THE PERFORMANCE OF CONVENTIONAL AND HUMID-CLIMATE VAPOR-COMPRESSION SUPERMARKET AIR-CONDITIONING SYSTEMS

by

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

MASTER OF SCIENCE
(Mechanical Engineering)

at the

UNIVERSITY OF WISCONSIN--MADISON

1988

ACKNOWLEDGEMENTS

Funding for this project was provided by the Electric Power Research Institute. Bob Parkes of Parkes Associates, Ken Bowlen of Cargocaire, Carl Laverrenz of Tyler, Keith Oliver and Bob Stevens of Kroger, David Knebel of Turbo, Peter Jenkins of Trane and Massoud Neshan of Hill supplied numerous difficult-to-find pieces of information. Several also reviewed model descriptions.

I would like to thank my advisors John Mitchell and Bill Beckman for their enthusiasm and careful guidance, even while on the other side of the Atlantic or fighting illness. Chuck Dorgan consistently knew the right value for the parameter I needed, Jack Duffie smoothed away many obstacles, and Sandy Klein taught me that "thermo" could indeed be fun.

My fellow students and the Solar Lab. staff created a comfortable, helpful environment and provided many much-needed diversions. The Jims deserve special thanks: Jim Braun for helping me out of numerous dead-ends, and Jim Kummer for making TRNSYS and the computers behave. A special thank you also to Byron Pritchard for writing the ice-storage component to be used in part two of this project.

Friends near and far shared with me cassette tapes, Thursday morning breakfasts, letters and phone calls from distant places, silence, sunny vacations, contra dances, meals and community; making my world wide and satisfying this past year and a half.

And finally, my love goes to my family. Regardless of the venture, they have always backed me.

Urban

ABSTRACT

Reducing supermarket energy costs is important both for the store owner and the general public. This study evaluates the energy consumption patterns of two types of vapor-compression air-conditioning systems (a conventional system and a humid-climate system) in a 30,000 ft^2 Miami, FL store.

Supermarket, refrigerated case, air-conditioning units, reheat and fan models were developed and implemented as TRNSYS computer simulation components. These components were used to build conventional and humid-climate air-conditioning system models, which were then used to perform annual simulations. The effects of varying climate, humidity setpoint, ventilation and circulation flow rates, refrigerated case capacity and internal sensible load were studied.

An annual air-conditioning energy consumption savings of 63% (37,100 kWh) was found with the humid-climate system in Miami. However, as a result of the smaller demand by the humid-climate system for reheat energy reclaimed from the refrigerated case condenser(s), the refrigerated case COP decreased. The increased refrigerated case energy consumption decreased the total annual energy consumption savings to 1% (9,700 kWh). This represents a savings on the order of $1,000. Given a payback period of 2 to 3 years, the cost of a humid-climate system cannot exceed the cost of a conventional system by more than $600 to $1,800. The energy requirements of both systems are sensitive to climate, humidity setpoint, circulation flow rate and refrigerated case capacity; but are not highly sensitive to ventilation flow rate or internal sensible load.

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NOMENCLATURE

BF	air conditioner coil bypass factor
C	capacity, either refrigeration or air conditioner
C_{equip}	initial cost of equipment
cdb	dry-bulb temperature of air entering air conditioner condenser
cfm	cubic feet per minute
COP	coefficient of performance
d	discount rate
ewb	wet-bulb temperature of air entering air conditioner evaporator
f_{BF_cfm}	air conditioner bypass modifying function for flow rate
f_{C_cfm}	air conditioner capacity modifying function for flow rate
f_{C_T}	air conditioner capacity modifying function for dry-bulb temperature
f_{med}	medium temperature refrigerated case credit modifying function
f_{low}	low temperature refrigerated case credit modifying function
f_{1/COP_cfm}	air conditioner 1/COP modifying function for flow rate
f_{1/COP_plr}	air conditioner 1/COP modifying function for partial-load-ratio
f_{1/COP_T}	air conditioner 1/COP modifying function for dry-bulb temperature
F_{pw}	present worth factor
h	refrigerated case height
i	general inflation rate
i_F	fuel inflation rate
L	refrigerated case linear length
$\dot{m}$	mass flow rate

N_e	number of years in economic analysis
plr	air conditioner partial load ratio
P_r	total power consumption of refrigerated cases
P_f	fan power
P_1	ratio of life cycle savings due to operating cost savings over the first year operating cost savings
P_2	ratio of life cycle cost due to equipment cost over the equipment cost
$Q_{l,c}$	latent refrigerated case credit
q_s	anti-sweat heater power
$Q_{s,c}$	sensible refrigerated case credit
R	thermal resistance
T	dry-bulb temperature
t_i	effective income tax rate
t_p	property tax rate
$S_{first\ year}$	first year operating cost savings
S_{LC}	life cycle savings
β	latent-to-total refrigerated case credit ratio
ΔP	static pressure drop
γ	refrigerated case on-time fraction
ρ	density
ω	absolute humidity

Subscripts:

actual	indicates actual value
air	pertaining to store air
c	pertaining to refrigerated case condenser(s)
case	pertaining to refrigerated case(s)
design	indicates design value
dp	dew point
e	pertaining to refrigerated case evaporator(s)
exit	pertaining to exiting air
i	pertaining to i^{th} component
in	pertaining to incoming air
low	pertaining to low temperature refrigerated cases
med	pertaining to medium temperature refrigerated cases
rated	indicates value at ARI rating conditions
ref	reference value
surf	pertaining to air state at surface of air conditioner evaporator coil
store	pertaining to store air state

Superscripts:

''	indicates per area value
+	indicates value used only if positive, otherwise replaced with 0

CHAPTER 1: INTRODUCTION

Approximately 4% of the annual U.S. electricity consumption occurs in supermarkets (FMI, 1979). The cost of energy in a supermarket can be equal to the store profit (Cargocaire, 1985, np) and, until recently, the cost of energy was one of the most pressing problems of supermarket owners and managers (Progressive Grocer, 1985, 38). For these reasons, this study and others have attempted to develop methods of reducing supermarket energy costs. The emphasis in this study was on reducing supermarket air-conditioning costs.

1.1: Scope of Study

Due to the presence of the refrigerator and freezer display cases, supermarket air-conditioning systems present design problems not found with other commercial building air-conditioning systems. The display cases perform part of the necessary air-conditioning, making the remaining cooling load smaller than that of a comparably-sized non-supermarket building. The cases remove proportionately more of the sensible load than the latent load, resulting in an air conditioner load with a higher than usual latent-to-total load ratio. Additionally, the display cases require the careful control of store humidity levels. Excess moisture causes increased refrigeration energy consumption and the deterioration of the appearance of frozen food packages (Cargocaire, 1985, np).

Supermarket air conditioning is most commonly done with conventional vapor-compression systems which process a mixture of fresh and return air. Vapor-compression air-conditioning systems which dehumidify only the in-coming outside air have been designed for humid-climates ("humid-climate systems"). Hybrid desiccant air-conditioning systems are being used in a few stores. These systems use a solid desiccant wheel to dehumidify, perform sensible cooling with a vapor-compression air-conditioning unit, and use reclaimed heat or gas heat to regenerate the desiccant wheel. Finally, ice-storage systems have been proposed for supermarket air-conditioning. While these systems would not reduce energy consumption they would allow energy use to be shifted to off-peak hours. Given an electricity rate schedule that favors off-peak electricity use, energy costs could be reduced.

This study compares conventional and humid-climate vapor-compression supermarket air-conditioning systems for a representative store in Miami, Florida. Computer models of the store and the two systems were created with the TRNSYS simulation program (Klein et. al., 1983). Using Typical Meteorological Year (Hall et. al., 1978) weather data for Miami, annual simulations for both systems were compared. Annual simulations were also made to determine the effects of varying climate, store humidity setpoint, ventilation and circulation flow rates, refrigerated case capacity and internal sensible load. Sample energy rate schedules were used to compare the operation costs of both kinds of systems in both Miami and in Madison, WI. A simplified economic analysis was then performed to determine how much higher the first cost of a humid-climate system could be assuming a payback period of 2 to 3 years.

1.2: Literature Review

At least two studies compared conventional supermarket air-conditioning systems with hybrid desiccant systems, using computer simulations either in whole or in part. In addition to literature published by manufacturers of humid-climate systems, at least one professional paper explains the benefits of such systems.

In June 1984, the Gas Research Institute (GRI) published the final report of its two-year comparison of conventional and hybrid desiccant systems (GRI, 1984). This study included both the experimental comparison of the systems installed in two supermarkets and the analysis of system computer simulations.

The field study confirmed that reduced store humidity levels decreased the refrigeration energy consumption. It also showed that hybrid desiccant systems allow the use of circulation flow-rates lower than those required by conventional vapor-compression systems. This can result in a considerable savings in fan operating costs. Due to numerous technical difficulties, the field tests did not demonstrate hybrid desiccant system operating cost savings in both stores. A number of areas for improvement were suggested, however.

Computer modelling in the GRI study showed that hybrid desiccant systems were most energy efficient at conventional store humidity levels in spite of reduced refrigeration energy consumption at lower humidities and are most promising in areas of moderate and high humidity. For a modern Miami, FL store with a store humidity setpoint of 0.009 $lbm_{water}/lbm_{dry\ air}$, an annual savings of $2400 to $4900 was predicted.

Richard Burns, working with the Solar Energy Lab. of the University of Wisconsin, wrote his M.S. thesis on "An Analysis of Hybrid Desiccant Cooling Systems in Supermarket Applications" (Burns, 1985). Burns' work relied exclusively on computer modelling and was based, in part, on the supermarket and refrigerated case models developed for the GRI study. He looked at a number of different hybrid desiccant system configurations and predicted a savings in air-conditioning costs of 50-70% (relative to a conventional vapor-compression system). He also found that, given a hybrid desiccant system, the air-conditioning energy consumption required to maintain lower store humidities exceeds the resulting reduction in refrigeration energy consumption. Burns found the greatest potential savings in regions with extended periods of high humidity, such as the southeastern states and the Gulf Coast.

A paper by E.R. Whitehead (Whitehead, 1985) describes the design and benefits of humid-climate air-conditioning systems. An experimental study detailed in the paper showed that a 39,000 ft^2 store with a humid-climate system used 83% of the energy used by a store of similar size equipped with a conventional air-conditioning system. Both stores were in Texas and the total annual store electricity consumptions were compared.

CHAPTER 2: SUPERMARKET AIR-CONDITIONING SYSTEMS

Two kinds of vapor-compression supermarket air-conditioning systems are considered in this study. This chapter describes both of these systems, and then describes the structure of the general vapor-compression air-conditioning system model used for annual computer simulations of both types of systems.

2.1: Conventional Vapor-Compression System

Most supermarket air conditioning is done with conventional vapor-compression systems processing all of the circulation air (a mixture of return and make-up air). Figure 2.1.1 diagrams such a system. Make-up air enters the system, is mixed with return air, and is cooled and dehumidified by the air-conditioning coil. In order to control store humidity, the processed air must be cooled below the saturation temperature. Under many conditions, the sensible cooling resulting from adequately dehumidifying exceeds the space sensible load. To maintain a comfortable store dry-bulb temperature, the air must be reheated. This is most often done with reclaimed heat from the air conditioner or refrigerated cases condensers. A supplementary heater may also be necessary in some cases if the reclaimed heat is not sufficient for both reheating and for winter heating. Processed air is often supplied at the front of the store, meeting the heavy load in this area due to checkout equipment, higher customer/employee density and infiltration through the entrance doors. The recommended place at which to remove the return air is underneath the freezer and refrigerated display cases (to be referred to collectively as "refrigerated cases"). This minimizes uncomfortably cold aisle temperatures near the cases. The resulting return air is cooler and drier than the

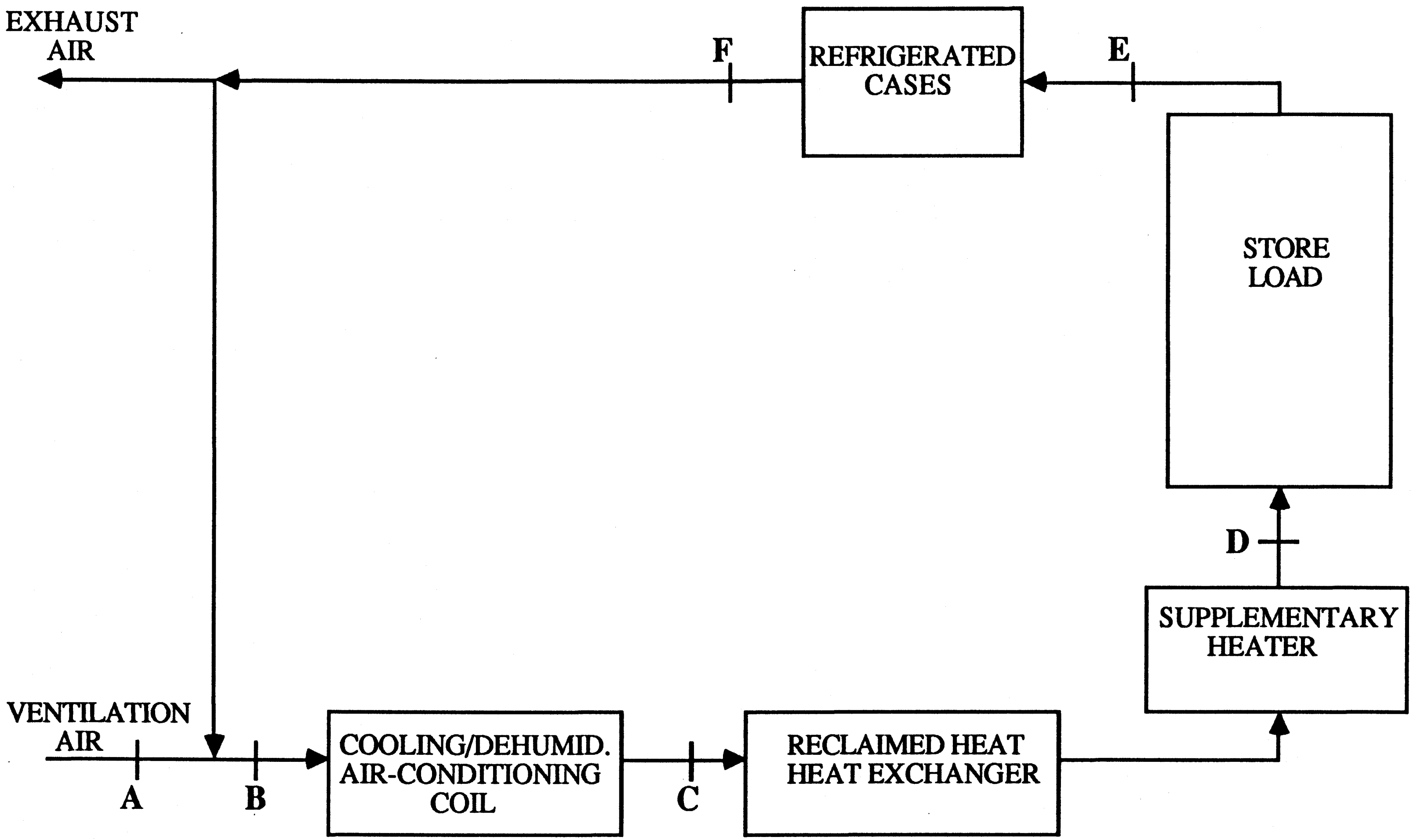

Fig. 2.1.1: Conventional Air-Conditioning System

store air. Because ventilation requirements demand that a given amount of outside air be continuously added to the system, an equal amount of return air must be exhausted.

Air states for a conventional air-conditioning system, given Miami design conditions, are shown on the psychrometric diagram of Fig. 2.1.2. Air state labels correspond to position labels in Fig. 2.1.1. Although the enthalpy difference per pound-mass of dry air across the air conditioner evaporator coil is small (approx. 6.2 Btu/lbm), the large amount of air processed (18,000 cfm or 78,840 lbm/hr) makes the total enthalpy change large (490,000 Btu/hr). At off-design conditions, it is often necessary to over-cool the air to control humidity. If adequate reclaimed heat is not available for reheating the over-cooled air, the system will be more costly to operate.

