
UNIT 22 TYPE 15  ALGEBRAIC OPERATOR FOR #6
INPUTS 5
0,0  6,1  7,1  8,1  9,1
.882 .0   .0   .0   .0
PARAMETERS 16
0  0  1  -4  -11  0  1  -4  -11  0  1  -4  -11  0  1  -4
*
UNIT 25 TYPE 19  ZONE #1
PARAMETERS 19
1 1856.25 0 506.25 3 1.17E5 1680. 0 0 30 0 0 0 21.5 25.0 23.25
0 0 .012
INPUTS 10
0,0  0,0  0,0  5,6  5,8  9,1  20,1  11,3  7,1  8,1
0.0  0.0  0.0  .0   .0   .0   .0    .0    .0   .0
*
UNIT 26 TYPE 19  ZONE #2
PARAMETERS 19
1 1856.25 0 506.25 3 1.17E5 1680. 0 0 30 0 0 0 21.5 25.0 23.25
0 0 .012
INPUTS 10
0,0  0,0  0,0  5,6  5,8  9,1  20,2  12,3  7,1  8,1
0.0  0.0  0.0  .0   .0   .0   .0    .0    .0   .0
*
UNIT 27 TYPE 19  ZONE #3
PARAMETERS 19
1 1856.25 0 506.25 3 1.17E5 1680. 0 0 30 0 0 0 21.5 25.0 23.25
```

```
0 0 .012
INPUTS 10
0,0  0,0  0,0  5,6  5,8  9,1  20,3  13,3  7,1  8,1
0.0  0.0  0.0  ,0   ,0   .0   ,0    ,0    ,0   ,0
*
UNIT 28 TYPE 19  ZONE #4
PARAMETERS 19
1 1856.25 0 506.25 3 1.17E5 1680. 0 0 30 0 0 0 21.5 25.0 23.25
0 0 .012
INPUTS 10
0,0  0,0  0,0  5,6  5,8  9,1  20,4  14,3  7,1  8,1
0.0  0.0  0.0  ,0   ,0   .0   ,0    ,0    ,0   .0
*
UNIT 29 TYPE 19  ZONE #5
PARAMETERS 19
1 825. 0 225. 3 5.19E4 422.8 0 0 0 0 0 0 21.5 25.0 23.25 0 0
.012
INPUTS 10
0,0  0,0  0,0  5,6  5,8  21,4  17,2  17,3  21,2  21,3
0.0  0.0  0.0  ,0   ,0   .0    ,0    ,0    ,0    .0
*
UNIT 30 TYPE 19  ZONE #6
PARAMETERS 19
1 1650 0 450 3 1.04E5 0 0 0 0 0 0 0 21.5 25.0 23.25 0 0 .012
INPUTS 10
0,0  0,0  0,0  5,6  5,8  22,4  0,0  0,0  22,2  22,3
0.0  0.0  0.0  ,0   ,0   ,0    ,0   ,0   ,0    .0
*
UNIT 31 TYPE 15  ALGEBRAIC OPERATOR...SENSIBLE
INPUTS 6
25,1  26,1  27,1  28,1  29,1  30,1
 .0    ,0    .0    .0    .0    .0
PARAMETERS 12
0  0  3  0  3  0  3  0  3  0  3  -4
*
UNIT 32 TYPE 15  ALGEBRAIC OPERATOR...LATENT
INPUTS 6
25,4  26,4  27,4  28,4  29,4  30,4
 .0    ,0    .0    .0    ,0    .0
PARAMETERS 12
0  0  3  0  3  0  3  0  3  0  3  -4
*
UNIT 33 TYPE 16  SOLAR RADIATION PROCESSOR OUTPUT
PARAMETERS 7
5  1  1  33.43  4871.  0  -1
INPUTS 7
5,5  5,4  5,19  5,20  0,0  0,0  0,0
.0   ,0   ,0    .0    ,2   20   .0
*
UNIT 34 TYPE 25  PRINTER
PARAMETERS 4
```

```
1  1.0  8760.0 8
INPUTS 5
5,6  5,8  31,1  32,1  33,6
TA   WA  QSENS  QLAT   HT
*
END
```

HOWE*TRNSYS.STRATEGY1

```
  SIMULATION 1.0  8760.0 1.0
*
**********************************
*                                *
*   DESICCANT AIR CONDITIONING   *
*          SYSTEM MODEL          *
*                                *
*   STRATEGY #1 :                *
*       CONDENSER-AUXILIARY      *
*                                *
**********************************
*
NOLIST
*
UNIT 1 TYPE 9  DATA READER
PARAMETERS 4
6  1  -1  1
(F10.4,5(1PE11.3))
*
UNIT 2 TYPE 14  FORCING FUNCTION...MVENT, KG/HR
PARAMETERS 12
0  0  8  0  8  9185.4  18  9185.4  18  0  24  0
*
UNIT 3 TYPE 14  FORCING FUNCTION...QLITES, KJ/HR
```

