AN ANALYSIS OF HYBRID DESICCANT COOLING SYSTEMS IN SUPERMARKET APPLICATIONS

by

PHILIP RICHARD BURNS

A thesis submitted in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE
(Engineering)

at the

UNIVERSITY OF WISCONSIN-MADISON

1985

AN ANALYSIS OF HYBRID DESICCANT COOLING SYSTEMS IN SUPERMARKET APPLICATIONS

Philip Richard Burns

MASTER OF SCIENCE
(Engineering)

UNIVERSITY OF WISCONSIN-MADISON

1985

ACKNOWLEDGMENTS

I would like to express my thanks to Professors John Mitchell and W. A. Beckman for their assistance and encouragement throughout this project. Their patience and understanding were often relied upon and greatly appreciated. My thanks to Professors J. A. Duffie and S. A. Klein for their friendship and support throughout my stay. To all four, goes my gratitude for creating an environment as pleasant as that in the Solar Lab.

As for my fellow graduate students, I can only wish I had more time to spend with you. I leave greatly enriched largely due to what you have imparted on me. Thank you for sharing your knowledge, your opinions, your spontaneity, your time and your friendship. Special thanks must go to Hank and Alan upon whose shoulders fell the large part of teaching me a new subject. Your assistance and friendship, particularly during that first semester, is deeply appreciated.

I would like to thank Andy Levine, Rohit Arora, and Diane Manley of Thermo Electron and Nancy Banks of Cargocaire for their help and interest in this work. Funding for this project was provided by the Solar Heating and Cooling Research and Development Branch, Office of Conservation and Solar Applications, U.S. Department of Energy.

Lastly, to my family, thank you for your support and encouragement over the years which has made all this possible.

ABSTRACT

Supermarkets present a unique air-conditioning situation. Open refrigerated cases inside a supermarket provide the majority of a store's sensible cooling needs. The primary role of a air-conditioning system becomes one of dehumidification. Traditional vapor compression cooling does not remove this type of load efficiently.

Desiccant dehumidifiers can be combined with vapor compression systems to perform the required moisture removal. The vapor compression unit only provides sensible cooling. Hybrid desiccant cooling systems eliminate cooling to the saturation line and subsequent reheating. The desiccant must be regenerated with a heat source.

Numerous possible system configurations are studied. Heat exchange, indirect evaporative cooling, solar energy and condenser heat all have potential benefits in these cycles. Mathematical models for individual components are developed.

Fixed condition studies explore the energy trade-offs among system components and between the refrigeration cases and the air conditioning system. Annual simulation studies are performed for a variety of U.S. locations. Results presented suggest potential reductions in air-conditioning costs of 50-70%.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

Figure		Page

CHAPTER 1

INTRODUCTION

The highly competitive supermarket industry operates on a low margin basis, relying on high sales volume to generate profits. Minimizing expenses is extremely important. The energy cost of maintaining a suitable environment for food products and customers concerns industry executives. In a 1981 survey, groups of chain executives, wholesale executives, independent owners and chain managers each listed energy costs as their number one worry (1). The industry as a whole consumes four percent of the United States annual electrical energy usage (2).

Methods for reducing electrical energy consumption can expect to receive considerable attention from the supermarket industry. Many research efforts are being conducted to explore ways to reduce energy consumption. One area where electrical energy may be saved is in air conditioning. The work reported in this thesis evaluates the potential savings obtained by alterations in the store's air conditioning system. In particular, hybrid desiccant cooling systems are studied in typical supermarket applications.

In the mid-1960's, an Australian researcher, Robert V. Dunkle proposed an alternative air conditioning system which utilized solar energy instead of the electrical energy of the traditional vapor compression cycle (3). The proposed system involved the use of a desiccant for dehumidification and heat exchangers and evaporative coolers for sensible cooling. Considerable research has been performed on the development of desiccant systems for residential applications (4-8). Jurinak (9) provides an outstanding overview of this research. Sheridan and Close (10) extend the concept to include commercial sized loads. Systems combining desiccant dehumidifiers and vapor compression machines have been proposed for commercial applications. Hybrid desiccant cooling systems studied by Sheridan and Mitchell (11), and Howe (12) have shown potential electrical energy savings of up to 40%. In addition, hybrid systems perform well in high latent load situations and provide close humidity control.

Being large consumers of electrical energy with favorable load characteristics, supermarkets have been targeted as a possible candidate for hybrid desiccant systems. Supermarkets have high latent load ratios and require strict humidity control primarily due to the presence of open refrigeration cases within the store. Thermo Electron, under the sponsorship of the Gas Research Institute, monitored a field development installation in a Jewel supermarket outside Chicago, Illinois, and performed much analytical work (13). CargoCaire, after establishing a couple of test installations in Texas, now markets a hybrid desiccant system designed for supermarket applications (14).

The current study evaluates the potential benefits of installing hybrid cooling systems in supermarkets. The specific project objectives are:

1) Develop or adopt load and component models suitable to the study of these systems utilizing computer simulation methods.
2) Determine the feasibility of applying desiccant systems in supermarkets.
3) Explore various system configurations and methods of obtaining regeneration heat and free cooling. Condenser heat, heat exchange, indirect evaporative cooling and solar energy all hold possible benefits.
4) Determine geographic regions well suited to this application.

Understanding the benefits of using hybrid desiccant systems needs some discussion of air conditioning processes and supermarket cooling requirements provided in Chapters 2 and 3. Chapter 4 describes the models used to simulate hybrid cooling systems. Chapters 5 and 6 report results obtained from fixed condition and yearly simulation studies. Conclusions are discussed in Chapter 7.

CHAPTER 2

HYBRID DESICCANT COOLING SYSTEMS

Section 2.1 Air Conditioning and Vapor Compression Cooling

The typical air conditioning process consists of two distinct components; moisture removal (latent cooling) and temperature reduction (sensible cooling). Figure 2.1.1 illustrates this breakdown on a psychrometric diagram. The process of receiving air at state A and supplying air at state B requires that enough moisture be removed to reduce the humidity ratio to that at state B. The energy required for this process can be expressed as,

$$\dot{Q}_l = \dot{m} * h_{fg} * (w_a - w_b) \qquad (2.1.1)$$

where $\dot{m}$ is the air mass flow rate, h_{fg}, the heat of vaporization, and w, the absolute humidity ratio. The air must also be sensibly cooled to temperature B requiring an energy expenditure of,

$$\dot{Q}_s = \dot{m} * C_p * (T_a - T_b) \qquad (2.1.2)$$

where C_p is the specific heat of air, and T, the air temperature. Adding the two components of the load together provides the minimum amount of energy required to bring air from state A to state B.

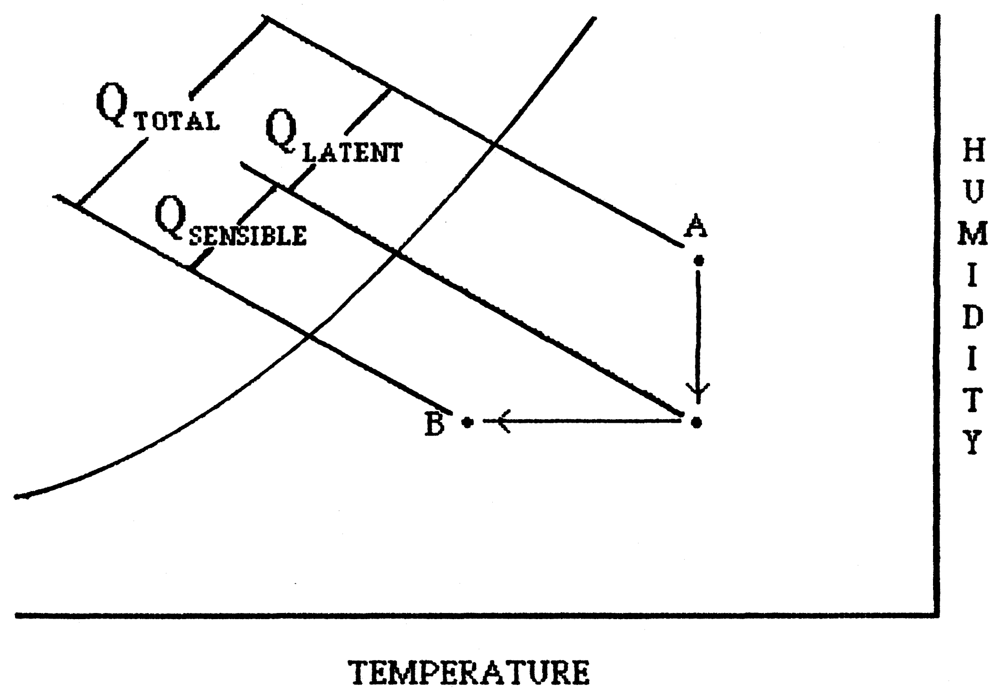


Figure 2.1.1 A typical air conditioning situation broken down into sensible and latent loads

The fraction of the total load required for moisture removal defines the latent load ratio,

$$LLR = \dot{Q}_l/\dot{Q}_t = \dot{Q}_l/(\dot{Q}_s + \dot{Q}_l) \tag{2.1.3}$$

The LLR describes the line on which supply air states must fall in order to meet both components of the load. Figure 2.1.2 shows load lines for LLR's of 0, .2, .65, and 1. The load line with a LLR of 0 will be a horizontal line, requiring strictly sensible cooling. For a LLR of 1 the load line is vertical, requiring only dehumidification. The length of the load line depends on the amount of air processed. A given cooling load may be expressed as

$$\dot{Q}_t = \dot{m} * (h_r - h_s) \tag{2.1.4}$$

where h_r is the room air state enthalpy, and h_s, the supply state enthalpy. On a psychrometric chart the length of the load line relates to the enthalpy difference between the room and supply air. If the air conditioning system processes larger amounts of air, a smaller enthalpy difference is needed.

The traditional means of air conditioning is by vapor compression. Vapor compression is a four step cycle in which a refrigerant vapor is compressed to a higher temperature and pressure, condensed at this higher temperature, throttled to a lower

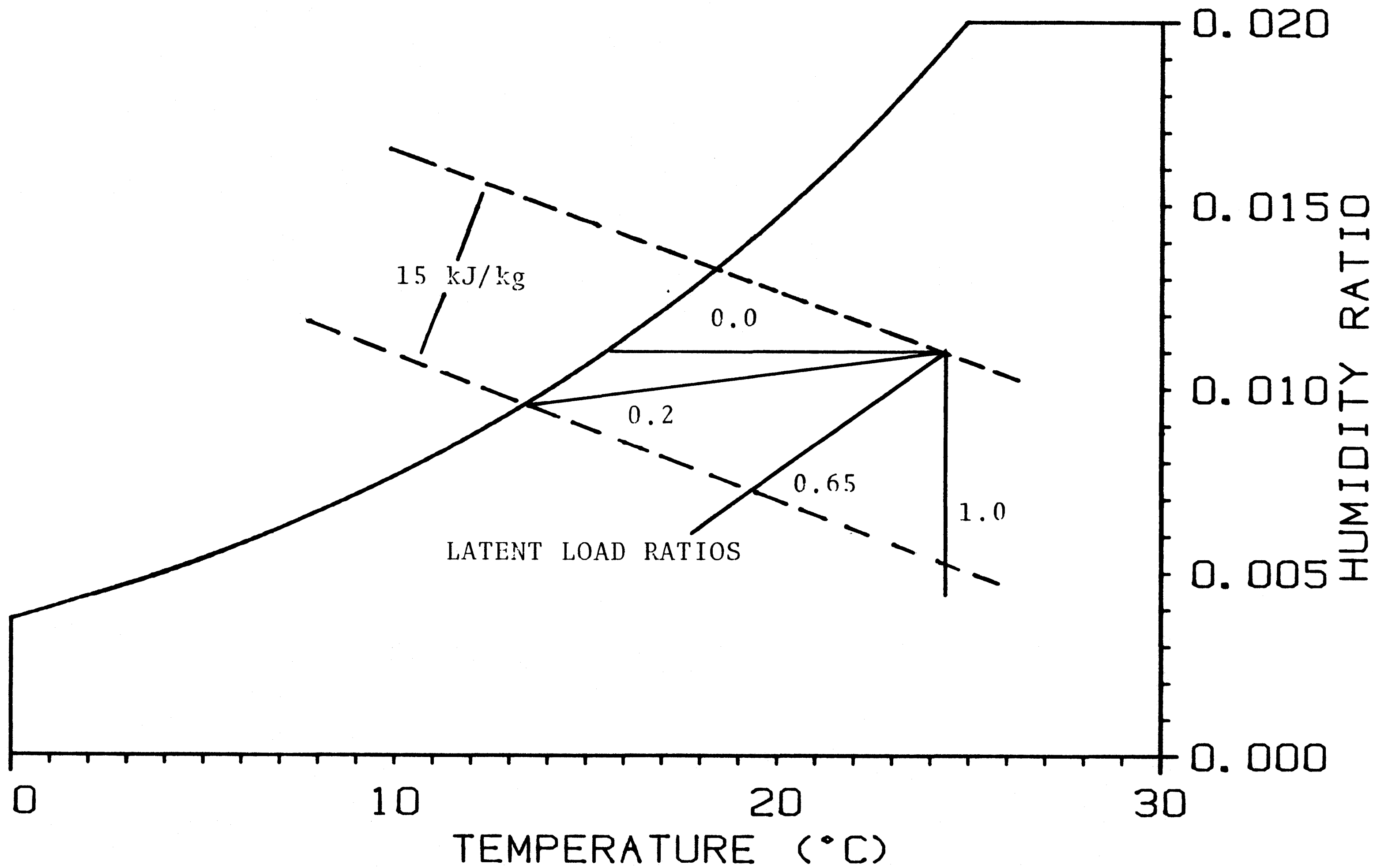


Figure 2.1.2 Load lines representing various latent load ratios

pressure and temperature, and finally evaporated at the lower temperature. The compression process requires mechanical work. Heat is rejected to the ambient during condensation and extracted from the conditioned space during evaporation. For these heat transfer processes to occur the evaporating temperature must be lower than the temperature of the conditioned air and the condensing temperature higher than the air receiving the condenser heat. The work requirement for compression increases with the difference in the evaporator and condenser temperatures.

Ideally, air conditioning should use the least amount of energy possible. Vapor compression, however, is inherently a means of cooling only. To dehumidify using a vapor compression machine, air must be cooled to its saturation temperature (dew point) and then further cooled to condense the water vapor out of the air. Often the saturation temperature at the desired humidity level is colder than the desired supply temperature. If this is the case, the air must be reheated to the supply temperature. Figure 2.1.3 shows a schematic diagram and a psychrometric chart of a typical vapor compression system. Return air mixes with outside air to meet minimum ventilation requirements (state 2). The mixed air passes over the evaporator of the vapor compression machine cooling it to state 3. If any reheat is required, the air is then heated to state 4 on the load line and supplied to the building maintained at state 5. Supply air states near saturation provide the most efficient

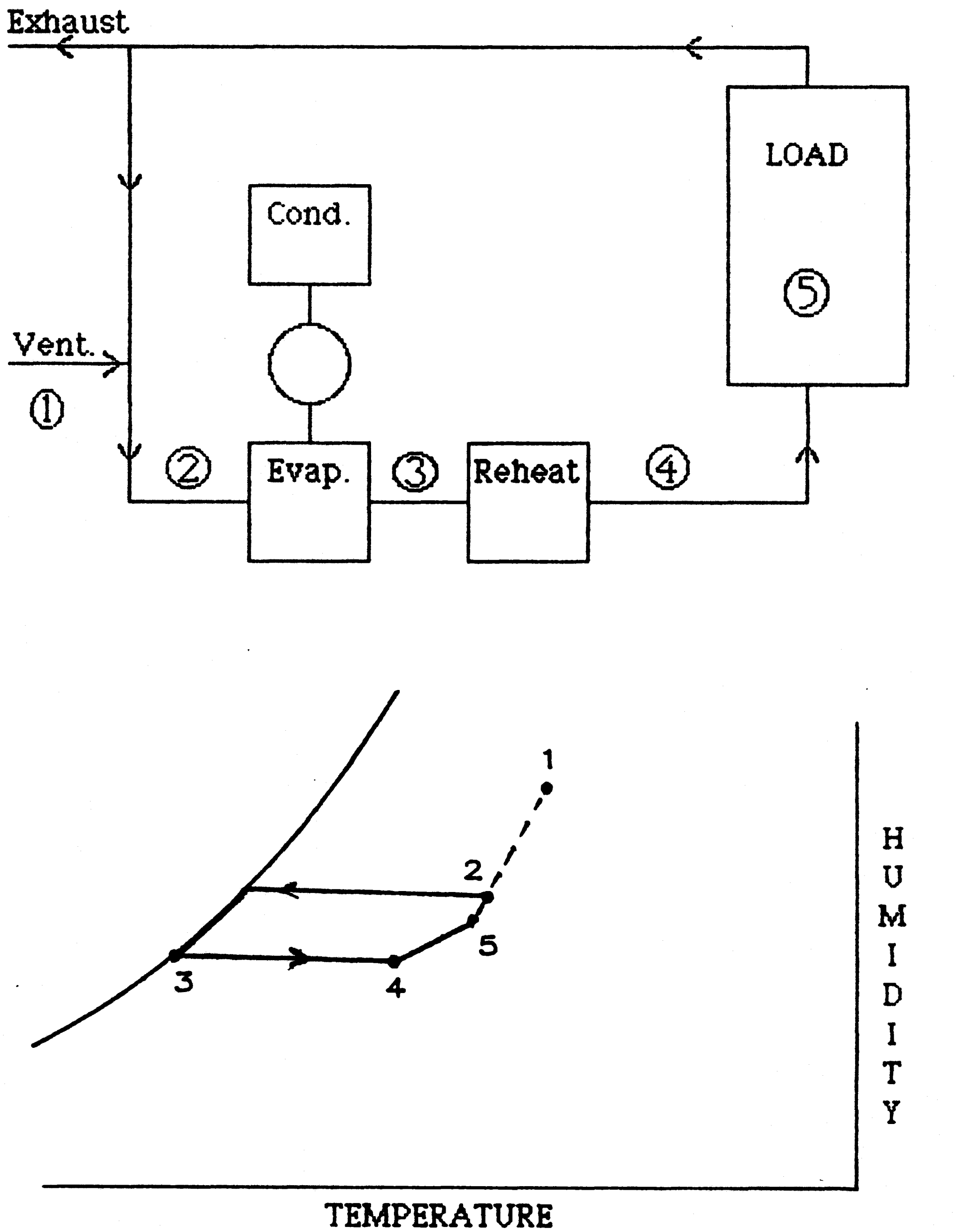


Figure 2.1.3 Schematic diagram and psychrometric chart of a conventional vapor compression air conditioning system

operation of a vapor compression system. The amount of heat supplied to the evaporator is equal to the cooling load. As supply temperatures increase away from saturation, the evaporator requirements remain unchanged but the cooling load decreases. That is, the heat requirements in the evaporator become greater than those required to meet the load.

There exists a general rule of thumb in air conditioning which recommends that no more than 400 cubic feet per minute (cfm) of air be processed for every ton of cooling required. This suggests a minimum enthalpy difference of about 15 kJ/kg (6.5 Btu/lb) between the supply air state and the room air state be maintained. In typical residential or commercial air conditioning situations latent load ratios tend to be around 0.20. A 15 kJ/kg difference on a load line with an LLR of 0.2 falls very close to the saturation line. In this typical type of cooling situation, a vapor compression system operates very efficiently.

A load line with an LLR of 0.65 never intersects the saturation line. Maintaining the 15 kJ/kg difference means not only a substantial increase in the evaporator requirement to cool air to a lower dew point, but also a decrease in the coefficient of performance of the vapor compression unit as a lower evaporator temperature would be needed. This COP decrease makes supply states at lower humidity levels undesirable. Processing larger amounts of air will meet loads with higher LLR's and maintain supply humidity ratios at practical levels. The enthalpy difference between supply and room air states is decreased.

In situations with high LLR's, supply air states will lie well away from the saturation line. Since air must still be cooled to the dew point, the evaporator energy requirement is larger than the size of the load and reheat energy is required. As a method of comparison, the coefficient of performance of an air conditioning system is often used to evaluate the relative performance of various systems. The cooling COP of a vapor compression machine is normally defined as the

$$\mathrm{COP} = \frac{\text{Heat removed}}{\text{Work input}} \qquad (2.1.5).$$

More descriptive in energy cost would be the following definition,

$$\mathrm{COP}_{\mathrm{eff}} = \frac{\text{Load}}{\text{Work}} = \frac{(\text{Heat removed - reheat})}{\text{Work}} \qquad (2.1.6)$$

No matter how good the performance of a vapor compression machine, if the desired supply state lies away from the saturation line the effective COP of the cooling system will be substantially less than the COP of the cooling unit.

We can imagine a situation where the moisture removal is performed prior to sensible cooling. On Figure 2.1.4, this refers to a change in air state from A to B. For a desired supply state on the saturation line, state C, the evaporator load is reduced only by

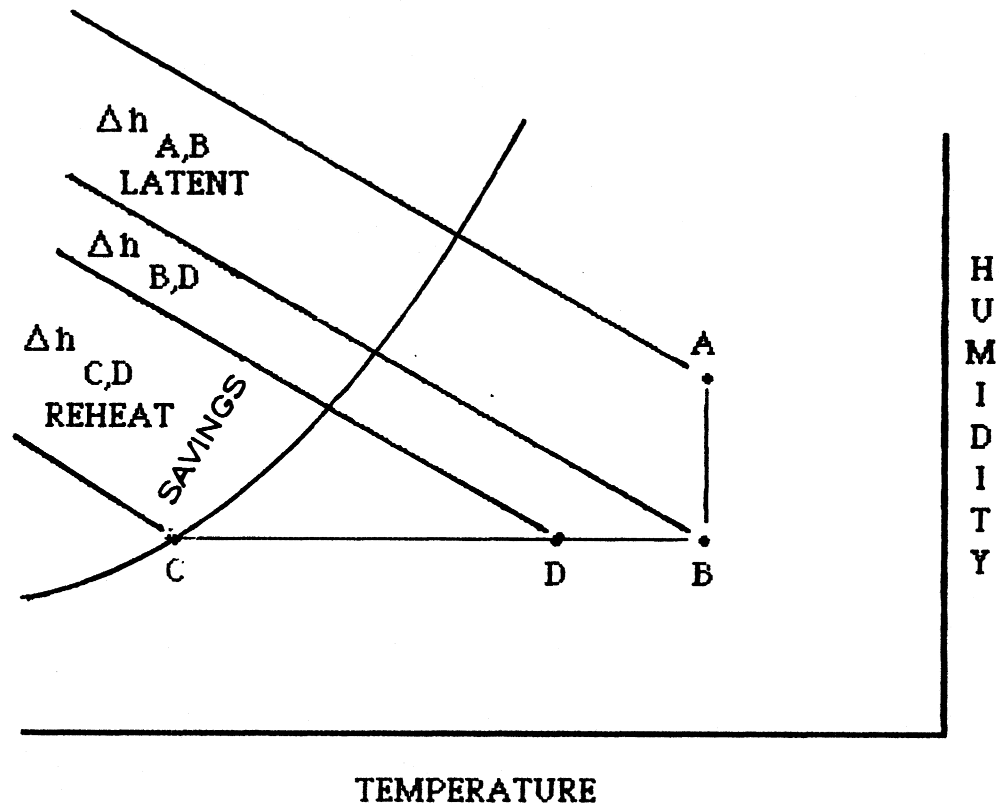


Figure 2.1.4 Energy savings possible by performing moisture removal before sensible cooling

the size of the latent load. However, for a desired supply state away from the dew point, state D, the evaporator load is reduced by both the latent load and the amount of reheat previously performed. Since the evaporator supplies higher temperature air, the evaporator temperature may be elevated, increasing the COP of the vapor compression unit. While moisture removal is not free, this example illustrates the potential reduction in cooling requirements when using an alternative to vapor compression for supply states away from saturation.

Section 2.2 Desiccant Cooling Systems

A possible method to remove water vapor from the air is adsorption by a desiccant material. During this process water vapor is adsorbed on the surface of the desiccant. This process is approximately a constant enthalpy process releasing the heat of adsorption to the air. This results in no reduction in the total cooling load, however the latent load has been replaced by an addition to the sensible load. Since the resulting air is both hotter and dryer than the ambient, the additional sensible load can be reduced by the use of heat exchangers and evaporative coolers. The desiccant, however, cannot adsorb infinite amounts of moisture, and must periodically be regenerated. This is accomplished by passing hot air over the desiccant to drive off the adsorbed water.

Figure 2.2.1 illustrates this process using a rotary wheel configuration in which the desiccant matrix passes alternately between process and regeneration air streams. Warm, damp process air (state 1) passes through the desiccant where the air is dried and heated to state 2. On the regeneration side, air must be heated to the required regeneration temperature (state 3). This heat can come from any available thermal energy source, such as a gas burner, solar energy, or a waste heat source like condenser heat. This hot air passes through the desiccant desorbing the water and exits, warm and very wet (state 4).

Two of the more extensively studied cycles which utilize a desiccant dehumidifier have been called the ventilation and recirculation cycles (9). Figure 2.2.2 depicts a schematic diagram and psychrometric chart illustrating the ventilation cycle. The ventilation cycle, as its name suggests, introduces ambient air as the process stream. Ambient air (state 1) passes through the dehumidifier where it is dried and heated by the adsorption process (state 2). This hot, dry air is sensibly cooled by a rotary heat exchanger using evaporatively cooled room air as the heat sink (state 3). The process air coming out of the heat exchanger can be further cooled by an evaporative process which brings it to the load line where it is supplied to the room (state 4). On the regeneration side, heat must be added to the air coming off the rotary heat exchanger to reach the regeneration temperature. The

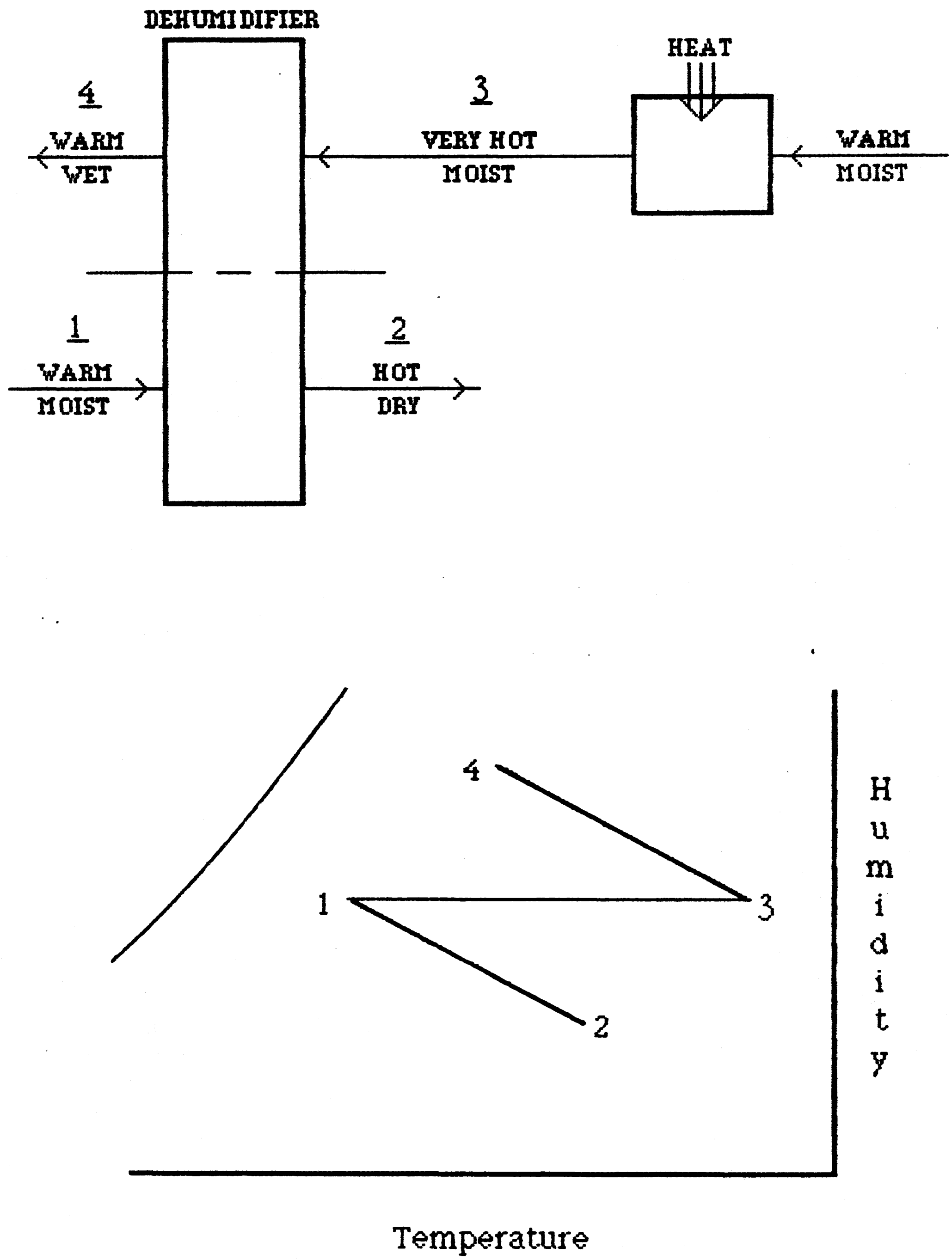


Figure 2.2.1 Representation of the air states resulting from desiccant dehumidifier processes

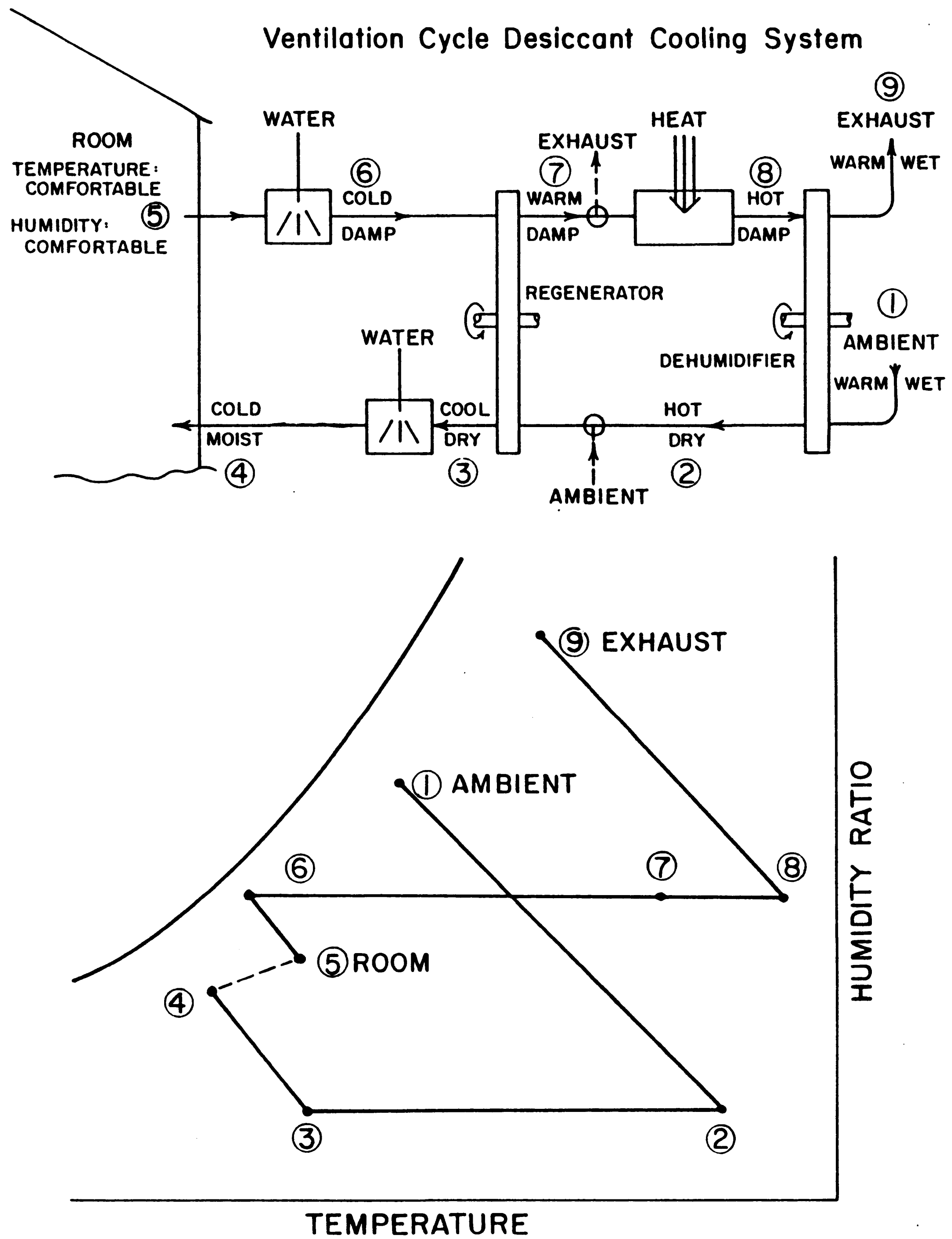
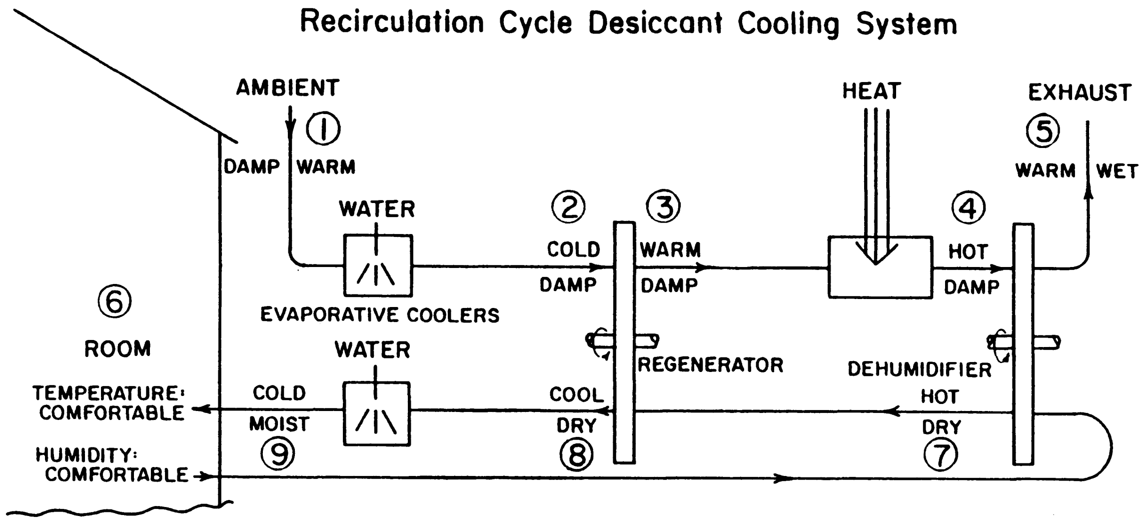


Figure 2.2.2 Schematic diagram and psychrometric chart of a ventilation cycle desiccant cooling system (from reference 9)

recirculation cycle shown in Figure 2.2.3 is similar to the ventilation cycle, the only difference being the sources of the process and regeneration air streams. Except for the parasitic power required to turn the wheels and operate the fans the only energy required to operate these systems is the auxiliary heat needed for regeneration. Thus thermal energy completely replaces the high grade electrical energy consumed by the vapor compression system.