2.2: Humid-Climate Vapor Compression System

A different kind of vapor-compression air-conditioning system, sometimes known as a humid climate system, is shown in Fig. 2.2.1. The operation of this system is much like that of the conventional systems except that the return air is handled differently. Ventilation air only is cooled and dehumidified by a primary air conditioning unit. This processed air is then mixed with the recirculated return air. Because it handles only the ventilation air, the humid-climate primary air conditioner has a smaller capacity than the conventional system air-conditioning unit.

Many humid-climate systems have secondary air-conditioning units to sensibly cool recirculated return air. This allows the sensible load to be met even when the primary air conditioner can not do all of the necessary sensible cooling. It also allows the sensible and latent loads to be more efficiently met: the primary air conditioner unit provides only enough cooling to meet the latent load while the secondary unit, which has a higher evaporator temperature and is therefore more efficient, meets the remaining

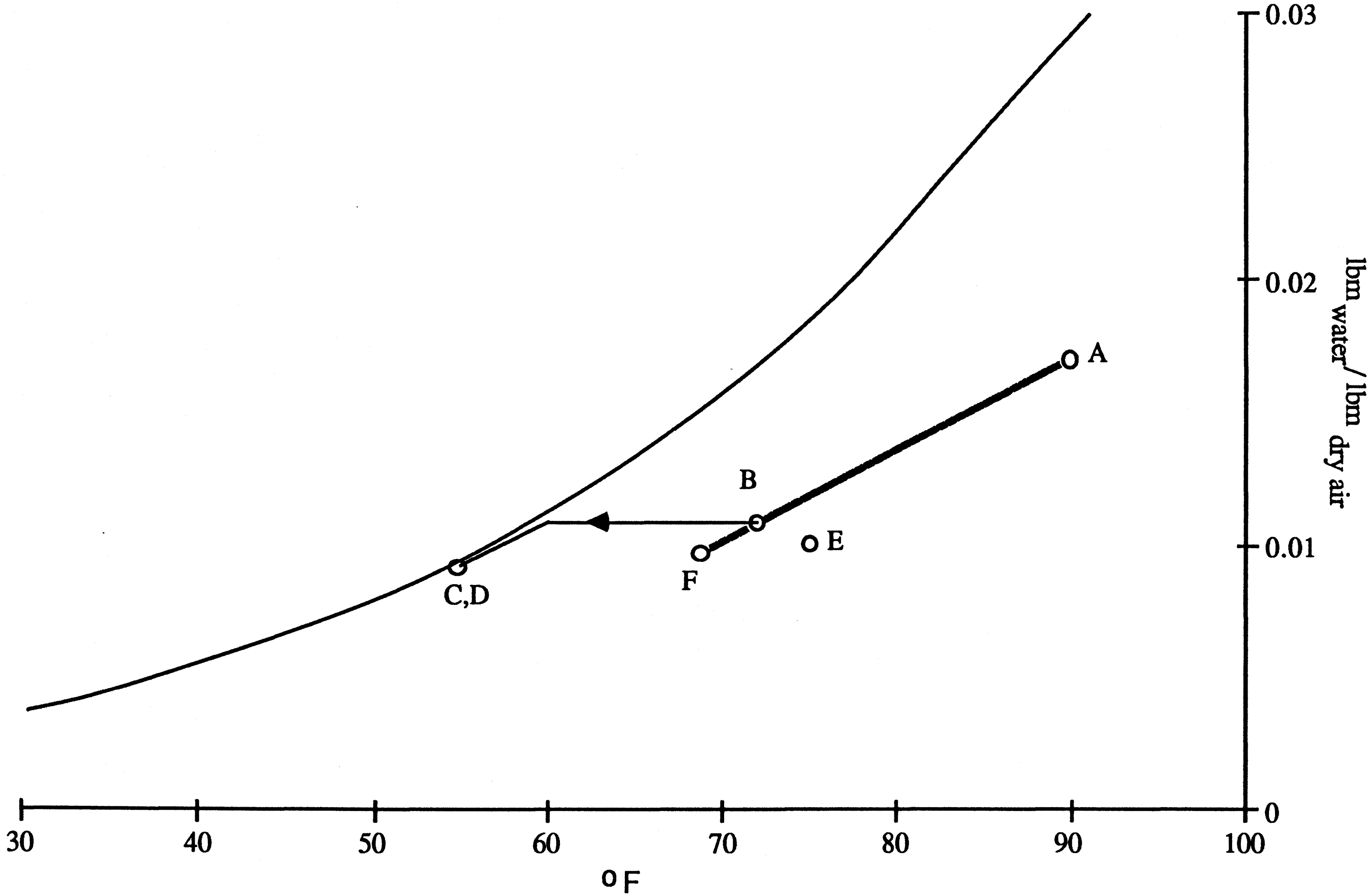

Fig. 2.1.2: Psychrometric Diagram for Conventional System

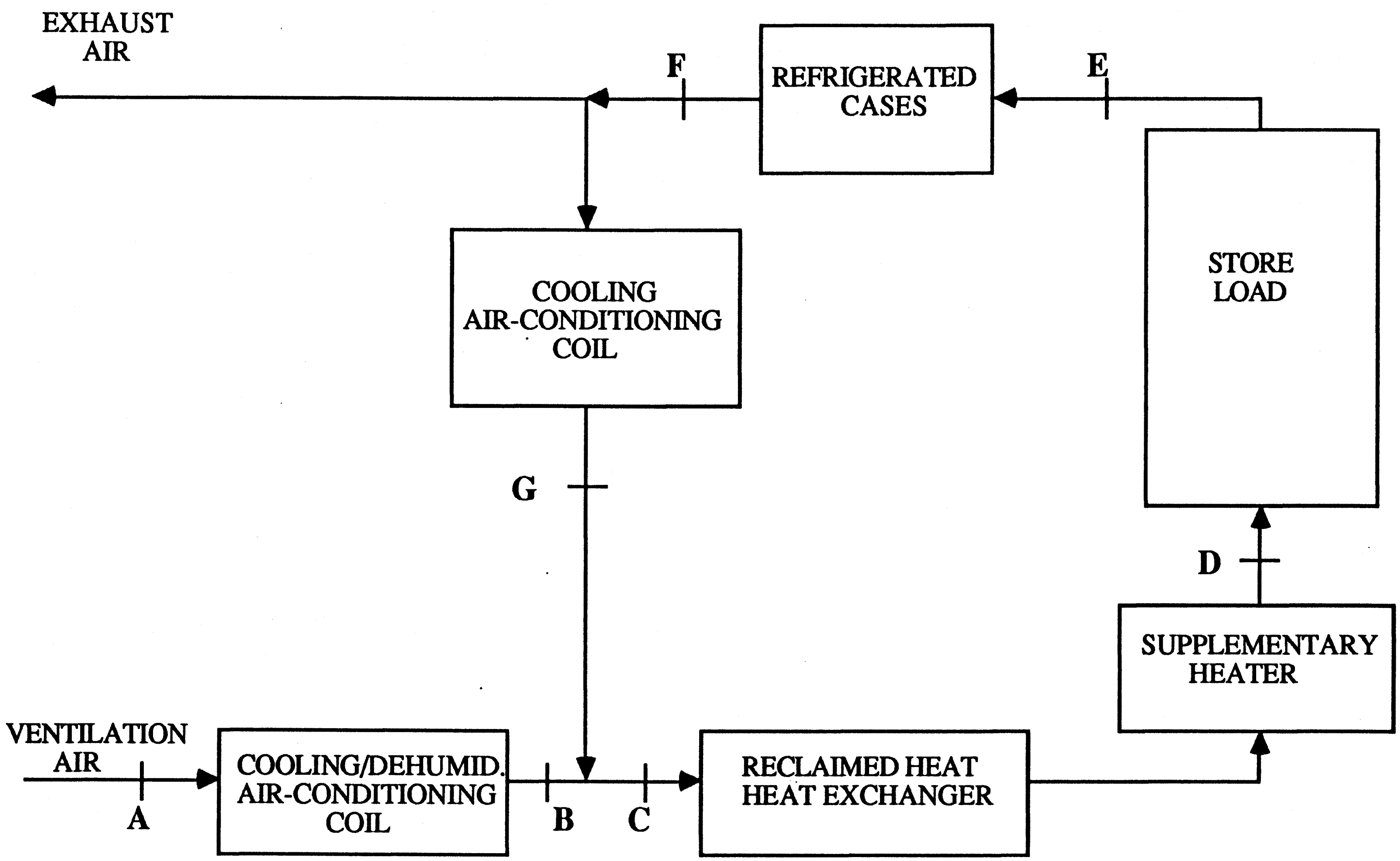

Fig. 2.2.1: Humid-Climate Air-Conditioning System

sensible load. If make-up air intake is substantially reduced at night, latent loads are high, or installed refrigeration capacity is low, it may be necessary to route some of the return air through the primary air conditioner to provide adequate humidity control.

Figure 2.2.2 shows the air states for a humid-climate system at Miami design conditions on a psychrometric diagram. Again, the air state labels of Fig. 2.2.2 correspond to the position labels of Fig. 2.2.1.

The enthalpy difference per pound of dry air across the humid-climate system primary evaporator coil is much larger (approx. 24.2 Btu/lbm) than that for the conventional system. However, the amount of air processed is considerably less (2700 cfm or 11,826 lbm/hr) giving a much smaller overall enthalpy difference (286,000 Btu/hr). The enthalpy difference per pound of dry air across the secondary coil is approximately 3 Btu/lbm. With a 15,300 cfm or 67,014 lbm/hr flow rate, this is a total enthalpy difference of 200,000 Btu. The combined total enthalpy difference across the two coils is equal to that of the conventional system. At off-design conditions, when the conventional system must overcool and then reheat all of the circulation air, the humid-climate system only overcools the ventilation air. The humid-climate system does much less total cooling and is therefore more efficient.

2.3: General Vapor-Compression System Model

One of the primary purposes of this study was to compare the humid-climate air-conditioning system to the conventional air-conditioning system. It was important not to confound the effects of the different methods of handling and processing air with the effects of different air-conditioning equipment. For this reason, a general vapor-compression air-conditioning system model was developed (Fig. 2.3.1). By properly setting the diverters, this model can emulate either a conventional system or a

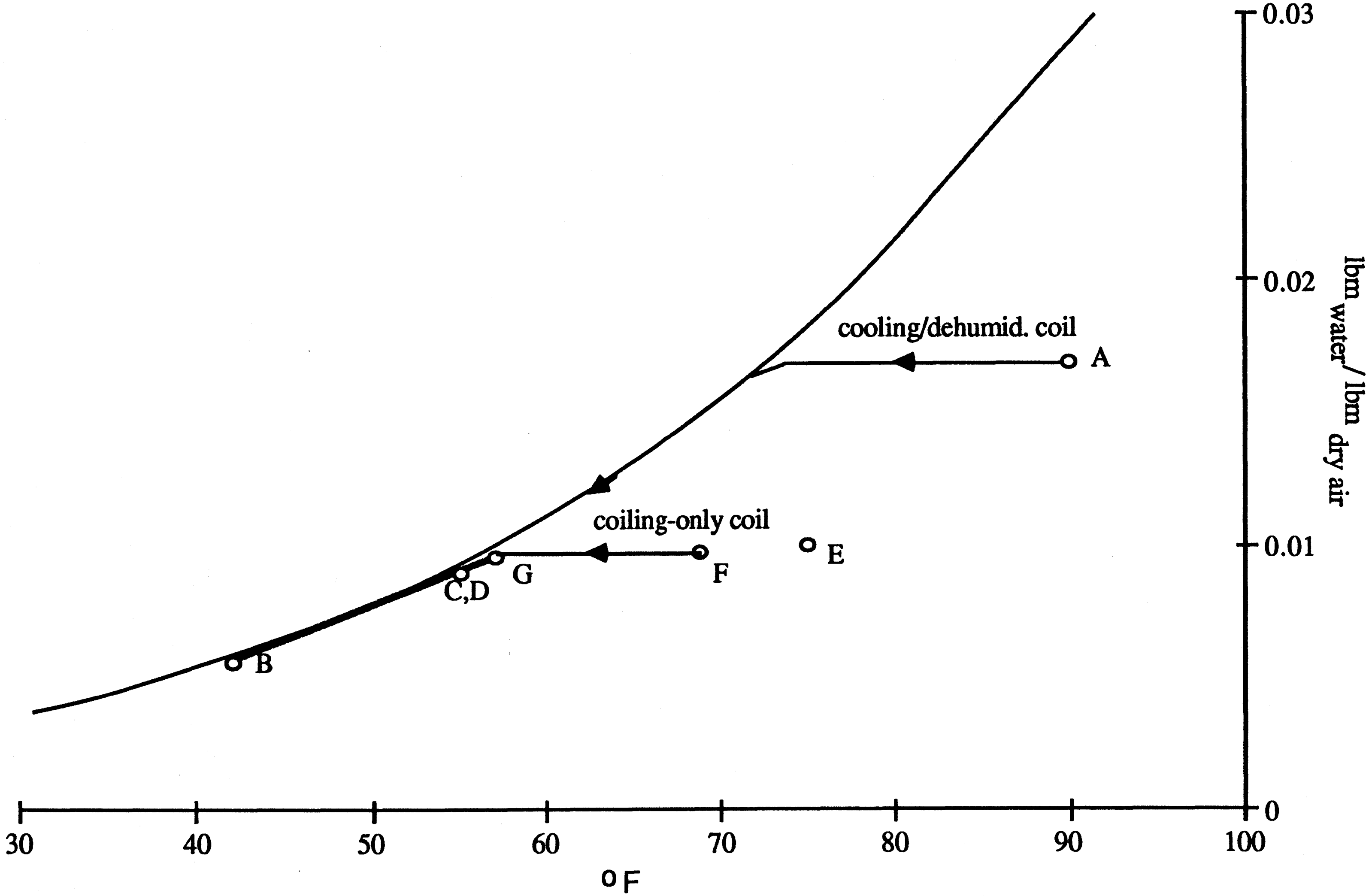

Fig. 2.2.2: Psychrometric Diagram for Humid-Climate System

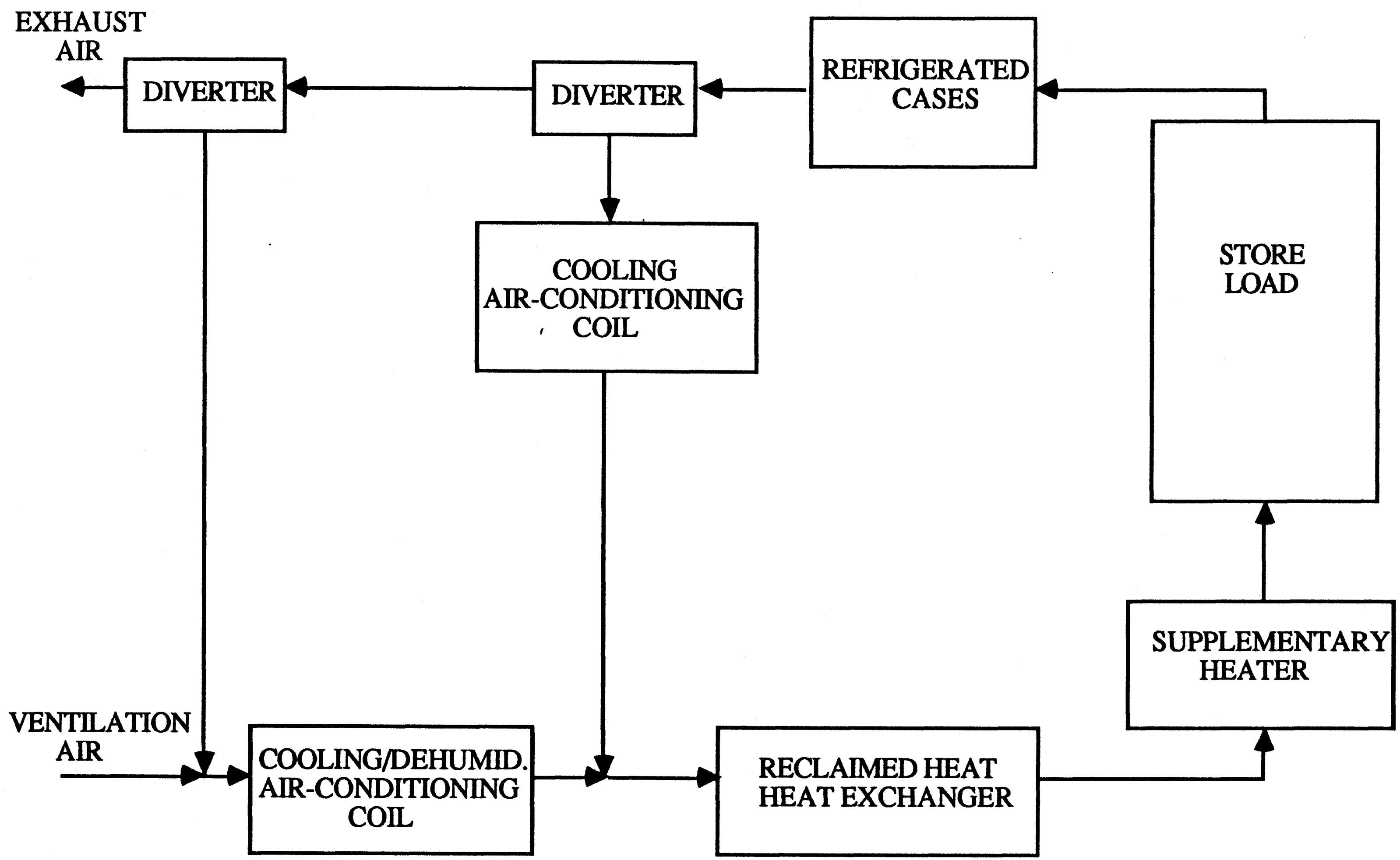

Fig. 2.3.1: General Vapor-Compression Air-Conditioning Model

humid-climate system. The air-conditioning coils are sized according to the design conditions and air-flow rates for each individual system, however the rated coefficients of performance and by-pass factors have been selected to be identical for all air-conditioning coils (see section 3.3). In practice, it is possible that the rated coefficients of performance (COP's) will not be identical, as the flow rates across the coils are quite different, and hence the temperature differences between the entering air and the evaporator may be different. The assumption that the COP's are identical implies that the coils are equally "well-designed" for the system-specific operating conditions.