```
PARAMETERS 12
0  34900  6  34900  8  209300  18  209300  20  34900  24  34900
*
UNIT 4 TYPE 30  NEWCON
PARAMETERS 5
.40  .60  .012  1.0  59450.0
INPUTS 7
0,0  1,2  1,3  1,4  1,5  2,1  3,1
25.0  20.  .012  .0  .0  .0  .0
*
UNIT 5 TYPE 31  MIXER
INPUTS 7
4,6  4,7  1,2  1,3  4,3  2,1  4,5
24.  .012  20.  .012  .0  .0  .0
*
UNIT 6 TYPE 32  DEHUMIDIFIER
PARAMETERS 2
.08  .95
INPUTS 8
4,3  5,1  5,2  4,2  1,3  4,4  1,2  4,8
.0  25.  .012  .009  .012  0  20.  .0
*
UNIT 7 TYPE 33  FAN #1
PARAMETERS 10
500  150  100  50  60  32  0  0  .50  30000
INPUTS 5
4,3  6,1  6,2  4,4  0,0
.0  25.  .009  0  1
*
UNIT 8 TYPE 34  INDIRECT EVAPORATIVE COOLER
PARAMETERS 2
.90  .90
INPUTS 4
7,1  7,2  1,2  1,3
25.  .009  20.  .012
*
UNIT 9 TYPE 35  DIRECT EVAPORATIVE COOLER
PARAMETERS 1
.90
INPUTS 2
8,1  8,2
.0  .0
*
UNIT 10 TYPF 36  VAPOR COMPRESSION UNIT
PARAMETERS 1
30.0
INPUTS 10
4,3  8,1  8,2  9,1  4,1  4,2  1,2  1,3  4,8  0,0
.0   15.  .009  10.  12.  .009  20.  .012  .0  1
*
UNIT 11 TYPE 33  FAN #2
```

```
PARAMETERS 10
100  150  50  60  32  0  0  0  .50  30000
INPUTS 5
4,8  10,4  10,5  4,4  10,6
.0  25.  .012  0  0
*
UNIT 12 TYPE 37  AUXILIARY HEATER
INPUTS 4
4,8  11,1  6,3  11,2
.0  25.  30.  .012
*
UNIT 13 TYPE 33  FAN #3
PARAMETERS 10
50  100  32  0  0  0  0  0  .50  30000
INPUTS 5
4,3  8,3  8,4  0,0  0,0
.0  10.  .012  1  1
*
UNIT 20 TYPE 28  SIMULATION SUMMARY
PARAMETERS 21
-1  1.0  8760.0  -1  1  -11  -4  -12  -4  -13  -4  -14  -4
-15  -4  -16  -4  -17  -4  -18  -4
INPUTS 8
10,7  7,6  11,6  13,6  12,1  10,3  11,3  12,2
LABELS 8
WORK  ELEC1  ELEC2  ELEC3  QAUX  QCOND  QFAN  EXCESS
*
UNIT 21 TYPE 28  SIMULATION SUMMARY
PARAMETERS 17
-1  1.0  8760.0  -1  1  -11  -4  -12  -4  -13  -4  -14  -4
-15  -4  -16  -4
INPUTS 6
12,3  12,4  12,18  4,4  4,5  10,6
LABELS 6
QAUXST  USE  EXSTOR  ION  IVENT  IONAC
*
UNIT 22 TYPE 27  HISTOGRAM
PARAMETERS 8
2  -1  -1  1.0  8760.0  0  24  24
INPUTS 1
12,1
QAUX
*
END
```

HOWE*TRNSYS.NEWCON

```
      SUBROUTINE TYPE30(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
C
C  ********************************************
C  *                                          *
C  *  PRIMARY BUILDING CONTROLLER MODEL.  IT  *
C  *  DETERMINES THE APPROPRIATE SUPPLY AIR   *
C  *  TEMPERATURE AND HUMIDITY RATIO TO MEET  *
C  *  THE BUILDING LOAD.  ALSO DETERMINES THE *
C  *  REQUIRED SYSTEM FLOW RATE.              *
C  *                                          *
C  ********************************************
C
C
      DIMENSION XIN(7),PAR(5),OUT(19),INFO(9)
      REAL MSYS,MVENT,MREG,ML2,MB
C
C
      DATA A,B,C/8.087E-05, 1.6005, 4.377E-03/
      DATA CPA,CPV,HFG/1.0, 1.805, 2501./
C
C
      TR       = XIN(1)
      TA       = XIN(2)
      WA       = XIN(3)
      QSENS    = XIN(4)
      QLAT     = XIN(5)
      MVENT    = XIN(6)
      QLITES   = XIN(7)
C
C
      FLITES   = PAR(1)
      BAL      = PAR(2)
      WMAX     = PAR(3)
      DELT     = PAR(4)
      MB       = PAR(5)
C
C
      INFO(6) = 19
      INFO(9) = 0
      IF(INFO(7) .EQ. -1) THEN
         CALL TYPECK(1,INFO,7,5,0)
         WR    = .008
         ELSE
         IF(INFO(7) .EQ. 0) THEN
            WR       = OUT(18)
```

```
            OUT(19) = WR
            ELSE
            WR      = OUT(19)
         ENDIF
      ENDIF
C
C
      IF(QSENS .LE. 0.0) THEN
         TD     = TR
         WD     = WR
         MSYS   = MVENT
         ION    = 0
         IVENT  = 0
         TLITES = TR + FLITES*QLITES/(MSYS*CP)
         WLITES = WR
         MREG   = BAL*MSYS
         WRN    = WR
         GO TO 98
         ELSE
         CONTINUE
      ENDIF
      IF(QLAT .LT. 0.0) QLAT = 0.0
      CP       = CPA + CPV*WR
      HA       = CPA*TA + WA*(HFG + CPV*TA)
      HR       = CPA*TR + WR*(HFG + CPV*TR)
      HD       = HR - 15.
      SHR      = QSENS/(QSENS + QLAT)
      TD       = TR - 15.*SHR/CP
      MSYS     = QSENS/(CP*(TR-TD))
      IF(MSYS .LT. MVENT) THEN
         MSYS = MVENT
         TD   = TR - QSENS/(MSYS*CP)
         ELSE
         CONTINUE
      ENDIF
      MREG     = BAL * MSYS
      ML2      = QLAT/HFG
      AEXP     = EXP(-MSYS*DELT/MB)
      WD       = (WMAX-ML2/MSYS+AEXP*(ML2/MSYS-WR))/(1-AEXP)
C
C
      TLITES   = TR + FLITES*QLITES/(MSYS*CP)
      WLITES   = WR
      HLITES   = CPA*TLITES + WLITES*(HFG + CPV*TLITES)
      TRA      = (MVENT*TA + (MSYS-MVENT)*TLITES)/MSYS
      WRA      = (MVENT*WA + (MSYS-MVENT)*WLITES)/MSYS
      HRA      = CPA*TRA + WRA*(HFG + CPV*TRA)
C
C
      IF(WD .GT. WA) THEN
         WD    = WA
```