Section 2.3 Hybrid Desiccant Systems

A hybrid desiccant cooling system utilizes both a desiccant dehumidifier and a vapor compression machine to meet the cooling load. The desiccant removes the latent load while the vapor compression unit handles the portion of the sensible load remaining after any heat exchange or evaporative cooling processes. As discussed in section 2.1, removing the latent load previous to passing the air over the evaporator coil reduces the electrical energy requirements of the vapor compression unit. This is especially true when supplying air away from the saturation line as occurs in high latent load situations. The evaporating temperature may be raised and the need for reheat due to excess cooling eliminated. Hybrid desiccant cooling systems separate the process of humidity control from temperature control. In the vapor compression system humidity may only be controlled by reducing the



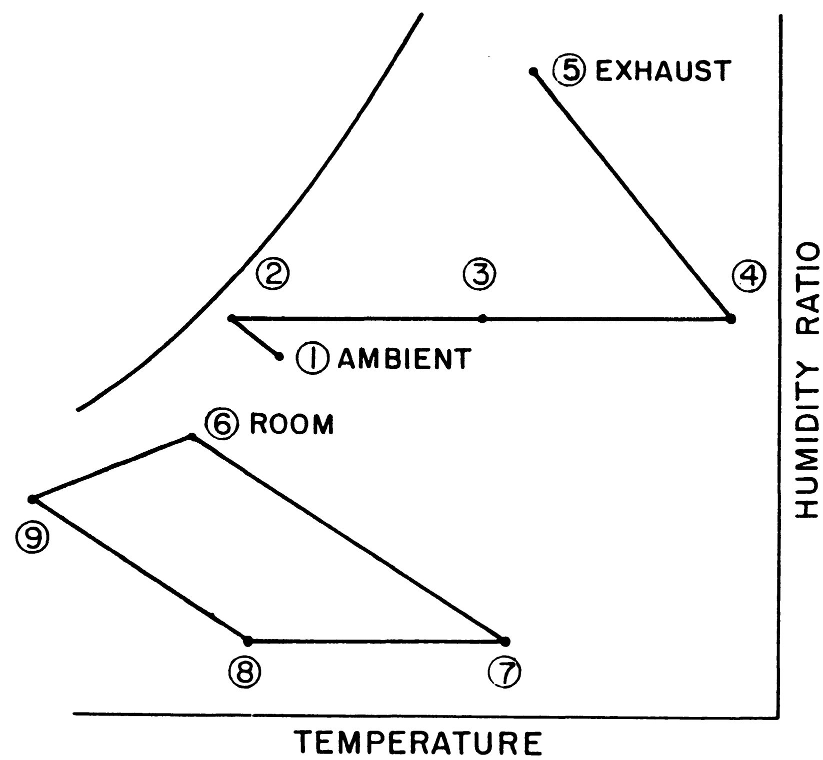


Figure 2.2.3 Schematic diagram and psychrometric chart of a recirculation cycle desiccant cooling system (from reference 9)

temperature. In hybrid systems, the desiccant dehumidifier regulates humidity and the vapor compression machine regulates temperature.

Figure 2.3.1 illustrates a hybrid cycle receiving considerable attention in previous work (11, 12). This cycle, which processes all of the recirculation air through the desiccant and utilizes heat rejected in the condenser to preheat the regeneration air stream, will be called the recirculation/condenser cycle. In this cycle, return air from the building mixes with ventilation air (state 1)and passes through the desiccant. The adsorption process in the desiccant releases latent heat causing hot, dry air to leave the dehumidifier (state 2). The air stream is then cooled with an indirect evaporative cooler (state 3). The vapor compression unit performs the remainder of the sensible cooling and the conditioned air stream is supplied to the building (state 4). To regenerate the desiccant, waste condenser heat is used to preheat ambient air (state 7). An auxiliary heat source provides any further heating necessary (state 8). The regeneration air stream is cooled and humidified as it passes through the desiccant and then exhausted to the outside (state 9).

Two other possible configurations will also be studied in some detail in this work. Both of these cycles only process the outside ventilation air through the desiccant. One, the ventilation/ condenser cycle shown in Figure 2.3.2, is similar to the

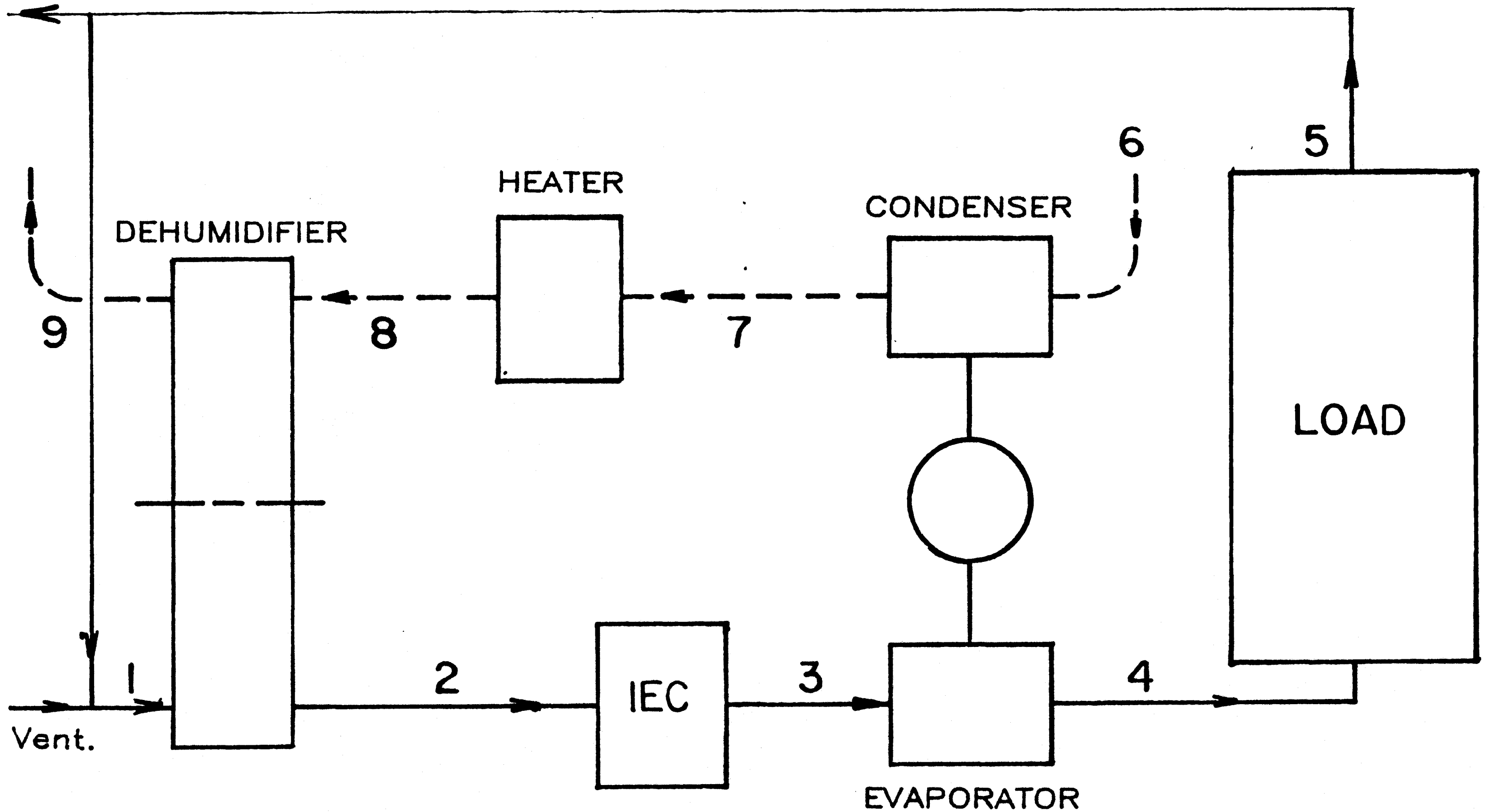


Figure 2.3.1 Schematic diagram of recirculation/condenser hybrid desiccant cooling system

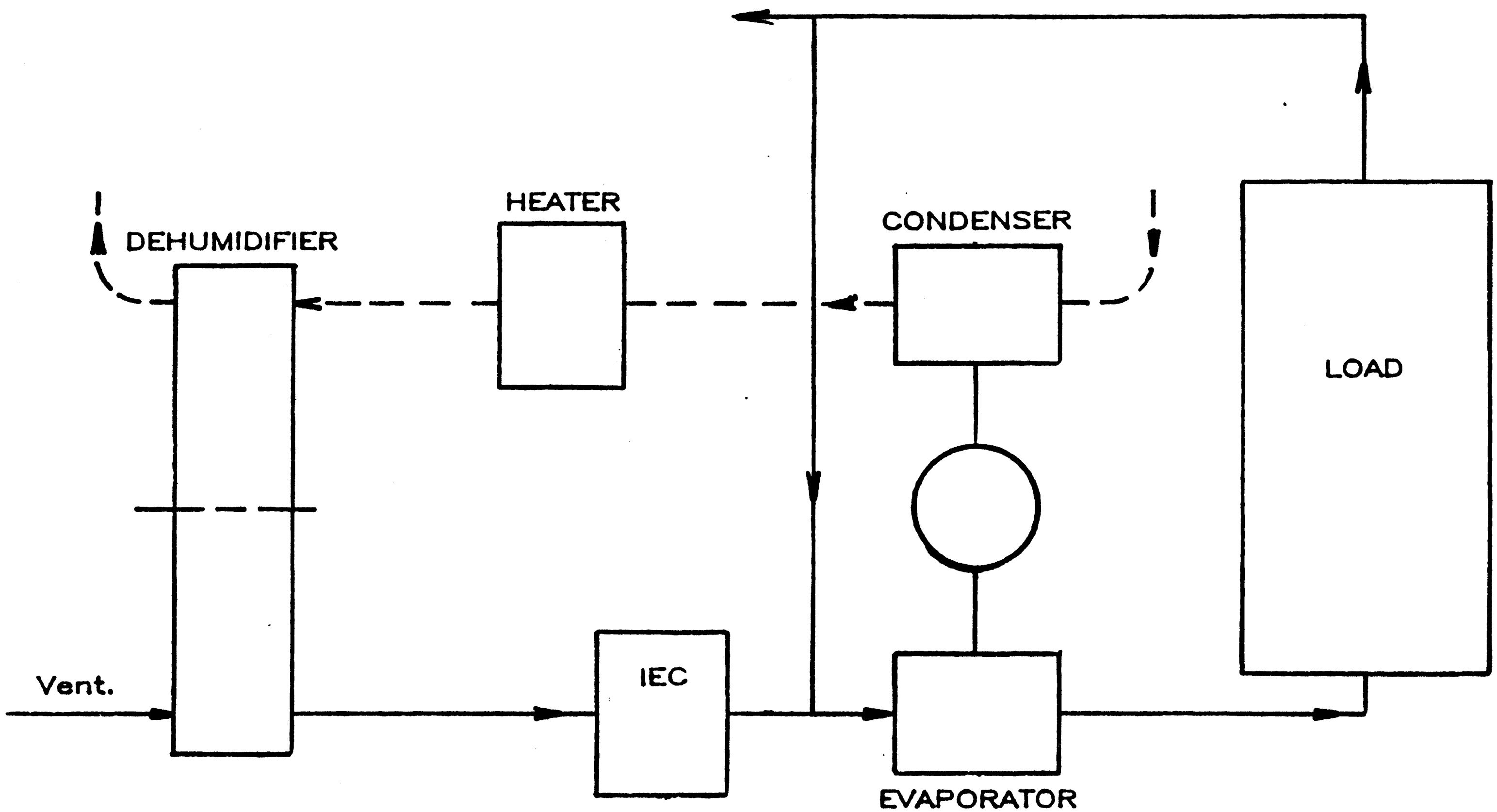


Figure 2.3.2 Schematic diagram of the ventilation/condenser hybrid desiccant cooling system

recirculation/condenser cycle with the only difference being the air processed. Condenser heat is utilized to preheat the regeneration air stream and an indirect evaporative cooler provides some free cooling on the process side. The third cycle considered, the ventilation/heat exchanger cycle shown in Figure 2.3.3, places a rotary heat exchanger between the process and regeneration streams to provide both cooling and heating to the respective streams. The use of the name ventilation might cause some confusion. Ventilation cycles often refer to systems which supply only processed ambient air to the space, where here it refers to systems which dehumidify make up air only before mixing with the return air from the store.

Various comments and comparisons may be made about these systems. The recirculation/condenser cycle processes much more air through the dehumidification components than the ventilation cycles. This results in smaller humidity drops and lower regeneration temperatures for the recirculation cycle, but also much larger equipment (larger initial costs) and larger fan power requirements. Various energy trade-offs exist within the cycles which utilize condenser heat. Cooling performed by the indirect evaporative cooler reduces the amount of electrical work performed by the vapor compression unit. This in turn means that less condenser heat will be available to heat the regeneration air stream. Another trade-off occurs with the use of condenser heat.

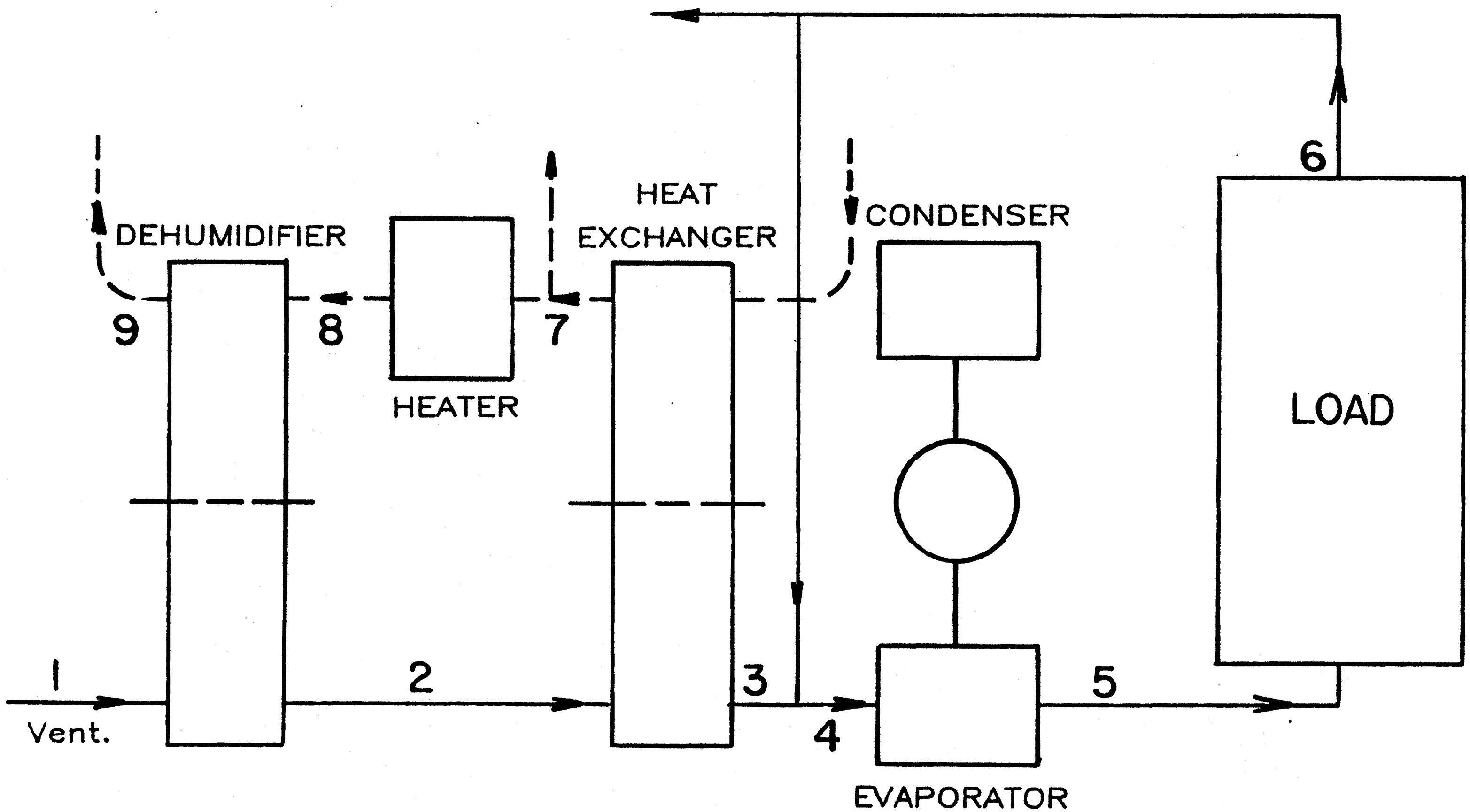


Figure 2.3.3 Schematic diagram of the ventilation/heat exchanger hybrid desiccant cooling system

Achieving high air temperatures leaving the condenser means that the condensing temperature must be even higher. This elevation in condensing temperature degrades the performance of the vapor compression unit and increases the electrical energy consumption required to meet the load on the evaporator. Both these situations present choices between more efficient cooling on the process side and more efficient heating on the regeneration side.

The previous discussion has introduced hybrid desiccant cooling systems and briefly touched on their configuarions and under what conditions they would be effective. Hybrid desiccant systems are effective in situations which require strict humidity control and supply states which lie away from the saturation line. The work that follows applies these systems to supermarket applications and details the interactions between the loads and the various components which make up the systems.

CHAPTER 3

SUPERMARKETS

Section 3.1 Supermarket Energy Consumption

Supermarkets consume tremendous amounts of electrical energy. This industry alone is responsible for four percent of this country's electrical energy usage, about 88 billion kWh annually (2). At an average of $0.07/kWh electricity cost the supermarket industry spends over $6 billion a year for electricity. This gives some impression of the potential benefit of any energy saving methods introduced into the supermarkets. Table 3.1.1 shows a breakdown of the annual electrical energy usage in an average supermarket (13). The air conditioning percentage increases during the cooling season. For stores in southern areas with longer cooling seasons, the annual percentage will be higher.

The major consumer, as could be expected, is the open refrigeration cases within the store. The relative magnitude of the case consumption, however, is a little misleading as these cases perform a large portion of both the heating and cooling in the store as well as the refrigeration requirements. The exposure of these refrigerated cases to the store environment cools the store air reducing the cooling requirments of an air conditioning system. In

Table 3.1.1
Average Supermarket Annual Electrical Energy Usage

	%	KWhr
Refrigeration	55	1,270,000
Lighting	23	530,000
Heating	8	180,000
Air Conditioning	6	140,000
Miscellaneous	8	180,000
Total	100	2,300,000

addition, heat available from the condensers of these cases meets most of the heating load. The fact remains however that the majority of the energy consumption lies with the refrigeration cases. Improvements in case performance have the largest energy saving potential, and work being done in this area runs from putting doors on the cases to developing improved compressors (15).

Despite being a relatively small part of the supermarket energy bill, considerable savings can be attained in air conditioning costs. The presence of the refrigerated cases alters the cooling loads in such a way that a traditional vapor compression system cools a supermarket very inefficiently. Hybrid desiccant systems meet these loads in a more effective manner. A detailed discussion of the loads found in the supermarket and the refrigerated cases effects on these loads provides a fuller picture of the potential of hybrid cooling systems in supermarkets.

Section 3.2 Refrigerated Cases

A typical supermarket may have up to fifty tons in refrigeration capacity within a store. Generally refrigerated cases are left open to facilitate shoppers' removal of food items from display shelves. Maintaining the desired refrigeration temperature with the constant exposure to the warm and moist store environment consumes a large amount of energy in the refrigerated cases. Customers' hands entering the cases to remove items, mix store and refrigerated air further aggravating the case loads. Open refrigerated cases cool and dehumidify the store air. Conduction and entrainment gains, which make up most of the case load, provide the store with sensible cooling. A latent load on the cases is also present. Entrained air generally has a water vapor content substantially above the saturation level of the case temperature. Cooling to the case temperature requires that moisture be condensed out of the air. This condensation creates a frost buildup in the cases which must be removed by periodic defrost cycles.

Since much of the refrigerated case load is due to the store conditions, strict control over these conditions must be maintained. If store temperatures and humidities exceed certain levels, case loads exceed the capacity of the refrigeration unit with the consequence being product spoilage. Refrigerated cases are designed to operate under certain maximum design levels. By industry standards, these levels are 24°C (75°F) and 55% rh (0.0104 kg/kg). To prevent overloading the cases both temperature and humidity must stay below these levels. Unlike a standard commercial building where the humidity level may float, a strict ceiling exists on the humidity level in a supermarket.

Since excess store humidity increases the refrigerated case loads, it follows then that a decrease in humidity will decrease the case loads. When less water vapor is contained in air entering the cases, the evaporator expends less energy for condensation. In addition, less frost buildup occurs reducing the amount of defrost energy required. Little information is available as to the actual reduction in energy consumption at lower humidity levels. Tyler Refrigeration Co. has published a limited amount of data (16), to which Thermo Electron fitted curves, and verified the trends at their field test sites (13). Figure 3.2.1 illustrates this reduction, for both low-temperature (frozen) cases and medium temperature (refrigerated) cases. The regression equations used are,

$$Z_{med} = 0.146 * (W_{sto} * 7000)^{0.452} \qquad (3.2.1)$$

$$Z_{med} = 0.302 * (W_{sto} * 7000)^{0.281} \qquad (3.2.2)$$

where Z is the fractional multiplier of the load at a standard humidity ratio (0.01 kg/kg). As the store humidity level decreases the amount of latent cooling performed by the cases also decreases. Figure 3.2.1 also illustrates the change in the latent load ratio as store humidity levels change. This follows the relationship,

$$LLR = 0.016 * \exp(0.035 * 7000 * W_{sto}) \qquad (3.2.3).$$

As was noted earlier, refrigeration consumes the largest portion of the store energy requirement. An HVAC system which could maintain

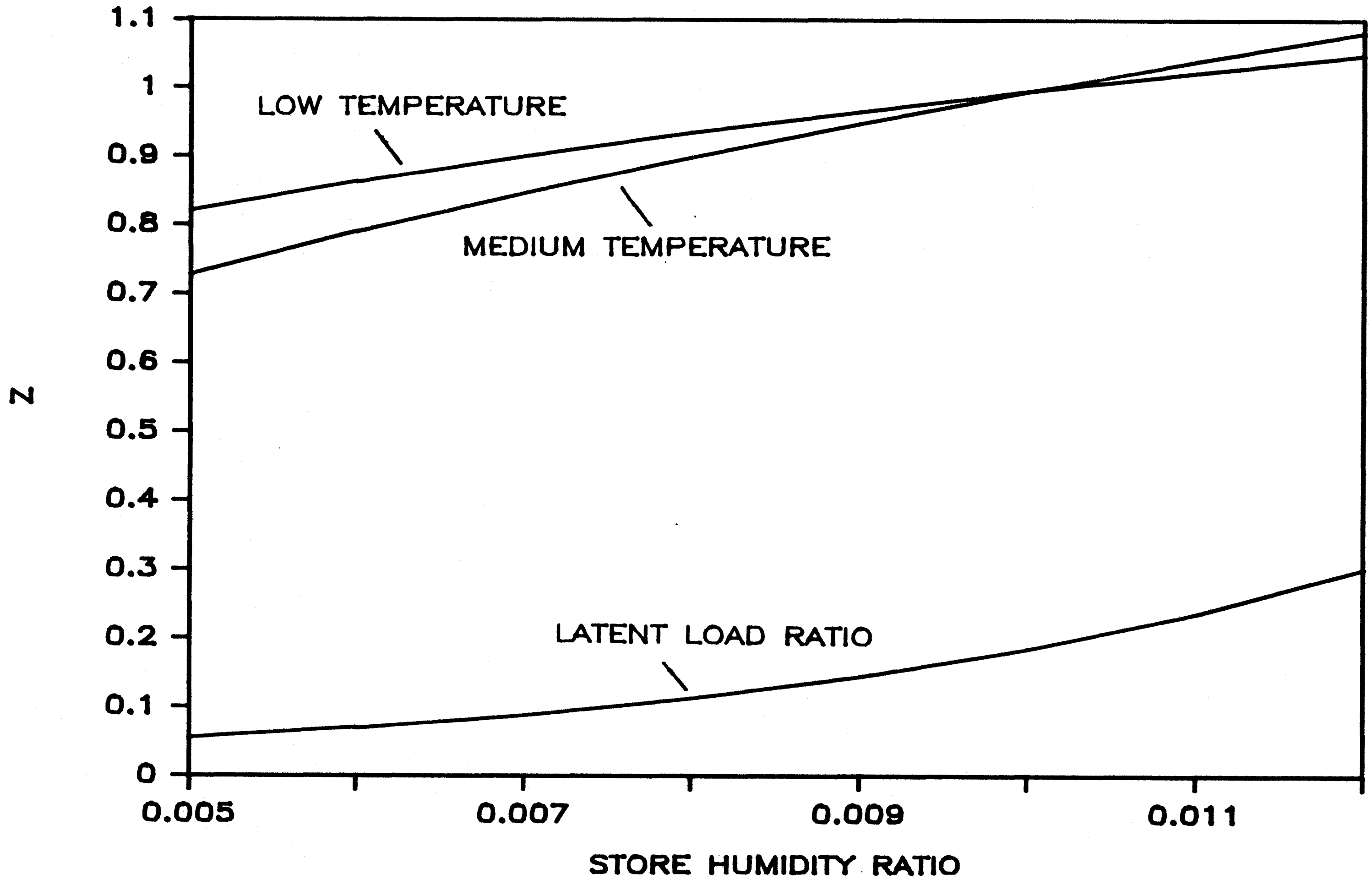


Figure 3.2.1 Refrigerated case energy consumption and latent load ratio as a function of store humidity level

lower store humidities would reduce the refrigeration energy consumption.

Section 3.3 Supermarket Loads

Sensible cooling loads come from a variety of sources. Among the major sources are internal generation (lights and equipment), people, transmission through the building envelope, ventilation, and infiltration. Similarly latent loads develop from internal generation, people, ventilation, and infiltration. Loads from internal generation and occupancy are independent of the ambient conditions. The large negative cooling load provided by the refrigerated cases differentiates supermarkets from most types of buildings. Figure 3.3.1 shows a typical internal load condition for a supermarket. Before ambient effects are considered there is a negative sensible cooling load and only a very small latent requirement. This load is substantially altered from that of a building where the refrigerated cases are not present. In many commercial buildings, the internal load dominates the final cooling load. In a supermarket those components dependent on the ambient conditions, transmission, ventilation and infiltration, determine the magnitude of the cooling load.

The refrigerated cases have a distinct impact on cooling loads in two ways. They substantially reduce the overall amount of

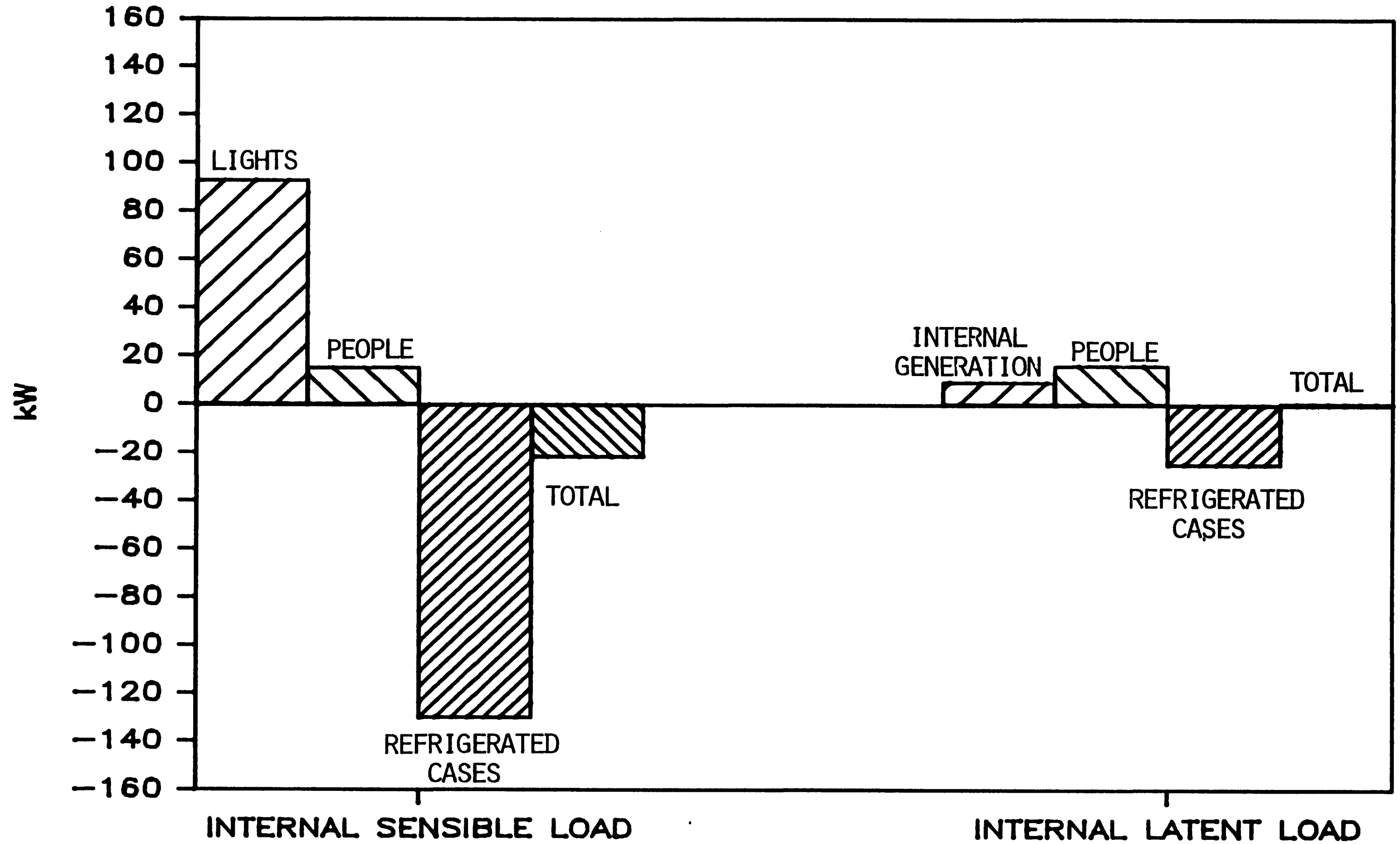


Figure 3.3.1 Typical composition of the internal cooling loads in a supermarket

cooling required. The air conditioning requirements found in supermarkets are much smaller than those in similar sized standard commercial buildings. Also, since the cases perform more sensible than latent cooling the relative composition of the load is altered. As less latent cooling is performed by the cases, the latent ratio of the remaining load will increase. The result being that supermarket loads are smaller and have a larger latent portion than loads found in a similarly sized commercial building.

Generally, when small loads exist, small amounts of air can be processed. However, a minimum amount of circulation air must be maintained. Standard supermarket practice calls for 0.006 Kg/m^2 (1 cfm/ft^2) of store floor space. Assuming a 6 m (20 ft) ceiling this provides three air changes per hour, which is a moderate amount of circulation air. For typical supermarket loads, this circulation rate requires an enthalpy difference between room and supply air of only 2-3 kJ/kg, much less than the 15 kJ/kg (6.4 Btu/lb) suggested by conventional air conditioning practice. This small enthalpy difference combined with the high latent load ratio results in conditioned air being supplied to the room at a state well away from saturation. When vapor compression is used, this requires cooling to a low temperature to remove moisture and reheating. While reheat energy may be supplied by condenser heat rejected from the refrigerators, the excess cooling must be purchased.

Supermarket air conditioning presents an ideal situation for hybrid desiccant systems. The strict humidity control requirement can be met without excess cooling. Cooling capacity requirements would be reduced. Vapor compression performance would increase with increases in evaporator temperature. Lower circulation rates can be maintained without the detrimental effects of cooling further down the saturation line.

CHAPTER 4

COMPONENT MODELS

The analysis of hybrid desiccant cooling systems by computer simulation requires the develoment of mathematical models which describe system performance. The transient simulation program, TRNSYS (17), is designed to link components together to form a system and solve the various mathematical models simultaneously. Mathematical models have been developed which describe the various components making up hybrid desiccant systems.

Section 4.1 Desiccant Dehumidifier

The dehumidifer considered is a rotary wheel consisting of silica gel which alternately passes through process and regeneration streams. Jurinak (9) and Van den Bulck (18) have discussed in some detail the modeling of desiccant dehumidifiers. The model used in this thesis to simulate the performance of the desiccant is an effectiveness-NTU model developed by Van den Bulck. Based on an analytical solution to the governing heat and mass transfer equations for an ideal dehumidifier, this model correlates enthalpy and moisture effectivenesses to real resistances to heat and mass transfer. The model compares favorably with the finite difference code, MOSHMX (19), and utilizes significantly less computational effort.

Many parameters affect desiccant performance. Desiccant mass, wheel speed, process and regeneration flow rates are all design variables which affect system performance. Van den Bulck (20) has studied the optimum operating values over a variety of inlet conditions. A dimensionless flow rate $\Gamma 1$ may be defined as,

$$\Gamma 1 = \frac{\text{mass of desiccant/time in period}}{\text{mass flow rate of process air}} \tag{4.1.1}$$

For systems operating in the ventilation mode, minimum auxiliary energy requirements occur where $\Gamma 1$ is 0.15 and the regeneration flow rate is 80% of the process flow rate. For recirculation cycles $\Gamma 1 =$ 0.1 and the flow rate ratio is 0.60. These values are used throughout the thesis. In theory maintaining a constant Γ requires a good deal of control. If the process flow rate is varied the wheel rotation speed will have to be altered correspondingly. Van den Bulck has shown however that deviations from an optimum wheel speed have only a small effect on the performance of the dehumidifier. The wheel modeled is assumed to be a high performance dehumidifier with high heat and mass transfer coefficients. The number of transfer units for heat transfer (UA/C_{min}) is assumed to be 15 on the process side. A Lewis number of one is also assumed. Table 4.1.1 summarizes the desiccant parameters used.