A number of non-essential features of vapor-compression air-conditioning systems have not been modelled. Refrigerator fluid sub-cooling has not been modeled as it would improve both the performance of the conventional system and that of the humid-climate system. The option of an economizer cycle for the conventional system (i.e. taking all circulation air from the outside when profitable) has not been included. It is expected that such an option's performance would be highly climate-dependent, however consideration should be given to future testing of this configuration. Variations on the humid-climate system have also not been included. As an example, some humid-climate-type systems do not have a secondary (sensible cooling-only) air-conditioning unit. These systems process a fraction of the return air, in addition to all of the outside air, through the primary air-conditioning coil if processing only the outside air is not sufficient to meet the load. It is expected that the performance of such a system would fall between the performance of the conventional system and that of the humid-climate system as modelled, and that the equipment cost would be close to that of a conventional system.

CHAPTER 3: SYSTEM COMPONENTS

The implementation of the general vapor-compression air-conditioning system model described in section 2.3 required the development of models for each of the system components. This chapter describes these models and their TRNSYS implementations.

3.1: Supermarket Building

The supermarket building model used in this study is representative of many large modern supermarkets. It was initially based on the store models used in the GRI (GRI, 1984) and Burns (Burns, 1985) studies, however it was extensively reviewed and revised by supermarket consultants. While it is important that the store model generate representative sensible and latent loads, it is not necessary, for the purposes of this study, to replicate a particular store or class of stores. Fig. 3.1.1 diagrams the energy and mass flows for this building. These quantities are explained below.

The supermarket is assumed to be open 24 hours a day, 7 days a week. The sales floor area is 30,000 ft^2 and is considered to be square. The conditioned air volume is 480,000 ft^3. The building walls have a U-value of 0.103 Btu/hr-ft^2-^oF while the roof has a U-value of 0.092 Btu/hr-ft^2-^oF. The store front faces south and is assumed to be 30% window. The lighting and equipment load is assumed to be 2 W/ft^2. Internal moisture (from vegetable spraying, floor washing, etc.) gain is taken to be 10 lbm/hr, exclusive of latent gain due to people. The circulation flow rate depends on the loads at design conditions but is constrained to be no less than 15 cfm/person (the ventilation requirement). The normal store setpoint temperature is 75^oF and the normal store absolute humidity setpoint is 0.01 $lbm_{water}/lbm_{dry\ air}$. For simulation

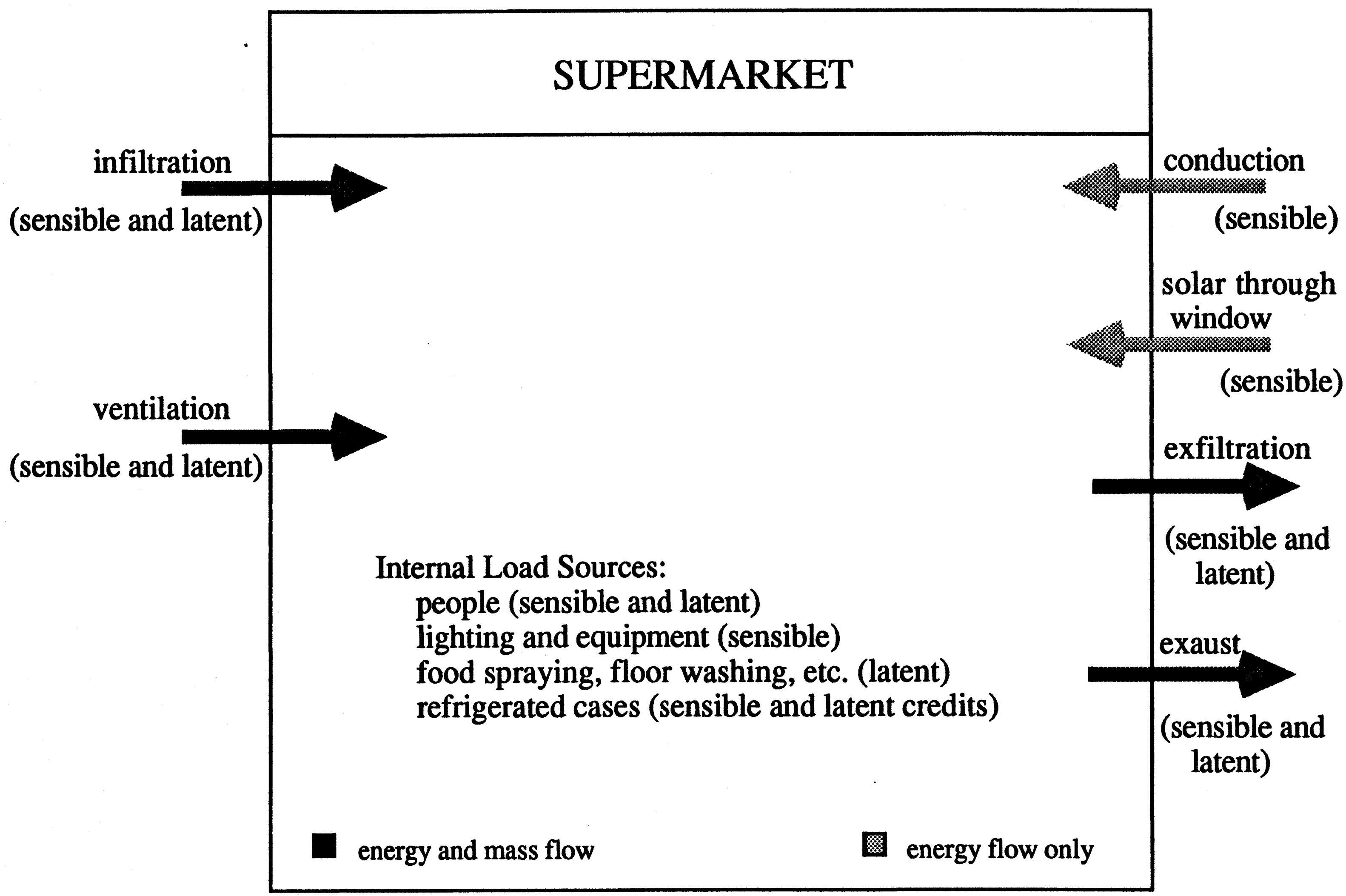

Fig. 3.1.1: Store Model Energy and Mass Flows

runs in which the humidity level is lowered, the temperature setpoint is adjusted to the equivalent ASHRAE comfort condition (ASHRAE, 1981, 8.21). Refrigerated cases are divided into two categories: medium-temperature cases and low-temperature cases. Closed-door and open cases are treated identically. The installed refrigerated case capacities and total linear lengths are: 42 tons medium-temperature capacity with 510 ft.; 15 tons low-temperature capacity with 300 ft. Table 3.1.1 summarizes the above information while Figures 3.1.2-3.1.5 describe the ventilation, infiltration, number-of-people, and lighting and equipment gain schedules.

This store model was implemented with a TRNSYS single zone component (Type 19) which uses the ASHRAE transfer function approach to model walls, roofs and ceilings. In addition to the store description given above, the use of this component required assuming the following values, which are summarized in Table 3.1.2: The store thermal capacitance is taken to be 7.8×10^5 Btu/^{o}F, based on the estimate of 130 lbm building material per ft^2 floor space for heavy construction (ASHRAE, 1985, 26.3). The moisture capacitance of the building contents is accounted for by multiplying the conditioned air volume by 20, giving an effective air volume of 9,600,000 ft^3. The reflectance of the inner building surfaces to solar radiation is assumed to be 0.5 while a value of 0.7 is used as the absorptance of solar radiation by the exterior surfaces. The convection coefficient for all inside surfaces, is taken to be 1.4556 Btu/hr-ft^2-^{o}F, the value used in determining ASHRAE transfer coefficients (ASHRAE, 1985, 26.11). The window diffuse solar radiation transmittance value used is 0.8 and the overall solar radiation transmittance is taken to be 0.87. The window U-value (not including inside or outside convection resistances) is estimated as 14.16 Btu/hr-ft^2-^{o}F based on 1/2 in. thick glass. The window dimensions are 10 ft x 83 ft, centered horizontally on the store front with the

Parameter	Value
Hours of operation	24 hours/day
Peak hours	7 am to 10 pm
Conditioned air volume	480,000 ft^3
Sales floor space	30,000 ft^2
Building walls U-value	0.103 Btu/hr-ft^2-F
(ASHRAE 8" concrete block, filled insulation:	A0,A1,C17,E1,E0)
Building roof U-value	0.093 Btu/hr-ft^2-F
(ASHRAE #17, steel sheet with 2" insulation)	
Storefront window U-value	0.917 Btu/hr-ft^2-F
Storefront window area (30% of south wall area)	831.4 ft^2
Circulation flow rate	approx. 0.6 cfm/ft^2
(based on design conditions with minimum of 15 cfm/person)	
Internal moisture generation	10 lbm/hr

Table 3.1.1: Store Model Description (continued on p. 18)

Parameter	Value
Temperature setpoints (extremes)	$75^{o}F$-$75.4^{o}F$
Humidity setpoints (extremes)	0.008-0.01 lbm_{water}/lbm_{air}
(ASHRAE equivalent comfort guide will be used)	
Installed refrigerated case capacity	
Medium temperature	42 Tons
Low temperature	15 Tons
Refrigerated cases linear length	
Medium temp.	510 ft
Low temp.	300 ft
Ventilation flow rate	scheduled (see Fig. 3.1.2)
Infiltration flow rate	scheduled (see Fig. 3.1.3)
Number of people in store	scheduled (see Fig. 3.1.4)
Internal sensible gain due to lighting and equipment	scheduled (see Fig. 3.1.5)

Table 3.1.1: Store Model Description (continued)