```
         ION    = 0
         ICHG   = 1
         IVENT = 1
         ELSE
         IF(WD .GT. WRA) THEN
            WD   = WRA
            ION = 0
            ICHG= 1
            IVENT = 0
            ELSE
            ION = 1
            ICHG= 0
            IF(HA .LT. HRA) THEN
               IVENT = 1
               ELSE
               IVENT = 0
            ENDIF
         ENDIF
      ENDIF
C
C
      FSAT    = A*TD**B + C - WD
      IF(FSAT .LT. 0.0) THEN
         WD   = A*TD**B + C
         ICHG = 1
         ION  = 1
         ELSE
         CONTINUE
      ENDIF
C
C
      IF(ICHG .EQ. 1) THEN
         WRN  = (ML2+WD*MSYS)/MSYS + (WR-(ML2+WD*MSYS)/MSYS)*AEXP
         ELSE
         WRN  = WMAX
      ENDIF
C
C
98    OUT(1)  = TD
      OUT(2)  = WD
      OUT(3)  = MSYS
      OUT(4)  = ION
      OUT(5)  = IVENT
      OUT(6)  = TLITES
      OUT(7)  = WLITES
      OUT(8)  = MREG
      OUT(18) = WRN
C
C
      RETURN
      END
```

HOWE*TRNSYS.MIXER

```
      SUBROUTINE TYPE31(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
C
C   ********************************************
C   *                                          *
C   *  BUILDING ECONOMIZER MODEL.  INPUT IVENT  *
C   *  FROM .NEWCON DETERMINES THE AMOUNT OF    *
C   *  OUTSIDE AIR TO BE INTRODUCED INTO THE    *
C   *  BUILDING.                                *
C   *                                          *
C   ********************************************
C
C
      DIMENSION XIN(7), OUT(2), INFO(9)
      REAL MSYS, MVENT
C
C
      DATA CPA,CPV/1.0, 1.805/
C
C
      TLITES = XIN(1)
      WLITES = XIN(2)
      TA     = XIN(3)
      WA     = XIN(4)
      MSYS   = XIN(5)
      MVENT  = XIN(6)
      IVENT  = XIN(7)
C
C
      INFO(6)= 2
      INFO(9)= 0
      CALL TYPECK(1,INFO,7,0,0)
      CP     = CPA + CPV*WLITES
      IF(IVENT .EQ. 1) THEN
         TRA = TA
         WRA = WA
         ELSE
         TRA = (MVENT*TA + (MSYS - MVENT)*TLITES) / MSYS
         WRA = (MVENT*WA + (MSYS - MVENT)*WLITES) / MSYS
      ENDIF
C
C
      OUT(1) = TRA
      OUT(2) = WRA
```

```
C
C
      RETURN
      END
```

HOWE*TRNSYS.DEHUM

```
      SUBROUTINE TYPE32(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
C
C   ********************************************
C   *                                          *
C   *  MODEL OF A ROTARY DESICCANT DEHUMIDIFIER *
C   *  CONTAINING NOMINAL SILICA GEL.           *
C   *                                          *
C   *  PERFORMANCE BASED ON EQUATIONS FOR F1-F2 *
C   *  POTENTIALS DEVELOPED BY JURINAK          *
C   *                                          *
C   *  MODEL DETERMINES THE REGENERATION        *
C   *  TEMPERATURE AT AMBIENT HUMIDITY RATIO    *
C   *  WHICH WILL DEHUMIDIFY EXACTLY TO THE     *
C   *  SUPPLY HUMIDITY RATIO.  THE PROCESS      *
C   *  STREAM OUTLET TEMPERATURE IS ALSO        *
C   *  DETERMINED.                              *
C   *                                          *
C   ********************************************
C
C
      DIMENSION TEMP(4),W(4),XIN(8),PAR(2),OUT(6),INFO(9)
      REAL MSYS, MREG
C
C
      DATA A1,A2,A3/2865., 4.344, .8624/
      DATA B1,B2,B3/6360., 1.127, .07969/
      DATA ZZ,CPA,CPV/1.490, 1.0, 1.805/
      DATA HFG/2501./
```

```
C
C
      F1(I) = -A1/((TEMP(I)+273.15)**ZZ)+A2*(W(I)**A3)
      F2(I) = ((TEMP(I)+273.15)**ZZ)/B1 - B2*(W(I)**B3)
      H(I)  = CPA*TEMP(I) + W(I)*(HFG + CPV*TEMP(I))
C
C
      MSYS    = XIN(1)
      TEMP(1) = XIN(2)
      W(1)    = XIN(3)
      W(2)    = XIN(4)
      W(3)    = XIN(5)
      ION     = XIN(6)
      TA      = XIN(7)
      MREG    = XIN(8)
C
C
      ETAF1   = PAR(1)
      ETAF2   = PAR(2)
C
C
      INFO(6)  = 6
      INFO(9)  = 0
      CALL TYPECK(1,INFO,8,2,0)
      IF(ION .EQ. 0) THEN
         TDEH     = TEMP(1)
         WDEH     = W(1)
         TREG     = TA
         WREG     = W(3)
         TEMP(4)  = TREG
         W(4)     = WREG
         GO TO 100
         ELSE
         CONTINUE
      ENDIF
      F11= F1(1)
      F21= F2(1)
      F12I= F11
      TEMP(2)= (A1 / (A2 * W(2) ** A3 - F12I)) ** (1. / ZZ) - 273.15
      F22I= F2(2)
      F23I= F22I
      TEMP(3)= (B1 * (F23I + B2 * W(3) ** B3)) ** (1. / ZZ) - 273.15
      GO TO 40
 30   TEMP(3)= TEMP(3) + 0.10
      IF(TEMP(3) .LT. 150.) GO TO 40
      WRITE(*,35)
 35   FORMAT('0 REGENERATION TEMPERATURE DID NOT CONVERGE')
      GO TO 99
 40   F13= F1(3)
      F23= F2(3)
      F12= F11 + ETAF1 * (F13 - F11)
```