While this model provides an accurate description of the desiccant states with far less computational effort than previous finite difference models, it is still sufficiently time consuming to be undesirable for use in yearly simulations. A further simplification utilizes a moisture effectiveness defined by Van den Bulck (18),

$$\varepsilon_m = \frac{(w_i - w_o)}{(w_i - w_{ideal})} \qquad (4.1.2)$$

where w_i is the inlet process humidity ratio, w_o is the outlet process humidity ratio, and w_{ideal} is the ideal outlet humidity ratio if there were zero resistance to heat and mass transfer. If the design parameters are being held constant this moisture effectiveness can be correlated with respect to the inlet air conditions. For the ventilation mode, inlet conditions are ambient termperature, regeneration temperature and ambient humidity ratio. For use in yearly simulations the following first order correlation was developed for ventilation cycles

$$\varepsilon_m = 0.898 - 0.0035\ T_i - 7.54\ w_i\ 0.0025\ T_{reg} \qquad (4.1.3)$$

Figures 4.1.1 and 4.1.2 show values of m for various inlet conditions. The correlation is applicable over this range of

Table 4.1.1
Desiccant Parameters

Γ_1	=	0.15	Ventilation cycles
Γ_1/Γ_2	=	0.80	
Γ_1	=	0.10	Recirculation cycles
Γ_1/Γ_2	=	0.6	
Ntu_1	=	15.0	
Ntu_2	=	$Ntu_1/(\Gamma_1/\Gamma_2)$	
L_e	=	1	

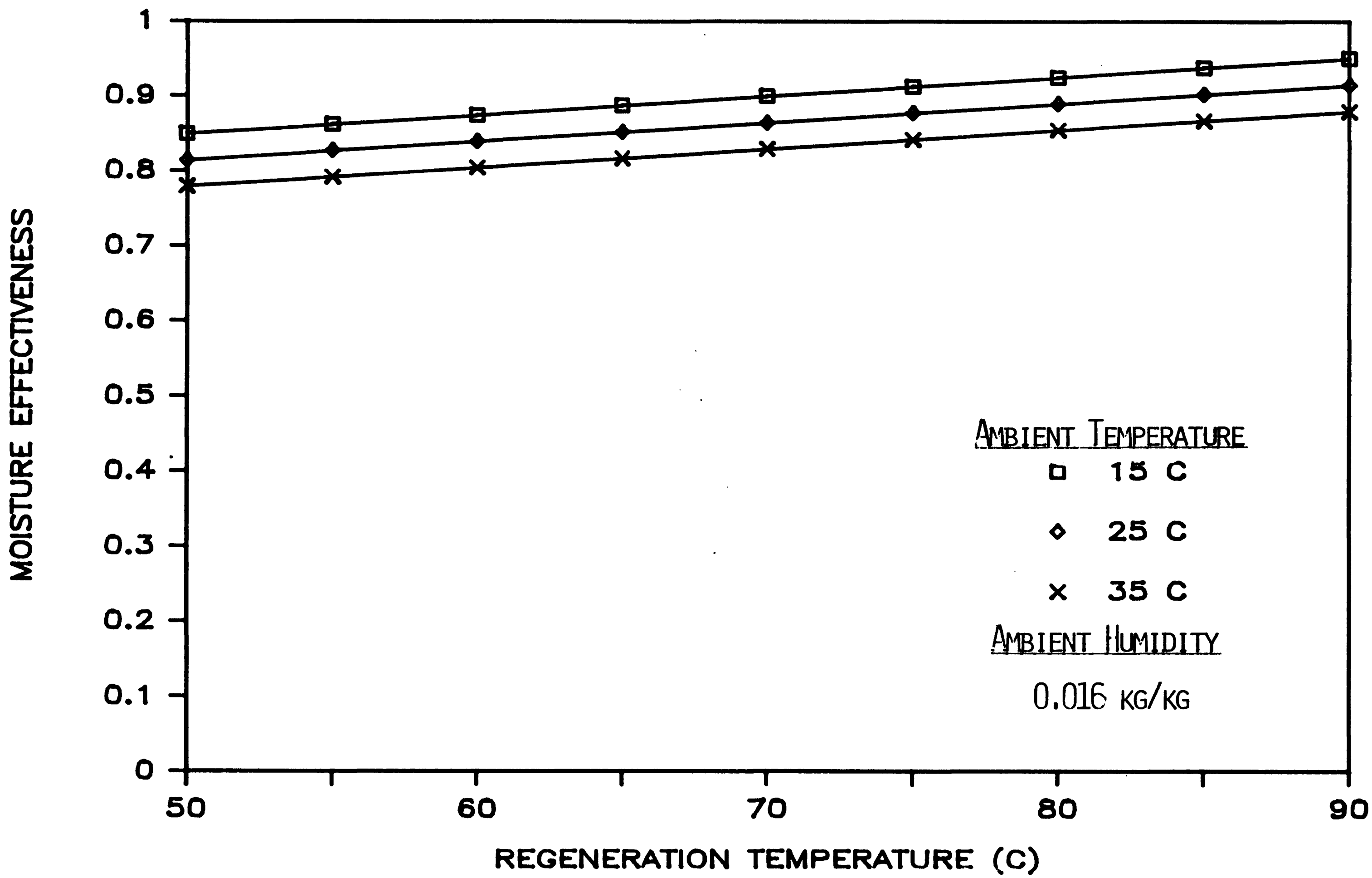


Figure 4.1.1 Moisture effectiveness correlation as a function of regeneration temperature at various ambient temperature and an ambient humidity ratio of 0.016 kg/kg

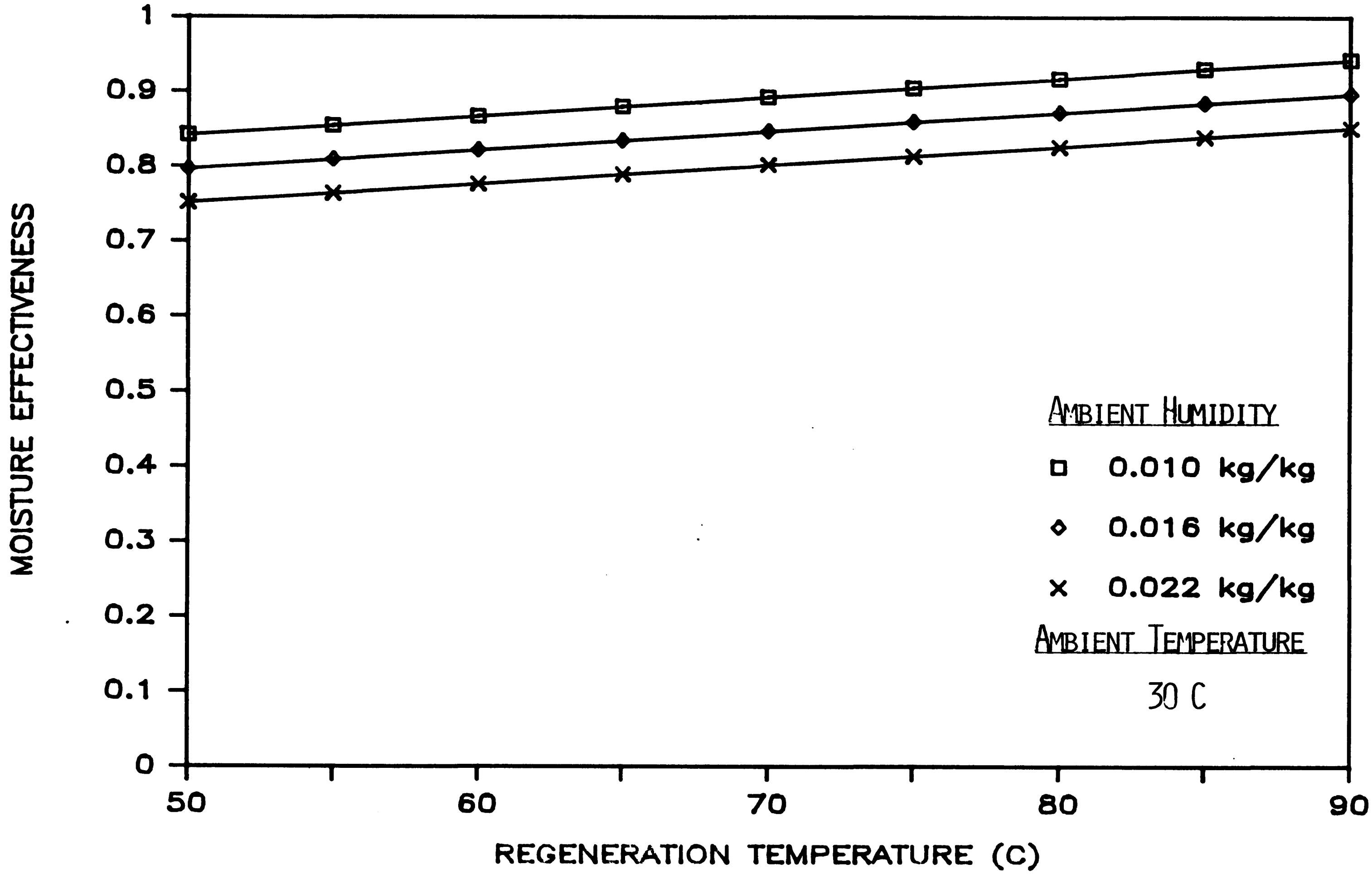


Figure 4.1.2 Moisture effectiveness correlation as a function of regeneration temperature at various ambient humidity ratios and an ambient temperature of 30°C

inlet conditions.

An enthalpy effectiveness is also needed to determine the process outlet state. Defined in the same manner as the moisture effectiveness, the enthalpy effectiveness has a value very close to one (18). The ideal outlet state can be computed quickly and use of the equation 4.1.3 reduces simulation run times substantially.

This effectiveness correlation method is used for the annual simulations discussed in Chapter 6. Figure 4.1.3 and 4.1.4 show sample results from the two methods in estimating system energy consumption. Figure 4.1.3 compares hourly energy requirements calculated by the two methods for a typical supermarket located in Ft. Worth for the month of August. Figure 4.1.4 compares monthly energy requirements for the Ft. Worth store over an entire year. Results using this method agree quite well with those obtained with the full model.

Section 4.2 Vapor Compression Model

Two of the hybrid cycles discussed in this thesis utilize heat from the condenser to preheat the regeneration air stream. This utilization of a vapor compression unit is not standard practice. Non-standard flow rates and condensing temperatures result from this application and the model must take these effects into account.

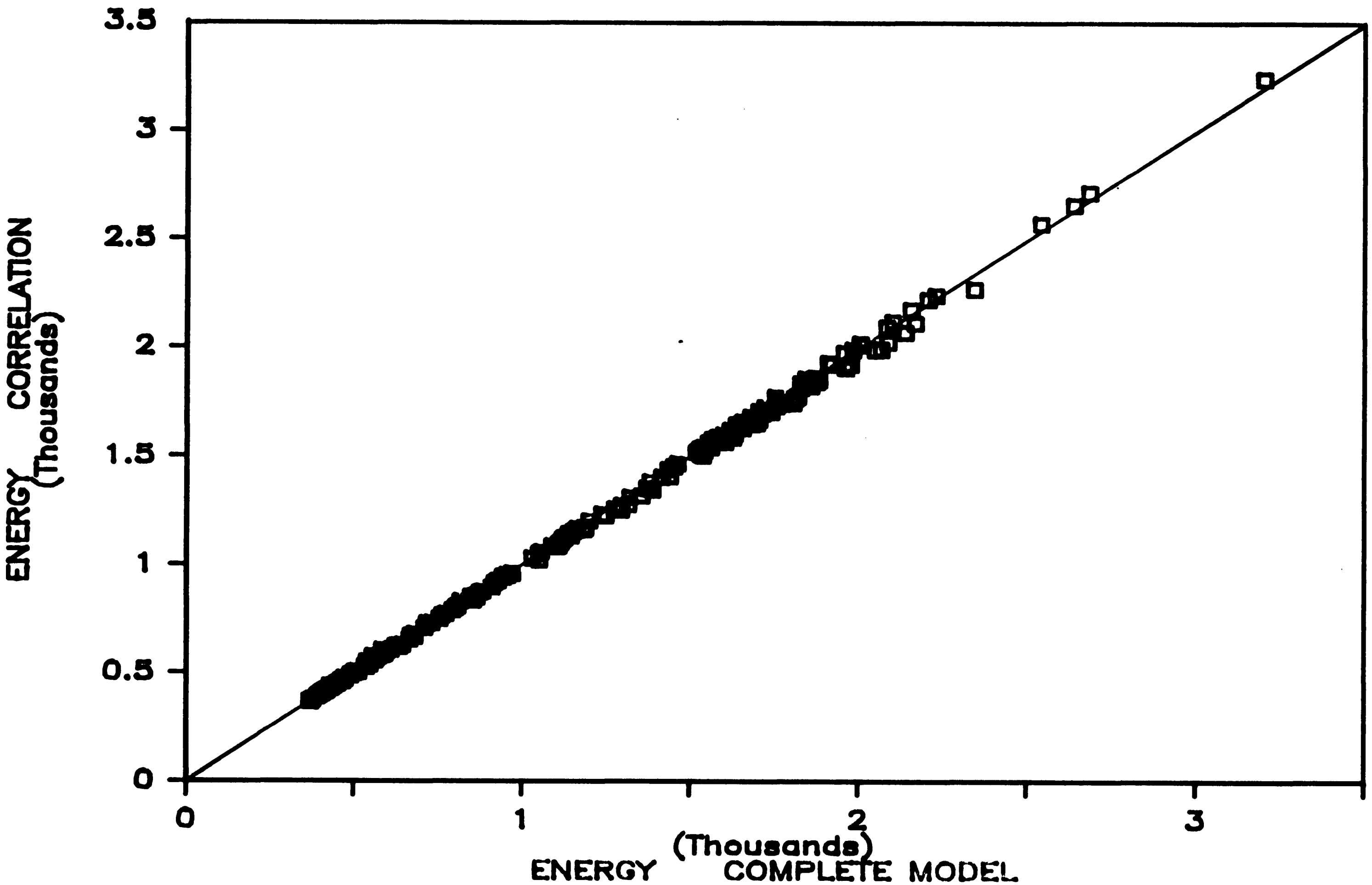


Figure 4.1.3 Comparison of hourly energy use for a typical supermarket in Ft. Worth for the month of August calculated using the complete desiccant dehumidifier model and the moisture effectiveness correlation

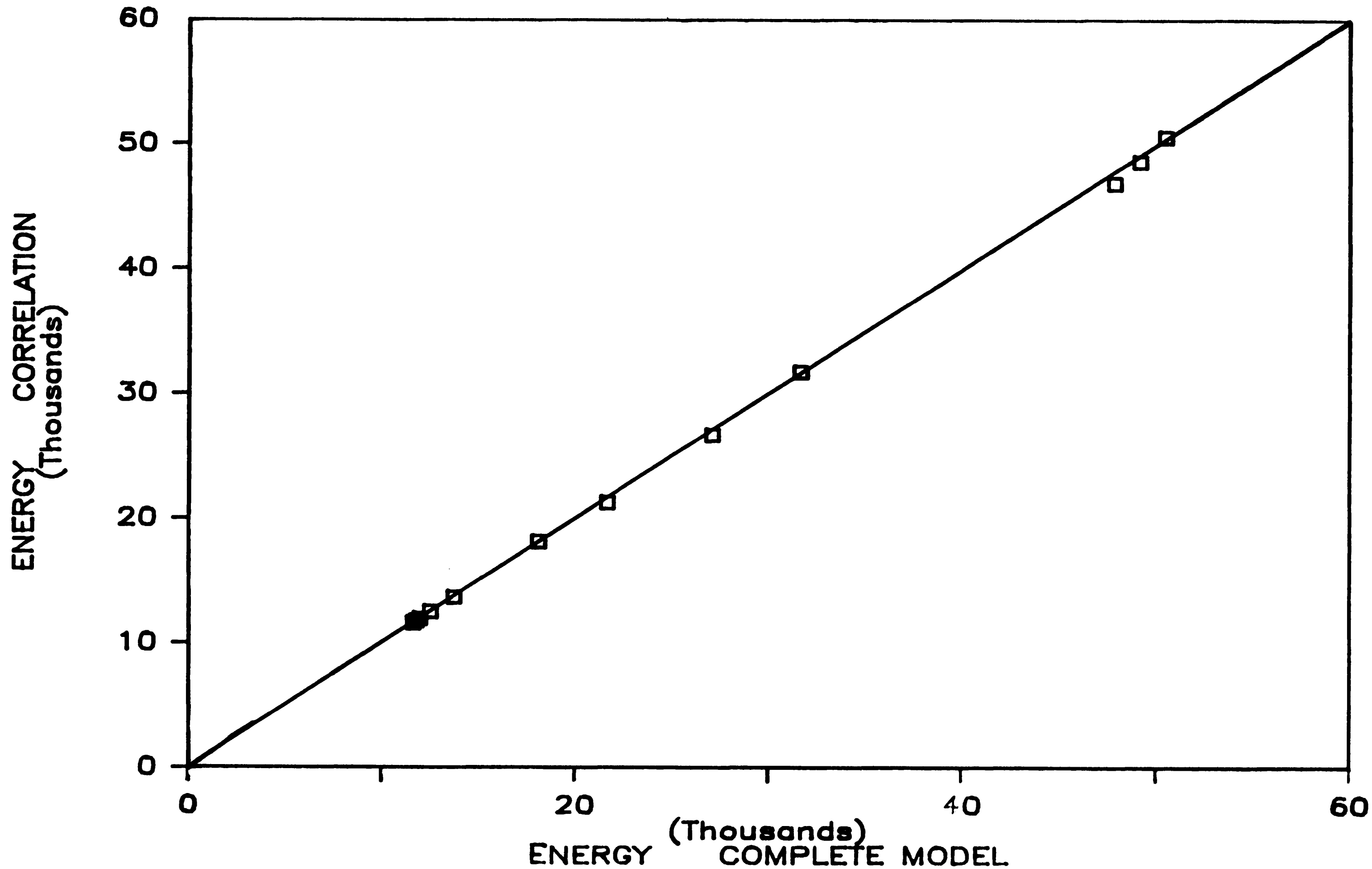


Figure 4.1.4 Comparison of monthly energy use of a year for a typical supermarket located in Ft. Worth using the complete desiccant model and the moisture effectiveness correlation

The vapor compression model calculates the evaporator requirement and any reheat needs given the inlet temperature and humidity (T_i, w_i), the supply temperature and humidity (T_s, w_s) and the mass flow rate, m. When dehumidification is required, the cooling process is assumed to cool all the way to the dew point of the desired store humidity level. Actual air states only approach this state. This model therefore presents a maximum energy bound in determining evaporator loads.

If the inlet humidity ratio is higher than the outlet, the usual situation in a standard vapor compression system, the dew point temperature of the supply air ($T_{d,s}$) is determined. The evaporator load is calculated from,

$$\dot{Q}_e = \dot{m} \, (h_i - h_{d,s}) \tag{4.2.1}$$

where h_i is the enthalpy of the inlet air and $h_{d,s}$ is the enthalpy at the supply dew point. Reheat requirements can be calculated as

$$\dot{Q}_r = \dot{m} \, (h_s - h_{d,s}) \tag{4.2.2}$$

where h_s is the supply enthalpy. If the inlet humidity, w_i, is equal to or less than the supply humidity, w_s, and T_i is greater than T_s, then the cooling requirement is

$$\dot{Q}_e = \dot{m}\,(h_i - h_s) \qquad (4.2.3).$$

When T_s is greater than T_i heating is required and the cooling load is zero.

While the evaporator loads are calculated in a very straight forward manner, the heat released in the condenser presents some difficulties. Since some of the cycles studied utilize condenser heat to preheat the regeneration air stream it is important to know how much heat is available and at what temperature it is available.

Performance data on vapor compression machines are usually presented in terms of the ambient air temperature entering the condenser with an assumed standard flow rate passing through the condenser. Actually, the performance of a vapor compression unit depends on the condensing temperature rather than the ambient air temperature. In utilizing condenser heat to preheat the regeneration stream, low air flow rates are used to increase the temperature of the air leaving the condenser. To enable the heat transfer to process to occur the condensing temperature must be increased. Some data are available which relate COP to condensing temperature (21). Extrapolation to higher condensing temperatures has been made by relating this data to the Carnot COP,

$$COP = k * \frac{T_e}{T_c - T_e} \qquad (4.2.4)$$

where k is a constant determined from,

$$k = \frac{COP_{data}}{COP_{Carnot}} \qquad (4.2.5)$$

Te is the evaporator temperature, and Tc is the condensing temperature. The value for the constant, k, was found to be 0.46 from the data. Figure 4.2.1 illustrates the relationship between COP and condensing temperature for an evaporator temperature of 4^oC.

The overall conductance-area product, UA was assumed to stay constant for an individual unit and was determined from the data by assuming a 4.4 C (10 F) log mean temperature difference at ARI standard condition. This model can be extended to different size machines by holding U constant and varying the area in proportion to the capacity of the unit.

The condensing temperature can be determined by solving the following set of equations. The condenser heat rejection may be calculated from the energy balance for the unit. In terms of COP and evaporator heat flow, the condenser heat flow is

$$\dot{Q}_{cond} = \dot{Q}_{evap}(1 + 1/COP) \qquad (4.2.6)$$

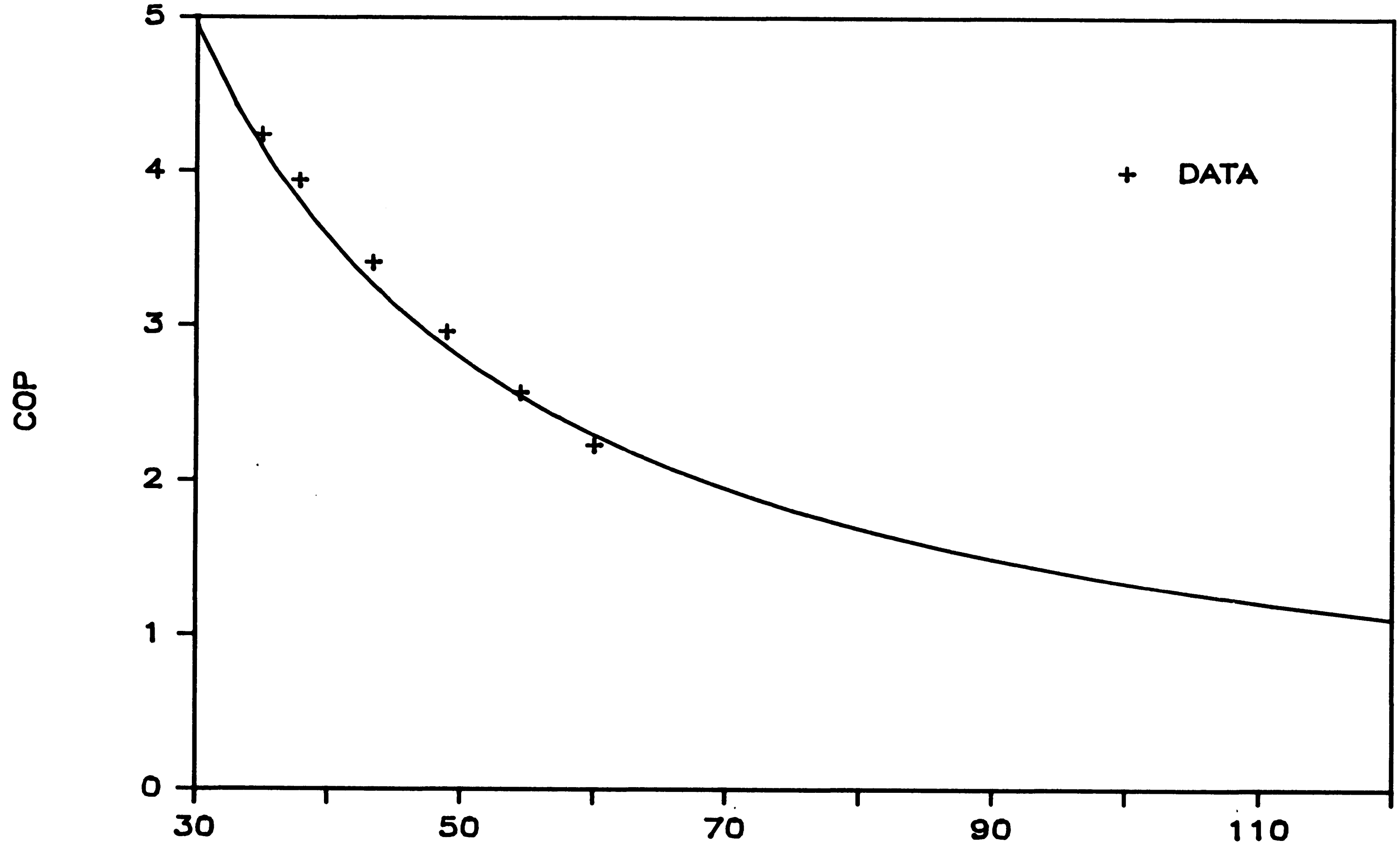


Figure 4.2.1 COP as a function of condenser temperature at an evaporator temperature of 4°C

The outlet temperature of the air stream follows from an energy balance on the air stream

$$T_o = T_i + \dot{Q}_{cond}/(\dot{m}C_p) \qquad (4.2.7).$$

The LMTD relation for heat transfer provides another equation

$$\dot{Q}_{cond} = UA\ (T_o - T_i)/\ln((T_c - T_i)/(T_c - T_o)) \qquad (4.2.8).$$

Equations 4.2.5, 4.2.6, 4.2.7, and 4.2.8 may be solved directly to determine T_c, COP, Q_{cond}, and T_o.

Figure 4.2.2 shows COP as a function of the ratio of the actual flow rate and the standard flow rates for a variety of inlet temperatures. At flow rates substantially smaller than the standard, a severe COP penalty occurs due to an increased condensing temperature.

Section 4.3 Rotary Heat Exchanger

The rotary heat exchanger is modeled using a temperature effectiveness defined as,

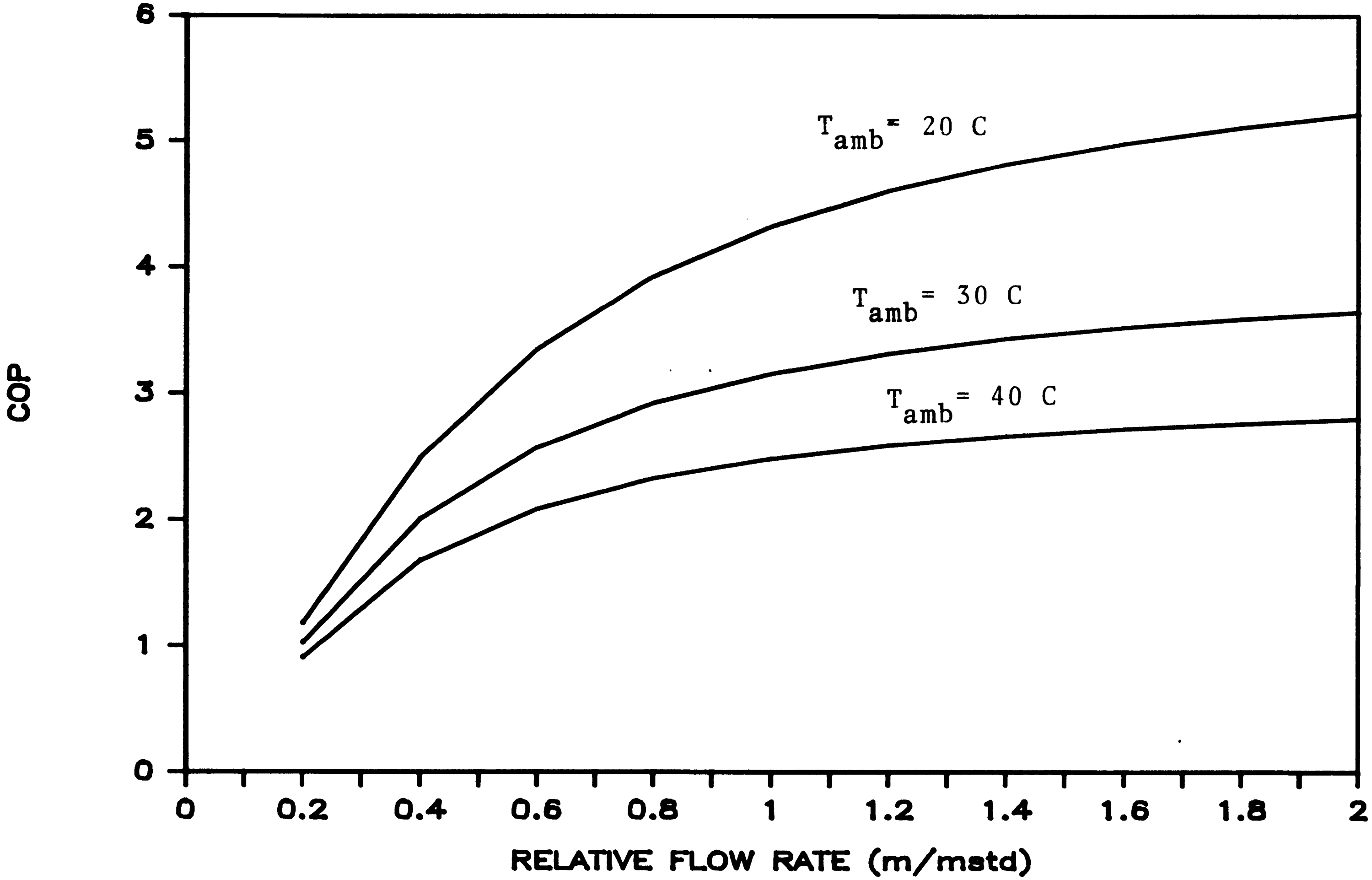


Figure 4.2.2 COP as a function of relative condenser air flow rate at various entering ambient temperatures

$$\varepsilon_{hx} = \frac{(T_h - T_o)}{(T_h - T_c)} \qquad (4.3.1)$$

where T_o is the temperature of the exiting process air, T_c is the temperature of the entering cool side air, and T_h is the temperature of the entering process (hot side) air. The effectiveness is a function of the design characteristics of the heat exchanger. Heat exchangers with high effectiveness are larger and more expensive. Throughout this study the effectiveness of the heat exchanger will be varied from zero to one. A practical upper limit is an effectiveness of 0.9. The largest heat exchange occurs at high effectiveness, however some systems will have minimum energy requirements at effectivenesses less than one.

Section 4.4 Indirect Evaporative Cooler

An indirect evaporative cooler is a heat exchanger in which water is sprayed into the secondary passage providing a low temperature sink due to evaporation. The effectivenss of the evaporative cooling process is defined as,

$$\varepsilon_{ec} = \frac{(T_i - T_c)}{(T_i - T_{wb})} \qquad (4.4.1)$$

where T_c is the temperature to which the air is cooled, T_i is the temperature of the incoming air, and T_{wb} the wet bulb temperature of the incoming air. This evaporative cooling effectiveness is always assumed to be 0.95.

Section 4.5 Auxiliary Heater

The auxiliary heat requirement may be calculated from,

$$\dot{Q}_a = \dot{m}_{reg} C_p (T_{reg} - T_i) \qquad (4.5.1)$$

where m_{reg} is the regeneration flow rate, T_{reg}, the regeneration temperature and T_i, the temperature of the air entering the heater after preheating. Since burners do not convert a fuel 100% efficiently the actual amount of the fuel supplied must be higher.

$$Q_f = \frac{Q_a}{\varepsilon_{heater}} \qquad (4.52).$$

Heater efficiencies vary depending on the type of heater used and the application. A typical heater efficiency in an industrial application is around 0.8, however all auxiliary heat requirements will be reported before burner inefficiencies are considered. The effect of furnace efficiency can be taken into account when considering the relative prices of electricity and heat. See Section 4.8, for a discussion of "energy weighting".

Section 4.6 Fan Power

Parasitic power required to move air through a desiccant system needs to be considered. The static pressure drop which must be overcome is calculated as,

$$\Delta P_{static} = \Sigma(\Delta P_i) * (\dot{m}/\dot{m}_{base})^2 \qquad (4.6.1)$$

where ΔP_i is the pressure drop across an individual component, at a base flow rate, $\dot{m}_{base}$, $\dot{m}$ is the actual flow rate through the component. The work required is calculated,

$$\dot{W}_{static} = \dot{m} * \Delta P_{static}/\rho \qquad (4.6.2)$$

where ρ is the density of the air. Fan inefficiencies which would increase the work requirement and add a small heat input to the air stream have been ignored.

Four fans are placed in the various hybrid systems. One each is placed in the process stream, the regeneration stream, the circulation stream and depending on the system, the condenser air stream for ventilation/heat exchanger cycle or the seconday stream of the indirect evaporative cooler for the cycles utilizing condenser heat (condenser cycles). Table 4.6.1 list the standard pressure drops assumed across each component.

Section 4.7 Supply States and Control

Given the sensible and latent components of the load the required supply states can be calculated from mass and energy balances.

$$w_s = w_r - \dot{Q}_l/(\dot{m}_s\ h_{fg}) \tag{4.7.1}$$

$$h_s = h_r - \dot{Q}_t/\dot{m}_s \tag{4.7.2}$$

The required outlet humidity ratio from the desiccant is

$$w_p = w_r - \dot{Q}_l(\dot{m}_p\ h_{fg}) \tag{4.7.3}$$

where $\dot{m}_p$ is the process air mass flow rate. Since the desiccant model calculates process outlet states given the inlet conditions, an iterative approach involving Newton's method calculates the required regeneration temperature. The regeneration temperature is controlled such that the process outlet state always meets the desired outlet humidity level.

Table 4.6.1
Fan Pressure Drops

	Pa	Inches of Water
Fan #1 Process Air Stream		
Dehumidifier	250	1
Heat Exchanger (IEC)*	120	0.5
Fan #2 Circulation Air		
Evaporator	50	0.25
Reheat Coil	60	.25
Supply Ducts	500	2
Fan #3 Regeneration Air		
Heat Exchanger* (Condenser)	120 60	0.5 0.25
Dehumidifier	250	1
Fan #4		
Condenser* (IEC)*	60 120	0.25 0.5

* Ventilation/Heat Exchanger Cycle
(Condenser cycles)

There are a few instances when the amount of moisture in the process stream available for removal is insufficient to meet the load. In a ventilation cycle when no ventilation air is required (no occupancy) this is always the case. When this situation occurs, the outlet humidity ratio is set to 0.005 kg and air is added to the process stream from the recirculation air until enough moisture can be removed from the system to meet the load. The amount of recirculation air added is

$$\dot{m}_{rec} = (\dot{m}_p * w_p - \dot{m}_v * w_v)/w_{rec} \qquad (4.7.4)$$

where w_p = 0.005 kg/Kg and $\dot{m}_v$ is the ventilation mass flow rate, and

$$\dot{m}_p = \dot{Q}_l/(h_{fg} * (w_p - w_r)) \qquad (4.7.5).$$

Section 4.8 Energy Weighting

Desiccant systems utilize thermal energy as a substitute for electrical energy. Comparing system energy consumption presents difficulties since gas and electricity costs differ. Multiplying all electricity usage by a weighting factor indicative of the relative costs of electrical and thermal energies, provides results which may be used to compare energy costs of the various cycles studied. For example, system energy consumption consists of three components, auxiliary heat, fan power, and vapor compression work.The consumption in weighted units will be,

$$E = Q_A + Fan * weight + VC * weight \qquad (4.8.1)$$

The use of the weighting factor is an application of the concept of resouce energy (9). Resource energy considers the conversion efficiency of primary fossil fuel to end use energy. A weighting factor of two is used almost exclusively throughout this thesis. This accounts for a 35% conversion of fossil fuels to electricity and a 70% efficiency in the auxiliary heater.

CHAPTER 5

SYSTEM PERFORMANCE AT FIXED OPERATING CONDITIONS

Exploring system performance at fixed conditions provides much information about the operation of hybrid systems. This chapter studies the hybrid systems under a variety of conditions in an attempt to compare system performance and understand the interactions and trade-offs present. A base case situation establishes a comparison criteria from which the effect of changes in various parameters may be evaluated.

Section 5.1 Conditions and Loads

The supermarket considered in these fixed condition calculations is loosely patterned after the Jewel store in West Chicago, Illinois in which Thermo Electron installed their hybrid system (22). The store contains 2800 square meters (30,000 square feet) of floor space, and 176 kw (50 tons) of installed refrigeration capacity. The store will be maintained at the maximum design condition for the refrigerated cases, which is a store condition of 24 C (75^{o}F) and 0.0104 kg/kg absolute humidity ratio (55% rh). The outdoor ambient air condition is 30 C (86^{o}F) and 0.016 kg/kg (60% rh). Typically a store circulates 0.006 kg/s-m^2 (1cfm/ft^2) of floor space. This means a standard circulation flow rate of 16.67 kg/s (30,000 cfm). The outside ventilation requirement is assumed to be ten percent of the circulation flow, 1.67 kg/s (3000 cfm). The internal load met by the air conditioning system is 24.3 kW (828 MBtu) 65% of which is latent. Unless otherwise noted all energies will be reported in weighted units with a weighting of 2. These base case conditions were chosen as being representative of a typical supermarket cooling situation and are summarized in Table 5.1.1.

Section 5.2 Base Case Performance

The standard vapor compression cycle is the standard to which the hybrid systems are compared. Figure 5.2.1 presents a schematic diagram of a standard vapor compression cycle with the state points and energy flows labeled for the base case conditions. The evaporator energy requirement of 222 kW (756 MBtu) requires 70.3 kW of compression work at a COP of 3.16. The amount of cooling done in the evaporator to reduce the humidity ratio is four times greater than that required to maintain the store conditions. Reheat energy of 164 kW (561 MBtu) makes up this difference. The effective COP of this cycle is 0.83.