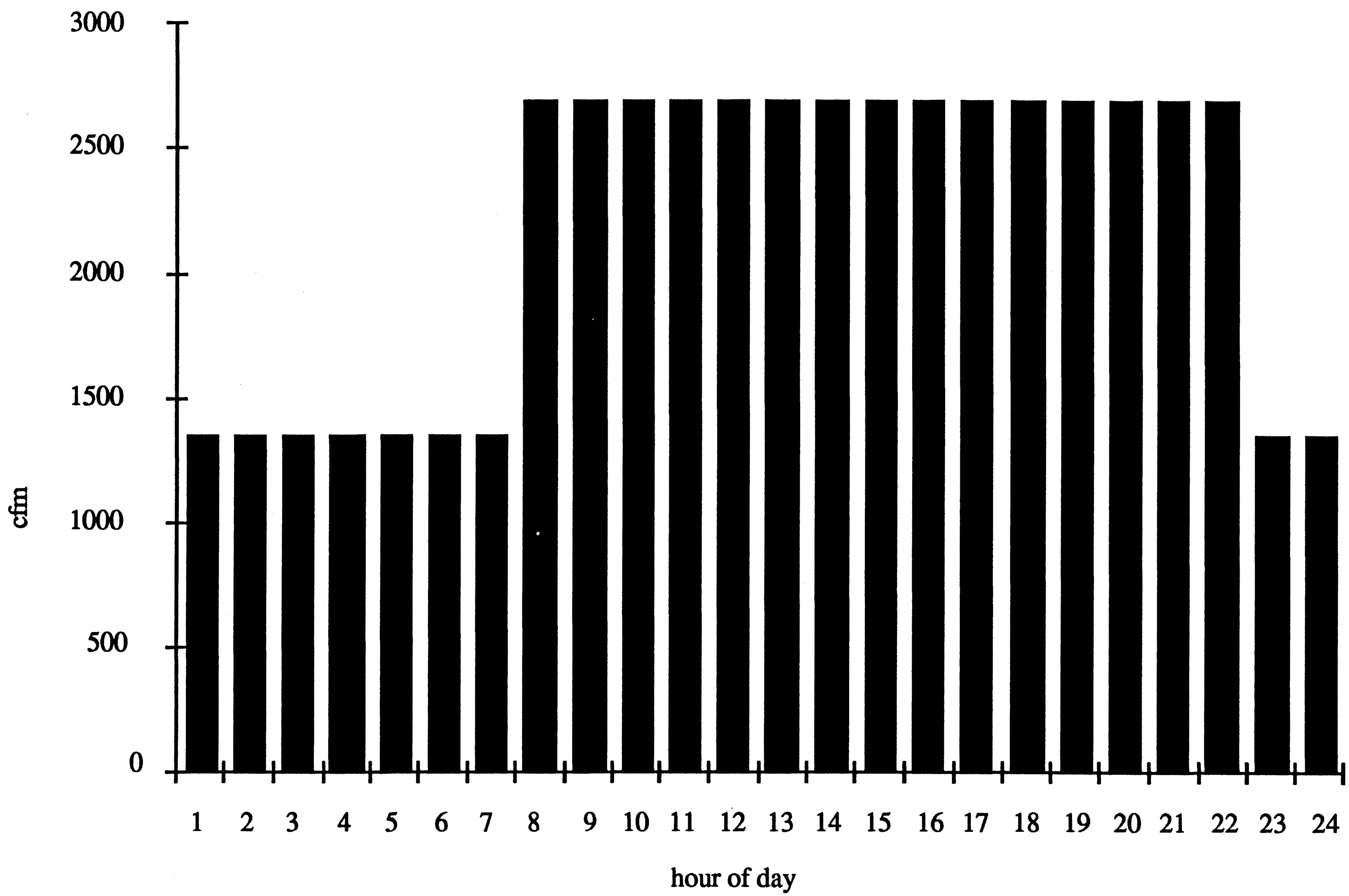

Fig. 3.1.2: Store Model Ventilation Flow Rate Schedule

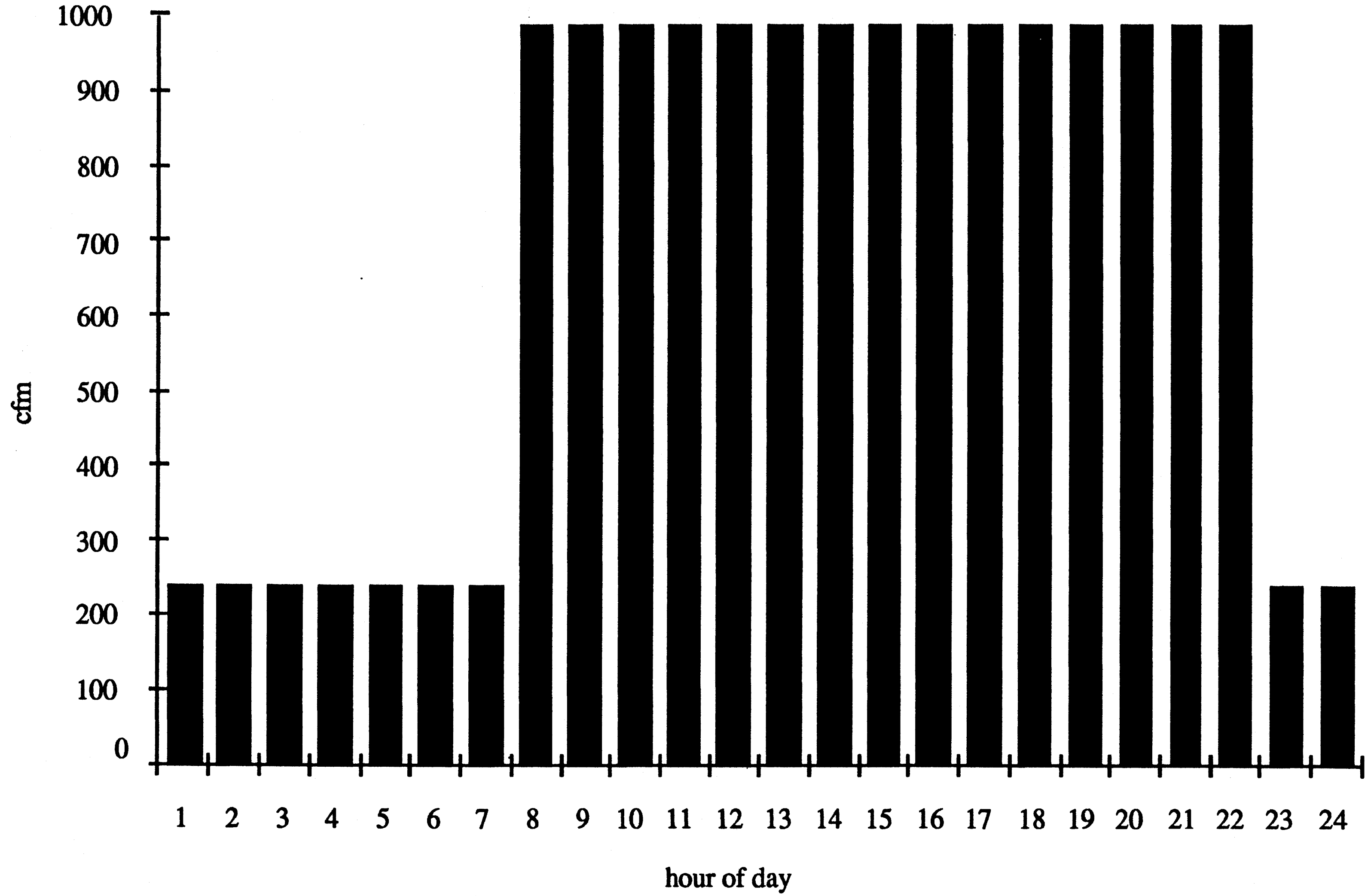

Fig. 3.1.3: Store Model Infiltration Flow Rate Schedule

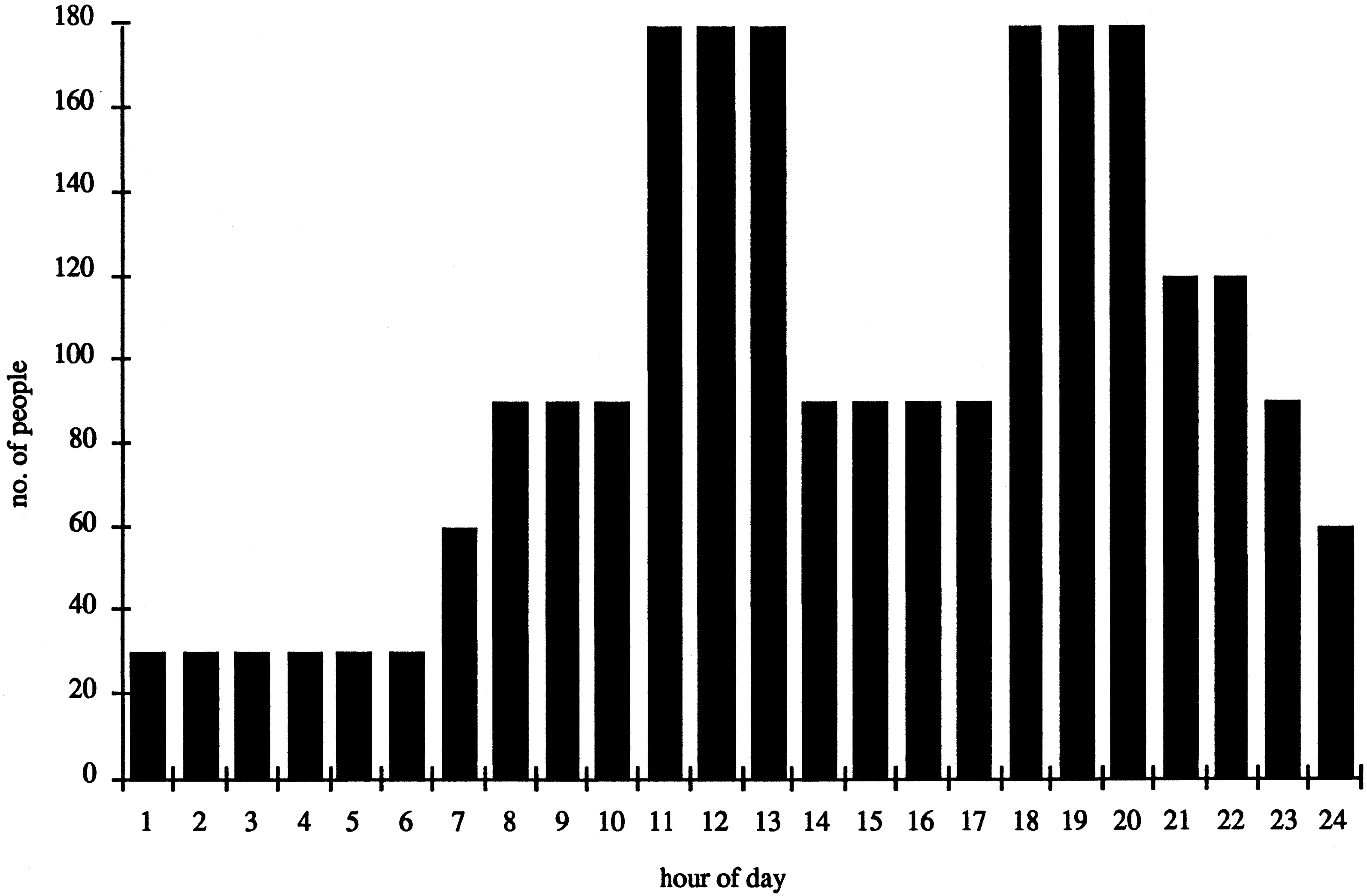

Fig. 3.1.4: Store Model Number-of-People-in-Store Schedule

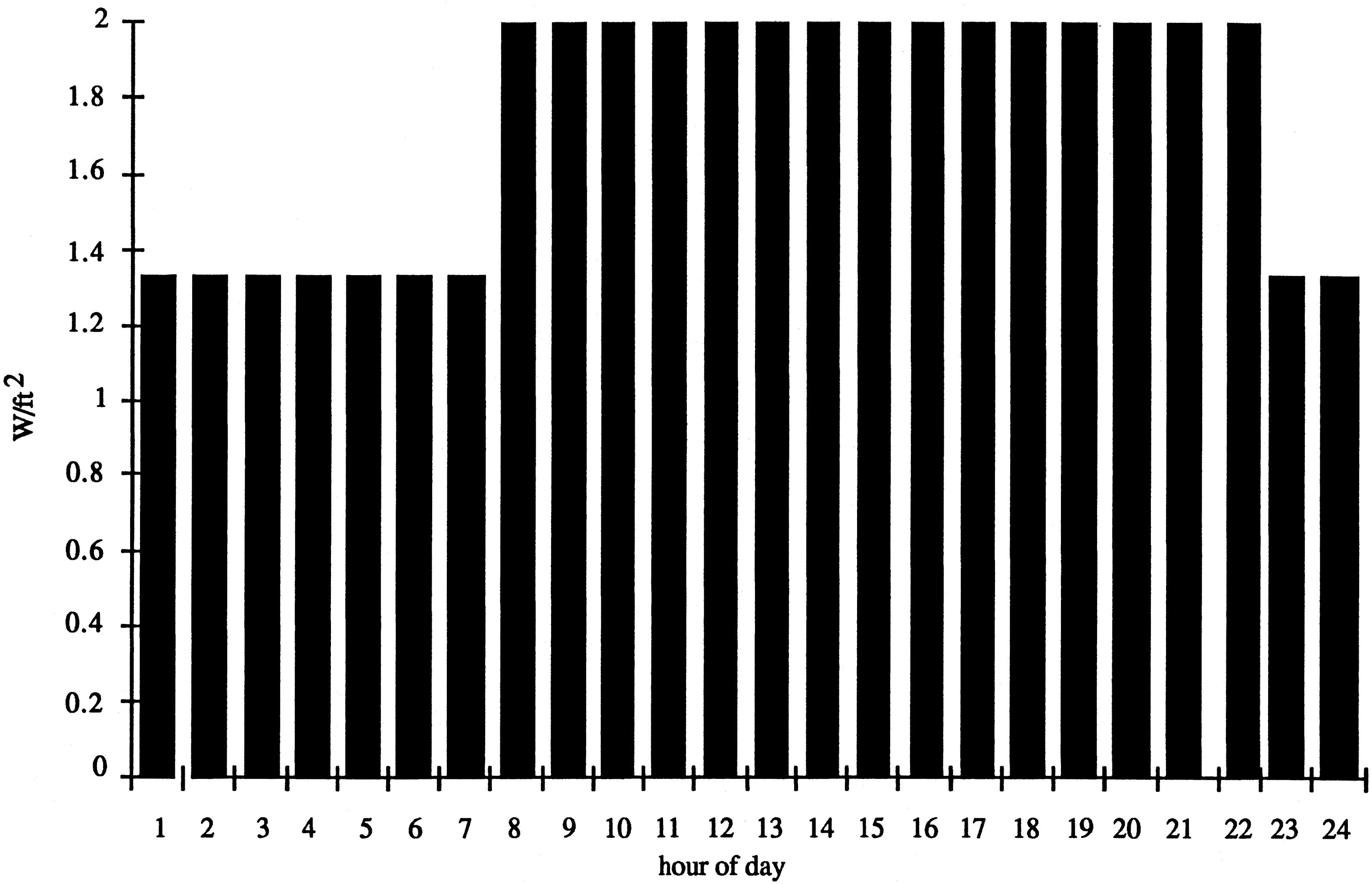

Fig. 3.1.5: Store Model Lighting and Equipment Gain Schedule

Parameter	Value
Store capacitance (ASHRAE heavy construction)	7.8×10^5 Btu/°F
Air volume used for moisture capacitance	9,600,000 ft^3
Walls and roof:	
Reflectance of solar radiation by inner surfaces	0.5
Absorptance of solar radiation by exterior surface	0.7
Inside convection coefficient	1.4556 Btu/hr-ft^2-°F
Window:	
Diffuse solar radiation transmittance	0.8
Overall solar radiation transmittance	0.87
U-value (not including inside or outside convection)	14.16 Btu/hr-ft^2-°F
Inside convection coefficient	1.4556 Btu/hr-ft^2-°F
Window dimension	10 ft high, 83 ft long
Window position	centered horizontally, up 3 ft from floor
Surfaces struck by beam radiation	floor

Table 3.1.2: Additional Store Model Parameters Required by TRNSYS Type 19 (continued on p. 24)

Parameter	Value
People gain:	
Sensible	315 Btu/hr-person
Latent	325 Btu/hr-person
Floor U-value	0.0238 Btu/hr-ft^2-^{o}F
ASHRAE 12" concrete slab with 12" insulation (A0, C11, B15, B15, E0)	

Table 3.1.2: Additional Store Model Parameters Required by TRNSYS Type 19 (continued)

lower edge 3 ft from the floor. Beam radiation transmitted by the window is assumed to strike the floor. The sensible and latent gains from people are taken to be 315 Btu/hr-person and 325 Btu/hr-person (ASHRAE, 1985, 26.21). The U-value of the floor is 0.02380 Btu/hr-ft^2-^{o}F. The floor is considered to be heavily insulated to simulate the minimal conductive losses through an actual floor due to the ground temperature below the floor approaching, given adequate time, the store temperature.

Sensitivity tests showed that store loads were not significantly effected by changing the following parameters: the store thermal capacitance, the reflectance of solar radiation by inner building surfaces, absorptance of solar radiation by the exterior building surfaces, the window diffuse solar radiation transmittance value, the overall solar radiation transmittance value, the window U-value, and the number of surfaces struck by beam radiation.

To avoid unnecessary complications, only the sales area of the store was modelled. Storage and work rooms, offices, the bakery, etc., have been neglected. As the heating and cooling needs of these areas are typically met by separate air-processing systems, the effect on the analysis of the principle air-conditioning system of neglecting these areas is to increase the conduction gains through the walls. However, the load due to conduction through the walls during the air-conditioning season was shown to be small compared to the loads from other sources (see Fig. 4.1.1.1).

Store temperature was not allowed to deviate from the temperature setpoint nor was the absolute humidity allowed to exceed the humidity setpoint. The TRNSYS Type 19 component capability to calculate infiltration loads was overridden to allow the use of separately calculated infiltration loads following the infiltration schedule of Fig. 3.1.2. It is assumed that 100% of the return air is taken from under the refrigerated cases, and that, as a result, one half of the sensible and latent refrigerated case credit directly affects the store air state while the remaining half acts on the return air (Cargocaire, n.d., 6). It is also assumed that the exfiltration rate is equal to the infiltration rate, making the supply air flow rate equal to the return air flow rate.

3.2: Refrigerated Cases

The refrigerated case model used for this study is based on that used by both GRI and Burns. It was modified to include anti-sweat heater energy consumption: the energy required for heaters which prevent condensation on metal strips, glass doors, etc.