```
      F22= F21 + ETAF2 * (F23 - F21)
      WOLD= .005
      DO 45 I=1,25
         FNW= B1*(F22 + B2*(WOLD ** B3)) + A1/(F12 - A2*(WOLD ** A3))
         DFNW= B1 * B2 * B3 * (WOLD ** (B3 - 1.)) + A1 * A2 * A3 *
     -         (WOLD ** (A3 - 1.)) / ((F12 - A2 * (WOLD ** A3)) ** 2)
         WNEW= WOLD - FNW / DFNW
         IF(WNEW.LE.0.0) WNEW= WOLD * .5
         IF(ABS((WNEW - WOLD) / WOLD).LT. .01) GO TO 50
         WOLD= WNEW
45    CONTINUE
      WRITE(*,47)
47    FORMAT('0 HUMIDITY RATIO DID NOT CONVERGE')
99    STOP
50    IF(ABS((WNEW - W(2)) / W(2)) .GT. .01) GO TO 30
      TEMP(2)= (A1 / (A2 * W(2) ** A3 - F12)) ** (1. / ZZ) - 273.15
C
C
      TEMP(4) = (MSYS*(H(1)-H(2)-HFG*(W(1)-W(2)))+MREG*(H(3)-
     -          HFG*W(3))) / (MSYS*CPV*(W(1)-W(2))+MREG*(CPA+CPV*
     -          W(3)))
      W(4)    = (MSYS*(W(1)-W(2))+MREG*W(3)) / (MREG)
C
C
      TMIX   = TEMP(1)
      WMIX   = W(1)
      TDEH   = TEMP(2)
      WDEH   = W(2)
      TREG   = TEMP(3)
      WREG   = W(3)
C
C
100   OUT(1) = TDEH
      OUT(2) = WDEH
      OUT(3) = TREG
      OUT(4) = WREG
      OUT(5) = TEMP(4)
      OUT(6) = W(4)
C
C
      RETURN
      END
```

HOWE*TRNSYS.FAN

```
      SUBROUTINE TYPE33(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
C
C  ****************************************
C  *                                      *
C  *  FAN MODEL CALCULATES THE TOTAL FAN  *
C  *  POWER CONSUMPTION.  A SINGLE        *
C  *  EFFICIENCY IS USED WHICH IS A       *
C  *  COMBINATION OF THE FAN STATIC       *
C  *  EFFICIENCY AND THE DRIVE-MOTOR      *
C  *  EFFICIENCY.                         *
C  *                                      *
C  ****************************************
C
C
      DIMENSION XIN(6), OUT(5), PAR(10), INFO(9), DP(8)
      REAL MSYS, MBASE
C
C
      DATA CPA,CPV,RHO/1.0, 1.805, 1.201/
C
C
      MSYS   = XIN(1)
      TFANIN = XIN(2)
      WFANIN = XIN(3)
      ION    = XIN(4)
      IONAC  = XIN(5)
C
C
      DO 100 I=1,8
         DP(I) = PAR(I)
100   CONTINUE
      EFAN   = PAR(9)
      MBASE  = PAR(10)
C
C
      INFO(6) = 6
      INFO(9) = 0
      CALL TYPECK(1,INFO,5,10,0)
      IF(ION .EQ. 0  .AND.  IONAC .EQ. 0) THEN
         TFAN   = TFANIN
         WFAN   = WFANIN
         QFAN   = 0.0
         WORKST = 0.0
         DPTOT  = 0.0
         ELEC   = 0.0
```

```
         GO TO 300
         ELSE
         CONTINUE
      ENDIF
      CP      = CPA + CPV*WFANIN
      DPSUM   = 0.
      DO 200 J=1,8
         DPSUM = DPSUM + DP(J)
200   CONTINUE
      DPTOT   = DPSUM * (MSYS/MBASE) ** 2
      WORKST  = MSYS * DPTOT / (RHO * 1000.)
      QFAN    = WORKST * (1.- EFAN)/EFAN
      TFAN    = TFANIN + QFAN/(MSYS*CP)
      WFAN    = WFANIN
      ELEC    = WORKST / EFAN
C
C
300   OUT(1) = TFAN
      OUT(2) = WFAN
      OUT(3) = QFAN
      OUT(4) = WORKST
      OUT(5) = DPTOT
      OUT(6) = ELEC
C
C
      RETURN
      END
```

HOWE*TRNSYS.IEC