To illustrate the operation and the energy usage of a hybrid system, the base case example with the ventilation/condenser system

Table 5.5.1
Store Parameters

Floor Space	2800 m^2	30,000 ft^2
Generated Load	24.3 kW	83 MBtuh
Latent Load (65%)	15.8 kW	54 MBtuh
Circulation Flow	16.7 kg/s	30,000 cfm
Ventilation Flow	1.67 kg/s	3,000 cfm
Refrigeration Capacity	176 kW	50 tons
Store Temperature	24^{o}C	75^{o}F
Standard Store Humidity	.0104 kg/kg	55% RH
Energy Weighting (Electric Gas)	2	

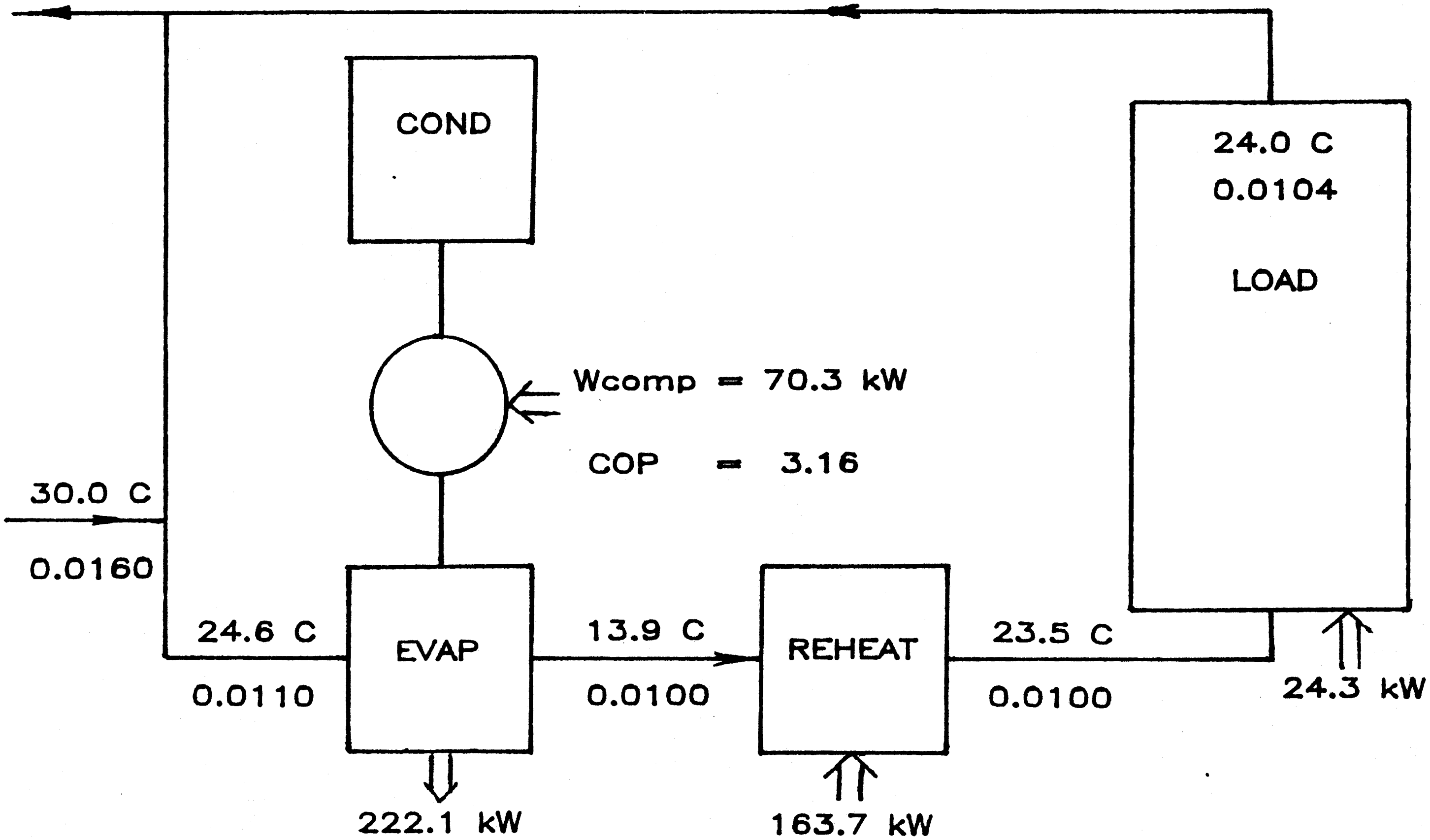


Figure 5.2.1 Base case state points and energy flows for standard vapor compression cycle

is described in some detail. The indirect evaporative cooler has an effectiveness of 0.8. Figure 5.2.2 summarizes the state points and energy flows on a schematic diagram and figure 5.2.3 depicts the process on a psychrometric chart. For the laten load, flow rates, and the ambient conditions, the desired outlet absolute humidity level of the process air is 0.0066. A regeneration temperature of 81.4 C provides this humidity level. The adsorption process heats the process air stream to 60 C. Indirect evaporative cooling provides 48.6 kW of free cooling, reducing the process temperature to 31.3 C. After mixing with the recirculated air the remainder of the sensible cooling is performed by vapor compression. No further dehumidification is needed. In this case, the amount of cooling is 20.5 kW which at a COP of 2.7 requires 7.6 kW of electrical energy consumption in the compressor. A heat flow of 28.1 kW is rejected to the condenser. The condenser heat raises the regeneration stream to a temperature of 51 C. The auxiliary heat requirement needed to produce the regeneration temperature of 81.4 C is 41.7 kW. The total energy cost is 58.3 weighted units, substantially less than the 141 weighted units consumed by the standard vapor compression system to meet this same load. The cycle has an effective weighted COP,

$$COP_{eff} = Load/(Q_a/2 + W_{vc}) \qquad (5.3.1)$$

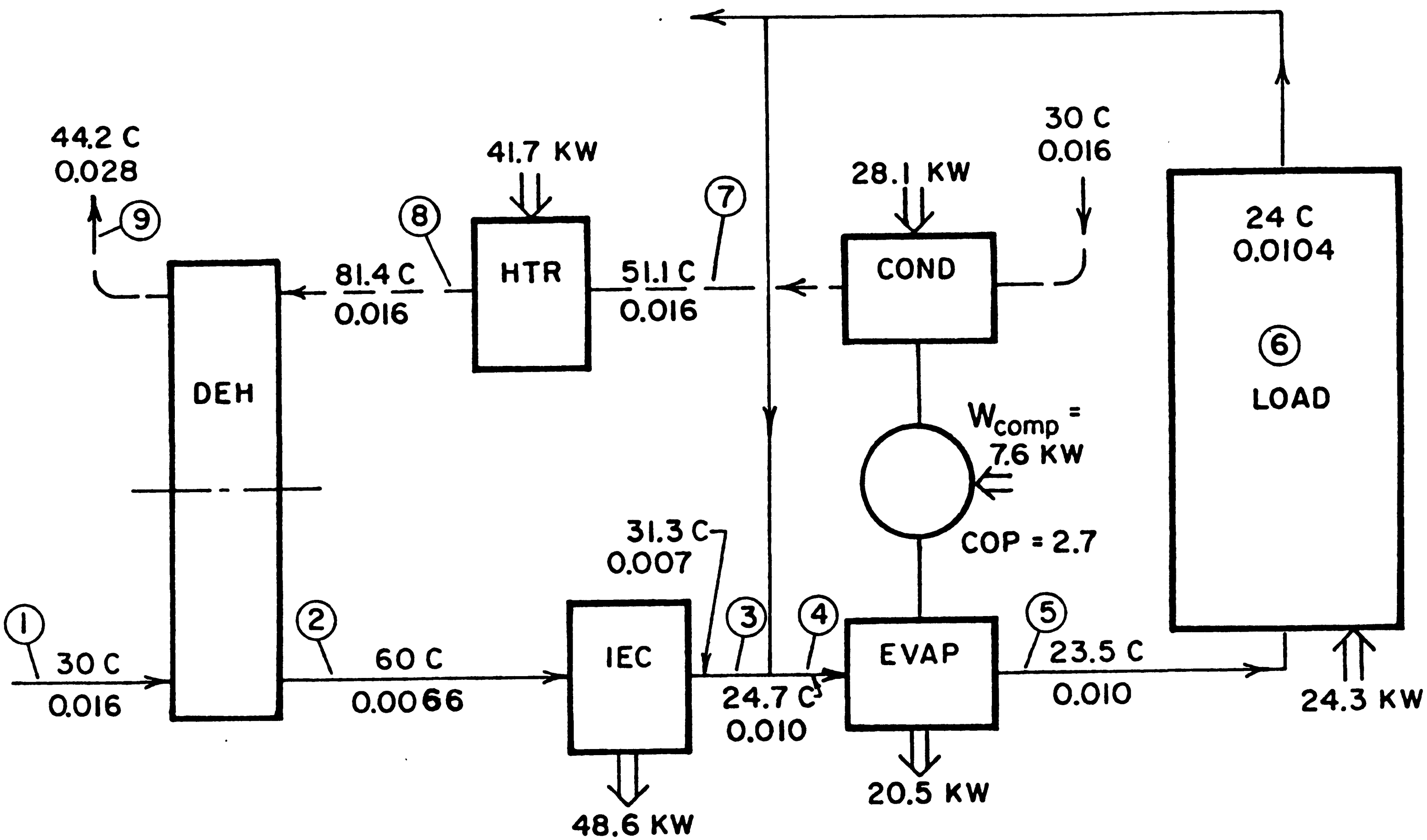


Figure 5.2.2 Base case state points and energy flows for ventilation/condenser cycle

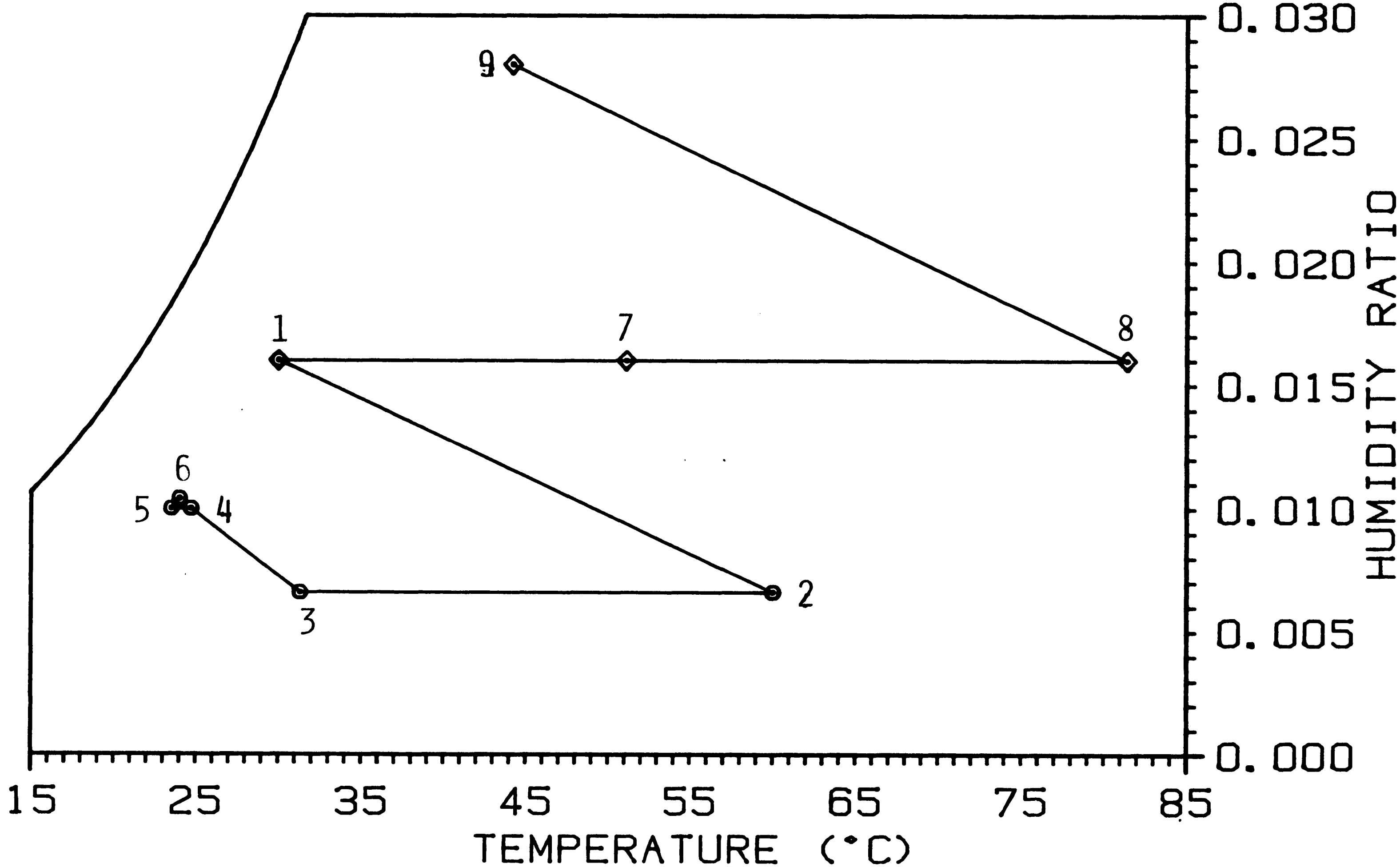


Figure 5.2.3 Psychrometric chart illustrating the base case state points of the ventilation/condenser cycle

of about 2. Figures 5.2.4 and 5.2.5 show the state points and energy flows for the two other cycles studied, the ventilation/heat exchanger and the recirculation/condenser cycles.

Some comment can be made from these base case calculations. Vapor compression work in the hybrid systems is one tenth of that for the standard vapor compression system. Among the desiccant systems, the two ventilation cycles consume similar amounts of energy. While the indirect evaporative cooler provides more free cooling than the heat exchanger, the heat exchanger provides more heat than the condenser on the regeneration side. The recirculation/condenser cycle requires more energy for both cooling and regeneration. The larger flow rate requires a smaller humidity reduction and lowers the regeneration temperature. However, the desiccant does not work as effectively in these conditions and the energy requirements in both streams increase.

Section 5.3 Variable Heat Exchanger Effectiveness

As noted before, a trade-off exists in cycles utilizing condenser heat between the amount of cooling performed by the indirect evaporative cooler (IEC) and the amount of heat available for regeneration from the condenser. By varying the effectiveness of the IEC from 0 to 1.0, which in effect regulates the amount of

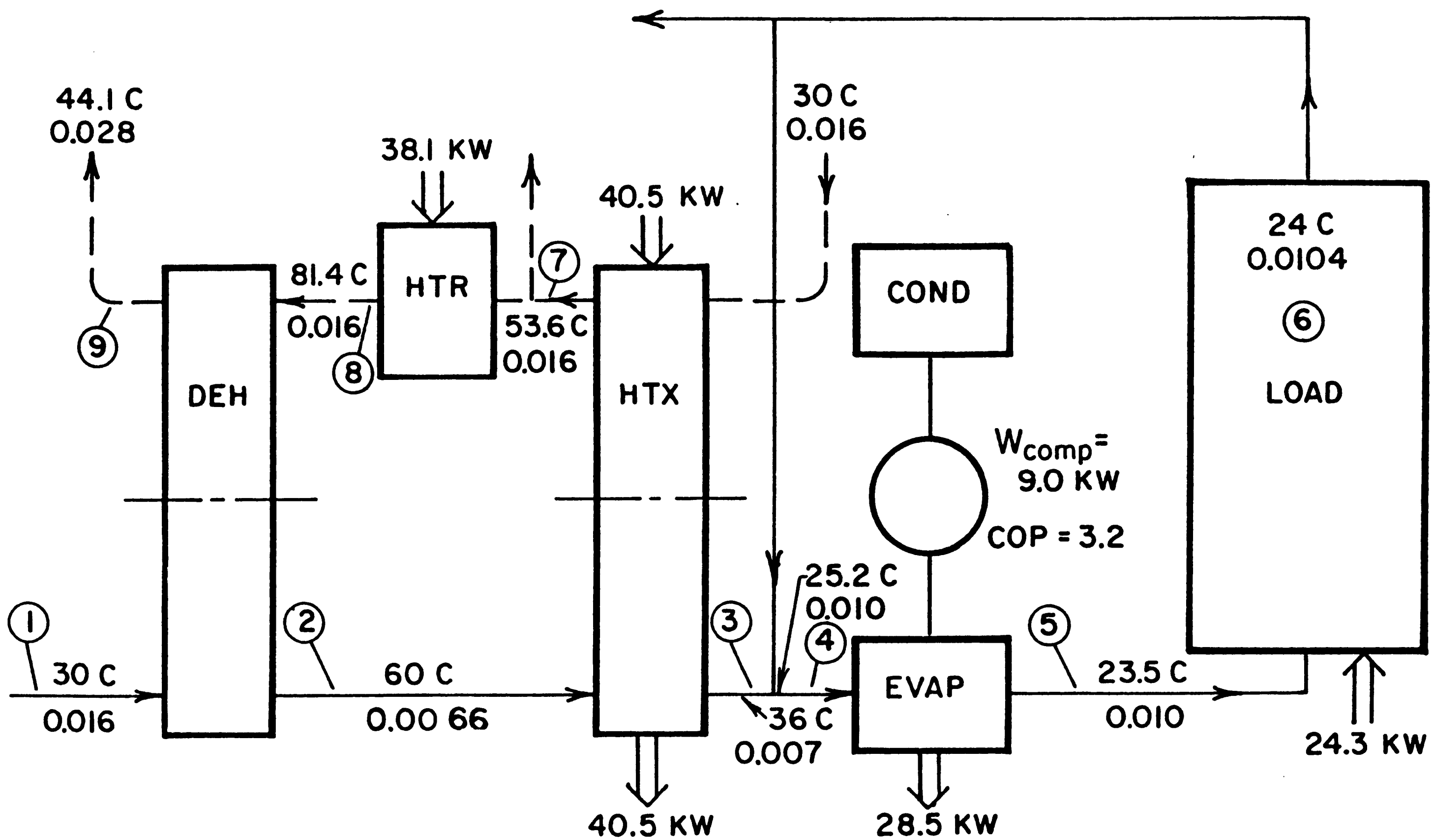


Figure 5.2.4 Base case state points and energy flows for ventilation/heat exchanger cycle

cooling done by that component, an optimum level of free cooling can be determined.

Figure 5.3.1 illustrates the breakdown of auxiliary heat, electrical energy, and total energy consumption expressed in weighted units for the ventilation/condenser configuration as a function of the IEC effectiveness. This figure indicates that the optimal amount of free cooling for this energy weighting is just enough so that the condenser heat available can completely regenerate the desiccant. If more free cooling is performed, the amount of auxiliary heat needed rises faster than the reduction in vapor compression work. Any less free cooling and the vapor compression unit performs more work without any further benefit on the regeneration side. The rapid increase in the vapor compression work, as heat exchanger effectiveness decreases, indicates the penalty taken in reclaiming condenser heat. As more heat is rejected, the condensing temperature rises. Analysis of the recirculation/condenser cycle results show similar trade-offs in the energy consumption of the system.

Figure 5.3.2 shows the total energy consumption of the three systems as a function of heat exchanger effectiveness. The ventilation/heat exchanger system performs best at a high effectiveness as there is no free cooling trade-off in the system. The ventilation/condenser and recirculation/condenser systems both have optimal points at intermediate effectiveness. The COP penalty

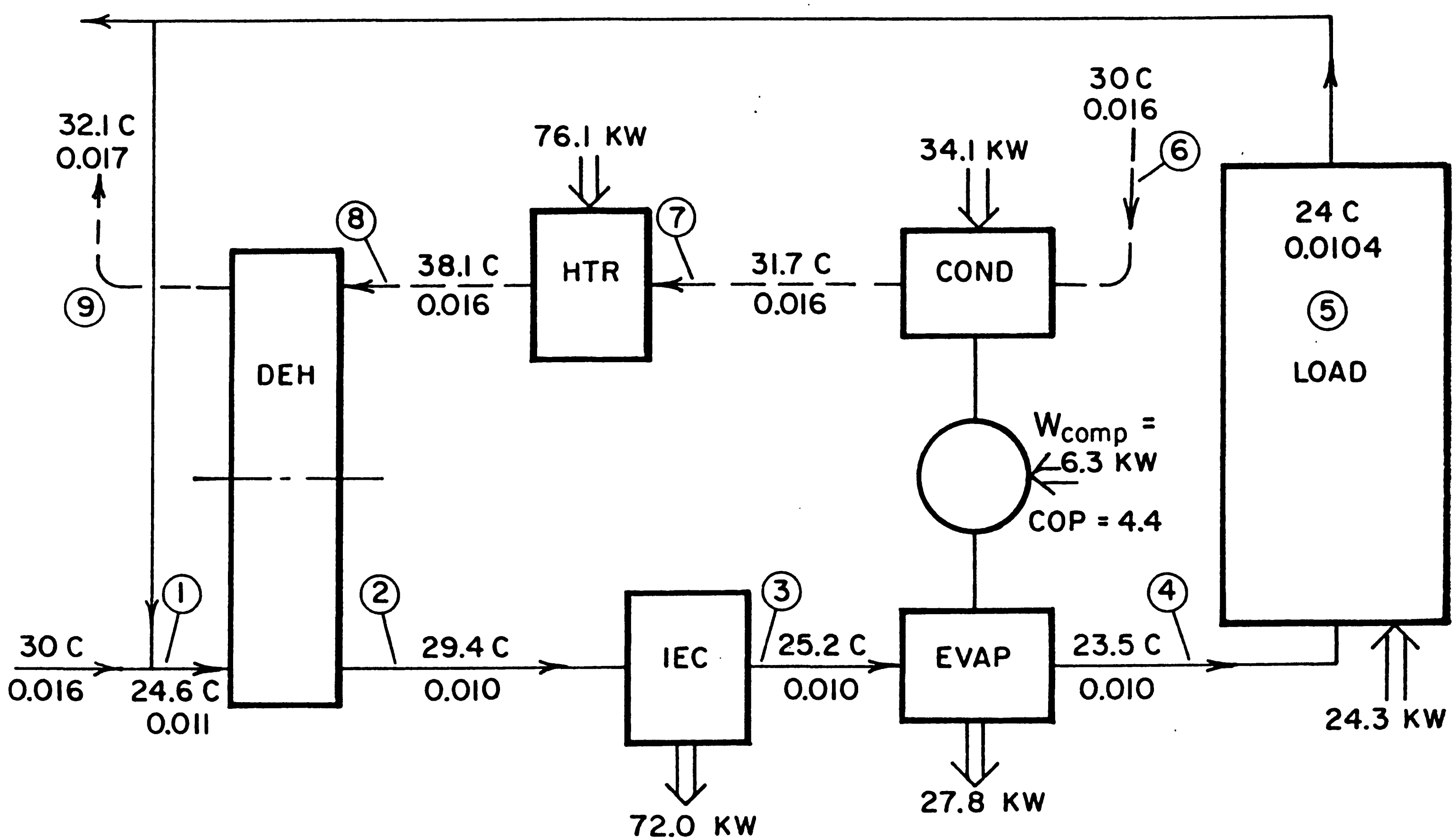


Figure 5.2.5 Base case state points and energy flows for recirculation/condenser cycle

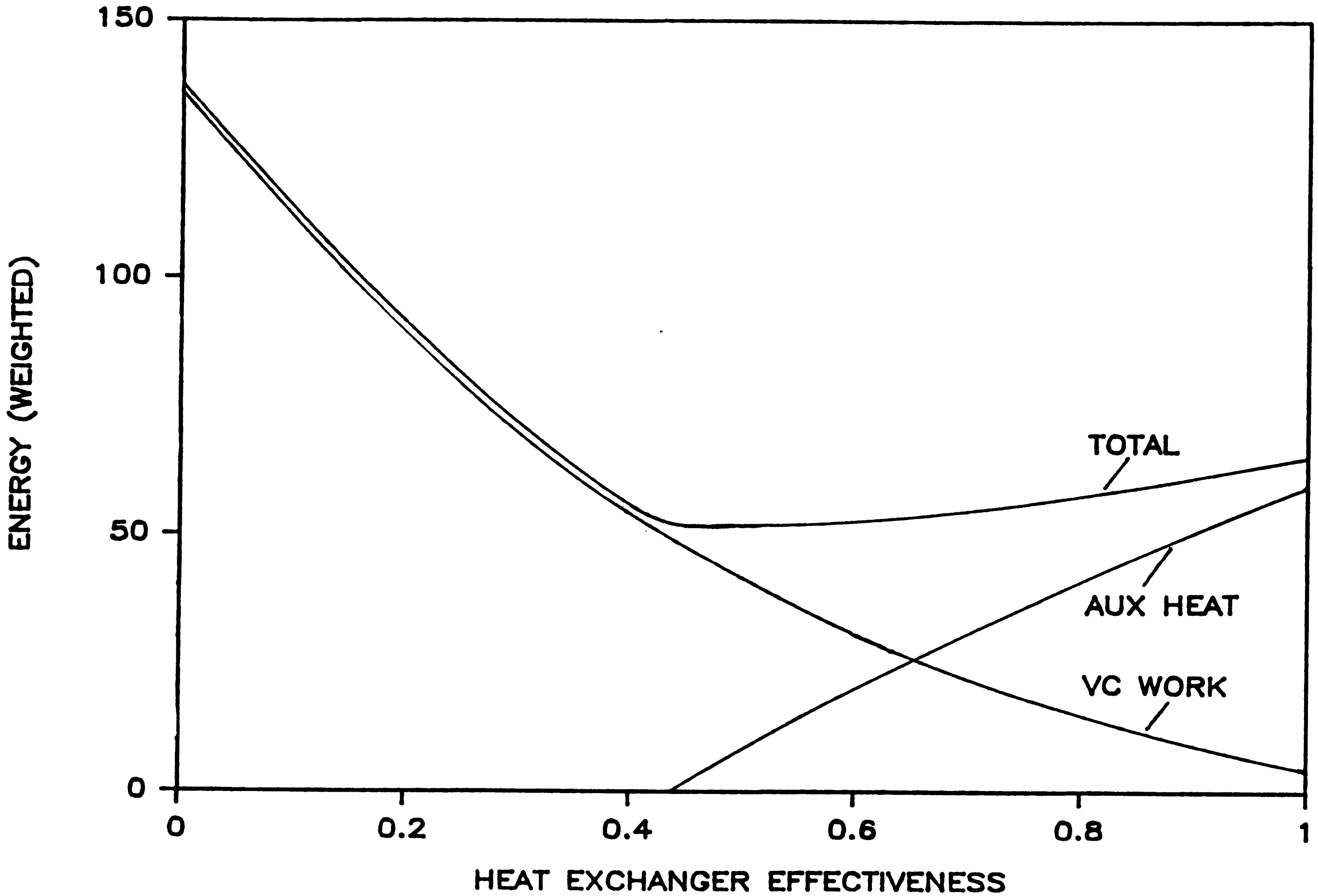


Figure 5.3.1 Breakdown in energy use for ventilation/condenser cycle as a function of heat exchanger effectiveness

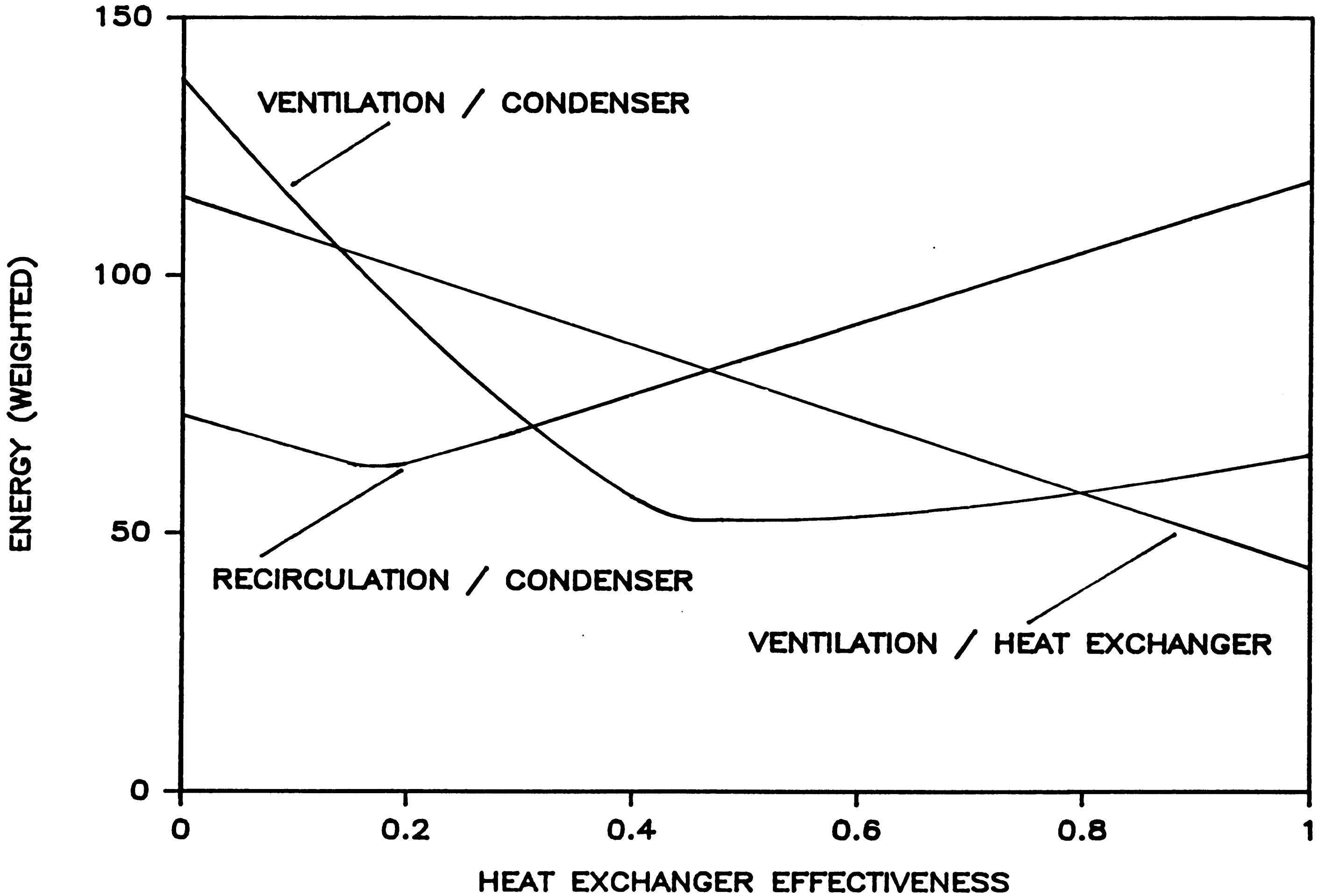


Figure 5.3.2 Comparison of total energy use of the three hybrid systems as a function of heat exchanger effectiveness

does not play as large a role in the recirculation/condenser system due to the larger flow rate. Lowering the heat exchanger effectiveness allows more heat to be released to the regeneration air stream without severely increasing the vapor compression work requirement. A significant decrease in energy cost is realized. All cycles are fairly close in total energy cost at their optimum heat exchanger effectiveness.

Section 5.4 Effect of Energy Weighting

In the previous section, it was shown that a heat exchanger effectiveness exists which minimizes energy cost. This optimum effectiveness will vary depending on the relative weights of thermal and electric energy. The results of section 5.3 suggest that for a relative weight of two, the optimum heat exchanger effectiveness occurs when no auxiliary heat is required. The optimum amount of condenser heat will decrease as heat cost increases, or electricity cost decreases. Figure 5.4.1 shows the ventilation/condenser cycle energy consumption for four different values of energy weightings. When electrical energy costs four times thermal energy the minimum vapor compression cooling requirement is desired. When the relative weighting equals one the minimum auxiliary requirement is desired.

This same trend does not appear as quickly in the recirculation/condenser cycle in figure 5.4.2. The large flow rate

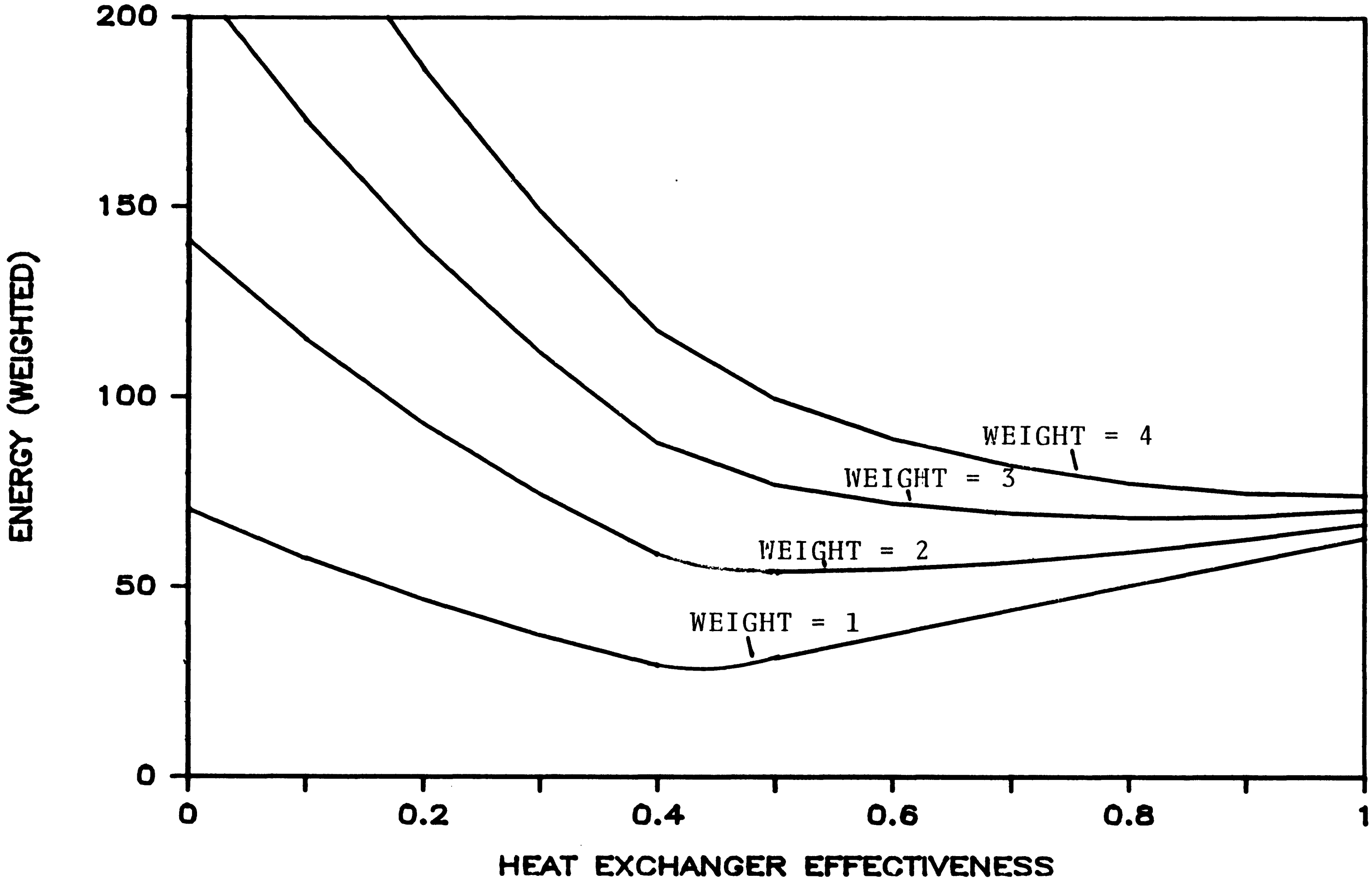


Figure 5.4.1 Weighted energy use for the ventilation/condenser cycle as a function of heat exchanger effectiveness at various energy weights

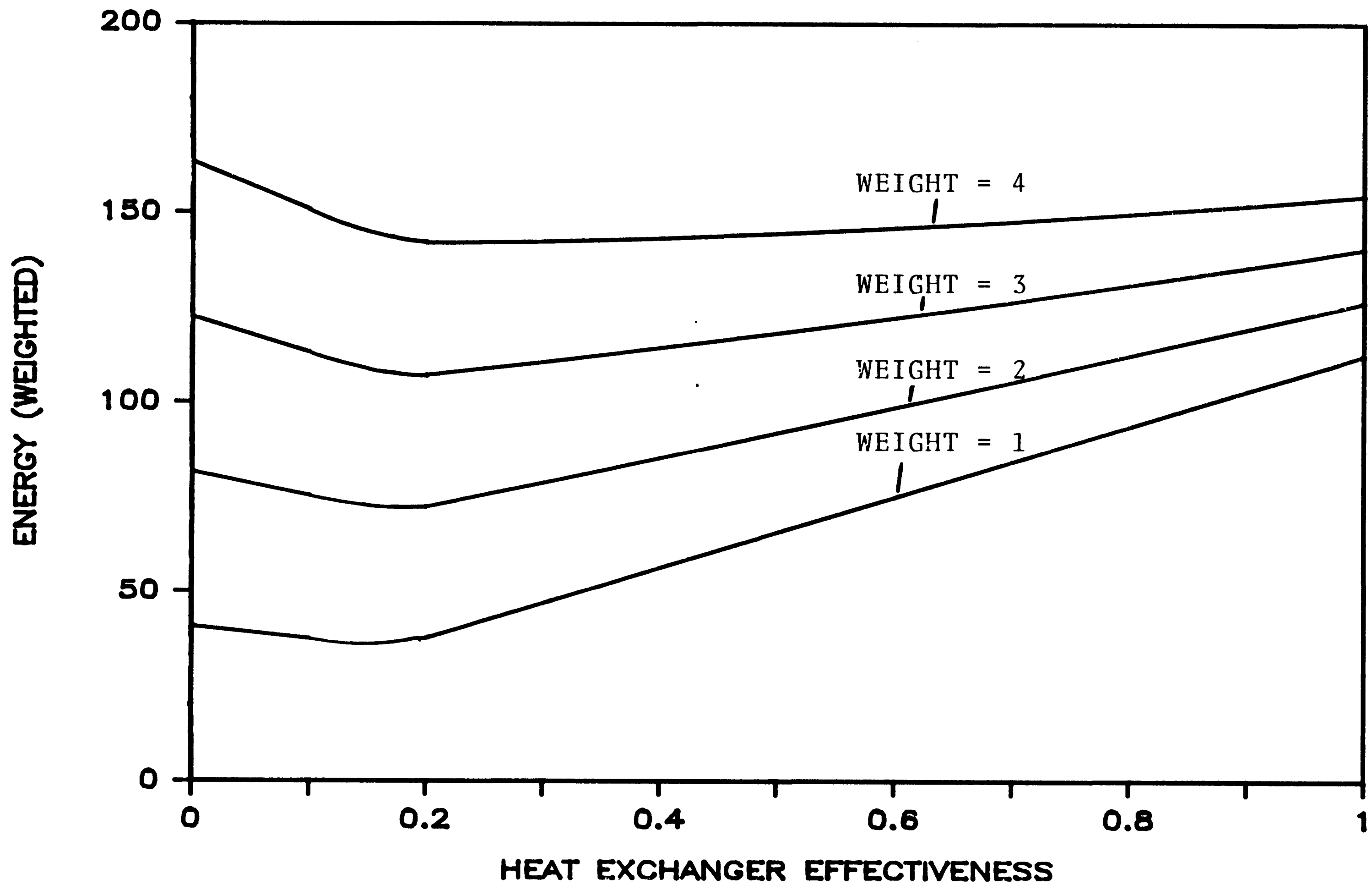


Figure 5.4.2 Weighted energy use for the recirculation/condenser cycle as a function of heat exchanger effectiveness at various energy weights

creates a large thermal energy requirement for regeneration. This larger flow rate through the condenser minimizes the COP penalty taken for using condenser heat. The result being that at an energy weighting of four, the optimum heat exchanger effectiveness still occurs where no auxiliary heat requirement exists. At still higher weightings an increase in the optimum heat exchanger effectiveness would occur.