The refrigerated cases produce a negative load, or credit, on the surrounding air. The sensible and latent case credits ($Q_{s,c}$ and $Q_{l,c}$) are calculated as follows:

$$Q_{s,c} = [C_{med} * f_{med}(\omega_{store}) + C_{low} * f_{low}(\omega_{store})\,] * (1 - \beta) * \gamma \qquad (3.2.1)$$

$$Q_{l,c} = [C_{med} * f_{med}(\omega_{store}) + C_{low} * f_{low}(\omega_{store})\,] * \beta * \gamma \qquad (3.2.2)$$

where ω_{store} is the absolute humidity of the store air in units of $lbm_{water}/lbm_{dry\ air}$, C_{med} and C_{low} are medium temperature and low temperature installed refrigerated case capacities, respectively; the latent-to-total credit ratio β is defined below, and γ is the fraction of time for which the cases are on as indicated in Fig. 3.2.1. The modifying functions f_{med} and f_{low} adjust the case credit for the absolute humidity level in the store and are defined below. These functions are based on curve fits to Tyler Refrigeration Company data (GRI, 1984, 108):

$$f_{med}(\omega_{store}) = 7.986 * \omega_{store}^{0.452} \qquad (3.2.3)$$

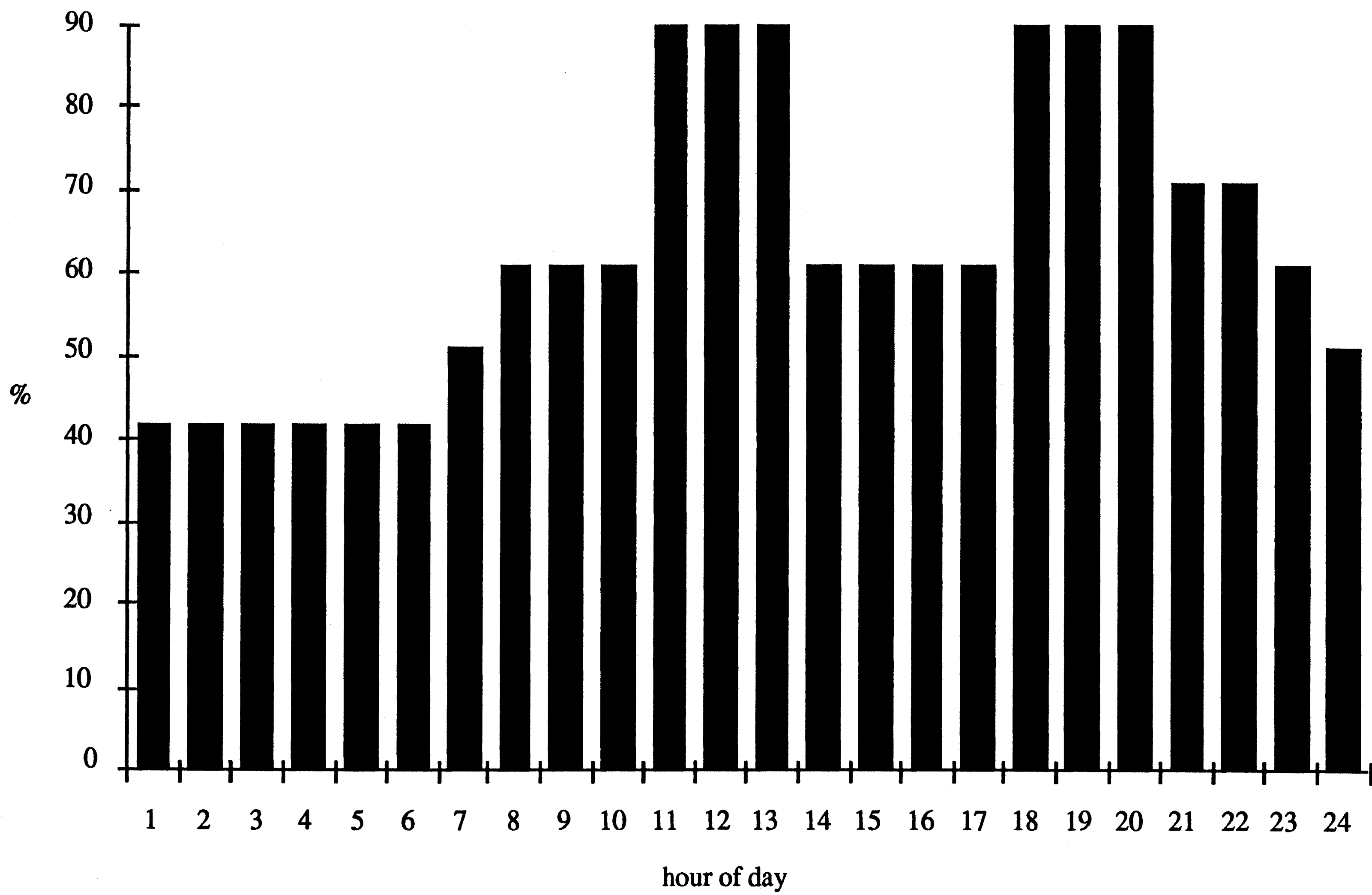

Fig. 3.2.1: Store Model Refrigerated Case On-Time

$$f_{low}(\omega_{store}) = 3.635 * \omega_{store}^{0.281} \tag{3.2.4}$$

$$\beta = 0.016 * \exp(245.0*\omega_{store}) \tag{3.2.5}$$

Anti-sweat heater power (q_s) was calculated using a simple resistance model and representative manufacturer's data. Anti-sweat heaters must raise the temperature of the outside surface of the case above the dew point temperature of the store air (T_{dp}).

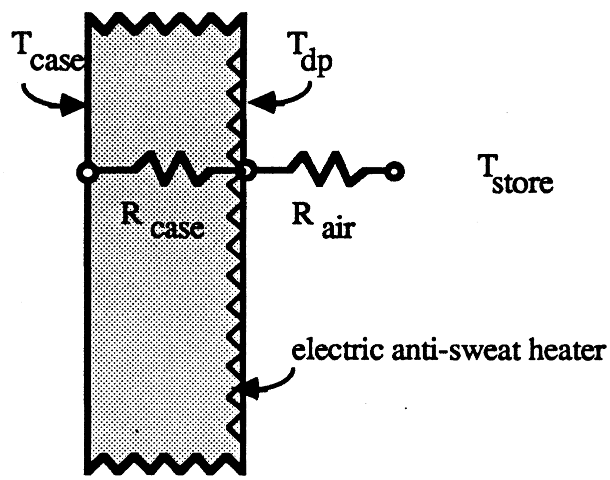

Fig. 3.2.2: Energy Balance on Case-Store Air Interface

A one-dimensional energy balance on the case-store air interface (see Fig. 3.2.2) requires:

$$q_s'' = \frac{T_{dp} - T_{store}}{R_{air}} + \frac{T_{dp} - T_{case}}{R_{case}} \tag{3.2.6}$$

For both low and medium temperature cases it is assumed that R_{air} = 1 hr-ft^2-^{o}F/Btu and that the case height (h) is 5 ft. Tyler refrigerated case data suggests that 115 V anti-sweat heaters on medium temperature cases use approximately 0.035 amps per linear foot of case, or 4 W/ft. If it is assumed that the heaters are continuously on when the store air has a temperature of 80^{o}F and a dew point temperature of 70^{o}F and the inside case surface temperature is 35^{o}F, $R_{case,med}$ = 2.7 hr-ft^2-^{o}F/Btu. For low temperature cases, 115 V heaters are taken to require 0.2 amps per linear foot of case or 23 W/ft. With the above store conditions and an inside case surface temperature of -10^{o}F, $R_{case,low}$ = 3.1 hr-ft^2-^{o}F/Btu.

Using the values for R_{case} calculated above, the anti-sweat heater energy consumption is calculated using the following equation:

$$q_s = \left[\frac{T_{dp} - T_{store}}{R_{air}} + \frac{T_{dp} - T_{case,med}}{R_{case,med}}\right]^+ * L_{med} * h_{med} +$$

$$\left[\frac{T_{dp} - T_{store}}{R_{air}} + \frac{T_{dp} - T_{case,low}}{R_{case,low}}\right]^+ * L_{low} * h_{low} \qquad (3.2.7)$$

where the "+" subscript signifies that the quantity in brackets is used only when positive and is otherwise replaced with 0, and L_{med} and L_{low} are the linear case lengths.

The total power consumption of the refrigerator cases (P_r) is defined by the following equation:

$$P_r = \left(\frac{Q_{s,c} + Q_{l,c}}{COP}\right) + q_s \qquad (3.2.8)$$

where $Q_{s,c}$ and $Q_{l,c}$ are found from eqs. 3.2.1 and 3.2.2. Since the evaporator temperature (T_e) remains nearly constant, the temperature of the air leaving the case condenser(s) (which is approximately the condenser temperature or T_c) will determine the COP. Using the Carnot COP to approximate this dependency,

$$COP = COP_{design} * \frac{(T_c - T_e)_{design}}{(T_c - T_e)} \tag{3.2.9}$$

Table 3.2.1 indicates the "design" values assumed. (Design COP's are those used in the GRI study (GRI, 1984, 112).)

	<u>medium temp.</u>	<u>low temp</u>
T_e	0 ^oF	-40 ^oF
T_c	100 ^oF	100 ^oF
COP_{design}	1.86	1.07

Table 3.2.1: Design Values Assumed for Refrigerated Case COP Determination

A new TRNSYS component was written to implement the above model. In addition to calculating the total power consumption, the exiting temperature and absolute humidity of the air passing under the refrigerated cases are calculated. It was assumed that 100% of the circulation air passes under the refrigerated cases before being either exhausted or reprocessed and that 50% of the sensible and latent refrigerated case credits act on this air (Cargocaire, n.d., 6). (The other 50% of the case credits is assumed to act on the store air.) It was also assumed that the refrigerated cases condenser(s) heat is reclaimed, and that the condenser temperature is approximately the temperature of the store supply air when reheat is done and 15°F higher than the outside air temperature when reheat is not done.

3.3 Vapor-Compression Air-Conditioning Units

Both conventional and humid-climate air-conditioning systems use vapor-compression air conditioners performing both sensible and latent cooling. A sensible-cooling-only air-conditioning unit is sometimes used in humid-climate systems. These units are modelled in a manner similar to that used in DOE-2 (LBL and LANL, 1981, IV.7-IV.16 and IV.62-IV.74).

The air state at the surface of an evaporator coil (T_{surf}, ω_{surf}) can be determined from the states of the air entering (T_{in}, ω_{in}) and leaving (T_{exit}, ω_{exit}) the coil:

$$T_{surf} = \frac{T_{exit} - T_{in} * BF}{1 - BF} \tag{3.3.1}$$

$$\omega_{surf} = \frac{\omega_{exit} - \omega_{in} * BF}{1 - BF} \tag{3.3.2}$$

where BF is the coil by-pass factor (the fraction of air that is unaffected by the coil) calculated for the given conditions. For a coil on which condensation occurs, T_{surf} and ω_{surf} describe an air state on the saturation line. Because there is only one absolute humidity for each temperature on the moist-air saturation line, the amounts of sensible and latent cooling performed by the coil are coupled.

This model requires the specification of coil performance at ARI rating conditions (67^{o}F wet-bulb entering evaporator temperature, 95^{o}F entering condenser dry-bulb temperature, and a flow-rate equal to or less than 37.5 cfm/1000 Btu/hr cooling capacity (ARI, 1966)). Coil performance at off-rated conditions is determined by multiplying the various rated performance indicators by one or more modifying functions:

$$BF = BF_{rated} * f_{BF\_cfm}(cfm_{actual}) \tag{3.3.3}$$

$$C = C_{rated} * f_{C\_T}(ewb,cdb) * f_{C\_cfm}(cfm_{actual}) \tag{3.3.4}$$

$$\frac{1}{COP} = \left(\frac{1}{COP_{rated}}\right) * f_{1/COP\_T}(ewb,cdb) * f_{1/COP\_cfm}(cfm_{actual}) * f_{1/COP\_plr}(plr) \tag{3.3.5}$$

where ewb is evaporator entering wet-bulb temperature (^{o}F), cdb is condenser entering dry bulb temperature (^{o}F), cfm_{actual} is the actual volume flow-rate (in cfm), C is the total capacity and plr is the part-load-ratio or (total load / C). The effect on the air conditioner COP of reclaiming heat from the condenser was not included in this study. Future studies should consider modelling this effect.

DOE-2 default functions have been used for the modifying functions (LASL, 1980, 72-73):

$$f_{BF\_cfm}(cfm_{actual}) = \frac{cfm_{actual}}{cfm_{rated}} \tag{3.3.6}$$

$$f_{C\_T}(ewb,cdb) = 0.418934 + 0.017421 * ewb - 0.00617 * cdb \tag{3.3.7}$$

$$f_{C\_cfm}(cfm_{actual}) = 0.69717199 + 0.39555 * \left(\frac{cfm_{actual}}{cfm_{rated}}\right) + 0.092727 * \left(\frac{cfm_{actual}}{cfm_{rated}}\right)^2 \tag{3.3.8}$$

$$f_{1/COP\_T}(ewb,cdb) = 0.282094 - 0.005832 * ewb + 0.01167 * cdb \tag{3.3.9}$$

$$f_{1/COP\_cfm}(cfm_{actual}) = 1.13318 - 0.13318 * \left(\frac{cfm_{actual}}{cfm_{rated}}\right) \tag{3.3.10}$$

$$f_{1/COP\_plr}(plr) = 0.29333333 + 0.70666667 * plr \tag{3.3.11}$$

It has been assumed that partial load capacity is achieved exclusively by cylinder unloading (i.e., hot-gas bypassing and compressor cycling are not used).

These default functions were used to predict manufacturer's published performance data (Carrier, 1986, 16). Predicted and published results agree well (Figs. 3.3.1-3.3.3). It is expected that the performance modifying functions will have slightly different shapes for different air-conditioning units, however for the purposes of this study, a representative, not a specific, air-conditioning unit is required.

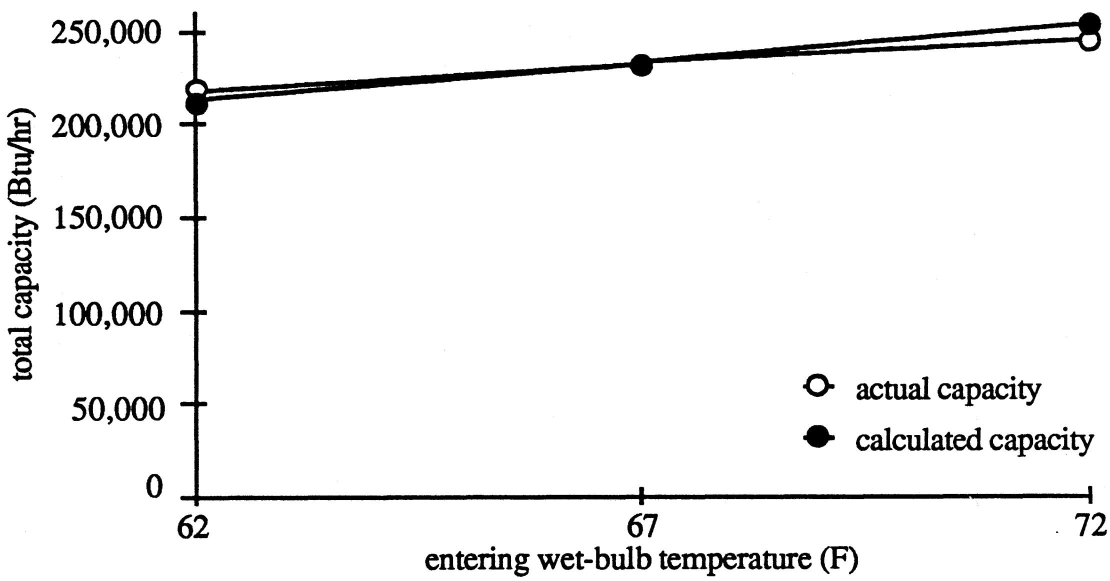

Fig. 3.3.1 (a): Total Capacity vs. Entering Wet-Bulb Temperature

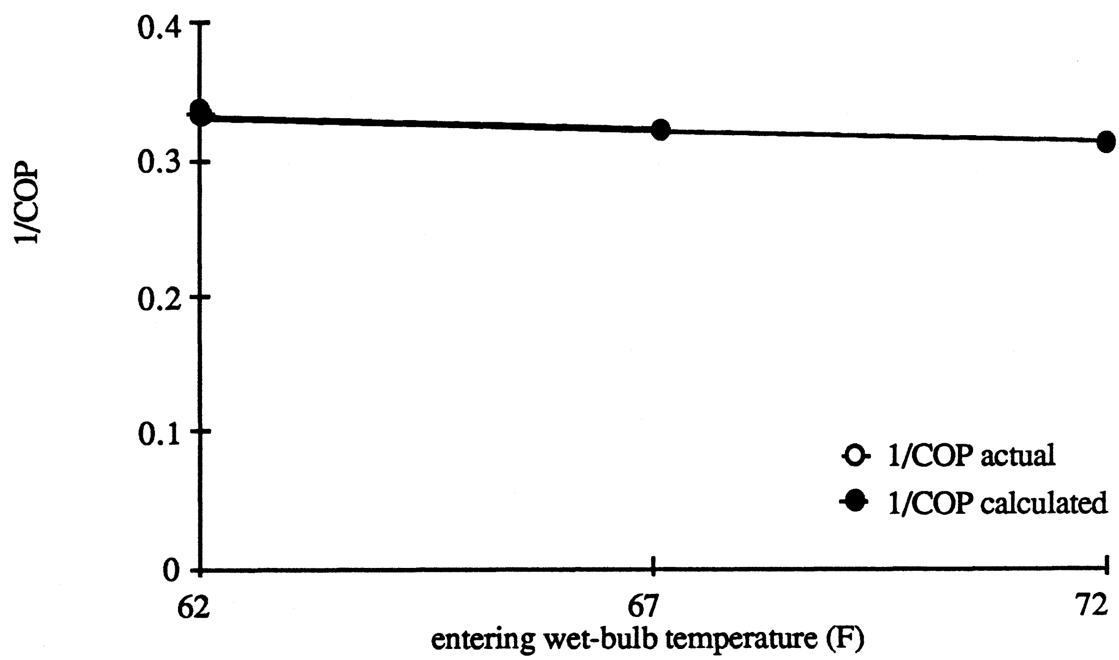

Fig. 3.3.1 (b): 1/COP vs. Entering Wet-Bulb Temperature

Fig. 3.3.1: Actual and Calculated 18-Ton Carrier Performances
(cond. dry-bulb: 95 F; flow rate: 8100 cfm)