```
      SUBROUTINE TYPE34(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
C
C  *************************************
C  *                                   *
C  *  MODEL DETERMINES THE OUTLET STATE   *
C  *  OF AN INDIRECT EVAPORATIVE COOLER.  *
C  *                                   *
C  *************************************
C
C
      DIMENSION XIN(4),OUT(6),PAR(2),INFO(9)
      INTEGER RETRN
C
C
      DATA CPA,CPV,HFG/1.0,1.805,2501./
      DATA EPS,PA/.001,101600./
      DATA C1,C2,C3/-5674.5354, 6.3925247, -0.9677843E-02/
      DATA C4,C5,C6/.62215701E-06,.20747825E-08,-.9484024E-12/
      DATA C7/4.1635019/
      DATA C8,C9,C10/-5800.2206, 1.3914993, -.048860239/
      DATA C11,C12,C13/.41764768E-04, -.14452093E-07, 6.54596731/
C
C
      P1(Z) = -7.90298 * (Z-1.)
      P2(Z) = 5.02808 * ALOG10(Z)
      P3(Z) = -1.3816E-07 * (10.**(11.344*(1.-1./Z))-1.)
      P4(Z) = 8.1328E-03 * (10.**(-3.49149*(Z-1.))-1.)
      P5(Z) = -9.09718*(Z-1.)
      P6(Z) = -3.56654*ALOG10(Z)
      P7(Z) = .876793*(1.-1./Z)
      P8    = -2.2199
C
C
      TWARM  = XIN(1)
      WWARM  = XIN(2)
      TDRY   = XIN(3)
      WDRY   = XIN(4)
C
C
      EIEC   = PAR(1)
      EEC    = PAR(2)
C
C
```

```
      INFO(6)= 6
      INFO(9)= 0
      CALL TYPECK(1,INFO,4,2,0)
      CP     = CPA + CPV*WDRY
      HDRY   = CP*TDRY + WDRY*(HFG+CPV*TDRY)
      IF(HDRY .LT. 9.67) GO TO 10
      Y      = ALOG(HDRY + 17.68)
      TWB    = 26.7153 + Y*(-43.44 + Y*(13.909 - Y*.977))
      GO TO 20
10    TT     = TDRY
      ASSIGN 30 TO RETRN
      GO TO 100
30    WSAT   = .62198 * PSAT/(1.- PSAT)
      TT     = (HDRY - HFG*WSAT) / (CPA + CPV*WSAT)
      ASSIGN 40 TO RETRN
      GO TO 100
40    WT     = .62198*PSAT / (1.- PSAT)
      IF(ABS(TT-TDRY) .GT. 0.01) GO TO 50
      TWB    = (TT + TDRY) * .5
      GO TO 20
50    SLOPE  = (WSAT-WT) / (TDRY - TT)
      A      = CPV * SLOPE
      B      = CPA + CPV*WSAT + SLOPE*(HFG - CPV*TDRY)
      C      = -HDRY + HFG*(WSAT - SLOPE*TDRY)
      IF(SLOPE .LT. 1.E-07) TWB = -C/B
      IF(SLOPE .GT. .99E-07) TWB = (-B + SQRT(B*B-4.*A*C))/(2.*A)
20    TG1    = TWB
      TG2    = TWB + .5
      NIT    = 0
      ASSIGN 60 TO RETRN
      TT     = TG1
      GO TO 100
60    WS1    = .62198*PSAT / (1.- PSAT)
      F1     = WDRY - ((HFG-2.381*TG1)*WS1 - (TDRY-TG1))/
     -          (HFG + CPV*TDRY - 4.186*TG1)
65    ASSIGN 70 TO RETRN
      TT     = TG2
      GO TO 100
70    WS2    = .62198*PSAT / (1.- PSAT)
      F2     = WDRY - ((HFG-2.381*TG2)*WS2 - (TDRY-TG2))/
     -          (HFG + CPV*TDRY - 4.186*TG1)
      IF(ABS(F1-F2) .LT. 1.0E-07) GO TO 95
      TGN    = TG2 - F2*(TG2-TG1) / (F2-F1)
      IF(ABS(TGN-TG2) .LT. EPS) GO TO 90
      TG1    = TG2
      F1     = F2
      TG2    = TGN
      NIT    = NIT + 1
      IF(NIT .GT. 100) GO TO 80
      GO TO 65
80    TWB    = TG2
```

```
      WRITE(*,210)
210   FORMAT(' FAILURE TO CONVERGE IN 100 ITERATIONS')
      GO TO 98
90    TWB     = TGN
      GO TO 98
95    TWB     = TG2
      NIT     = NIT + 1
      WRITE(*,220) NIT
220   FORMAT(' LOSS OF PRECISION AFTER',2X,I3,2X,'ITERATIONS')
98    TT      = TWB
      ASSIGN 99 TO RETRN
      GO TO 100
99    WWB     = .62198*PSAT / (1.- PSAT)
      GO TO 300
C
C
100   IF(TT .GT. 0.) GO TO 105
      Z       = 273.16/(TT+273.16)
      PSAT    = 10.**(P5(Z) + P6(Z) + P7(Z) + P8)
      GO TO RETRN, (30,40,60,70,99)
105   Z       = 373.16/(TT+273.16)
      PSAT    = 10.**(P1(Z) + P2(Z) + P3(Z) + P4(Z))
      GO TO RETRN, (30,40,60,70,99)
C
C
300   TWBK    = TWB + 273.15
      IF(TWB .LT. 0.0) THEN
         PPVS = C1/TWBK + C2 + C3*TWBK + C4*TWBK**2 + C5*TWBK**3 +
     -             C6*TWBK**4 + C7*ALOG(TWBK)
         ELSE
         PPVS = C8/TWBK + C9 + C10*TWBK + C11*TWBK**2 + C12*TWBK**3 +
     -           C13*ALOG(TWBK)
      ENDIF
      PVS     = EXP(PPVS)
      WWB     = .62198*PVS / (PA-PVS)
      TEC     = TDRY - EEC*(TDRY-TWB)
      WEC     = WDRY - EEC*(WDRY-WWB)
C
C
      TIEC    = TWARM - EIEC*(TWARM-TWB)
      WIEC    = WWARM
C
C
      OUT(1) = TIEC
      OUT(2) = WIEC
      OUT(3) = TEC
      OUT(4) = WEC
      OUT(5) = TWB
      OUT(6) = WWB
C
C
```