Figure 5.4.3 depicts the energy use of the various systems at an energy weighting of four. Under these conditions, the recirculation/cycle is clearly not competitive. Since less free cooling is available from the heat exchanger than the indirect evaporative coolers the ventilation/heat exchanger cycle suffers a little more than the ventilation/condenser cycle as electricity costs increase. However, their energy consumption remains very close to each other. High effectiveness heat exchangers are desired for both ventilation cycles at this energy weighting.

Section 5.5 Effect of Flow Rate Reduction

A reduction in the recirculation/condenser energy consumption is possible by lowering the system flow rate. Since the hybrid desiccant systems do not require low evaporator temperatures to remove large amounts of moisture it is possible to circulate less air through the store. This has the effect of reducing the fan

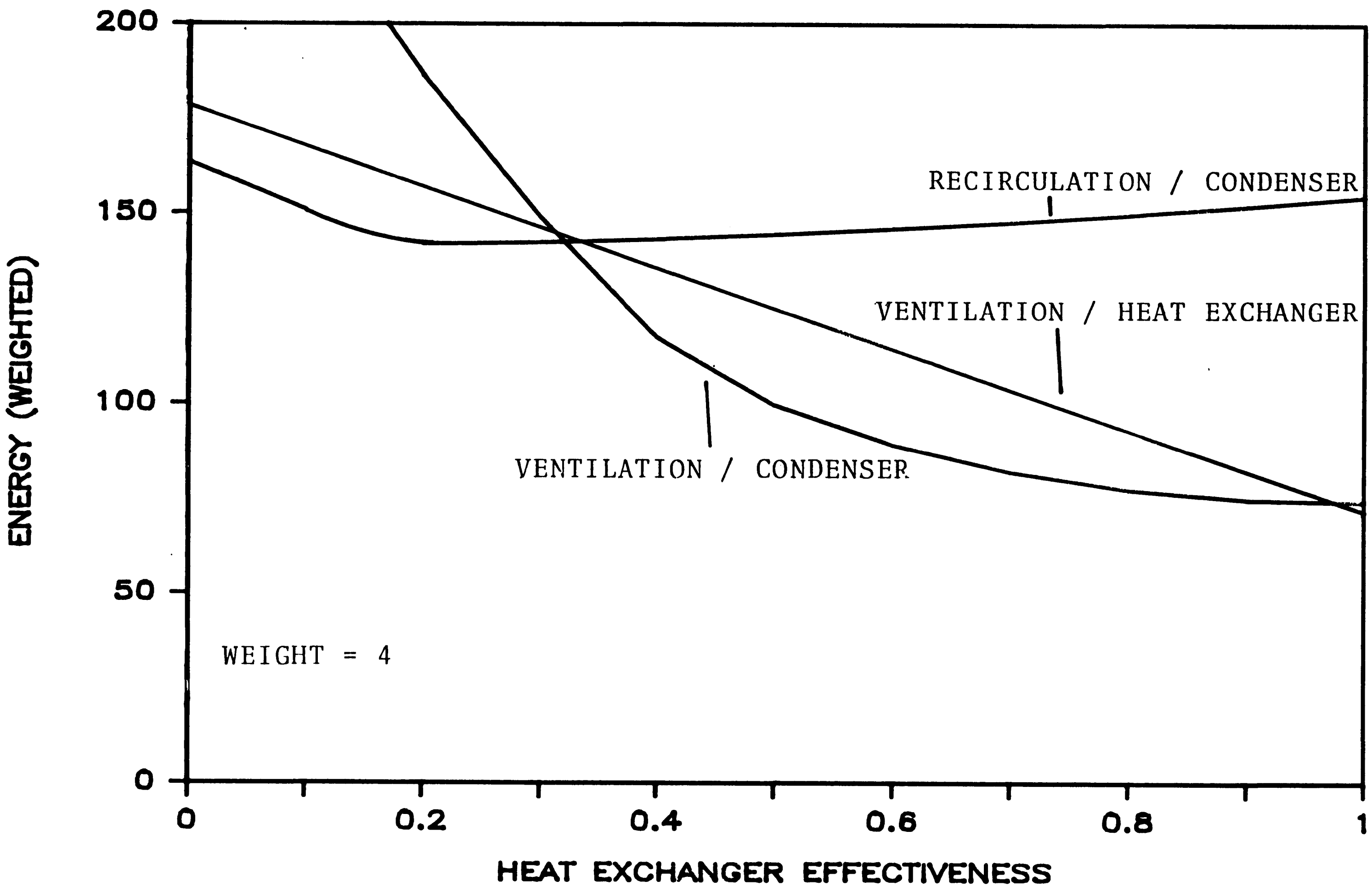


Figure 5.4.3 Comparison of the total energy use of the three hybrid systems at an energy weighting of four

power required for air circulation. In addition, a reduction in the flow rate decreases the energy consumption of the recirculation/condenser cycle. Figure 5.5.1 shows the reduction in total energy at different flow rates for the recirculation/condenser cycle. As the amount of air through the desiccant decreases both the amount of auxiliary heat required and the amount of vapor compression work decreases. Eventually at low enough flow rates the recirculation/condenser cycle consumes less energy than the ventilation cycles. The low air flow rate through the condenser associated with a system flow rate of 2.8 Kg/s causes the steep increase in weighted energy consumption at low heat exchanger effectiveness. The COP penalty taken when utilizing condenser heat becomes greater at low flow rates.

Except for fan power, the energy expenditure of the ventilation systems does not change as the circulation flow rate decreases. As the required amount of ventilation air is assumed to remain the same, the desiccant cycles require the same outlet humidity ratio and regeneration temperature, therefore the desiccant performance remains the same.

Figure 5.5.2 provides a comparison of the energy breakdowns in the various systems at their optimum system parameters. Table 5.5.1 lists the regeneration temperature and optimum heat exchanger effectiveness for each system. The hybrid systems all consume considerably less weighted energy than the standard vapor

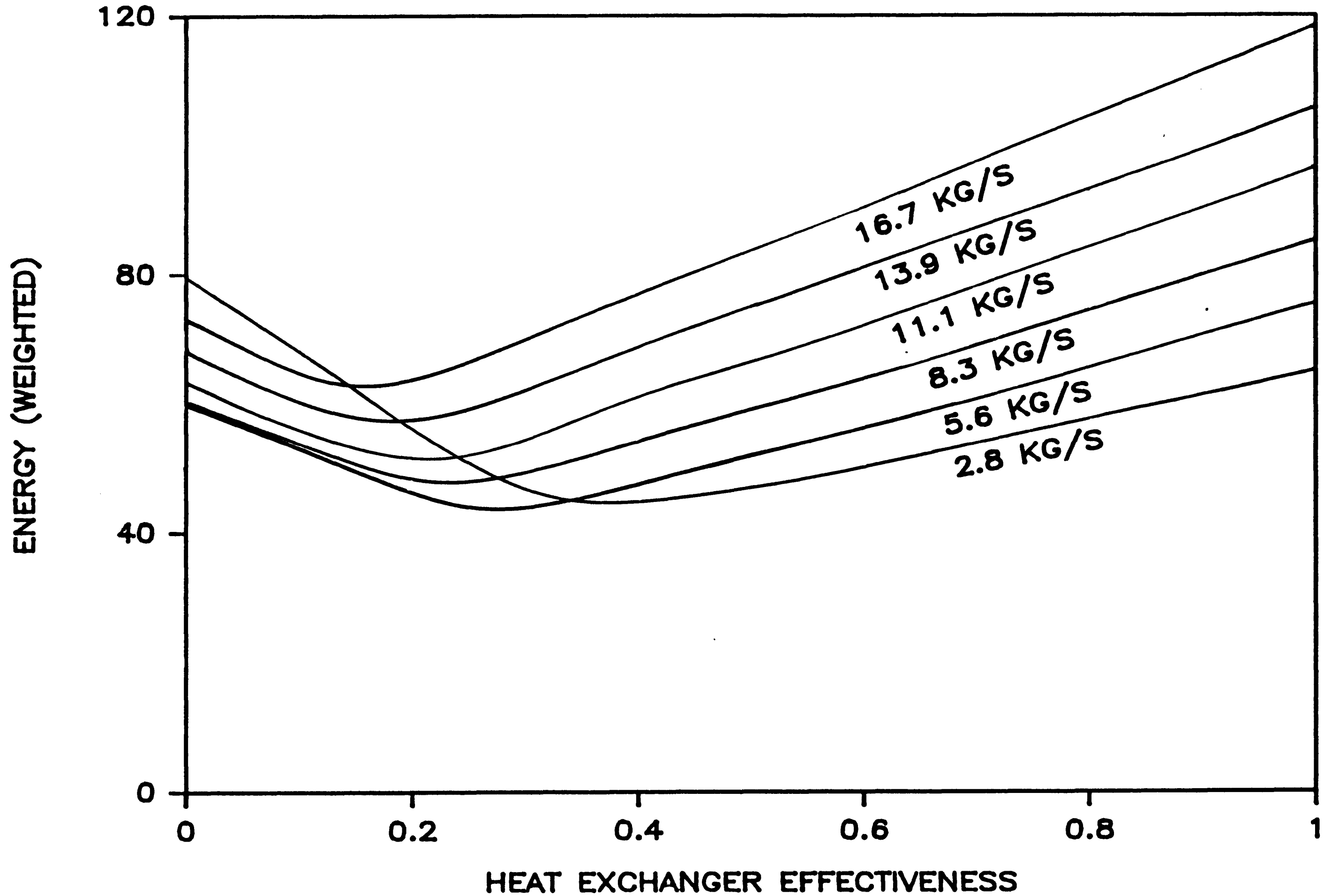


Figure 5.5.1 Recirculation/condenser cycle energy use at different system flow rates

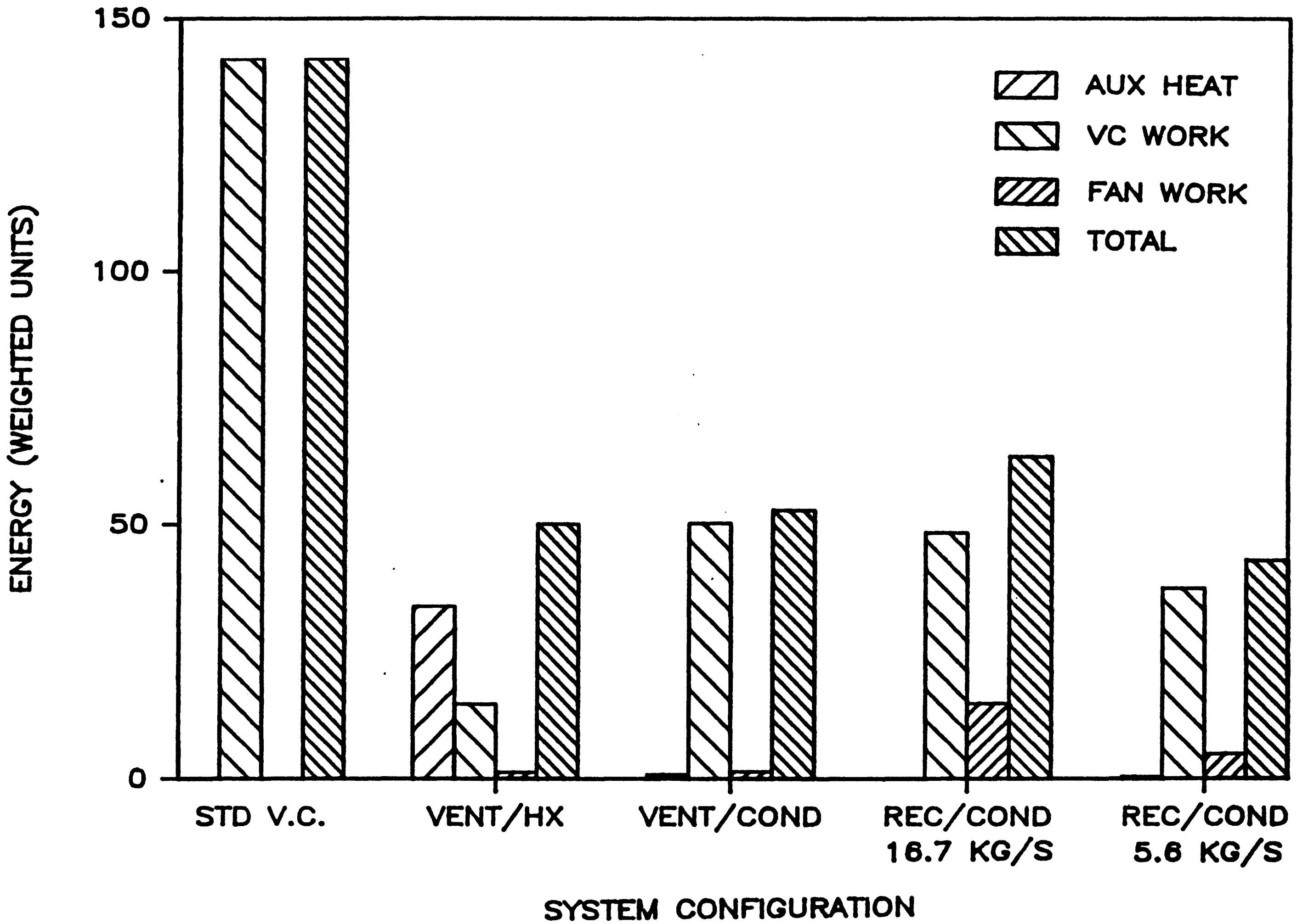


Figure 5.5.2 Breakdown of the energy use of various systems at optimum heat exchanger effectiveness

Table 5.5.1
Operating Conditions at Optimum ε_{hx}

	Std VC	Vent/HX	Vent/cond	Rec/Cond High Flow	Rec/Cond Low Flow
System Flow Rate	16.7 kg/s	16.7 kg/s	16.7 kg/s	16.7 kg/s	5.6 kg/s
Regeneration Temperature	---	81.4°C	81.4°C	38.1°C	46.8°C
Optimum ε_{hx}	---	0.9	0.44	0.18	0.27
System Energy Consumption (weighted units)	141	50	53	64	46

compression systems and they are all comparable in performance. At the optimum effectivenesses, the systems utilizing condenser heat do not require auxiliary heat. The regeneration heat requirements are met in the condenser. At standard system flow rates, despite performing less compressor work, the fan power required to circulate the large air flows through the desiccant make the recirculation/condenser cycles less favorable than the ventilation cycles. If the flow rate is decreased both vapor compression and fan work go down. The recirculation/condenser cycle requires a regeneration temperature of only 38 C. If a large source of moderate temperature heat were available (condenser heat from refrigerated cases or solar energy) the auxiliary heat requirement could easily be met by heat exchange with this source. With the heating requirement taken care of, the effectiveness of the IEC can be increased and more free cooling realized. The energy costs in this case would be quite low. The cost of auxiliary heat would still be zero (though the heating requirement is large) and the vapor compression work could be reduced to 6.3 kW (12.6 weighted units) at an IEC effectiveness of 0.8.

Section 5.6 Store Humidity Reduction

Since the electrical energy consumption of the refrigerated cases is the largest energy cost in the supermarket, one of the attractive ideas behind using a desiccant in this application is the potential to maintain lower store humidity levels at a lower cost than the vapor compression system. Lowering the humidity level in the store reduces the amount of water removal the cases must perform in order to maintain desired case temperatures. Again, a tradeoff exists between the reduction in case electrical energy consumption due to a lower humidity level and the energy cost of maintaining that lower humidity level.

Figure 5.6.1 displays the total air conditioning energy consumption of the ventilation/condenser cycle at various store humidity levels. These calculations are made considering the standard circulation flow rate, 16.67 kg/s, and an ambient condition of 30 C and .016 kg/kg absolute humidity ratio. Since the refrigerated cases do less work as humidity levels decrease (see Section 3.3), the internally generated load is increased accordingly. Removing sufficient moisture to maintain lower store humidity levels requires higher regeneration temperatures. The auxiliary heat required to meet these high regeneration temperatures is so large that the energy consumption of these systems increases considerably as the humidity levels go down. Figure 5.6.2 shows the weighted energy consumption of both the air conditioning system and the refrigerated cases as a function of store humidity level. Despite a refrigerated case energy use reduction of 15% at 0.007 kg/kg the savings are more than offset by the increase in auxiliary

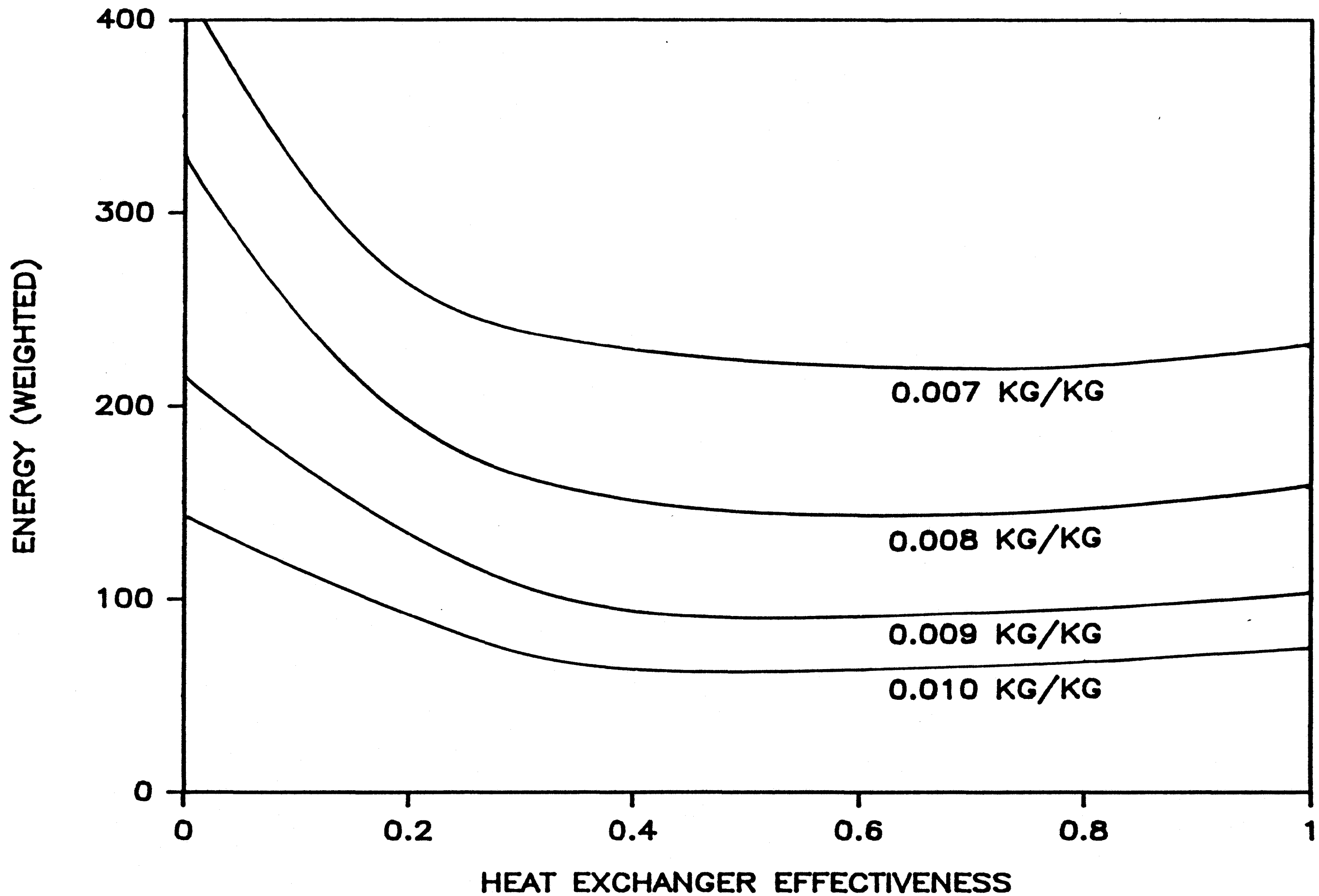


Figure 5.6.1 Ventilation/condenser cycle energy use at various store humidity levels

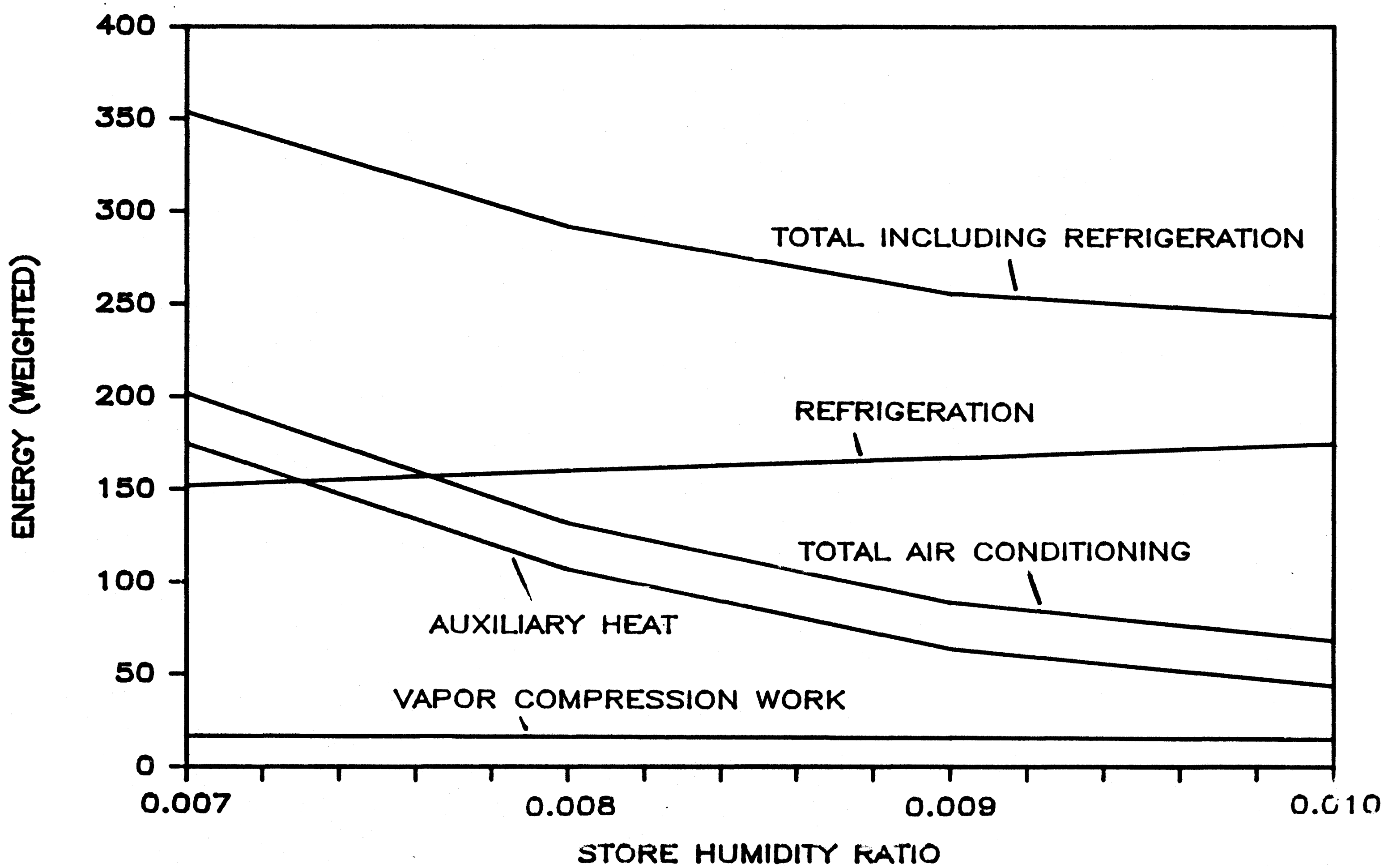


Figure 5.6.2 Breakdown of store energy consumption including refrigerated case consumption as a function of store humidity levels

heat requirements. While possible savings from reduced defrost periods has not been considered, this analysis indicates that lower store humidity levels would not decrease store energy costs.

Section 5.7 Performance Maps

Ambient conditions play a strong role in determining the performance of these cycles. Changes in the ambient conditions effect both the process and regeneration sides of the desiccant systems. In addition to contributing to the load through ventilation, ambient conditions affect the processes that drive the desiccant. Increases in the temperature and humidity of the process inlet state increase both the process outlet temperature and the required regeneration temperature. An increase in the humidity ratio of the regeneration air stream will also increase the regeneration temperature. Since ambient air is used for regeneration, the temperature of the air will affect the amount of heat required to meet the regeneration temperature. A high ambient air temperature will reduce the auxiliary heat requirement.

The effect of the ambient condition on system energy consumption is studied using the base case store conditions. The store load is held constant at 24kW (65% latent) before ventilation. While the ambient would unquestionably effect this load through transmission and infiltration, maintaining a constant load provides a good comparison of the relative abilities of the systems to remove the load at different ambient conditions. The total cooling load on the system will vary as ambient ventilation air is introduced.

Figures 5.7.1 and 5.7.2 show the required regeneration temperatures for the ventilation systems and the recirculation system respectively. The regeneration temperatures in the recirculation cycle are substantially less than those of the ventilation cycles. This is due to a higher process outlet humidity ratio and a lower temperature and humidity ratio in the inlet process stream.

At different ambient conditions a system consumes different amounts of weighted energy units. This energy consumption can be mapped by plotting contours of equal energy cost on temperature and humidity axes. Figure 5.7.3 shows a performance map for the standard vapor compression cycle. Energy consumption runs between 100 and 180 weighted units over the range of typical summer ambient conditions. As expected energy requirements increase as temperature and humidity increase.

Figure 5.7.4 presents the weighted energy contours for the ventilation/heat exchanger cycle. Throughout the region plotted, energy costs are substantially less than those for the standard vapor compression cycle. Energy use also increases as temperature and humidity increase, though humidity does play a stronger role in this cycle compared to the standard vapor compression cycle. The

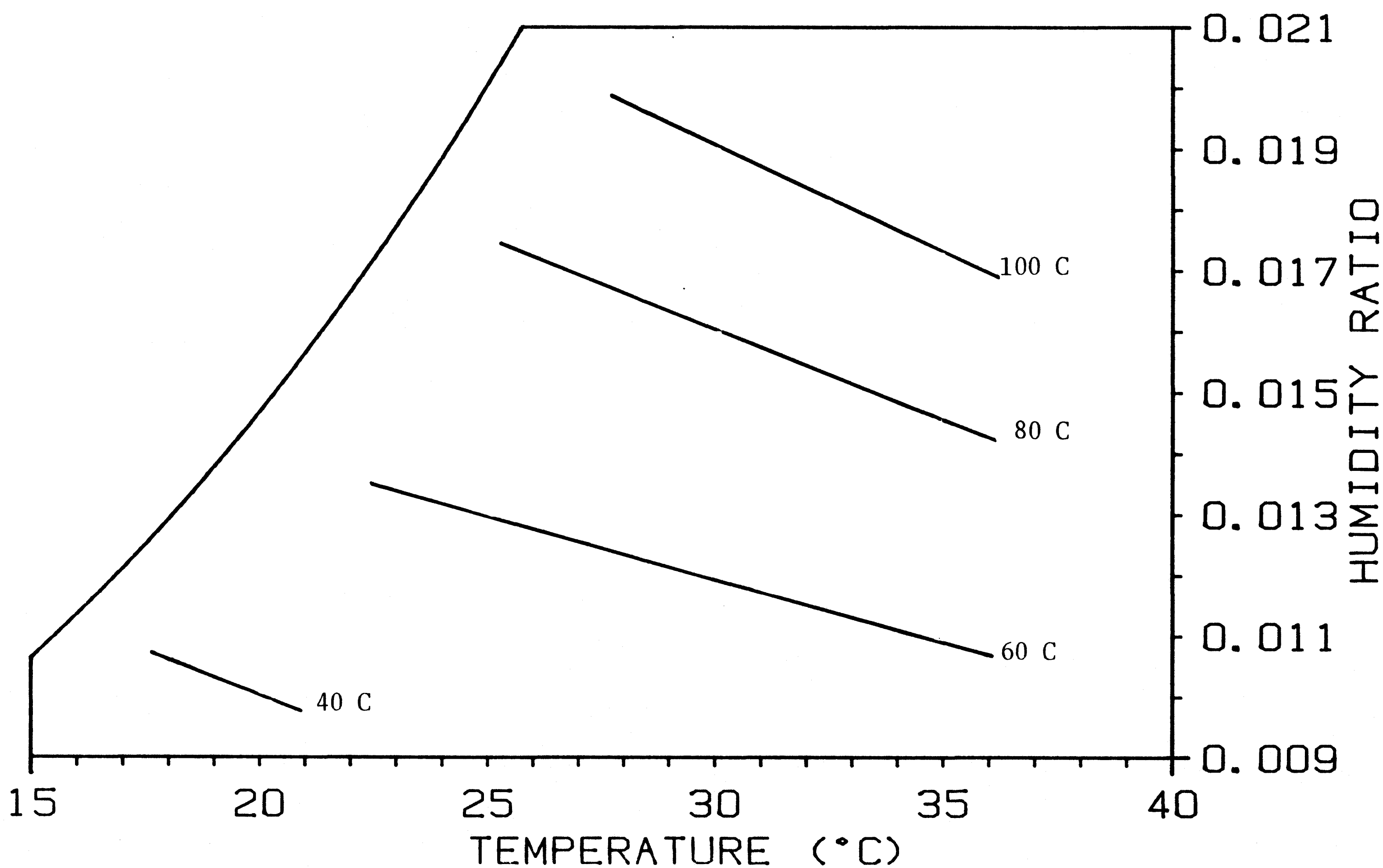


Figure 5.7.1 Regeneration temperatures required to meet the base case load for ventilation cycles as a function of ambient conditions

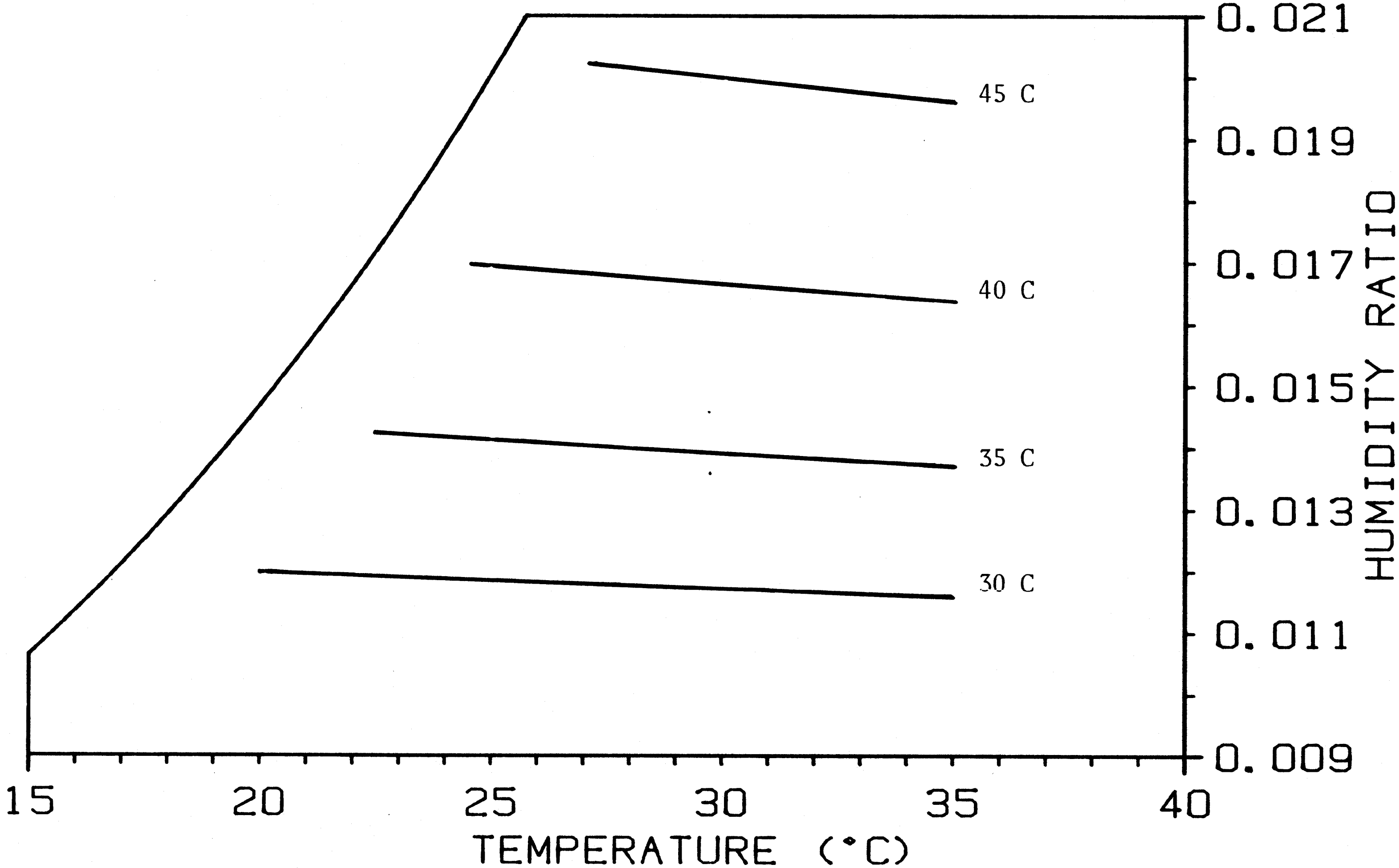


Figure 5.7.2 Regeneration temperatures required to meet the base case load for recirculation cycles as a function of ambient conditions