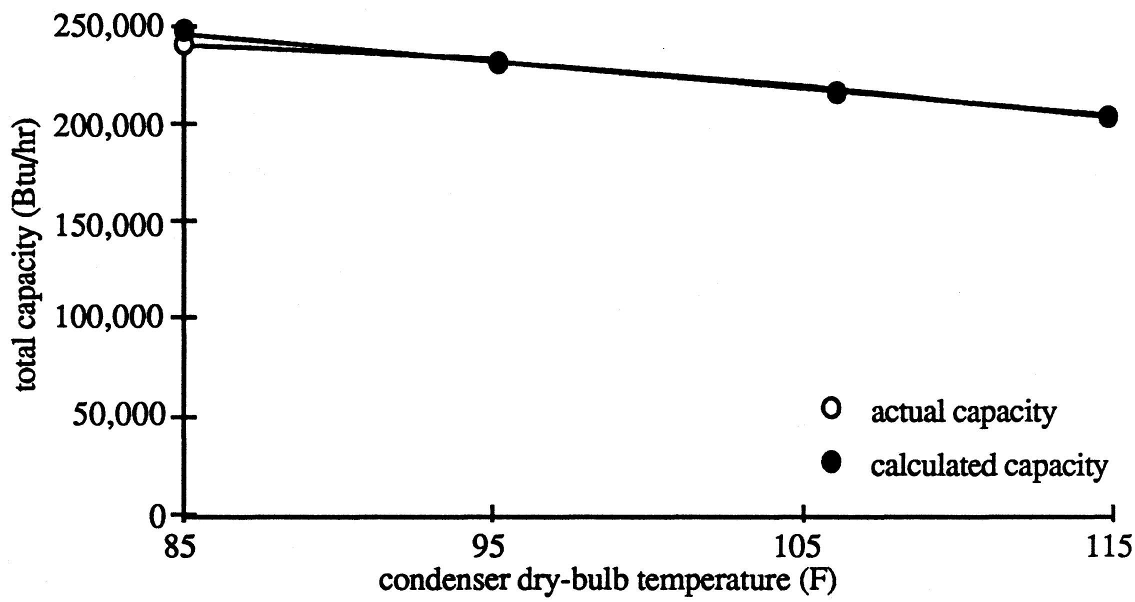

Fig. 3.3.2 (a): Total Capacity vs. Condenser Dry-Bulb Temperature

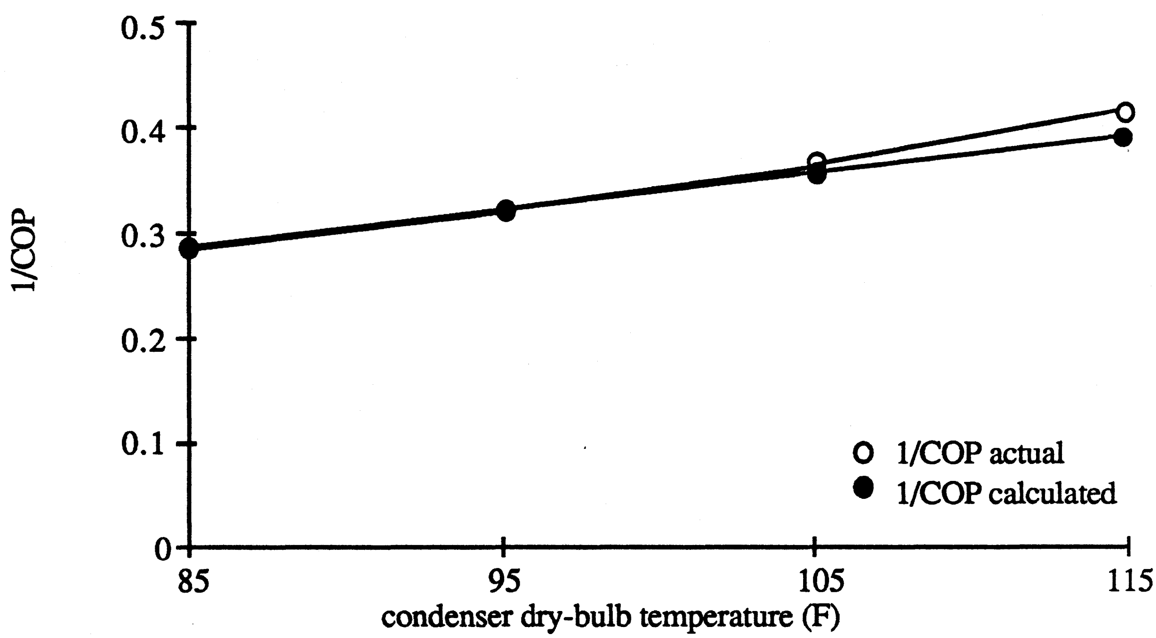

Fig. 3.3.2 (b): 1/COP vs. Condenser Dry-Bulb Temperature

Fig. 3.3.2: Actual and Calculated 18-Ton Carrier Performances
(entering wet-bulb: 67 F, flow rate: 8100 cfm)

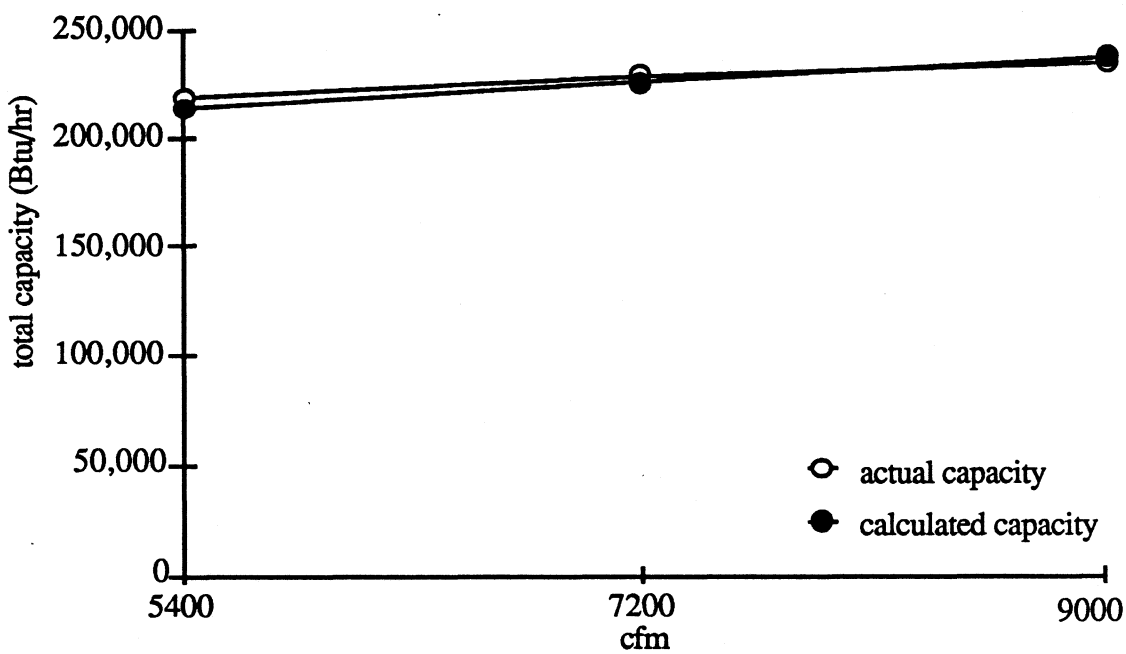

Fig. 3.3.3 (a): Total Capacity vs. Flow Rate

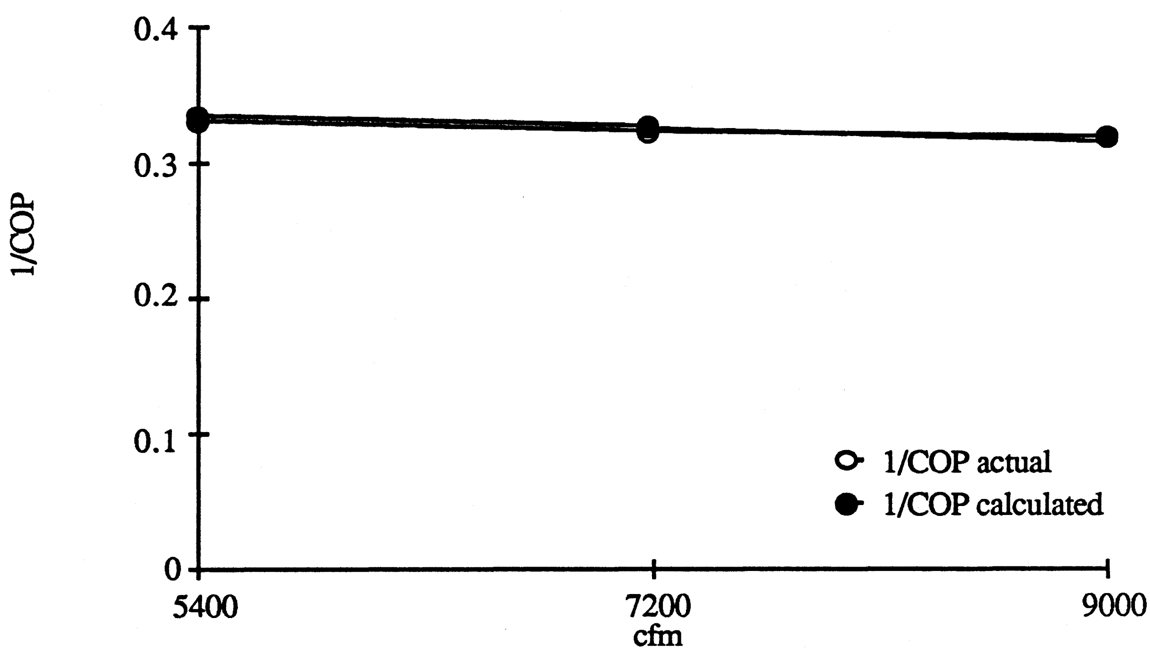

Fig. 3.3.1 (b): 1/COP vs. Entering Wet-Bulb Temperature

Fig. 3.3.3: Actual and Calculated 18-Ton Carrier Performances

(entering wet-bulb: 67 F; condenser dry-bulb: 95 F)

(continued on pg. 37)

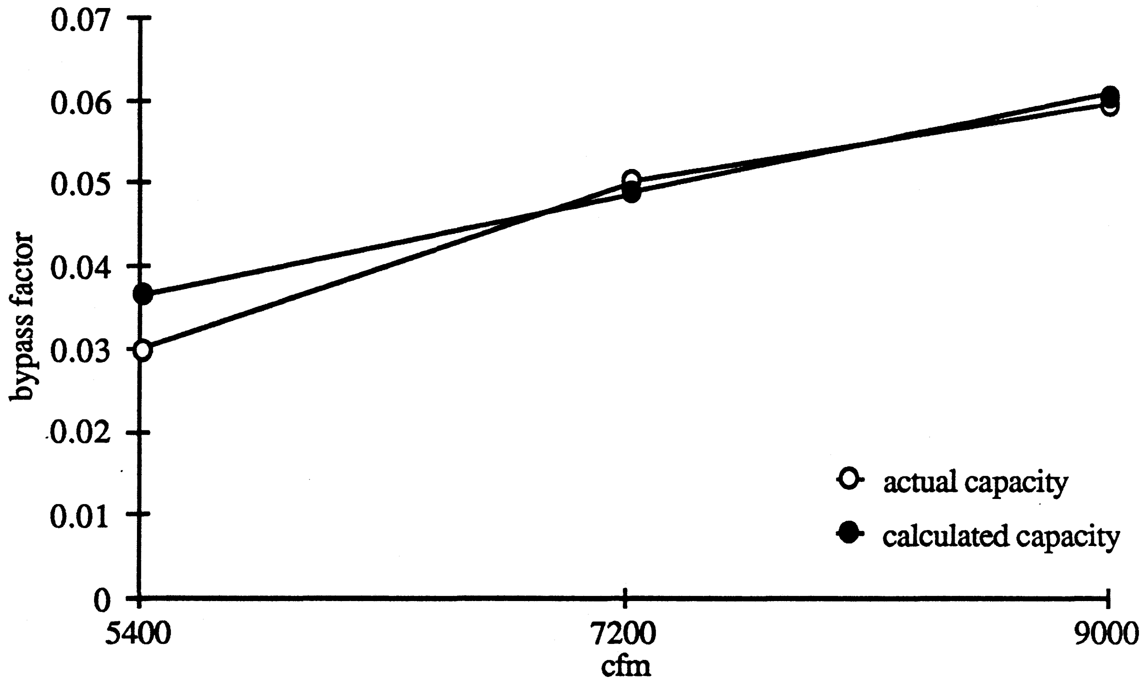

Fig. 3.3.3 (c): Bypass Factor vs.cfm

Fig. 3.3.3: Actual and Calculated 18-Ton Carrier Performances
(entering wet-bulb: 67 F; condenser dry-bulb: 95 F)
(continued)

3.3.1: Sensible and Latent Cooling Unit

A new TRNSYS component was used to implement the vapor-compression air-conditioner model described above. To avoid multiple system iterations for each time-step of a simulation, the control of the air conditioner was included in the component. For conventional systems the sensible load is first met. If this does not also meet the latent load, the latent load is met, sensibly over-cooling the air. For a humid-climate system, the latent load is met and it is assumed that the secondary air-conditioner will meet any remaining sensible load.

The TRNSYS psychometerics routine was used to determine ω_{surf} given T_{surf}, while a regression fit to moist air states on the saturation line was used to determine T_{surf} given ω_{surf} (see Fig. 3.3.1.1). T_{surf} is constrained to be no less than 33°F. The rated capacity was chosen to just meet the load at design conditions, the rated COP for all units was assumed to be 4 and the rated bypass factor for all units was assumed to be 0.05. Assuming that the temperature and humidity remain at or below the setpoints implies that the sensible and latent loads be met at all times. For this reason, for the few hours of the year that the load exceeds the capacity, the partial-load-ratio is set to 1 and the load is considered to be fully met.