```
      RETURN
      END
```

HOWE*TRNSYS.EC

```
      SUBROUTINE TYPE35(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
C
C  ***************************************
C  *                                     *
C  *  MODEL DETERMINES THE OUTLET STATE  *
C  *  OF A DIRECT EVAPORATIVE COOLER     *
C  *                                     *
C  ***************************************
C
C
      DIMENSION XIN(2), OUT(4), PAR(1), INFO(9)
      INTEGER RETRN
C
C
      DATA CPA,CPV,HFG/1.0, 1.805, 2501./
      DATA EPS,PA/.001, 101600./
      DATA C1,C2,C3/-5674.5354, 6.3925247, -0.9677843E-02/
      DATA C4,C5,C6/0.62215701E-06, 0.20747825E-08, -0.9484024E-12/
      DATA C7/4.1635019/
      DATA C8,C9,C10/-5800.2206, 1.3914993, -0.048860239/
      DATA C11,C12,C13/0.41764768E-04, -0.14452093E-07, 6.54596731/
C
C
      P1(Z) = -7.90298 * (Z-1.)
      P2(Z) =  5.02808 * ALOG10(Z)
      P3(Z) = -1.3816E-07 * (10. ** (11.344*(1.-1./Z))-1.)
      P4(Z) =  8.1328E-03 * (10.**(-3.49149*(Z-1.))-1.)
      P5(Z) = -9.09718 * (Z-1.)
      P6(Z) = -3.56654 * ALOG10(Z)
      P7(Z) =   .876793 * (1.-1./Z)
```

```
      P8     = -2.2199
C
C
      TDRY    = XIN(1)
      WDRY    = XIN(2)
C
C
      EEC     = PAR(1)
C
C
      INFO(6)= 4
      INFO(9)= 0
      CALL TYPECK(1,INFO,2,1,0)
      CP      = CPA + CPV*WDRY
      HDRY    = CP*TDRY + WDRY*(HFG + CPV*TDRY)
      IF(HDRY .LT. 9.67) GO TO 10
      Y       = ALOG(HDRY + 17.68)
      TWB     = 26.7453 + Y*(-43.44 + Y*(13.909 - Y*.977))
      GO TO 20
10    TT      = TDRY
      ASSIGN 30 TO RETRN
      GO TO 100
30    WSAT    = .62198 * PSAT/(1.-PSAT)
      TT      = (HDRY - HFG*WSAT) / (CPA + CPV*WSAT)
      ASSIGN 40 TO RETRN
      GO TO 100
40    WT      = .62198*PSAT/(1.-PSAT)
      IF(ABS(TT-TDRY) .GT. 0.01) GO TO 50
      TWB     = (TT + TDRY) * .5
      GO TO 20
50    SLOPE   = (WSAT - WT) / (TDRY - TT)
      A       = CPV * SLOPE
      B       = CPA + CPV*WSAT + SLOPE*(HFG - CPV*TDRY)
      C       = -HDRY + HFG*(WSAT - SLOPE*TDRY)
      IF(SLOPE .LT. 1.E-07) TWB = -C/B
      IF(SLOPE .GT. .99E-07) TWB = (-B + SQRT(B*B - 4.*A*C))/(2.*A)
20    TG1     = TWB
      TG2     = TWB + .5
      NIT     = 0
      ASSIGN 60 TO RETRN
      TT      = TG1
      GO TO 100
60    WS1     = .62198*PSAT/(1.-PSAT)
      F1      = WDRY - ((HFG - 2.381*TG1)*WS1 - (TDRY - TG1))/
     -          (HFG + CPV*TDRY - 4.186*TG1)
65    ASSIGN 70 TO RETRN
      TT      = TG2
      GO TO 100
70    WS2     = .62198*PSAT/(1.- PSAT)
      F2      = WDRY - ((HFG - 2.381*TG2)*WS2 - (TDRY - TG2))/
     -          (HFG + CPV*TDRY - 4.186*TG2)
```

```
      IF(ABS(F1-F2) .LT. 1.0E-07) GO TO 95
      TGN    = TG2 - F2*(TG2 - TG1) / (F2-F1)
      IF(ABS(TGN-TG2) .LT. EPS) GO TO 90
      TG1    = TG2
      F1     = F2
      TG2    = TGN
      NIT    = NIT + 1
      IF(NIT .GT. 100) GO TO 80
      GO TO 65
80    TWB    = TG2
      WRITE(*,210)
210   FORMAT(' FAILURE TO CONVERGE IN 100 ITERATIONS')
      GO TO 98
90    TWB    = TGN
      GO TO 98
95    TWB    = TG2
      NIT    = NIT + 1
      WRITE(*,220) NIT
220   FORMAT(' LOSS OF PRECISION AFTER',2X,I3,2X,'ITERATINS')
98    TT     = TWB
      ASSIGN 99 TO RETRN
      GO TO 100
99    WWB    = .62198*PSAT/(1.- PSAT)
      GO TO 300
C
C
100   IF(TT.GT. 0.) GO TO 105
      Z      = 273.15/(TT + 273.15)
      PSAT   = 10. ** (P5(Z) + P6(Z) + P7(Z) + P8)
      GO TO RETRN, (30,40,60,70,99)
105   Z      = 373.15/(TT + 273.15)
      PSAT   = 10. ** (P1(Z) + P2(Z) + P3(Z) + P4(Z))
      GO TO RETRN, (30,40,60,70,99)
C
C
300   TWBK   = TWB + 273.15
      IF(TWB .LT. 0.0) THEN
         PPVS = C1/TWBK + C2 + C3*TWBK + C4*TWBK**2 + C5*TWBK**3 +
     -          C6*TWBK**4 + C7*ALOG(TWBK)
         ELSE
         PPVS = C8/TWBK + C9 + C10*TWBK + C11*TWBK**2 + C12*TWBK**3 +
     -          C13*ALOG(TWBK)
      ENDIF
      PVS    = EXP(PPVS)
      WWB    = .62198*PVS/(PA - PVS)
      TEC    = TDRY - EEC*(TDRY - TWB)
      WEC    = WDRY - EEC*(WDRY - WWB)
C
C
      OUT(1) = TWB
      OUT(2) = WWB
```