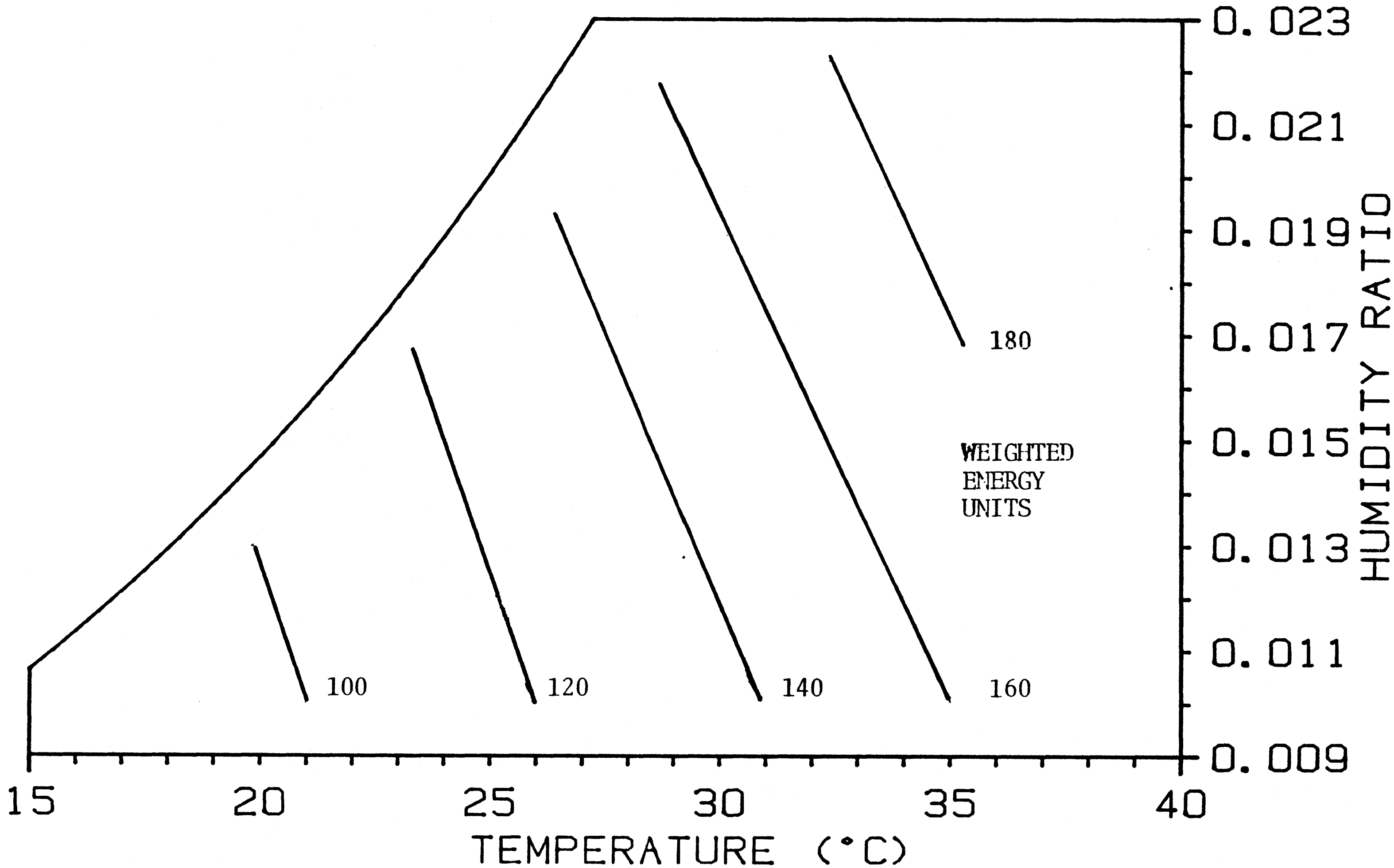


Figure 5.7.3 Weighted energy consumption of the standard vapor compression system over a variety of ambient conditions

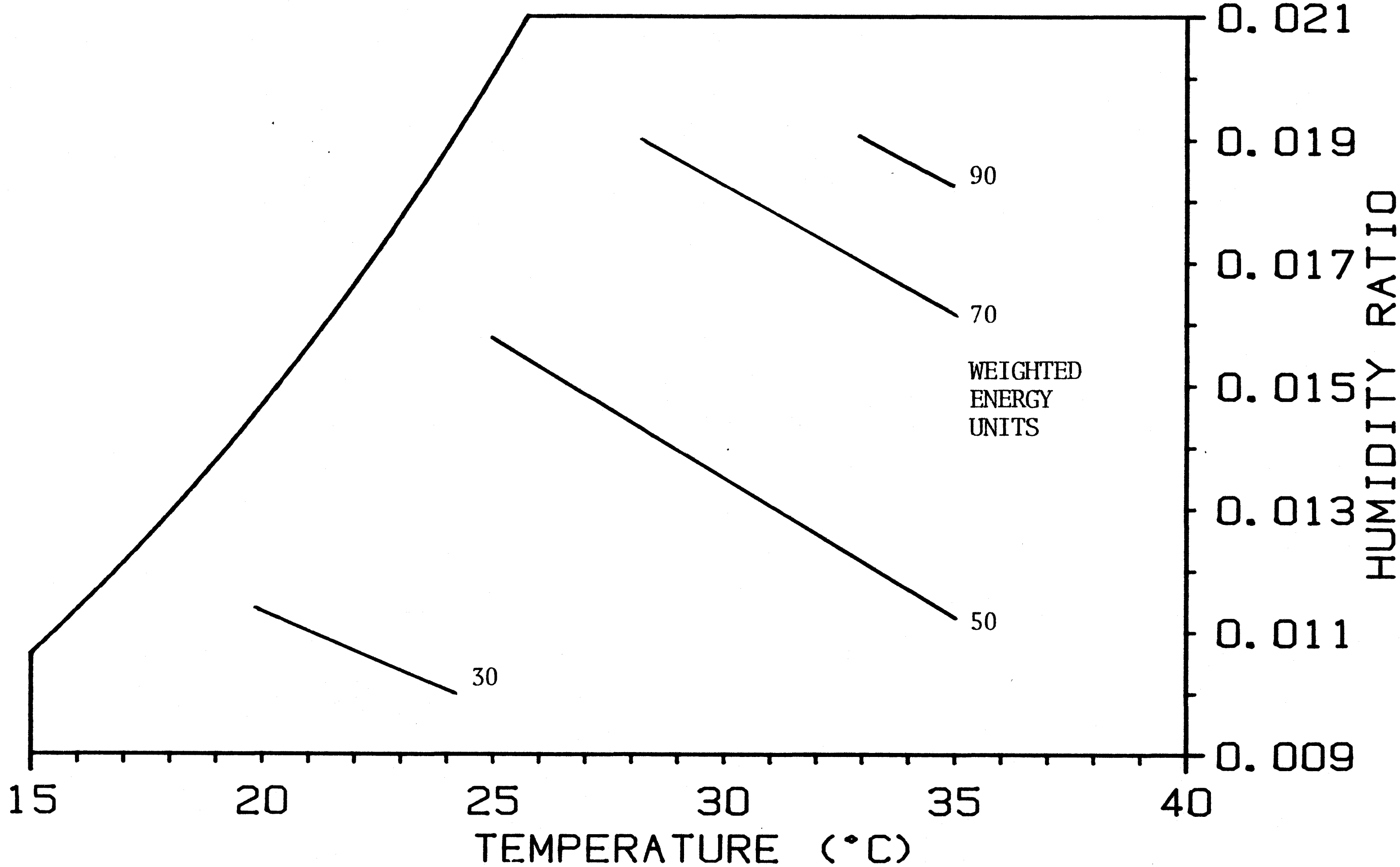


Figure 5.7.4 Weighted energy consumption of the ventilation/heat exchanger cycle over a range of ambient conditions

amount of auxiliary heat required is strongly influenced by the ambient humidity ratio. In the ventilation/condenser cycle as shown in Figure 5.7.5 the humidity ratio primarily determines the energy use. The increased cooling requirement caused by a higher ambient temperature is offset by a higher regeneration source and increased condenser heat.

The energy consumption for the recirculation/condenser cycle shown in Figure 5.7.6 decreases as ambient temperature increases. Since regeneration temperatures are quite low in this cycle, a few degrees difference in the ambient temperature can be a substantial portion of the regeneration requirement. At high ambient temperatures and low humidities, the ambient temperature is sufficient to regenerate this cycle. At low temperatures, very large amounts of heat are required for regeneration.

With the exception of the recirculation cycle near the saturation line all hybrid systems perform substantially better than a vapor compression system. The ratio of the energy costs of the ventilation/condenser cycle to that of the standard vapor compression cycle is presented in Figure 5.7.7. Savings are greatest at low humidities and high temperatures, and are more than 50% over the whole range of ambient conditions.

These performance maps suggest which systems might perform better at various ambient conditions. The recirculation/condenser cycle does not provide much savings over a vapor compression system

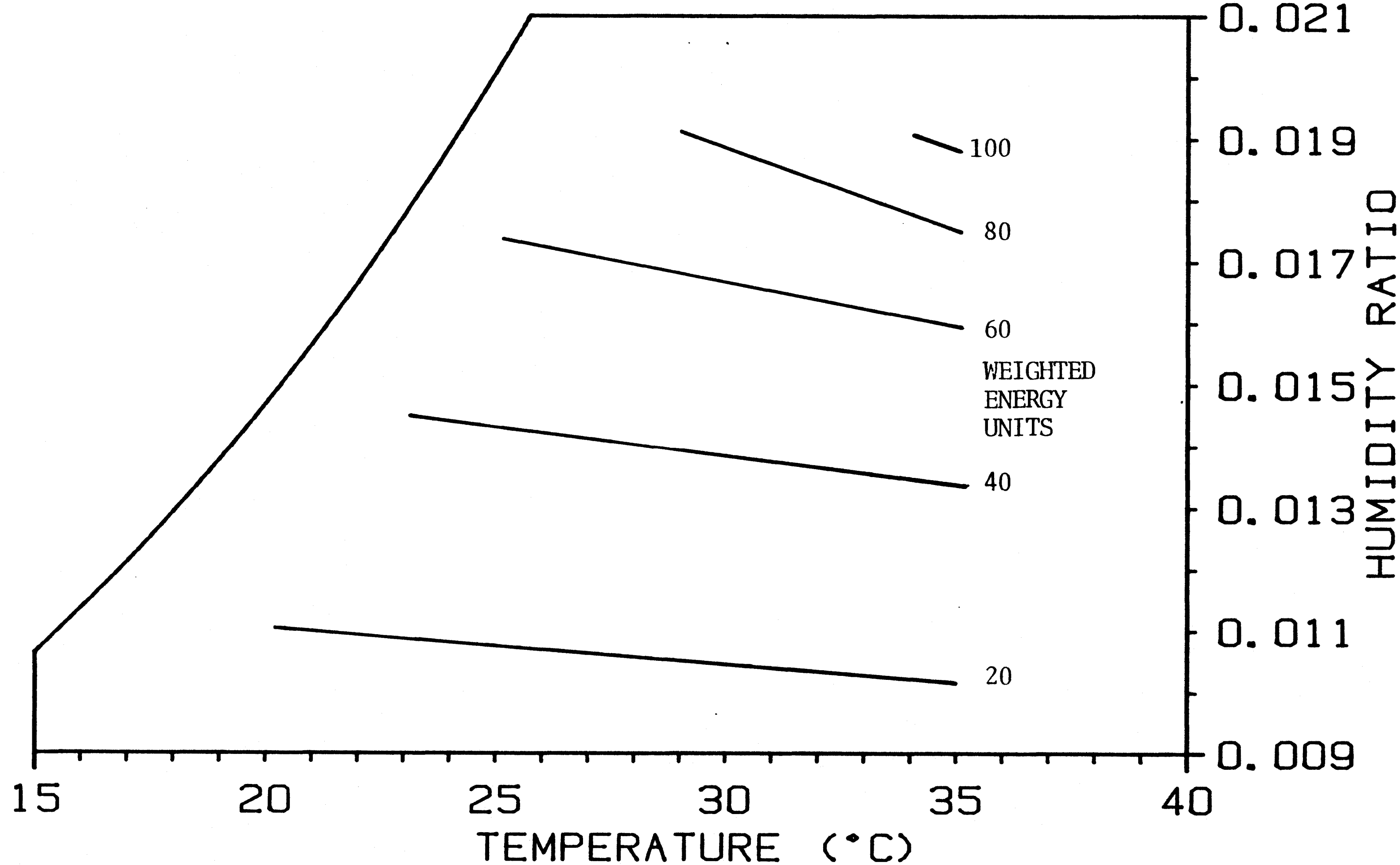


Figure 5.7.5 Weighted energy consumption of ventilation/condenser cycle

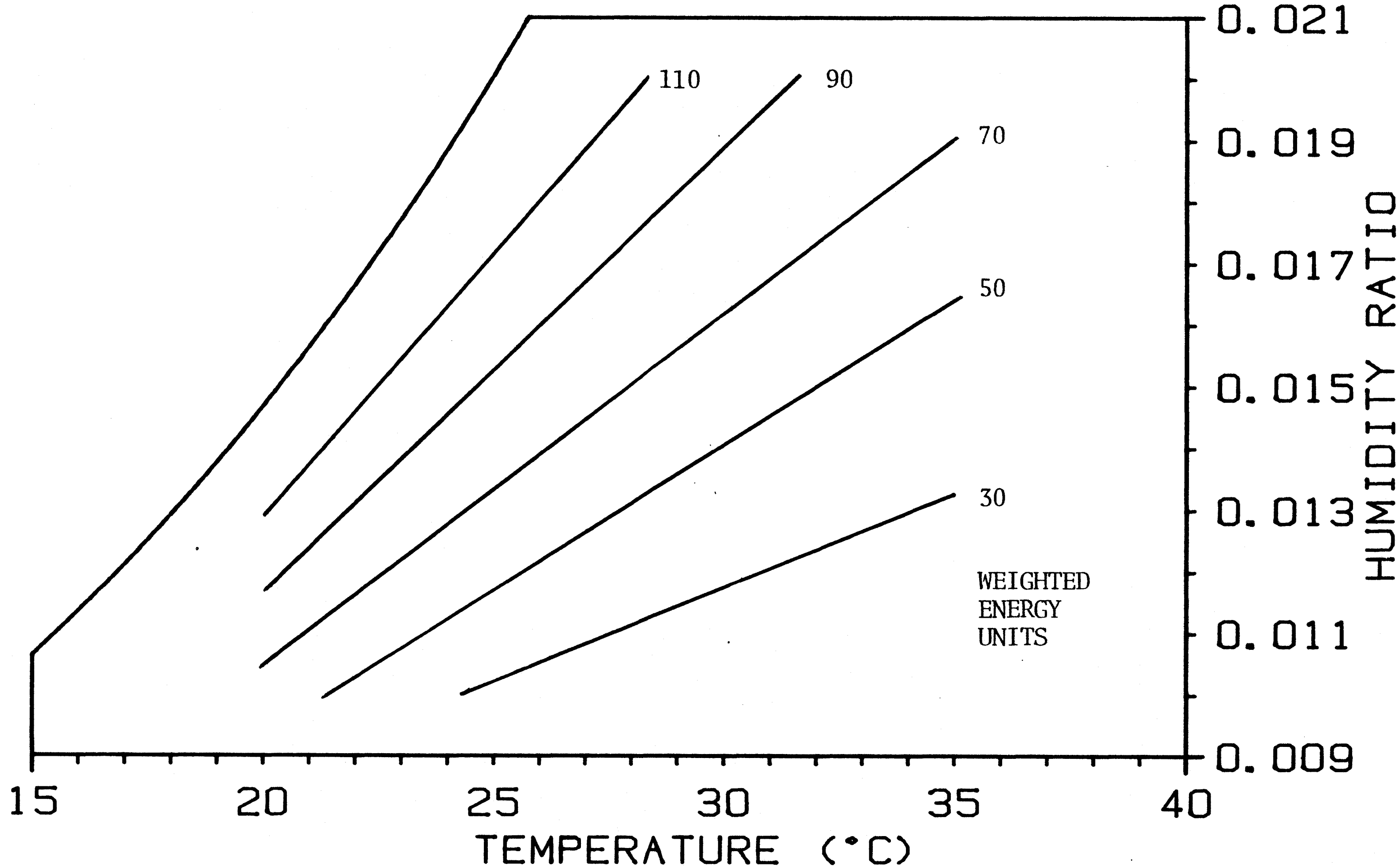


Figure 5.7.6 Weighted energy consumption of recirculation/condenser cycle

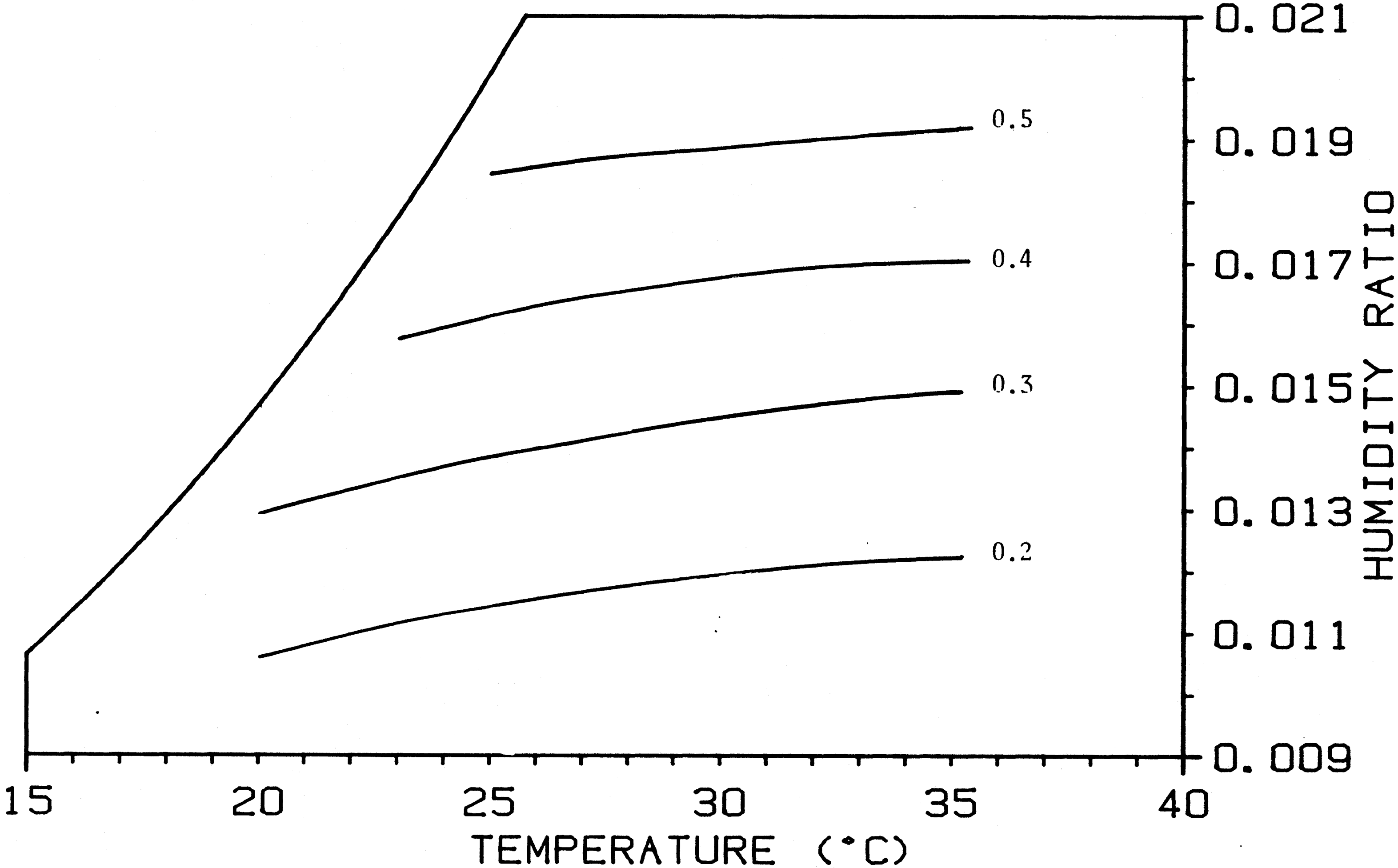


Figure 5.7.7 Ratio of the energy consumption of the ventilation/condenser cycle and the standard vapor compression cycle over a range of ambient conditions

at ambient conditions near saturation. At high temperatures and low humidities, where the recirculation/condenser cycle does perform well the large fan power requirements make it less atrractive than the ventilation cycles. In addition, the large flow rates involved will require larger equipment sizes increasing initial installation costs. The recirculation/condenser cycle therefore does not appear attractive for supermarket applications.

When high ambient humidity ratios exist the desiccant must remove large amounts of moisture from the air. This results in higher regeneration and process outlet temperatures. The ventilation/heat exchanger cycle's use of this high process outlet temperature for regeneration gives it advantage over the ventilation/condenser cycle when processing humid air. The ventilation/condenser cycle can achieve high regeneration temperatures leaving the condenser only with a reduction in the performance of the vapor compression unit. At lower humidity ratios less heat is available for the ventilation/heat exchanger cycle from the process outlet air and the lower regeneration temperatures can be achieved with condenser heat with less penalty. Figure 5.7.8 shows regions where each ventilation cycle consumes less energy than the other. At high humidities toward the saturation line the ventilation/heat exchanger cycle consumes less weighted energy than the ventilation/condenser cycle which does better at low humidities.

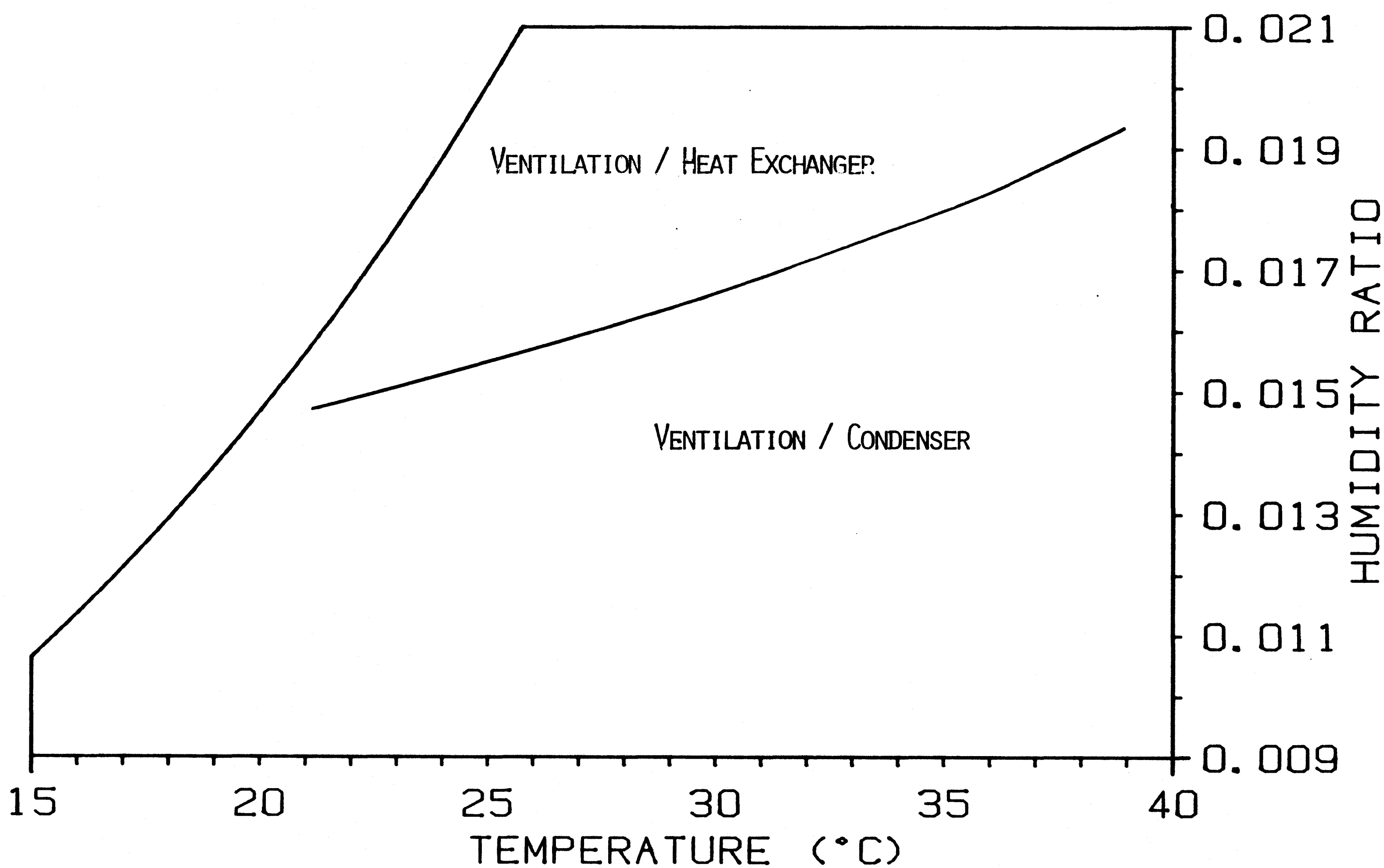


Figure 5.7.8 Ambient regions where each ventilation cycle maintains a performance advantage over the other

CHAPTER 6

SIMULATION RESULTS

Fixed condition calculations provide much information about hybrid system performance but are limited to arbitrary fixed ambient and load conditions. Hybrid systems performa under constantly changing conditions. Ambients and loads change with the time of day and with the season. A realistic idea of the actual benefits of a hybrid system requires an annual performance evaluation that accounts for these variations. This chapter reports annual simulation results of hybrid system operation in a typical supermarket.

Section 6.1 Store and Load Models

Thermo Electron has developed a description of a typical supermarket which has been adopted for use in annual simulations (13). The store is considered to be a single cooling zone which contains 2800 m^2 (30,000 ft^2)of floor space with a 7 m (23 ft) ceiling. The building walls, ASHRAE wall #12 (23), are light-colored, 8-inch concrete block with 2 inches of insulation and a U-value of 0.57 W/m^2 C (.08 Btu/hr ft^2 F). The store front faces south 60% of which contains a window with a U-value of 5.90 W/m^2 C (1.04 Btu/hr ft^2 F). A 4 m (13 ft) overhang shades the store front. Store parameters are summarized in Table 6.1.1.

Many of the factors contributing to internal loads depend on time of day schedules. The store modeled is open from 8 a.m. to 10 p.m., 365 days a year. Figure 6.1.1 shows the schedules used for calculating occupancy, ventilation, infiltration, lighting and refrigeration loads. People standing or walking slowly produce 90 Watts of sensible heat and 95 Watts of latent heat (24). The number of people inside the store varies throughout the day. Based on the 0.006 kg/m (1 cfm/ft^2) criteria, a standard circulation flow rate of 16.7 kg/s (30,000 cfm) is used. The ventilation requirement when the store is open is 1.67 kg/s (3000 cfm), 10% of the circulation flow rate. No air is ventilated during closed hours. An infiltration rate of .56 kg/s (1000 cfm) is assumed during open hours. When the store is closed, this is reduced to .14 kg/s (250 cfm). The store's lighting load is 32 W/m^2 (3 W/ft^2). Between the hours of 4 and 7 a.m. which is after restocking and before the store opens, this is reduced to 11 W/m^2 (1 W/ft^2). The store contains a total of 147 kW of installed refrigeration capacity divided between 102 kW in medium temperature cases and 44 kW in low temperature cases. The amount of cooling performed by the cases on the store environment is the product of the refrigeration capacity and the case on-time. The refrigeration on-time used 0.90 during

Table 6.1.1
Store Parameters

Dimensions	70 m	x	40 m	x	7 m	=	19,600 m^3
	230 ft	x	130 ft	x	23 ft	=	688,000 ft^3
Floor Area			2800 m^2				30,000 ft^2
Walls (ASHRAE #12) Light colored 8" concrete block 2" insulation			U = 0.57 W/m^2 C				0.1 Btu/hr -ft^2 -F
Roof (ASHRAE #1) Light colored 1" wood 2" insulation			U = 0.45 W/m^2 C				0.08 Btu/hr -ft^2 -F
South-facing storefront 60% window 4 m (13 ft) overhang			U = 5.9 W/m^2 C				1.04 Btu/hr -ft^2 -F
Installed Refrigeration Capacity			146 kW				42 tons
Medium Temperature			102 kW				29 tons
Low Temperature			44 kW				13 tons

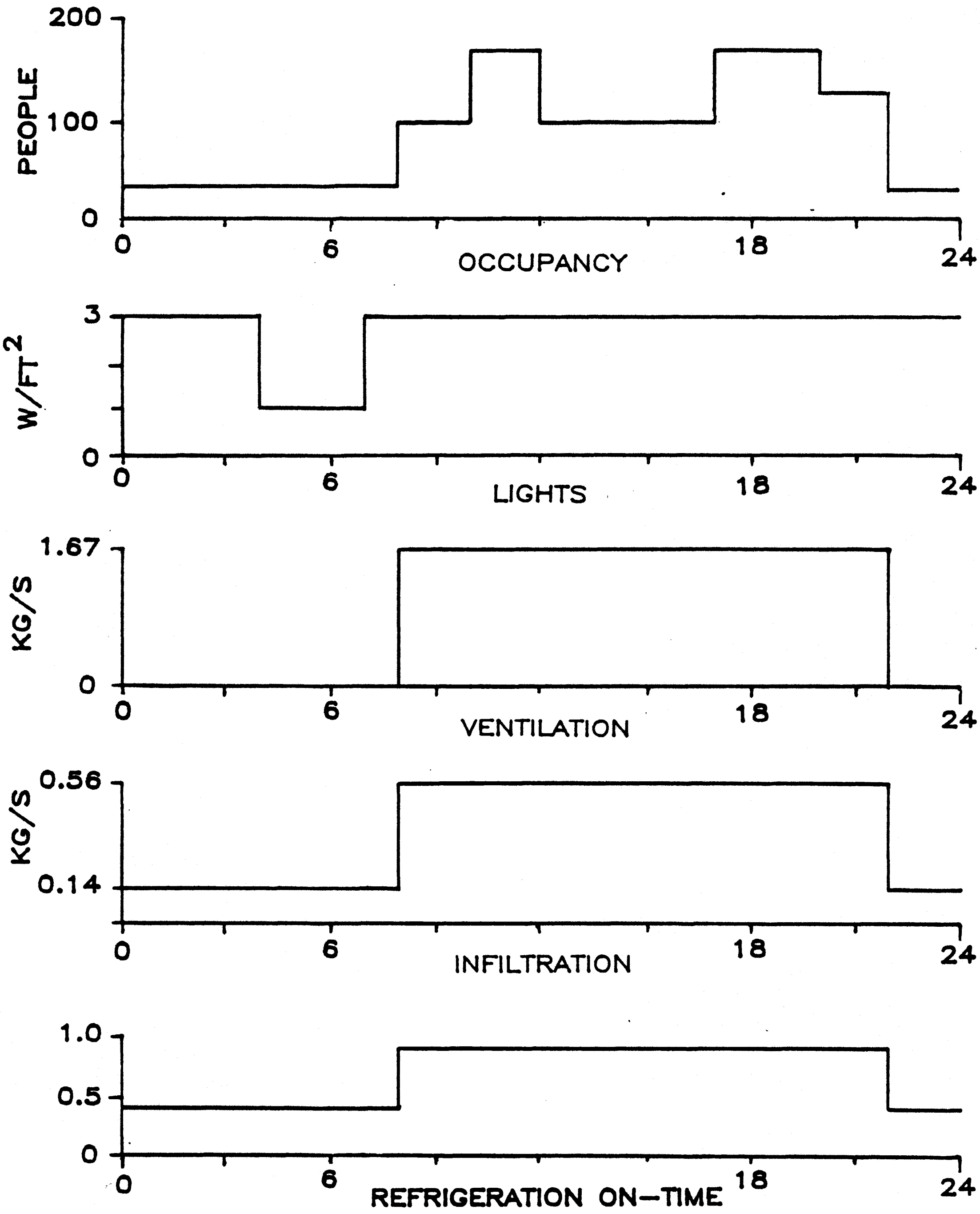


Figure 6.1.1 Schedules used in calculating store cooling loads

open store hours and 0.42 during closed hours. The lower on-time figure results from reduced air entrainment and lower case lighting loads. Further reductions in refrigeration loads due to lower store humidity levels are handled as described in Section 3.2.2. Internal moisture generation is assumed constant at 0.378 kg/s (30 lbs/hr).

The store cooling loads are calculated using the standard TRNSYS Type 19 Zone component (17). The cooling loads are calculated based on the Room Transfer Function Method described in ASHRAE (23). This method produces a time dependent sum of radiation, conduction and internal heat gains, which are used in determining the cooling load. The calculated cooling load is the energy removal necessary to maintain the store at the temperature and humidity set points which are 24 C and 0.01 kg/kg. The store conditions are not allowed to float on either side of the set points.

Ambient conditions and solar radition values for various locations are supplied by Solmet Typical Meterological Year (TMY) weather tapes (25.)

Section 6.2 Systems Studied

Yearly system simulations were performed for the standard vapor compression cycle, the ventilation/heat exchanger cycle and the ventilation/condenser cycle using the transient simulation program, TRNSYS (17). In addition, a system utilizing solar energy for regeneration heat is studied. From the fixed condition studies the energy use and cost of the recirculation/condenser system were so high that no annual studies were conducted with this system.

Various heat exchanger effectivenesses were used in the fixed condition studies to determine the optimum operating point. Searching for the optimum heat exchanger effectiveness at every time step is impractical in an annual simulation and a fixed heat exchanger effectiveness was chosen for each simulation. The ventilation/heat exchanger cycle always performs better at high effectiveness and therefore a heat exchanger effectiveness of 0.9 is chosen for this system. The ventilation/condenser cycle contains a trade-off between heat exchanger effectiveness and auxiliary heat requirements. To explore this trade-off, simulations using heat exchangers with effectivenesses of 0.6 and 0.9 are performed.

Since hybrid systems utilize thermal energy for regeneration, solar energy may be used to offset the auxiliary heat requirement. Figure 6.2.1 shows a schematic diagram of a cycle incorporating solar energy. Solar collectors are placed after the heat exchanger in the regeneration stream of a ventilation/heat exchanger cycle. The collector modeled using the standard Type 1 TRNSYS component is a commercially available air-type collector. The collector parameters listed in Table 6.2.1 were obtained from standard SRCC test results. (26) If the incident radiation on the collector does not exceed the critical level, the collector is bypassed.