3.3.2 Sensible Cooling Only Unit

A TRNSYS component similar to that described above was created to model the secondary (sensible-only) unit of the humid-climate system. Only the control of the unit differs: as this coil is not expected to do any latent cooling, T_{surf} is first determined. If the corresponding ω_{surf} implies that condensation occurs, the resulting exiting absolute humidity will be calculated, otherwise the incoming and the exiting absolute humidities will be identical. T_{exit} will be calculated in both cases. Once again, the rated capacity was chosen to just meet the load at design conditions, the rated

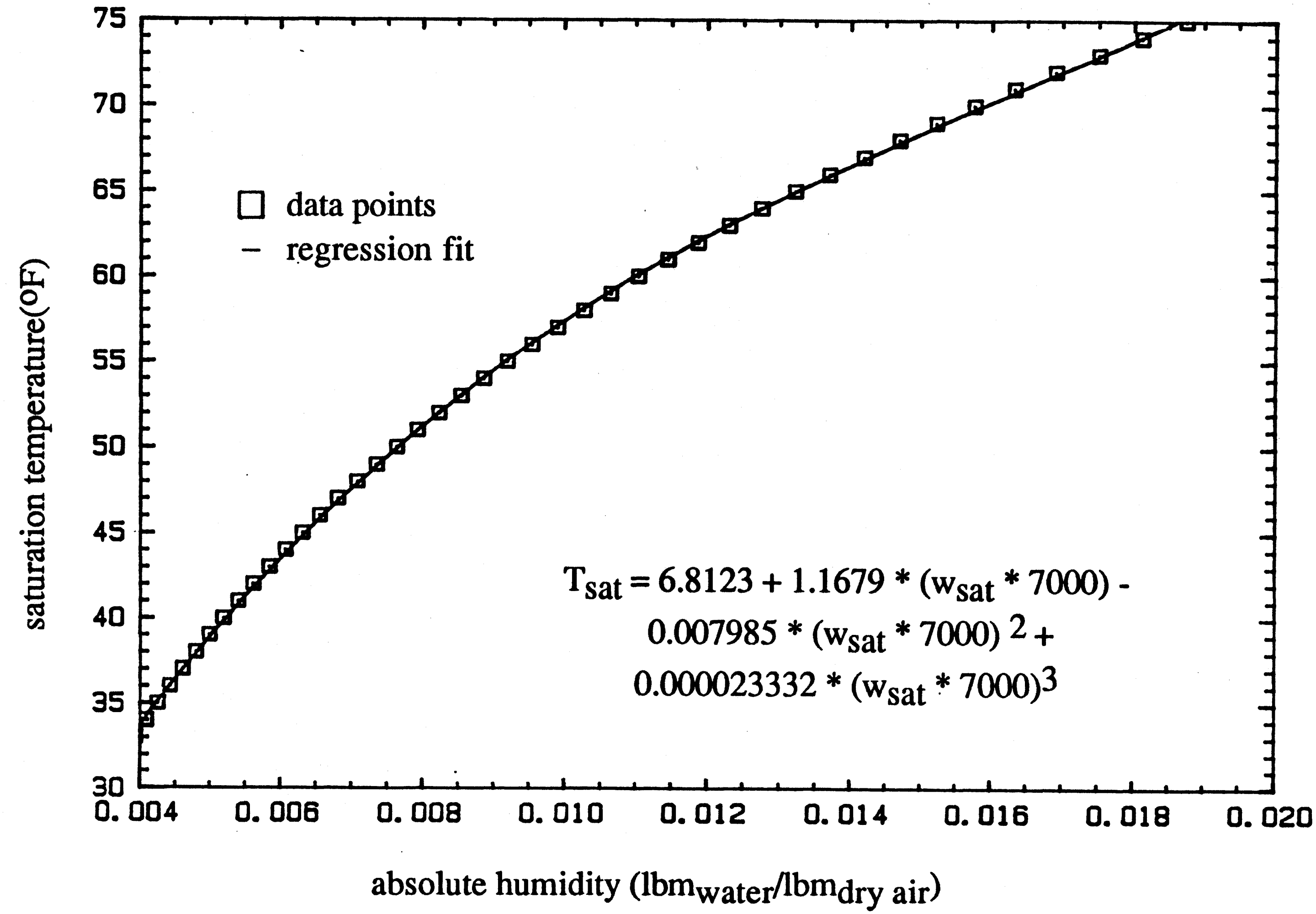

Fig. 3.3.1.1: Regression Fit for Saturation Temperature as a Function of Saturation Humidity

COP for all units was assumed to be 4 and the rated bypass factor for all units was assumed to be 0.05. If the load exceeds the capacity, the partial-load-ratio is set to 1 and the load is considered to be fully met.

3.4: Reheat Heat Exchanger

To control humidity, the primary air-conditioning unit in both the conventional and humid-climate systems often over-cools the air. This air must be reheated before returning it to the store. A typical 30,000 ft^2 store can produce 600,000-900,000 Btu/hr of reclaimed heat from the air-conditioner condenser (McQuay, 1979, 4). Heat may also be reclaimed from the refrigerated cases condenser(s). As reheat/winter heating requirements larger than 700,000 Btu/hr were not encountered in any Miami simulation, all heating was considered "free".

A TRNSYS algebraic unit (Type 15) was used to calculate required heat and exiting temperature.

3.5: Fan Power

The air flow rates through some components of the humid-climate system are considerably smaller than those in the conventional system. If these two systems are to be compared, fan power cost must be included.

Fan power is a function of volume flow-rate ($\dot{m}/\rho$) and static pressure drop (ΔP_i) for each component i of the system as given by:

$$P_f = \sum\left[\frac{\dot{m}_i}{\rho} * \Delta P_i * \left(\frac{\dot{m}_i}{\dot{m}_{i,ref}}\right)^2\right] \qquad (3.5.1)$$

where $\dot{m}_{i,ref}$ is the flow rate at which the pressure drop for the i^{th} component was determined, $\dot{m}_i$ is the actual flow rate through component i, and ρ is the density of the dry air. Fan inefficiencies which heat the air have been considered negligible.

Table 3.5.1 lists the assumed component pressure drops and reference flow rates. The duct pressure drops are those used by Burns (Burns, 1985, 54).

Neither the conventional system nor the humid-climate system is a variable volume system. As a result, there are two constant sets of flow rates for each system: one for the peak hours period and one for the off-peak hours when the ventilation is cut back. Fan power requirements were hand-calculated and then added to the computer-calculated power requirements for other components.

3.6: Miscellaneous Components

Several standard utility TRNSYS components were used in system simulations. Solar radiation data, outside temperature and humidity, and windspeed were read from a Typical Meteorological Year (TMY) data set with a Type 9 data reader. The radiation data from this component was processed by a Type 16 radiation processor to generate the solar energy received by the store walls, roof and window. Several units of the Type 14 time-dependent forcing function component were used to generate hourly values for scheduled quantities such as infiltration and ventilation rates. Finally, slightly improved Type 11 flow-diverter/mixer components were used to control air-flow. (See Appendix B.)

Component	ΔPi(in. H_2O)	$\dot{m}_{i,ref}$ (lbm/hr)
ducts	2.0	78,840
reheat coil:		
conventional (2 rows)	0.30	78,840
humid-climate (1 row)	0.24	78,840
conventional a.c. coil (4-6 rows)	1.4	78,840
primary humid-climate a.c. coil (8 rows)	1.7	11,830
secondary humid-climate a.c. coil (4 rows/sensible)	1.2	67,010
indirect evaporative cooler	0.5	78,840
outside air filter	1.0	11,830

Table 3.5.1: Pressure Drops Across Air-Conditioning System Components

CHAPTER 4: SYSTEM SIMULATIONS

The component models described in Chapter 3 were combined to form models of the conventional and humid-climate systems discussed in Chapter 2. This chapter reports and analyzes the results of the annual simulations of these models. Unless otherwise specified, all simulations were performed using Miami, FL TMY weather data.

4.1: Base Cases

Design loads for the model supermarket were calculated for Miami. The circulation flow rate and air conditioner capacity were sized to meet these loads, and annual simulations of these "base cases" were run.

4.1.1: Design Loads and System Sizing

Both the conventional and the humid-climate systems were sized using design loads calculated by the Trane Company's Load Design program (Trane, 1986). Fig. 4.1.1.1 shows the supermarket loads and credits at design conditions. The lighting load and the load due to conduction through the roof are the major components of the sensible load. The total latent load is dominated by the people-generated load and ventilation latent loads. For design purposes, the refrigerated case on-time was taken to have its lowest value (see Fig. 3.2.1). Even at this level, both the sensible and latent refrigerated case credits are substantial.

Given the internal (i.e. excluding ventilation) design loads of 381,600 Btu/hr sensible and 77,300 Btu/hr latent, the circulation flow rate was chosen to be 18,000 cfm, the lowest round number flow rate which gives a physically achievable supply

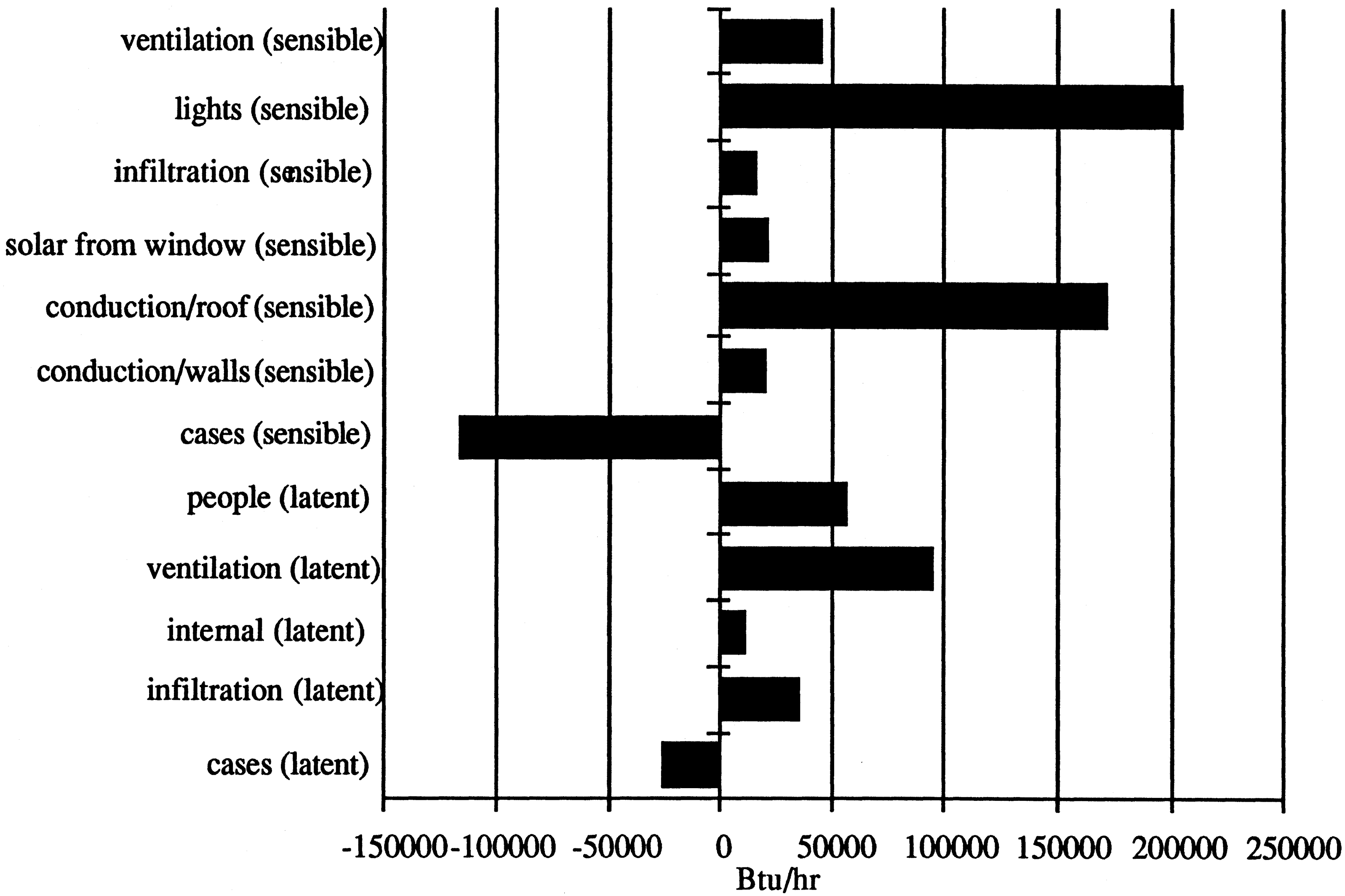

Fig. 4.1.1.1: Supermarket Loads/Credits at Design Conditions

air state. The rated air conditioner capacities of 495,000 Btu/hr (41.3 Tons) for the conventional system and 235,000 Btu/hr (19.6 Tons) and 215,000 Btu/hr (17.9 Tons) for the humid-climate system primary and secondary air conditioners, respectively, were chosen to just meet the design loads. The conventional and humid-climate systems configured with these values formed the base cases.

4.1.2: Energy Consumption Breakdown

Only those supermarket energy consumptions that vary with air conditioner system type were considered in this study. Into this category fall the electricity used by the air conditioner unit(s), the refrigerated cases and the fans, and the energy required by the reheat coil. Fig. 4.1.2.1 charts these quantities for each month of the conventional system base case simulation while Fig. 4.1.2.2 displays the same quantities for the humid-climate base case simulation. The reheat energy is considered to be "free" in this study (see section 3.4). Of the energy quantities which must be purchased, the refrigerated case compressor energy consumption is the greatest: between 5 to 230 times the air conditioner energy consumption, depending on the month and on the system. For both types of systems, the refrigerated case anti-sweat heater energy consumption is very small (~2%) compared to the compressor energy consumption This suggests that the Burns and GRI studies were justified in neglecting the anti-sweat heaters. For the conventional system, the energy necessary for the refrigerated cases compressor stays relatively constant throughout the year as the result of two different effects: lower store humidities during the winter tend to decrease this energy consumption, while a higher demand for heat/reheat energy during the summer (and hence a higher refrigerated case COP) also tends to decrease this energy consumption. The humid-climate system does not require as much heat/reheat energy

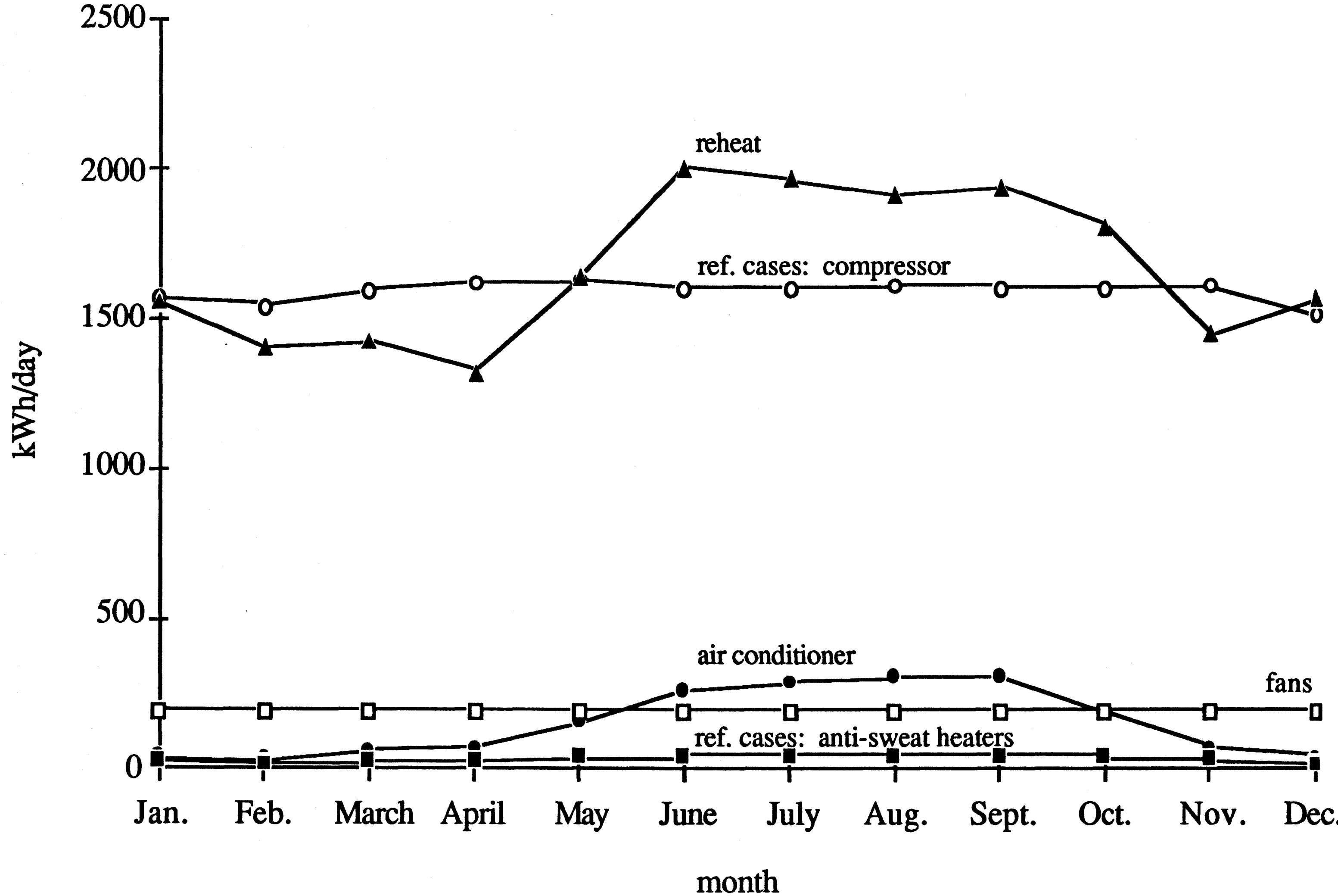

Fig. 4.1.2.1: Annual Energy Consumption Breakdown (Conventional System Base Case)