```
      OUT(3) = TEC
      OUT(4) = WEC
C
C
      RETURN
      END
```

HOWE*TRNSYS.AC

```
        SUBROUTINE TYPE36(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
C
C     *************************************************
C     *                                               *
C     *   PERFORMANCE MODEL OF A VAPOR COMPRESSION    *
C     *   UNIT.  INLET AND OUTLET AIR CONDITIONS ARE  *
C     *   ASSUMED KNOWN.                              *
C     *                                               *
C     *   THIS MODEL GIVES THE LOWER PERFORMANCE BOUND *
C     *   BY ASSUMING IDEAL COOLING TO THE SATURATION *
C     *   LINE AND THEN TO THE DEWPOINT OF THE        *
C     *   DELIVERED AIR STATE.  REHEAT REQUIRED.      *
C     *                                               *
C     *************************************************
C
C
      DIMENSION XIN(10), OUT(9), INFO(9), PAR(1)
      REAL MSYS, MREG
C
C
      DATA CPA,CPV,HFG,PA/1.0, 1.805, 2501., 101600./
C
C
      MSYS    = XIN(1)
      TIEC    = XIN(2)
      WIEC    = XIN(3)
```

```
      TIECWB  = XIN(4)
      TD      = XIN(5)
      WD      = XIN(6)
      TREJ    = XIN(7)
      WREJ    = XIN(8)
      MREG    = XIN(9)
      ICOND   = XIN(10)
C
C
      TON     = PAR(1)
C
C
      INFO(6) = 9
      INFO(9) = 0
      CALL TYPECK(1,INFO,10,1,0)
      CP      = CPA + CPV*WIEC
      IF(WIEC .GT. (WD+.0001)) THEN
         PVFANS = PA*WD/ (.62198 + WD)
         ALPHA  = ALOG(PVFANS)
         TDP    = -35.957 - 1.8726*ALPHA + 1.1689*ALPHA**2
         WDP    = WD
         HDP    = CPA*TDP + WDP*(HFG + CPV*TDP)
         HIEC   = CPA*TIEC + WIEC*(HFG + CPV*TIEC)
         QEVAP  = MSYS*(HIEC - HDP)
         TAC    = TDP
         WAC    = WDP
         FLR    = QEVAP/(TON*12000.*1.0548)
         IF(FLR .GT. 1.0) FLR = 1.0
         COPAC  = .162*FLR*TON*EXP(-.183*FLR*TON) - .753*FLR -.0073*TON
     -            + 3.68 - .03998*(TREJ - TIECWB - 15.)
         QCOND  = QEVAP*(1.+1./ COPAC)
         TCOND  = TREJ + QCOND/(MREG*CP)
         WORK   = QCOND - QEVAP
         IONAC  = 1
         ELSE
         IF(TIEC .GT. TD) THEN
            QEVAP  = MSYS*CP*(TIEC - TD)
            TAC    = TD
            WAC    = WIEC
      FLR     = QEVAP/(TON*12000.*1.0548)
      IF(FLR .GT. 1.0) FLR = 1.0
      COPAC   = .162*FLR*TON*EXP(-.183*FLR*TON) - .753*FLR - .0073*TON +
     -           3.68 - .03998*(TREJ - TIECWB - 15.)
            QCOND  = QEVAP*(1.+1./ COPAC)
            TCOND  = TREJ + QCOND/(MREG*CP)
            WORK   = QCOND - QEVAP
            IONAC  = 1
            ELSE
            QEVAP  = 0.0
            TAC    = TIEC
            WAC    = WIEC
```

```
            COPAC  = 1000.
            QCOND  = 0.0
            WORK   = 0.0
            TCOND  = TREJ
            IONAC  = 0
         ENDIF
      ENDIF
      WCOND   = WREJ
      IF(ICOND .EQ. 0) QCOND = 0.0
C
C
      OUT(1)  = QEVAP
      OUT(2)  = COPAC
      OUT(3)  = QCOND
      OUT(4)  = TCOND
      OUT(5)  = WCOND
      OUT(6)  = IONAC
      OUT(7)  = WORK
      OUT(8)  = TAC
      OUT(9)  = WAC
C
C
      RETURN
      END
```

HOWE*TRNSYS.AUXHTR