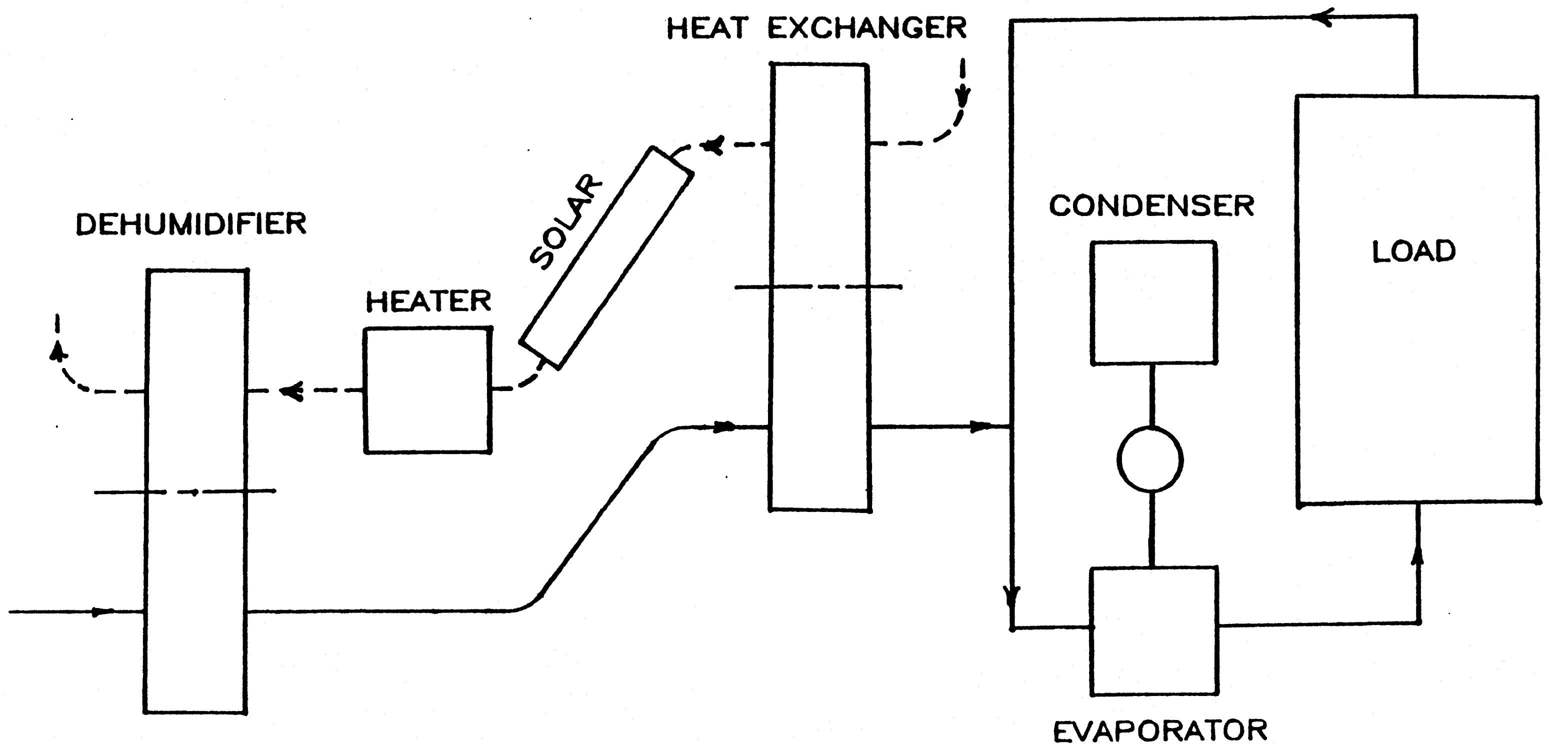


Figure 6.2.1 Schematic diagram of a cycle incorporating solar energy

Table 6.2.1
Solar Collector Parameters

Collector fluid	-	air
$F_R(\tau\alpha)_n$	=	0.6190
$F_R U_L$	=	-4.177 W/m^2-^oC (.58 Btu/hr -ft^2 -F)
b_o	=	-0.221
Test flow rate	=	0.0256 kg/s-m^2 (140 scfm)

Section 6.3 System Comparisons

The results of annual simulations for a store located in Ft. Worth are used to illustrate performance differences among the systems. Hourly loads were calculated and stored in a file for use in each simulation. Table 6.3.1 contains monthly averages of ambient temperature, ambient humidity ratio, and sensible and latent loads before the ventilation requirement is considered for Ft. Worth. Negative loads mean that heating or humidification are necessary to maintain store set point levels of 24 C and 0.010 kg/kg. The primary cooling requirements occur in the summer months of June, July, and August, with July being a little warmer and less humid than June and August.

Hybrid desiccant systems reduce the vapor compression energy needs substantially. Figure 6.3.1 depicts the monthly energy requirements of the vapor compression unit for the various systems over a year for a store located in Ft. Worth. The majority of the cooling requirement occurs during the three summer months of June, July and August. All hybrid systems consume at least 75% less electrical energy for compression than the standard vapor compression cycle. Among the hybrid cycles the ventilation/heat exchanger cycle uses the least energy. Vapor compression requirements in the ventilation/solar cycle are exactly those of the ventilation/heat exchanger cycle as solar energy is used only to

Table 6.3.1
Monthly Average Conditions in Ft. Worth

Month	$T_{ambient}$	$W_{ambient}$ kg/kg	$Q_{sensible}$ kJ/hr	Q_{latent} kJ/hr
January	7.0	.0047	-143,000	-20,000
February	9.7	.0057	-124,000	-16,000
March	12.0	.0052	-103,000	-19,000
April	17.7	.0090	- 55,000	- 6,000
May	21.5	.0120	- 11,000	5,000
June	26.9	.0152	35,000	15,000
July	30.1	.0133	69,000	9,000
August	28.7	.0142	51,000	11,000
September	23.6	.0186	20,000	4,000
October	19.5	.0099	36,000	- 2,000
November	13.7	.0069	- 84,000	-13,000
December	8.0	.0052	-135,000	-18,000

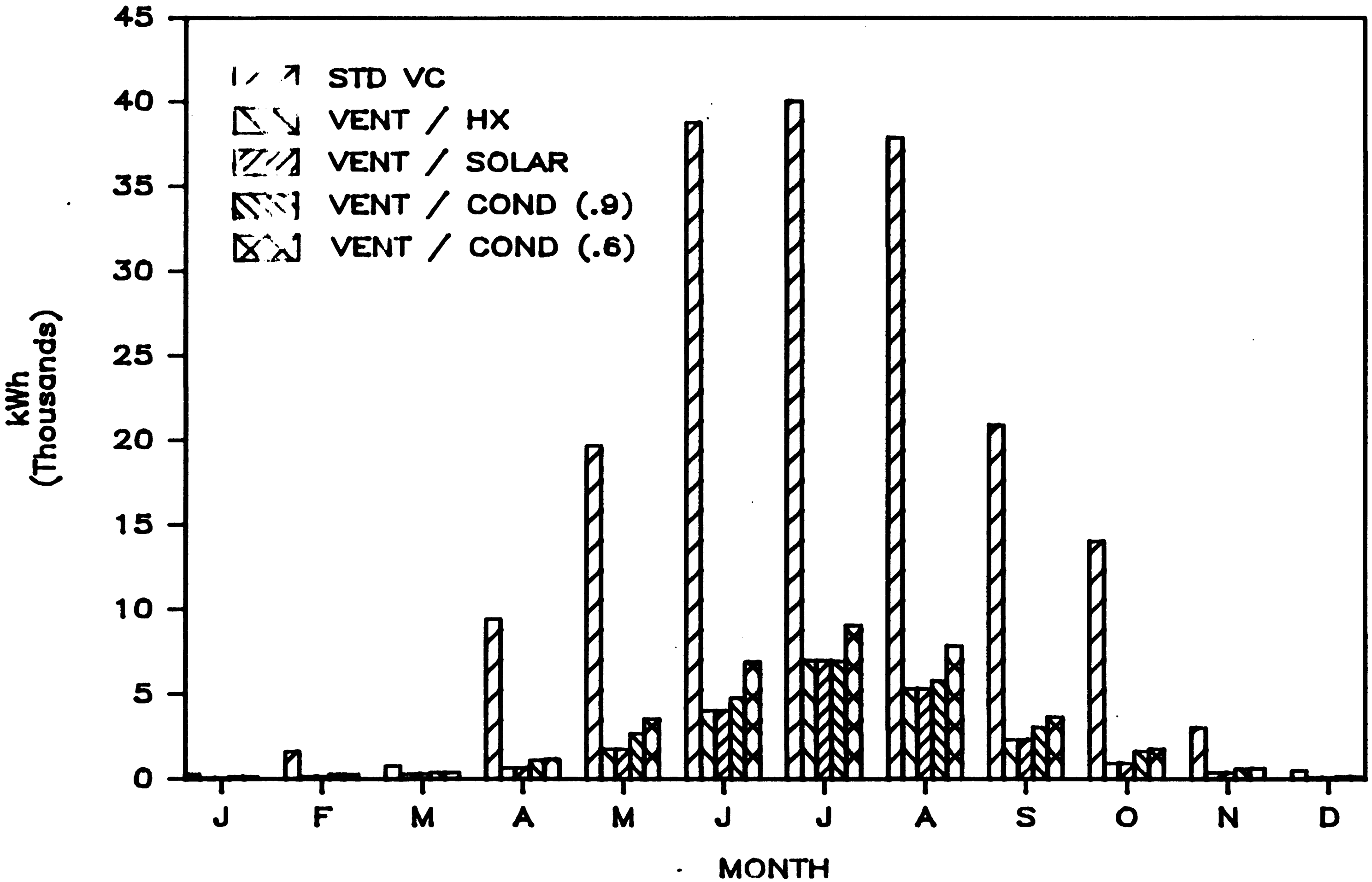


Figure 6.3.1 Comparison of the monthly vapor compression energy requirement of the various systems for a store located in Ft. Worth

offset auxiliary heat requirements. Despite receiving additional free cooling from an indirect evaporative cooler the ventilation/condenser cycle with an indirect evaporative cooler effectiveness of 0.9 requires more energy than the ventilation/heat exchanger cycle. This is due to the performance penalty taken when raising the condenser temperature to utilize condenser heat for regeneration. The expected increase in vapor compression requirements can be seen when the indirect evaporative cooler effectiveness is reduced to 0.6.

The vapor compression reduction comes at the expense of an auxiliary heating requirement. Figure 6.3.2 depicts the auxiliary heat consumption of the various systems. There is no auxiliary heat requirement for the standard vapor compression cycle. Generally, the heat requirements for the ventilation/condenser cycle with a 0.9 effectiveness exceed those of the ventilation/heat exchanger cycle. The warmer and drier conditions create the smaller heat requirements in July, and favor the ventilation/condenser cycles. The use of solar collectors cuts the auxiliary heat needed in half. The expected trade-off in auxiliary heat needs and indirect evaporative cooler effectiveness can be seen in the ventilation/condenser cycles.

Adding the vapor compression work, the auxiliary heat and fan power requirements together provides a total energy cost comparison among the systems. Since electricity and gas costs differ, Figure 6.3.3. presents monthly energy costs of the various systems in

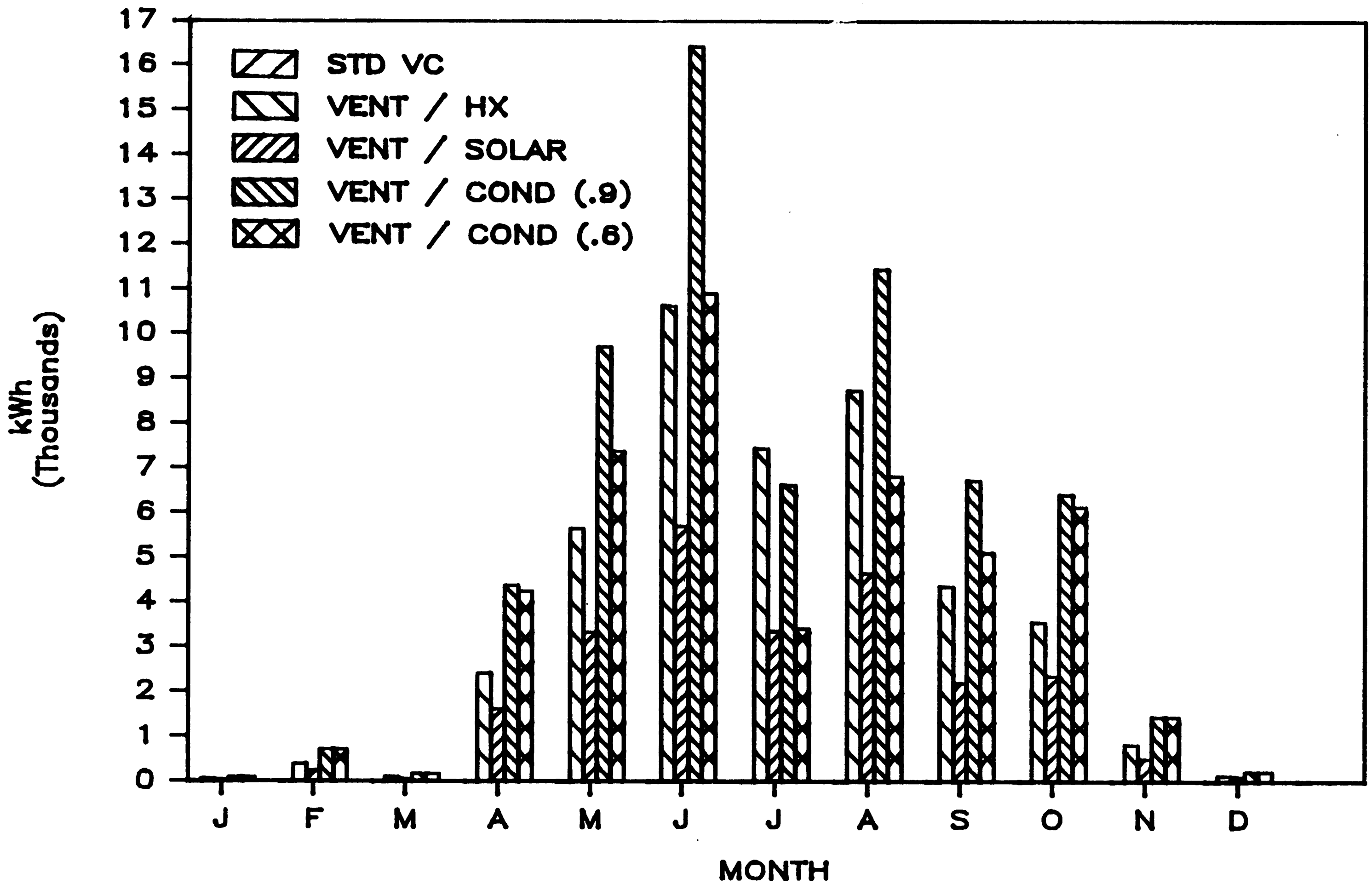


Figure 6.3.2 The monthly auxiliary heat consumption of the various systems for a store located in Ft. Worth

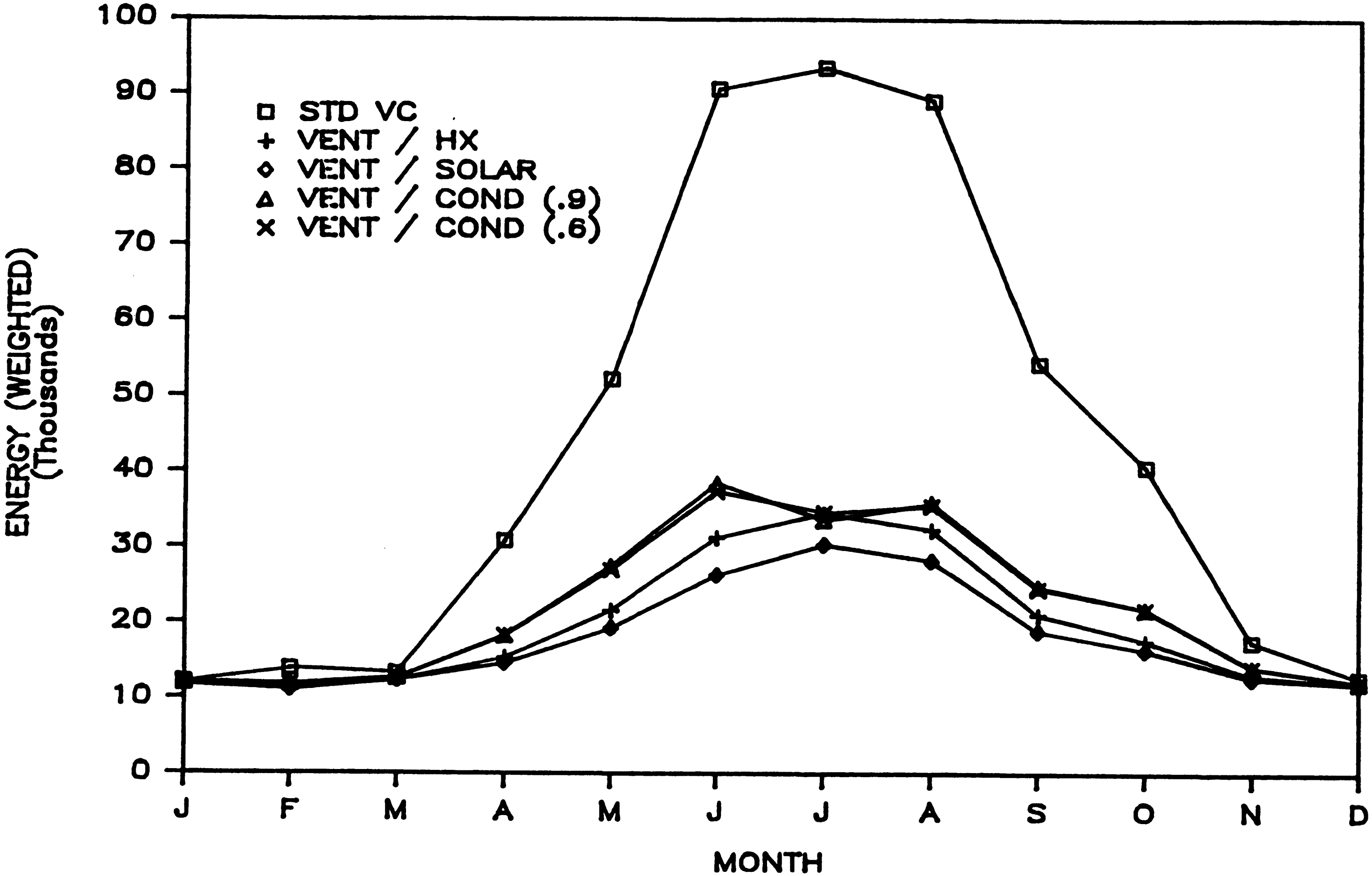


Figure 6.3.3 Total air conditioning energy use of the various system for a store located in Ft. Worth

weighted energy units with a weighting factor of 2. The energy cost during the winter months is primarily due to the fan power required to circulate air through the store. This is common to all the systems. The additional fan power needed for the desiccant streams is small in comparison amounting to about 10% of the total fan power. All the hybrid systems require considerably less energy than the standard vapor compression system. Among the desiccant cycles, the ventilation/heat exchanger cycle consumes consistently less energy throughout the year than the ventilation/condenser cycles. No significant difference in energy cost exists among the two ventilation/condenser cycles, with the heating-cooling trade-off evening out over the month. The indirect evaporative cooler with the lower effectiveness will have a lower initial cost.

During the three peak summer months the ventilation/heat exchanger cycle consumes about 60,000 less weighted energy units per month than the standard vapor compression system. Assuming a cost of 3.5 cents per weighted unit (7 cents per kWh electricity) this produces a monthly savings of $2100.

Figure 6.3.4 presents the monthly energy costs of the various systems for a store located in Miami. Miami has a year long dehumidification requirement which produces a large savings potential each month. Miami's climate is much wetter and a little cooler than Ft. Worth's. The ventilation/heat exchanger system maintains an advantage over the ventilation/condenser cycle under

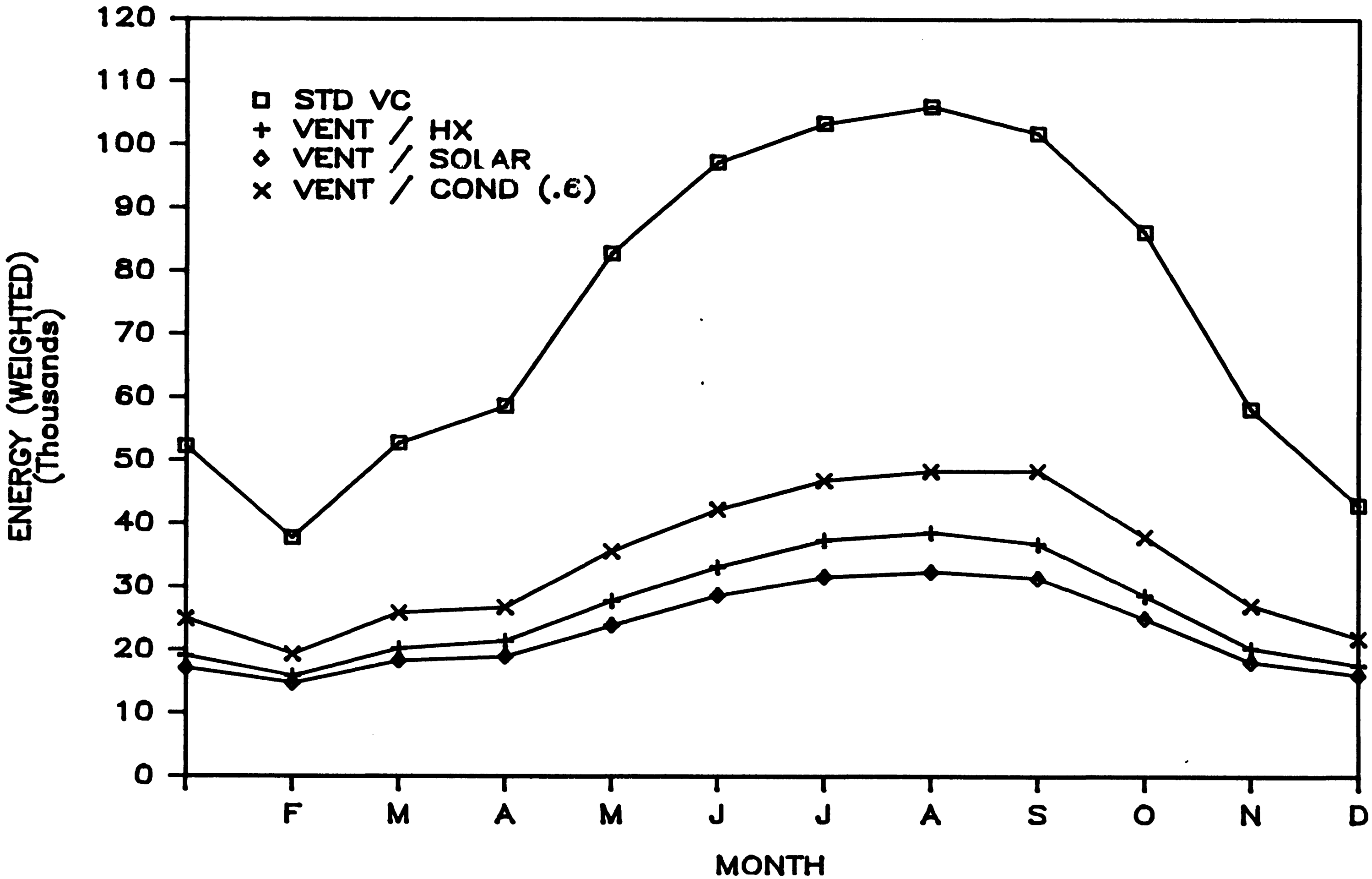


Figure 6.3.4 Total air conditioning energy use of the various systems for a store located in Miami

these ambient conditions. Again solar collectors reduce the energy cost of the ventilation/heat exchanger system

Section 6.4 Use of Solar Energy

Solar energy may be used to offset the auxiliary heat required for regenerating desiccant systems. To evaluate the potential effect of solar energy on system costs, different sized solar arrays are placed in the regeneration air stream. The amount of heat collected will increase as the area of the solar collectors increases. Not all of this additional heat can be used. After the regeneration requirement has been met, additional heat is of no use unless it is stored for a later time when it can be utilized. Figure 6.4.1 shows the auxiliary heat requirement for the ventilation/solar cycle in Ft. Worth at various collector areas. A collector area of 100 m^2 produces a large reduction in the heating requirement. Additional collector area has little effect. As collector area is increased there are fewer and fewer hours which can use the additional heat to reach the regeneration temperature at times when solar energy is available. In a no storage system, excess heat must be dumped. Figure 6.4.1 suggests that just under half the auxiliary heat requirement in Ft. Worth can be met by a no storage solar system.

Since dehumidification needs exist when solar energy is not

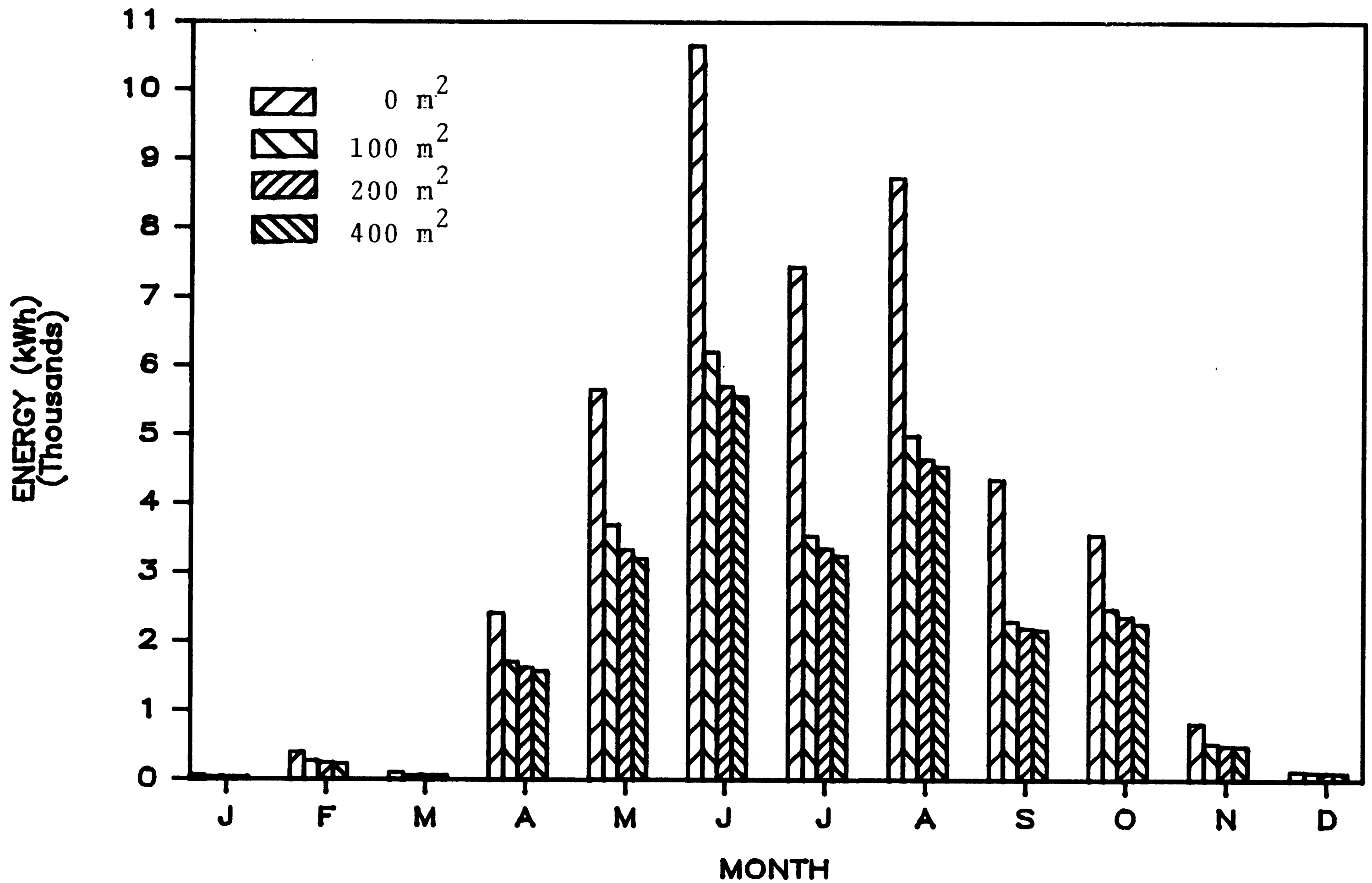


Figure 6.4.1 Monthly auxiliary/heat requirement for the ventilation/solar cycle in Ft. Worth at various collector areas

directly available, solar energy will not be able to meet all of the regeneration heat requirement without the addition of a storage system. To explore the ability of solar collectors to fully meet the regeneration needs the concept of infinite storage is introduced. Infinite storage implies that all energy leaving the collector in excess of the regeneration requirement can be stored with no losses and at the temperature it was collected. Despite being an ideal situation, this concept is useful in determining an upper performance boundary for solar energy.

By breaking down the auxiliary heat requirement into temperature bins, the amount of heat needed at each temperature level over a given time period can be determined. Similarly, excess heat available for storage may be broken into bins. This produces an idea of how much and at what temperature heat is needed, and how much and at what temperature heat is stored. Stored energy can be used to meet the auxiliary requirements at the same or lower temperatures, but not higher temperatures.

Figure 6.4.2 shows auxiliary heat and storage bins for 100 m^2 and 200 m^2 of collector area in Ft. Worth. As the collector area increases the auxiliary needs decrease slightly as more solar energy is used directly for regeneration. In addition more heat is available for storage and is available at an increased temperature, since the air flow rate through the collector stays constant. The algorithm used to determine the effect of the stored energy in

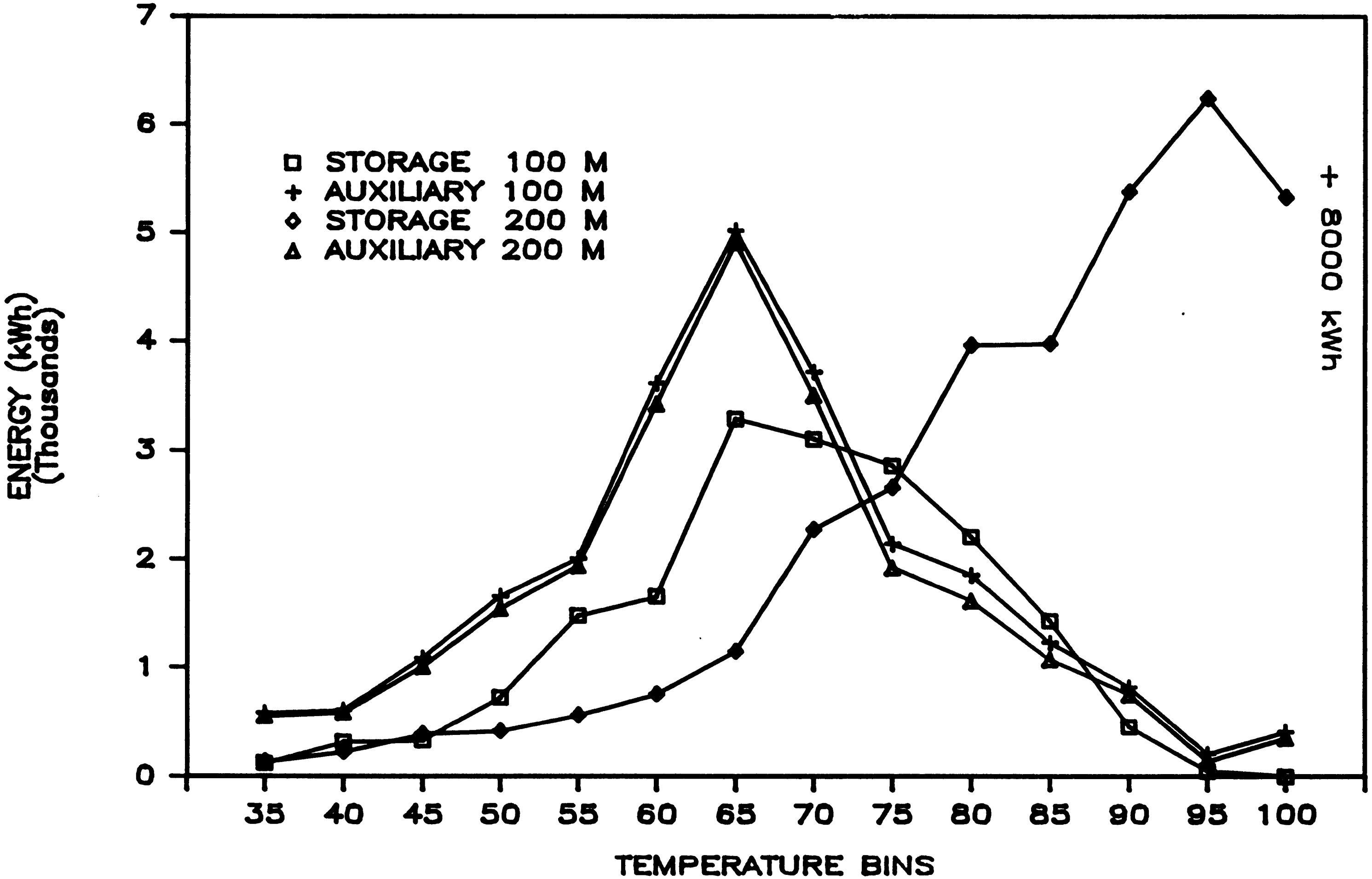


Figure 6.4.2 Auxiliary heat and storage bins for Ft. Worth at collector areas of 100 m^2 and 200 m^2

reducing the auxiliary heat needs starts with the energy stored in the highest temperature bin. This high temperature heat is used to meet the auxiliary requirements at this temperature. If any energy is left over it is used to meet the auxiliary heat in the next highest temperature bin and so on until the heat in that storage bin is exhausted. Each storage bin is successively stepped through from high temperatures to low temperatures. When the situation arises that the auxiliary heat requirement in a particular bin exceeds the amount stored at that temperature and that leftover from higher temperatures, the difference must be purchased.

This technique may be used over any time period, such as a day, week, month, or year. The larger the time period the larger the storage capacity must be. Yearly storage means that the storage tank is large enough to hold all the heat collected over a year, whereas daily storage only needs to hold collected over 24 hours. Obviously, smaller storage capacities are less expensive. Time periods are fixed. beginning at the first midnight of the day, month or year considered. It is assumed that any heat stored within the time period may be used at any other time. For a yearly storage system this can mean that heat stored in June has already been used in February. In actuality it would be used the following February.

Figure 6.4.3 shows the annual purchased auxiliary heat for various collector areas and storage sites in Ft. Worth. As area and storage sites increase, purchased energy goes down. A system with

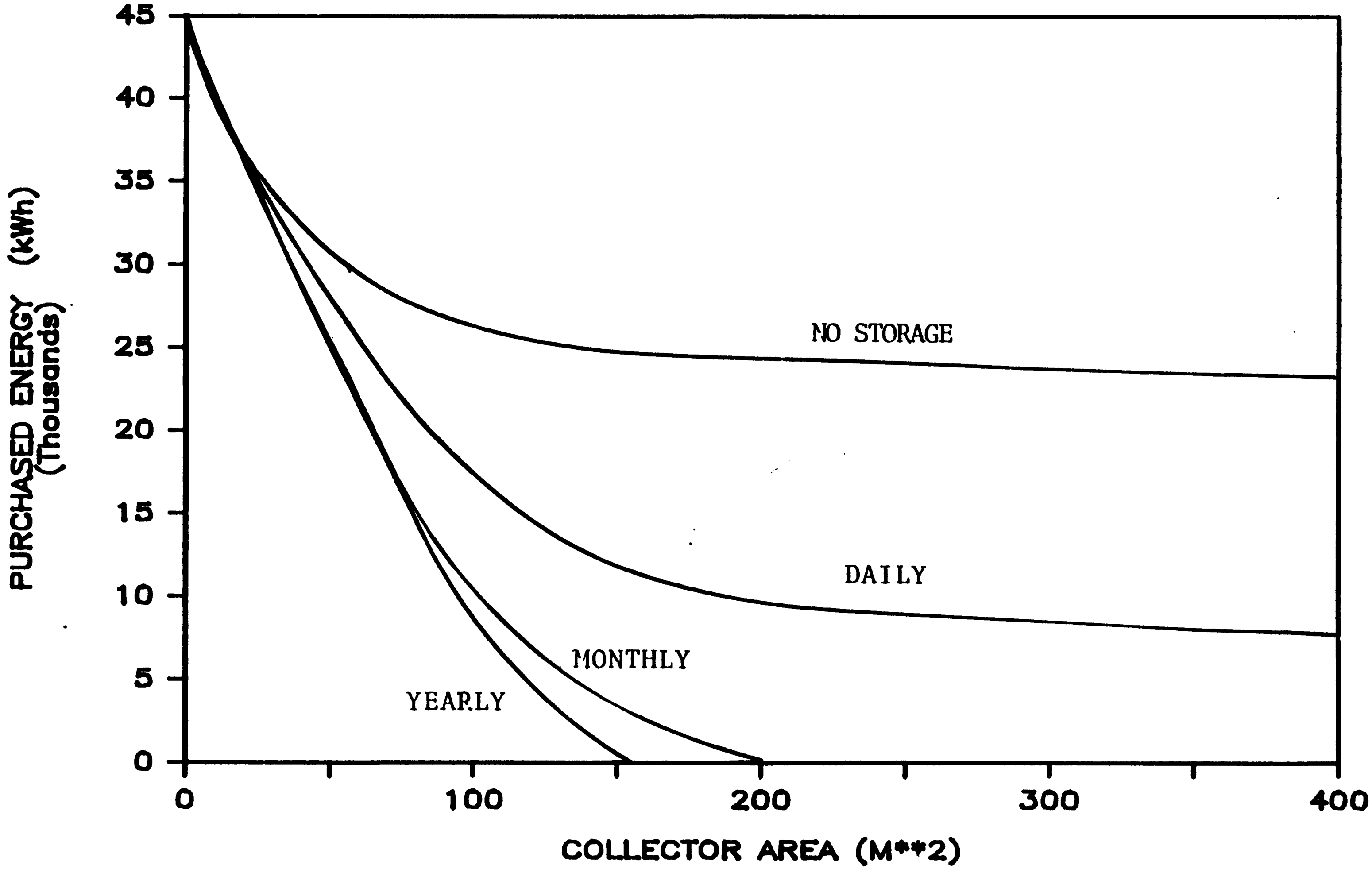


Figure 6.4.3 Auxiliary heat requirements for Ft. Worth at various storage sizes as a function of collector area

200 m^2 of collector area and a monthly storage capacity completely meets the auxiliary load. Less expensive daily storage reduces the load 75%. Figure 6.4.4 shows a similar figure for Miami. Without solar energy the auxiliary heat requirement is over twice that of Ft. Worth. A collector area of 400 m^2 and yearly storage are required to remove the total load.