```
      SUBROUTINE TYPE37(TIME,XIN,OUT,T,DTDT,PAR,INFO)
C
C
C  *********************************************
C  *                                           *
C  *  MODEL DETERMINES THE AMOUNT OF AUXILIARY *
C  *  ENERGY REQUIRED FOR REGENERATION ASSUMING *
C  *  BOTH AN INFINITE STORAGE AND A NO-STORAGE *
C  *  SYSTEM.                                  *
C  *                                           *
C  *********************************************
C
C
      DIMENSION XIN(4), OUT(19), INFO(9)
      REAL MSYS
C
C
      DATA CPA,CPV/1.0,1.805/
C
C
      MSYS   = XIN(1)
      THAVE  = XIN(2)
      TREQD  = XIN(3)
      WHAVE  = XIN(4)
C
C
      IF(INFO(7) .EQ. -1) THEN
         INFO(6) = 2
         INFO(9) = 0
         CALL TYPECK(1,INFO,4,0,0)
         EXSTOR  = 0.0
         ELSE
         IF(INFO(7) .EQ. 0) THEN
            EXSTOR = OUT(18)
            OUT(19) = EXSTOR
            ELSE
            EXSTOR  = OUT(19)
         ENDIF
      ENDIF
      CP     = CPA + CPV*WHAVE
      QAUX   = MSYS * CP * (TREQD - THAVE)
      IF(QAUX .LT. 0.0) THEN
         EXCESS = -QAUX
         EXSTOR = EXSTOR - QAUX
         QAUX   = 0.0
         QAUXST = 0.0
```

```
      USE    = 0.0
      ELSE
      EXCESS = 0.0
      QAUXST = QAUX - EXSTOR
      IF(QAUXST .LT. 0.0) THEN
         EXSTOR = -QAUXST
         QAUXST = 0.0
         USE    = QAUX
         ELSE
         USE    = EXSTOR
         EXSTOR = 0.0
      ENDIF
   ENDIF
C
C
   OUT(1) = QAUX
   OUT(2) = EXCESS
   OUT(3) = QAUXST
   OUT(4) = USE
   OUT(18)= EXSTOR
C
C
   RETURN
   END
```

REFERENCES

[1] Spielvogel, L.G., "Office Building Energy Use," ASHRAE Journal, Nov. 82, pp. 25-31.

[2] Dunkle, R.V., "A Method of Solar Air Conditioning," Mech. and Chem. Engr. Trans., Inst. Engrs. Australia, MCI, Vol. 73, May 1965.

[3] Nelson, J.S., et al., "Simulations of the Performance of Open Cycle Desiccant Systems Using Solar Energy," Solar Energy, 21, 273 (1978).

[4] Oonk, R.L., et al., "Performance of Solar-Desiccant Cooling Systems in New York City." Proceedings of the 1978 Annual Meeting of AS/ISES, 460, (1978).

[5] Jurinak, J.J. and Beckman, W.A., "A Comparison of the Performance of Open Cycle Air Conditioners Utilizing Rotary Desiccant Dehumidifiers," Proceedings of the 1980 Annual Meeting of AS/ISES, (1980).

[6] Schultz, K.J., and Mitchell, J.W., "The Performance of Solar-Desiccant Systems Using Cooled Desiccant Beds," Proceedings of the 1982 Annual Meeting of AS/ISES, 585-590 (1982).

[7] Close, D.J. and Sheridan, J., "Low Energy Cooling for Humid Regions," Australian Institute of Refrigeration, Air Conditioning and Heating, (AIRAH) Federal Conference, Tasmania, (1982).

[8] Sheridan, J.C. and Mitchell, J.W., "Hybrid Solar Desiccant Cooling Systems," Proceedings of the 1982 Annual Meeting of AS/ISES, (1982).

[9] Jurinak, J.J., "Open Cycle Desiccant Cooling--Component Models and System Simulations," Ph.D. thesis, University of Wisconsin-Madison, Aug. (1982).

[10] ASHRAE, ASHRAE Handbook 1981 Fundamentals, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, New York (1981).

[11] ASHRAE, ASHRAE Handbook 1978 Applications, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, New York (1978).

[12] Banks, P.J., "Prediction of Heat and Water Vapour Exchanger Performance from that of a Similar Heat Exchanger," Compact Heat Exchangers--History, Technological Advancement, and Mechanical Design Problems (ASME Winter Annual Meeting), HTD-10, 57, Nov. (1980).

[13] Pescod, D., "Unit Air Cooler Using Plastic Heat Exchanger with Evaporating Cooled Plates," Australian Refrigeration, Air Conditioning and Heating, 22, 9 (1968).

[14] Product literature, "Trane Rooftop Air Conditioners, 20 to 75 Tons," Form DS PKG-4, The Trane Company, La Crosse, WI (July 1981).

[15] Product literature, "Carrier Weathermaster III Heat Pump System," form 38HQ-1P, Carrier Corporation, Syracuse, NY.

[16] Ottenstein, J., "Application of Solar Energy in Multizone Buildings," M.S. thesis in Mechanical Engineering, University of Wisconsin-Madison (1979).

[17] Klein, S.A., et al., "TRNSYS--A Transient Simulation Program." University of Wisconsin-Madison, Engineering Experiment Station Report 38-10, Version 11.1 (1981).

[18] ASHRAE, ASHRAE Handbook 1972 Fundamentals, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, New York (1972).

[19] SOLMET Typical Meteorological Year, Tape Deck 9734, National Oceanic and Atmospheric Administration, Environmental Data Service, National Climatic Center, Asheville, North Carolina.

[20] Schultz, K.J., "The Performance of Desiccant Dehumidifier Air-Conditioning Systems Using Cooled Dehumidifiers," M.S. thesis in Mechanical Engineering, University of Wisconsin-Madison (1983).

[21] Howe, R.R., Beckman, W.A., and Mitchell, J.W., "Commercial Applications for Solar Hybrid Desiccant Systems," Proceedings of the 1983 Annual Meeting of ASES, (1983).

[22] Howe, R.R., Beckman, W.A., and Mitchell, J.W., "Factors Influencing the Performance of Commercial Hybrid Desiccant Air Conditioning Systems." To be published in the Proceedings of the 18th Intersociety Energy Conversion Engineering Conference, (1983).

[232 Duffie, J.A., and Beckman, W.A., Solar Engineering of Thermal Processes, Wiley-Interscience, New York, 1980.

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