With large collector areas and storage tanks, solar energy is capable of removing a good portion of the auxiliary heat requirements. Before solar collectors are installed in a hybrid system the fuel savings attributable to the collectors must be able to offset their additional cost in a reasonable period of time. Figures 6.4.5 and 6.4.6 show the annual fuel savings in dollars per square meter of collector as a function of collector area and storage size for Ft. Worth and Miami. The installation of 50 m^2 of collector in Ft. Worth produces an annual fuel savings amounting to \$10/m^2. Adding a large yearly storage tank increases the return per square meter to \$14 annually. The additional storage will increase the initial cost of the system. Adding more collector area does not sufficiently increase the fuel savings to increase the per area return. The fuel savings per area in Miami is \$15/m^2 at 50 m^2 and no storage. Again additional area decreases the return. Typical solar collectors cost about \$250/m^2. Even with favorable parameters in a life cycle analysis, the cost of the collectors is more than an order of magnitude greater than the expected return. The benefits

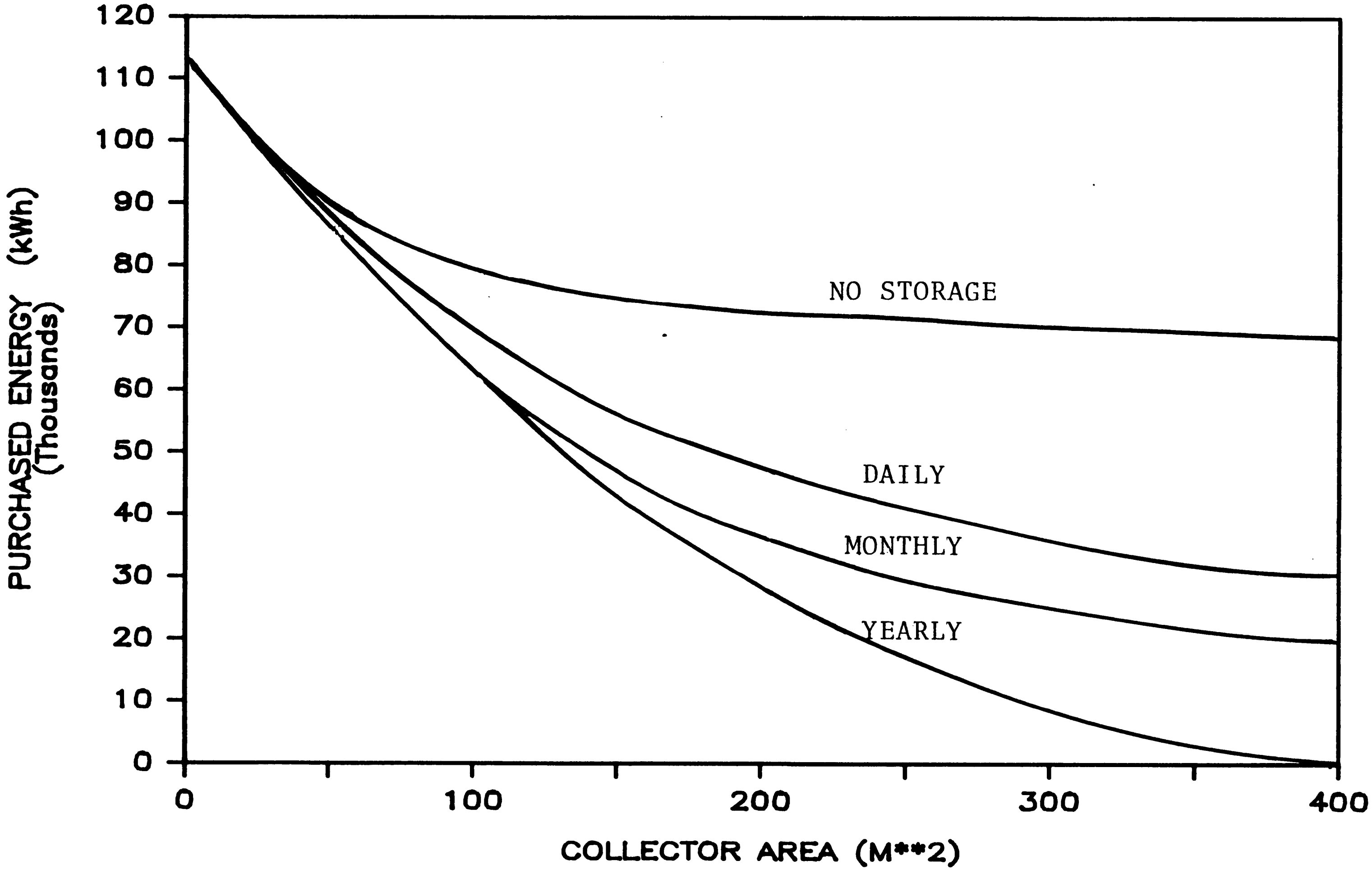


Figure 6.4.4 Auxiliary heat requirements for Miami at various storage sizes as a function of collector area

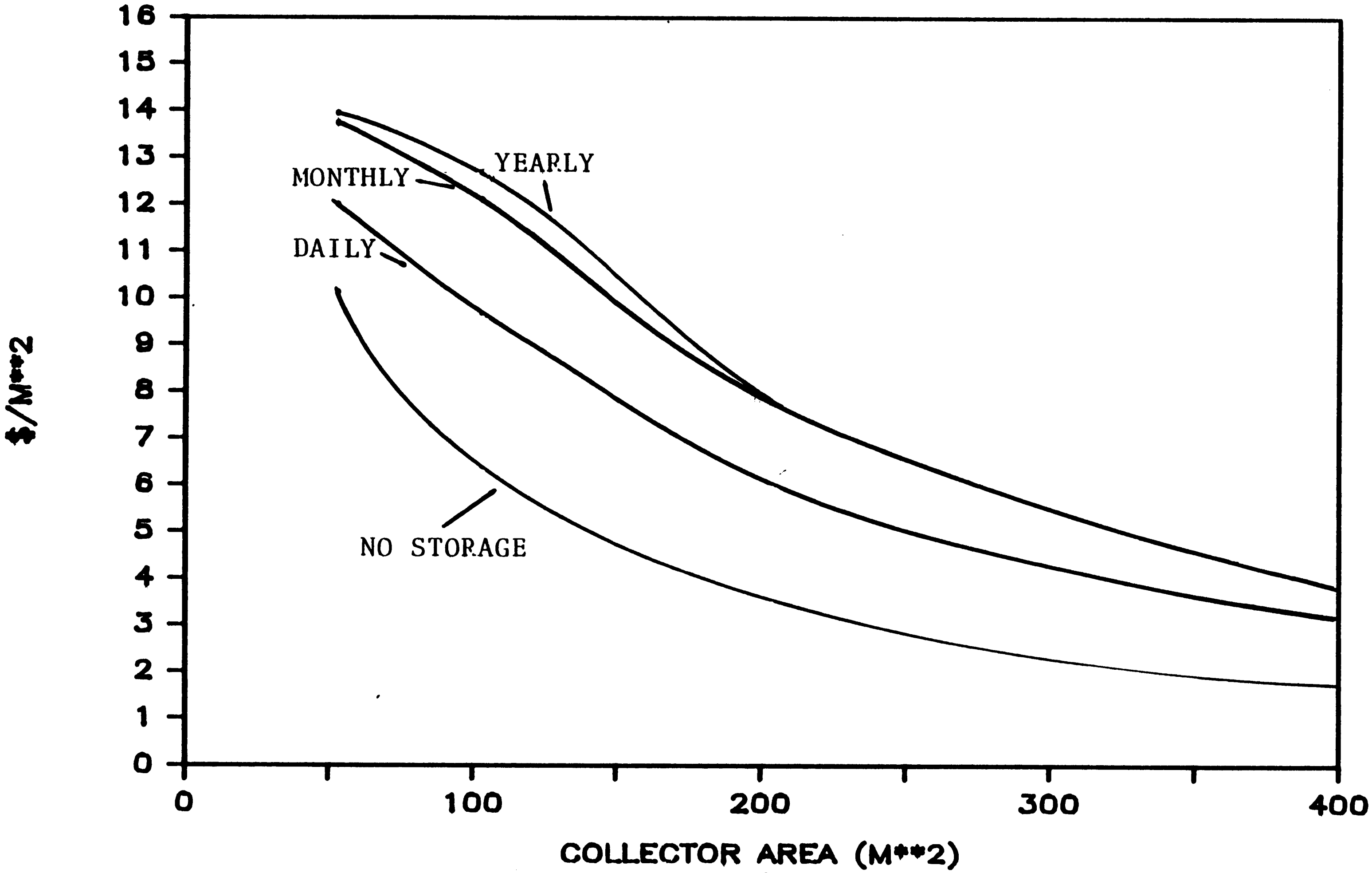


Figure 6.4.5 Annual energy savings per square meter of collector area in Ft. Worth as a function of collector area at different storage sizes

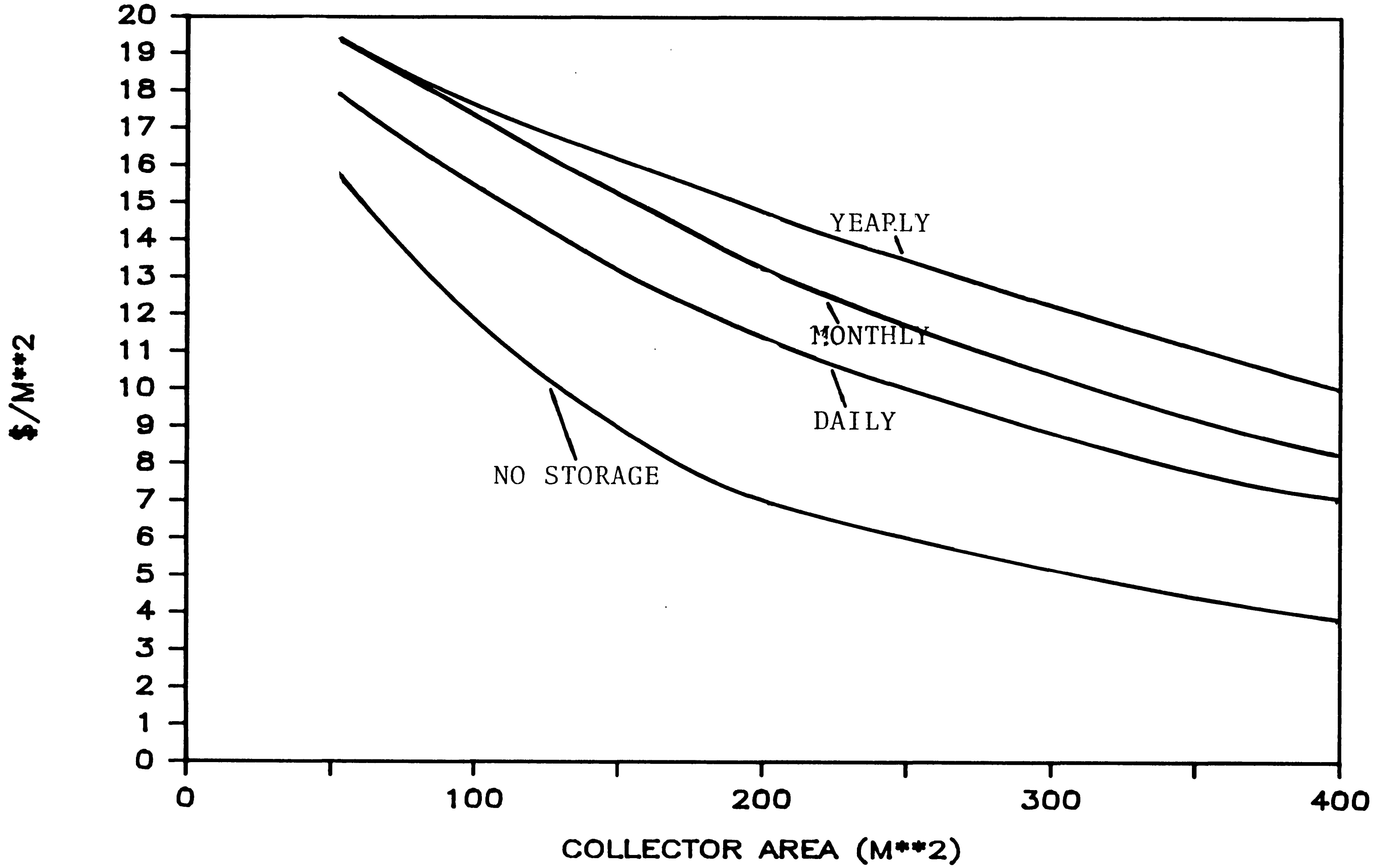


Figure 6.4.6 Annual energy savings per square meter of collector area in Miami as a function of collector area at different storage sizes

of using solar collectors in hybrid desiccant cycles for supermarkets do not justify their cost.

Section 6.5 Favorable Geographic Regions

Comparisons have been made between the standard vapor compression cycle and the ventilation/heat exchanger cycle for stores located in Ft. Worth, Miami, Madison, Washington D.C., and Phoenix. Figure 6.5.1 shows the monthly energy savings for each location, assuming an energy cost of 3.5 cents/weighted unit. As expected, a strong correlation exists between ambient humidity and energy savings. The hybrid system gains an advantage over the standard vapor compression system only when dehumidification is required. Miami, with year-round humidity, sees a large savings throughout the year. Phoenix, while also maintaining an extended cooling season, is sufficiently dry that the desiccant system has minimal impact throughout most of the year. Northern locations, like Madison and Washington only see sizeable savings during the humid summer months.

Figure 6.5.2 plots the monthly energy savings versus the monthly average humidity ratio. The monthly average humidity ratio is obtained from averaging hourly TMY values. Regardless of location, months with high average ambient humidity ratios produce the largest amount of savings. Regions with high humidity ratios for large

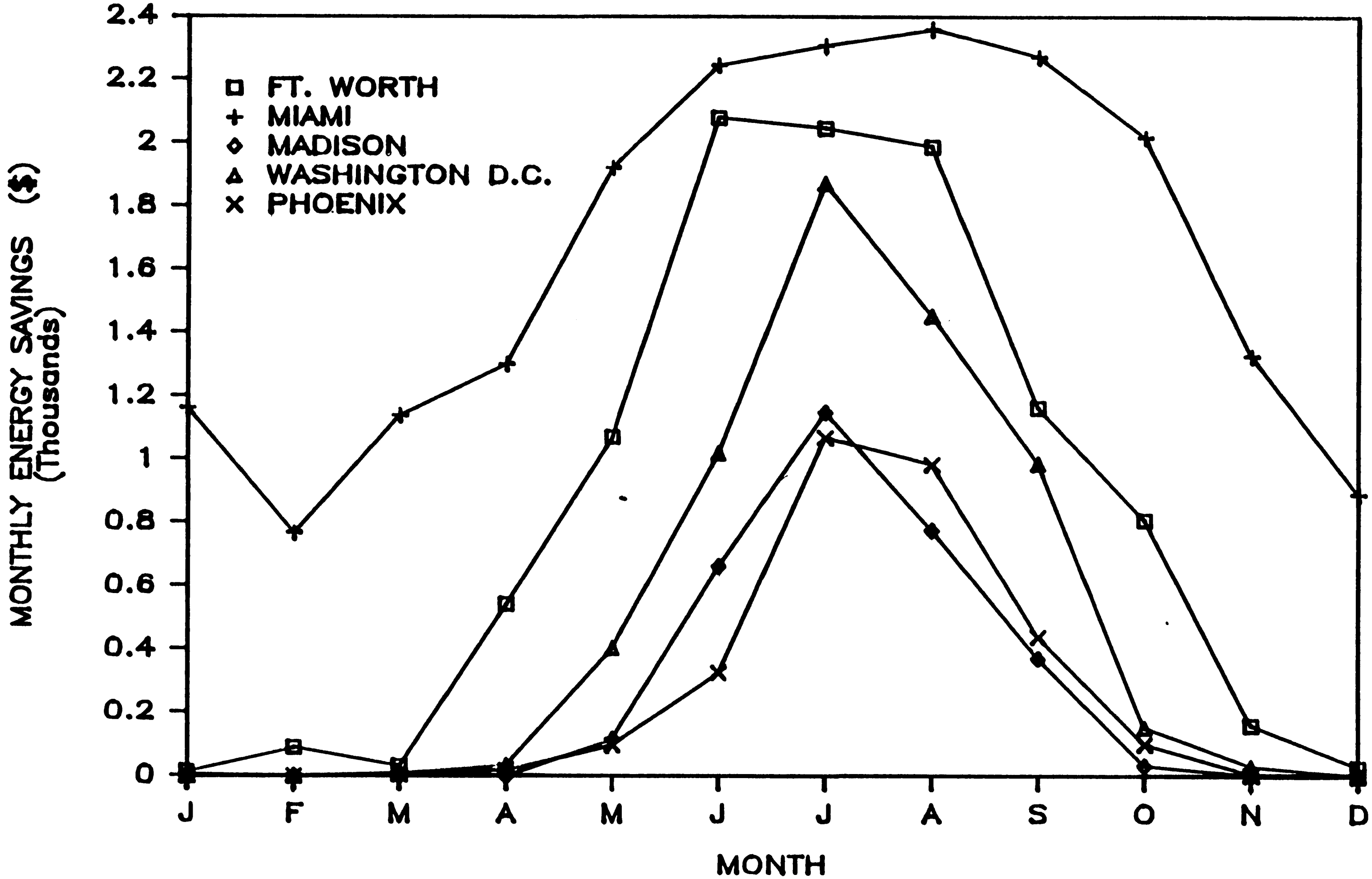


Figure 6.5.1 Monthly energy cost reduction of the ventilation/heat exchanger cycle over the standard vapor compression cycle for five U.S. locations

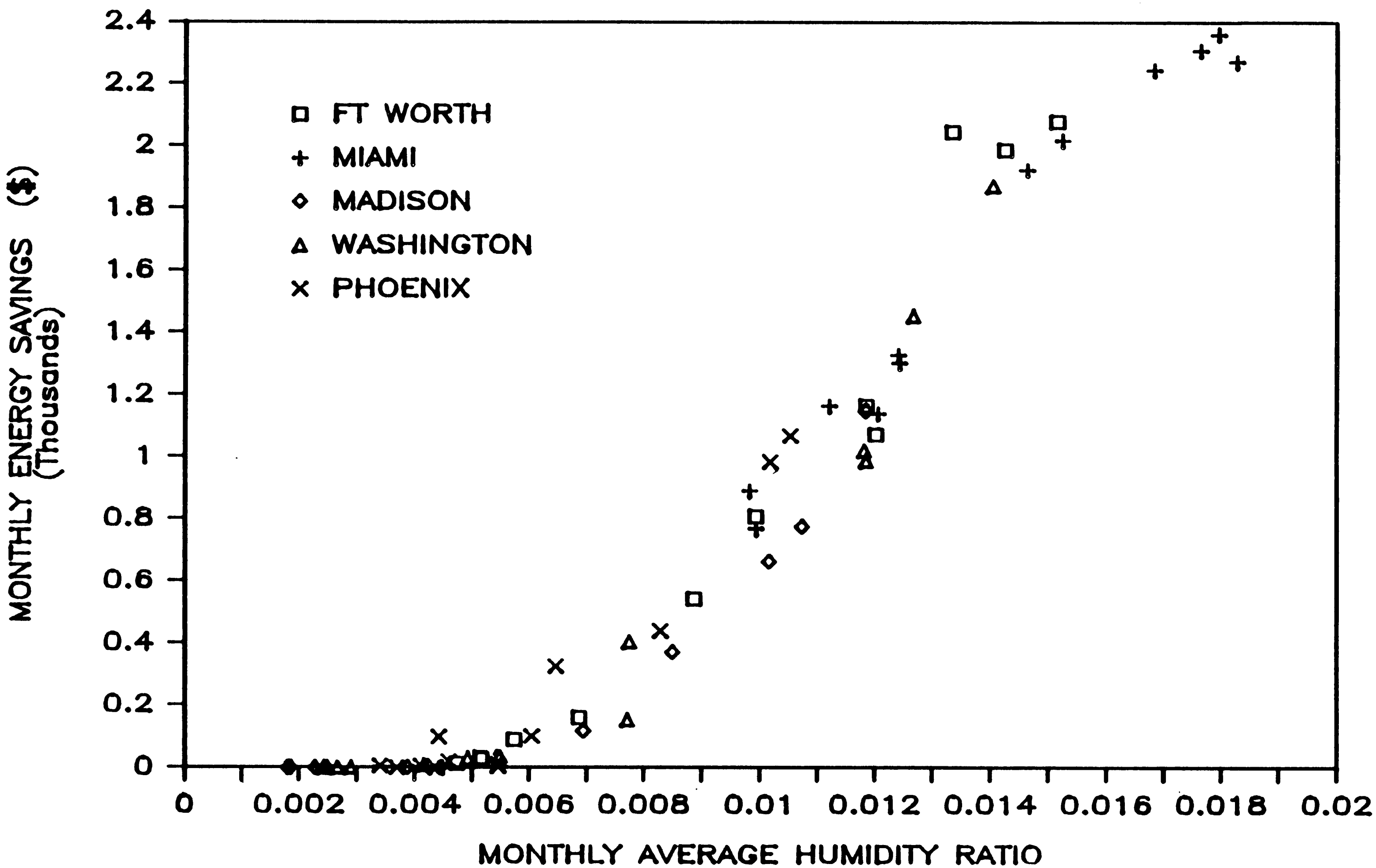


Figure 6.5.2 Monthly energy cost reduction as a function of the monthly average humidity ratio

portions of the year will receive the most benefit from these systems. Such conditions exist in the Southeastern United States and the Gulf Coast, making these ideal regions for the application of desiccant systems in supermarkets.

Section 6.6 System Economics

For hybrid desiccant cooling systems to attract the interest of supermarket executives they must provide economic advantages. The expected fuel savings must offset the cost of any additional equipment purchases. A simple payback period can be expressed as the incremental cost of additional equipment divided by the annual fuel savings of that additional equipment.

$$\text{Payback} = \text{Incremental Cost/Annual Fuel Savings} \qquad (6.6.1)$$

As the low margin basis of the supermarket industry discourages risk taking, it is expected that anticipated payback periods must be no more than two years to generate sufficient interest among the industry.

Desiccant cooling cycles contain much equipment not found in traditional cooling systems. The cost of purchasing the desiccant wheel, heat exchanger, auxiliary heater and other items needed to complete these systems can be substantial. However, since hybrid cycles reduce the load on the vapor compression machine the installed capacity requirement of these machines is significantly reduced. Table 6.6.1 lists the vapor compression capacity which meets the cooling needs in each cycle 95% of the time. Capacity reductions are in the neighborhood of 50 tons for all locations. Reductions of this magnitude can be expected for any location in which dehumidification is required. Using a desiccant system always removes the need to cool to the saturation line. Large capacity reductions can be expected even in relatively cool climates with short cooling seasons. The installed cost of a vapor compression machine is about \$750/ton (27). The increased cost of the desiccant equipment is at least partially offset by this reduction.

The reduced electrical energy needs of a hybrid system will decrease the peak demand charges the store will have to pay. Some utilities charge commercial customers for both the amount of energy consumed and for their peak capacity requirements. The monthly peak demand charge can be calculated as follows,

$$\text{Total demand cost} = \text{Peak kW usage} * \text{demand charge/kW} \qquad (6.6.2)$$

This calculation may be summed over each month to determine the annual demand costs. The magnitude of the demand charge varies considerably depending on location and time of year. To estimate the value of the demand reduction a typical price of \$7/kW (28) is used for each location. Since this is a recurring saving the demand

Table 6.6.1
Vapor Compression Capacity Requirements
(tons)

City	Std VC	Ventilation	Difference
Ft. Worth	62	12	50
Miami	67	11	56
Madison	54	7	47
Washington, D.C.	59	9	50
Phoenix	54	16	38

Table 6.6.2
Annual Fuel Savings

City	Energy Use Reduction ($0.035/Unit)	Demand Reduction (@ $7/kW)	Annual Savings
Ft. Worth	$10,034	$4,900	$14,934
Miami	$19,715	$5,504	$25,219
Madison	$ 3,129	$2,310	$ 5,439
Washington, D.C.	$ 5,964	$3,570	$ 9,534
Phoenix	$ 3,055	$2,287	$ 5,342

Table 6.6.3
Allowable Additional Cost to Meet Two Year Payback Period

City	Annual Fuel Savings	Payback Period x Annual Savings	Capacity Reduction (@ $750/ton)	Cost
Ft. Worth	$14,934	$29,868	$37,500	$67,368
Miami	$25,219	$50,438	$42,000	$92,438
Madison	$ 5,439	$10,878	$35,250	$46,128
Washington, D.C.	$ 9,534	$19,068	$37,500	$56,568
Phoenix	$ 5,342	$10,684	$28,500	$39,184

reduction can be added to the energy consumption reduction to determine the annual fuel savings. Table 6.6.2 shows the dollar value of the energy and peak demand reductions for the five locations presented here. Locations with long dehumidification periods receive the most benefit from hybrid cycles. Miami requires moisture removal 12 months a year and receives large reductions in both energy use and peak demand. Ft. Worth has large dehumidification needs for about half the year. Washington requires dehumidification for only a couple of months where Madison and Phoenix really only peak for one month. The energy use reductions reflect the length of time each location uses its air conditioning system for moisture removal. The spread in the demand reduction is not as great as for energy use as only one hour of dehumidification is needed to receive a large demand reducction benefit during a month. In cooler climates there can be many months when periods requiring moisture removal are very short. The energy cost reduction would be small in this case, however the demand reduction might be as large during a peak cooling month.

The amount which can be paid for the additional desiccant equipment and still achieve a set payback can be calculated as follows,

follows,

$$\text{Incremental dehumidifier cost} = \text{Payback period} * \text{Annual Fuel Savings} + \text{Reduction in VC cost} \quad (6.6.3)$$

The results of these calculations are given in Table 6.6.3. A supermarket in Ft. Worth could spend up to $67,368 on additional desiccant equipment and still receive a two year payback. The system would provide about $15,000 a year in energy savings. If the store is unwilling to reduce the size of its vapor compression machine in fear that the desiccant might fail, only the annual fuel savings can be used to offset the cost of additional desiccant equipment. If this is the case a two year payback would be achieved if the dehumidification equipment could be purchased for $29,868 or less. The reduction in vapor compression capacity provides a large fraction of the benefit in these systems, which suggests that even in locations with very short cooling seasons, like Madison, implementation of these systems could be beneficial. Reductions in vapor compression costs can conceivably offset additional equipment costs. The store will benefit from whatever annual fuel savings exists and from the improved environmental control provided by these systems.

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

The work reported in this thesis has used modeling techniques and computer simulation methods to evaluate the potential benefits of applying hybrid desiccant cooling systems in supermarkets. Some conclusions and recommendations drawn from this work follow.

Section 7.1 Conclusions

Due to the unique cooling situation produced by the presence of open refrigerated cases, the use of standard vapor compression cycles for air conditioning supermarkets is very inefficient. The high latent load ratios found in supermarkets provide an ideal application for hybrid desiccant cooling systems. Hybrid systems can reduce air conditioning energy costs between 50% and 70%.

Air conditioning energy costs however are not a large part of the supermarket's expenses. The refrigerated cases consume over 50% of the store's energy bill. Attempts to reduce the energy consumption of the refrigerated cases by maintaining lower store humidity levels with the desiccant system prove uneconomical. The increase in the auxiliary heat requirement necessary to maintain lower humidity levels more than offsets any reduction in refrigeration costs, regardless of the system configuration considered.

Various possible configurations of desiccant systems have been studied. The latent load in a supermarket, while being a large portion of the total load, is small enough so that dehumidification requirements can be met by dehumidifying only the ventilation air. In commercial buildings with larger loads this is not possible. Processing only ventilation air implies a substantial flow rate reduction through the dehumidification equipment, resulting in smaller equipment sizes and lower fan power requirements. For this reason, recirculation cycles probably cannot compete with ventilation cycles in this application.

In cycles utilizing condenser heat for regeneration, an energy trade-off exists between the free cooling available in the condenser. Often times obtaining the maximum amount of free coolings from the IEC does not result in the minimum energy cost of the system. The optimum may be at an intermediate heat exchanger effectiveness.

Annual simulations suggest that although consuming similar amounts of energy, the ventilation/heat exchanger cycle slightly out performs the ventilation/condenser cycle. In addition, the ventilation/heat exchanger cycle is a simpler configuration. The actual utilization of condenser heat could be very difficult.

Solar collectors with no storage capacity can reduce the auxiliary heat requirements of hybrid cycles by 50%. Additional auxiliary heat reductions occur with the use of storage. The amount of savings potential however does not appear to economically justify the additional cost of the solar collectors.

The required installed capacity of the vapor compression unit is decreased 80% with the use of hybrid systems. This results in reduced peak demand charges and lower initial costs. The reduction in initial cost offsets the additional cost of the desiccant equipment and may produce immediate paybacks.

A strong correlation exists between the ambient humidity ratio and the magnitude of the potential energy savings. Regions with extended periods of high humidity can expect to receive the largest reductions in energy costs. In the United States, areas with this characteristic include the Southeastern states and the Gulf Coast.

Section 7.2 Recommendations

This analysis has not evaluated the benefits of lower store humidity ratios in creating reduced defrost requirements and a better food storage environment. These effects are difficult to quantify; however some method of estimating them should be developed. The potential benefits might justify the increased fuel cost of maintaining lower store humidity levels.

The refrigerated cases produce a large amount of condenser heat which is currently used for reheating (summer) and heating (winter) needs. The heat might be used for regenerating the desiccant. MacDonald (29) has discussed this possibility. Further work as to the amount of heat available, at what temperature, and the effect on the performance of the refrigerated cases, could be incorporated in the models studied in this analysis.

Some of the cycles discussed in this work use condenser heat from the vapor compression unit for regeneration. Reclaiming condenser heat is not a standard practice and practical difficulties might develop in actual implementation. Some research should be undertaken exploring the feasibility of this idea.

The model of the vapor compression machine performance was developed from limited data taken for one particular unit and extended for use of any size machine. Further experimental data for COP's based on evaporating and condensing temperatures rather than inlet conditions would be very helpful in refining this model as would data reflecting the effect of air flow rate variations.

The simulation studies were performed assuming steady state performance and fixed store conditions. Loads were assumed met at every time step and store temperatures and humidities were not allowed to float. Simulation allowing for dynamic control and finite capacity of the air conditioning equipment would produce a more realistic idea of system performance. Allowing store conditions to drop below the maximum set points will decrease the cooling loads in periods of low ambient temperature and humidity ratio.

The cost of desiccant equipment should be quantified. Since only ventilation air needs to be processed, equipment sizes are small and should not be a prohibitive additional cost.

REFERENCES

1. "48th Annual Report of the Grocery Industry," <u>Progressive Grocer</u>, April (1981).

2. Guide to Energy Conservation for Grocery Stores, Federal Energy Administration FEA/D-76/096, January (1977).

3. Dunkle, R.V., "A Method of Solar Air Conditioning," <u>Mech. and Chem. Eng. Trans.</u>, Inst. Engns. Australia, MCI, 73, (1965).

4. Nelson, J.S., et. al., "Simulations of the Performance of Open Cycle Desiccant Systems Using Solar Energy," Solar Energy, <u>21</u>, 273 (1978).

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

6. Jurinak, J.J., Mitchell, J.W., and Beckman, W.A., "Open Cycle Desiccant Air Conditioning as an Alternative to Vapor Compression Cooling in Residential Applications," ASME Journal of Solar Engineering, August (1984).

7. Rousseau, J., "Development of a Solar Desiccant Dehumidifier: Phase II Technical Progress Report," AiResearch Manufacturing Company Report No. 81-17773, (1981).

8. Schlepp, D., "A High Performance Dehumidifier for Solar Desiccant Cooling Systems," SERF/TP-252-1979, (1983).

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

10. 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).

11. Sheridan, J.C., and Mitchell, J.W., "Hybrid Solar Desiccant Cooling Systems," <u>Proceedings of the 1982 Annual Meeting of AS/ISES</u>, (1982).

12. Howe, R.R., Beckman, W.A., and Mitchell, J.W., "Commercial Applications for Solar Hybrid Desiccant Systems," <u>Proceedings of the 1983 Annual Meeting of ASES</u>, (1983).

13. Cohen, B.M., et. al., "Field Development of a Desiccant-Based Space-Conditioning System for Supermarket Applications, Final Report" GRI Report No. GRI-84/0111, June (1984).

14. Banks, N.J., "Personal Communication," CargoCaire Engineering Corporation, Amesbury, MA, May (1984).

15. Toscano, William M., et. al., "Research Development of Highly Energy-Efficient Supermarket Refrigeration Systems" ORNL/SUB-88-61601/1, October (1981).

16. "Envelope of Operation: A Chart of Supermarket Ambient and Refrigeration Display Case Loads," Tyler Refrigeration Corporation, AC00444 Rev. 10/76 Special.

17. Klein, S.A., et. al., "TRNSYS - A Transient Simulation Program," University of Wisconsin-Madison, Engineering Experiment Station Report 38-12, Version 12.1 December (1983).

18. Van den Bulck, E., "Analysis of Solid Desiccant Rotary Dehumidifiers," M.S. thesis, University of Wisconsin-Madison, (1983).

19. MacLaine-Cross, I.L., "A Theory of Combined Heat and Mass Transfer in Regenerators," Ph.D. thesis, Department of Mechanical Engineering, Monash University, Australia, (1974).

20. Van den Bulck, E., Mitchell, J.W., and Klein, S.A., "The Design of Dehumidifiers for Use in Desiccant Cooling and Dehumidification System," ASME Paper 84-HT-32, ASME-AICHIE National Heat Transfer Conference, Niagara Falls, NY (1984).

21. Product Literature, "Model AB5530H Compressor Performance," Tecumseh Products Company, September (1981).

22. Cohen, B.M., et. al., "Field Development of a Desiccant-Based Space-Conditioning System for Supermarket Application, Annual Report" GRI Report No. TE 4308-42-83.

23. ASHRAE, <u>ASHRAE Handbook 1977 Fundamentals</u>, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, New York, (1977).

24. ASHRAE, <u>ASHRAE Handbook 1981 Fundamentals</u>, American Society of Heating, Refrigerating and Air-Conditioning Engineers, New York, (1981).

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

26. Directory of S.R.C.C. Certified Solar Collector Ratings, Solar Rating and Certification Corporation, Washington D.C., Fall (1983).

27. Means, R.S., "Means Mechanical Cost Data" Robert Snow Means Co., Kingston MA (1984).

28. "MGE Commercial Electric and Gas Service Rates," Madison Gas and Electric Brochure, September (1984).

29. MacDonald, N.J., "Utilization of Condenser Heat for Desiccant Dehumidifiers in Supermarket Applications" ASHRAE Transactions, V. 89, (1983).