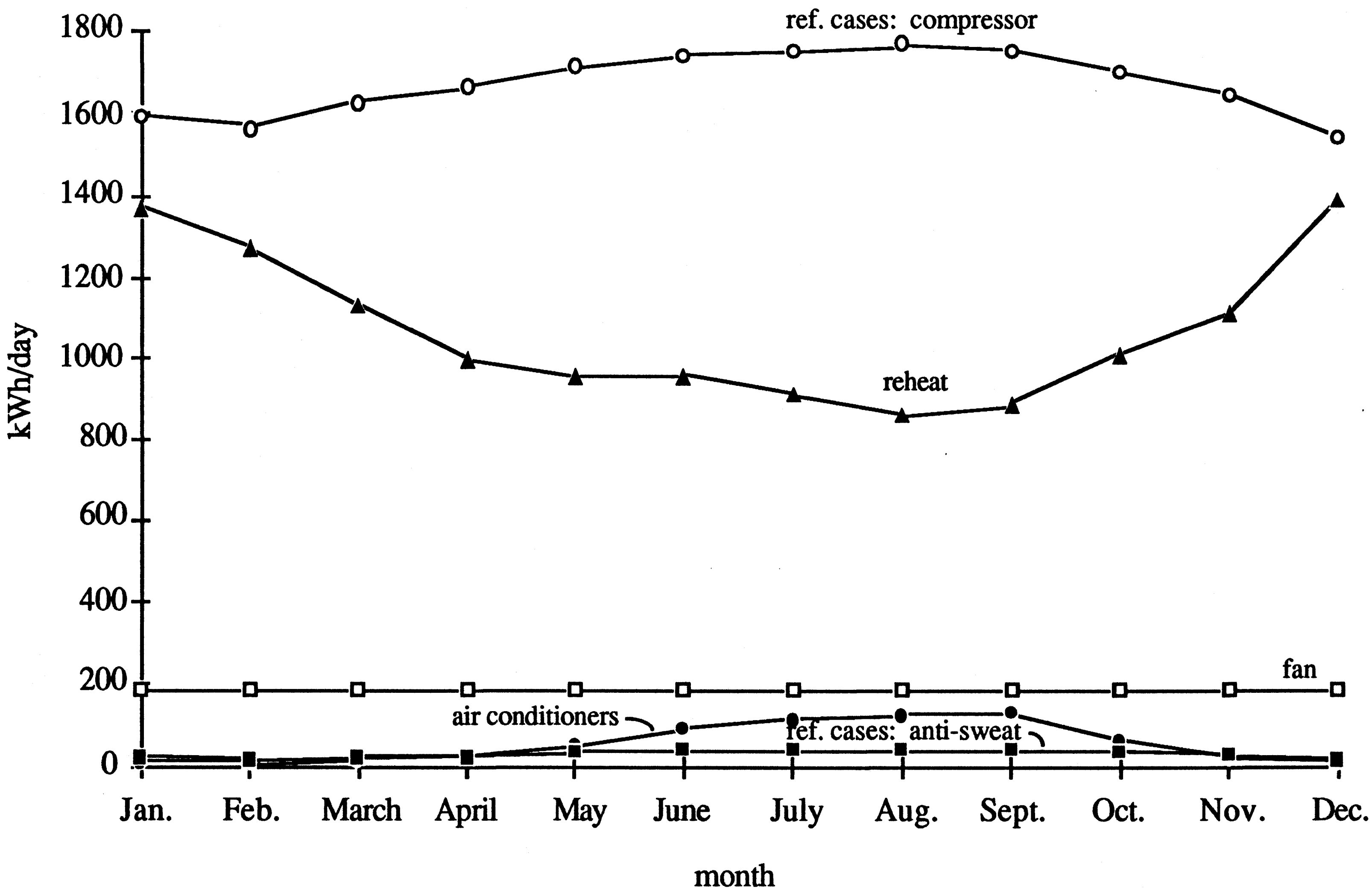

Fig. 4.1.2.2: Annual Energy Consumption Breakdown (Humid-Climate System Base Case)

during the summer as it does not over-cool the processed air to the same degree. The refrigerated case condenser is consequently more often cooled by the warmer outside air, reducing the case COP and increasing the refrigerated case energy consumption.

4.1.3: Conventional System vs. Humid-Climate System

Fig. 4.1.3.1 compares the air-conditioner, refrigeration and fan power requirements for the two systems The substitution of a humid-climate system for a conventional system resulted in a 63% annual air conditioner energy savings. The differences in the air handling of the two systems produces a 5% annual fan power savings for the humid-climate system. The humid-climate refrigeration energy consumption is 5% higher than that of the conventional system as a result of the effect of reclaimed reheat on the refrigerated case COP. The total (i.e. air-conditioner units(s), fans and refrigerated cases) annual energy savings is 9,700 kWh, or 1%. If the change in the refrigeration energy consumption for the humid-climate system is ignored, the annual savings would be 37,100 kWh or 5%. Clearly the effect on the refrigerated case COP of reclaiming heat from the refrigerated case condenser is important.

4.2: Sensitivity Analyses

A number of the base-case model parameters were varied to study what effect, if any, different store configurations have on the potential energy savings of a humid-climate system.

4.2.1: Climate

Comparing annual simulation results done with TMY weather data for a humid climate such as Miami with those for a colder, drier climate such as Madison, WI,

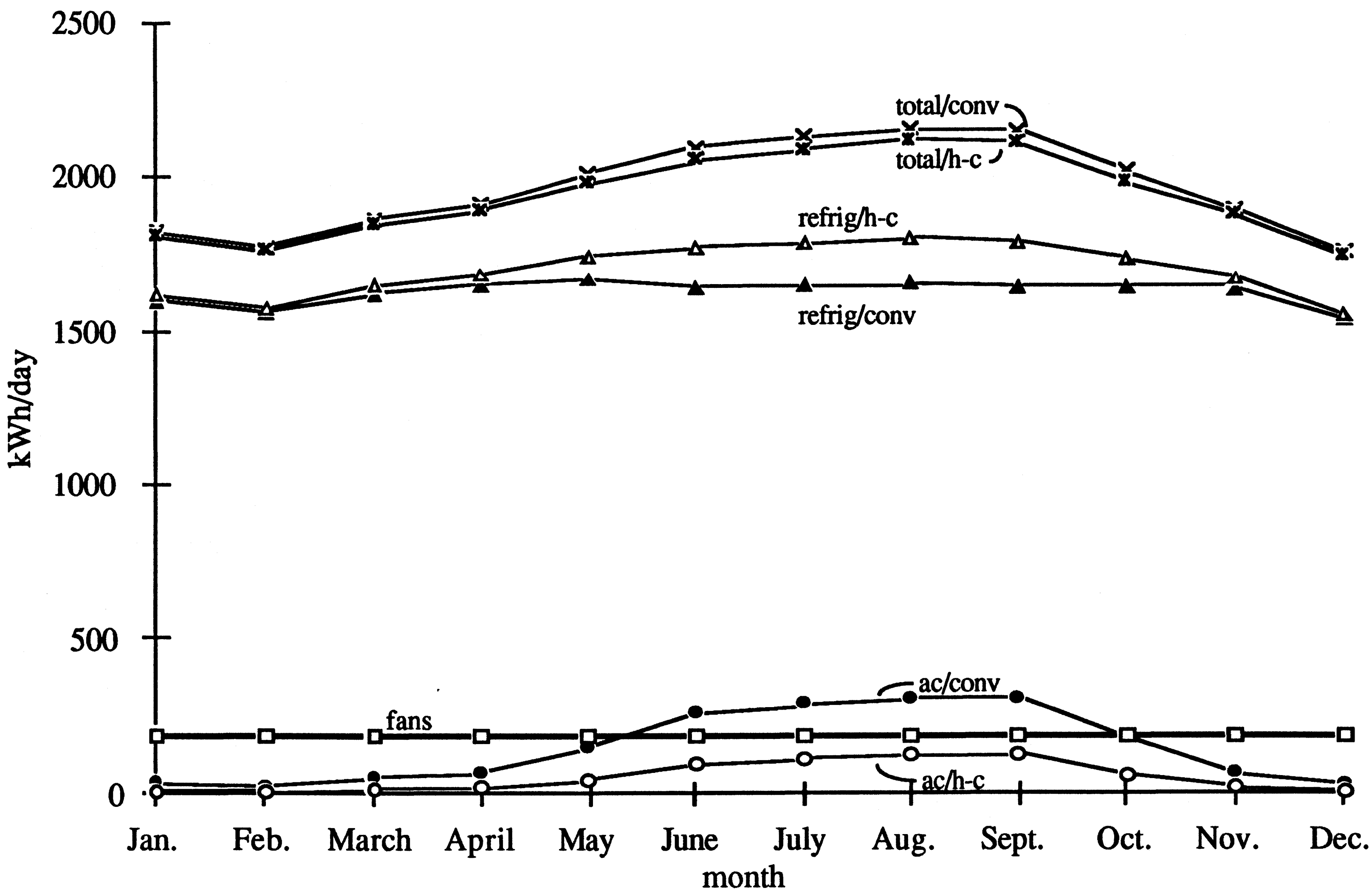

Fig. 4.1.3.1: Energy Consumption for Conventional and Humid-Climate System Base Cases

reveals the aptness of the name "humid-climate system". Fig. 4.2.1.1 is a bar graph showing the annual energy required for air conditioning, refrigeration and reheat/heating by both conventional and humid climate systems in Miami and in Madison. With the exception of the weather data, the simulations for both locations are identical.

As stated in section 4.1.3, the humid climate system uses 63% less air-conditioner energy in Miami than the conventional system. The percentage savings in Madison is similar: 65%. However, the climate in Madison requires only 5% of the air-conditioning energy necessary in Miami. Clearly the potential savings are much greater in a humid climate.

The refrigerated cases require 9% less energy in Madison in Miami. This is because in Madison the average store humidity is lower due to the drier climate and the average outside temperature is lower. The increase in refrigeration energy noted for the humid-climate system in Miami does not occur in Madison, as the outside air (i.e., the air used to cool the refrigerated case condenser(s) when reheat is not done) is generally cooler than the store supply air.

Finally, as expected, the reheat/heating requirements in Madison are greater. In Miami, the humid-climate system requires noticeably less reheat energy, whereas there is little difference between the conventional and humid-climate systems in Madison. This is due to the small air-conditioning requirements and large heating requirements in Madison. The heat/reheat requirements in Madison exceeded 1,000,000 Btu/hr during some portions of the year. It is unlikely reclaimed heat could meet this demand.

4.2.2: Humidity Setpoint

It is well accepted that lower store humidity levels result in lower refrigeration energy requirements (Tyler, 1978). As the refrigeration energy consumption is much

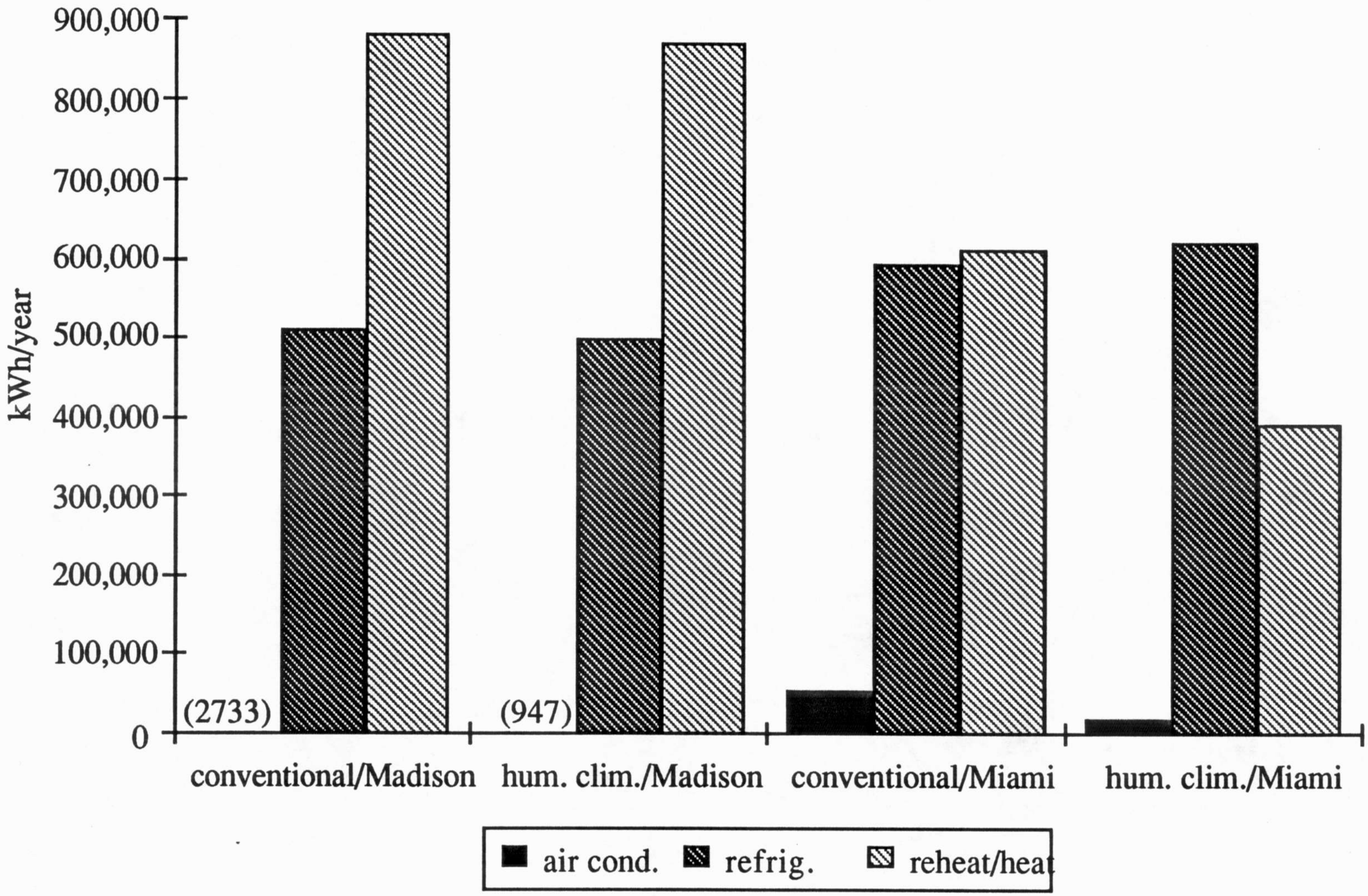

Fig. 4.2.1.1: Climate Dependence

greater than that of either the fans or the air-conditioning units, it might be possible for a lower store humidity setpoint to result in net energy conservation.

Fig. 4.2.2.1 shows the effect of lower store humidity setpoints on the total energy requirements of a conventional air-conditioning system. Only the store setpoints were changed between simulations. The temperature setpoint was increased as the humidity setpoint was decreased to give approximately the same comfort condition (ASHRAE, 1981, 8.21). Although the refrigeration energy consumption does decrease, this is more than compensated for by the increased air conditioner energy use. The net effect is greater annual energy consumption for lower humidity setpoints. A similar comparison was attempted for the humid-climate system, however lower store humidity levels could not be maintained with the base-case ventilation levels (see Fig. 3.1.2) as the ventilation air could not be cooled and dried sufficiently to meet the load.

To determine what would happen if the air-conditioning systems were sized for lower store humidity levels, design loads were calculated for the lower humidity setpoints and the circulation and air-conditioner capacities were recalculated. Fig. 4.2.2.2 plots the air conditioner, refrigeration, fan and total annual energy consumptions for resized conventional systems. Lower humidity setpoints do save energy, however this is the result of lower circulation flow rates (16,000 cfm for a humidity level of 0.009 $lbm_{water}/lbm_{dry\ air}$, and 13,000 cfm for a humidity level of 0.008 $lbm_{water}/lbm_{dry\ air}$). The lower humidity supply air can be cooler, allowing the sensible and latent load to be met with a lower circulation flow rate.

Fig. 4.2.2.3 shows the energy requirements for resized humid-climate systems. The lower humidity setpoints allow the same lower circulation flow rates found for the resized conventional systems, however the ventilation rates must be increased to provide adequate air to dehumidify. In the case of the system sized for a humidity

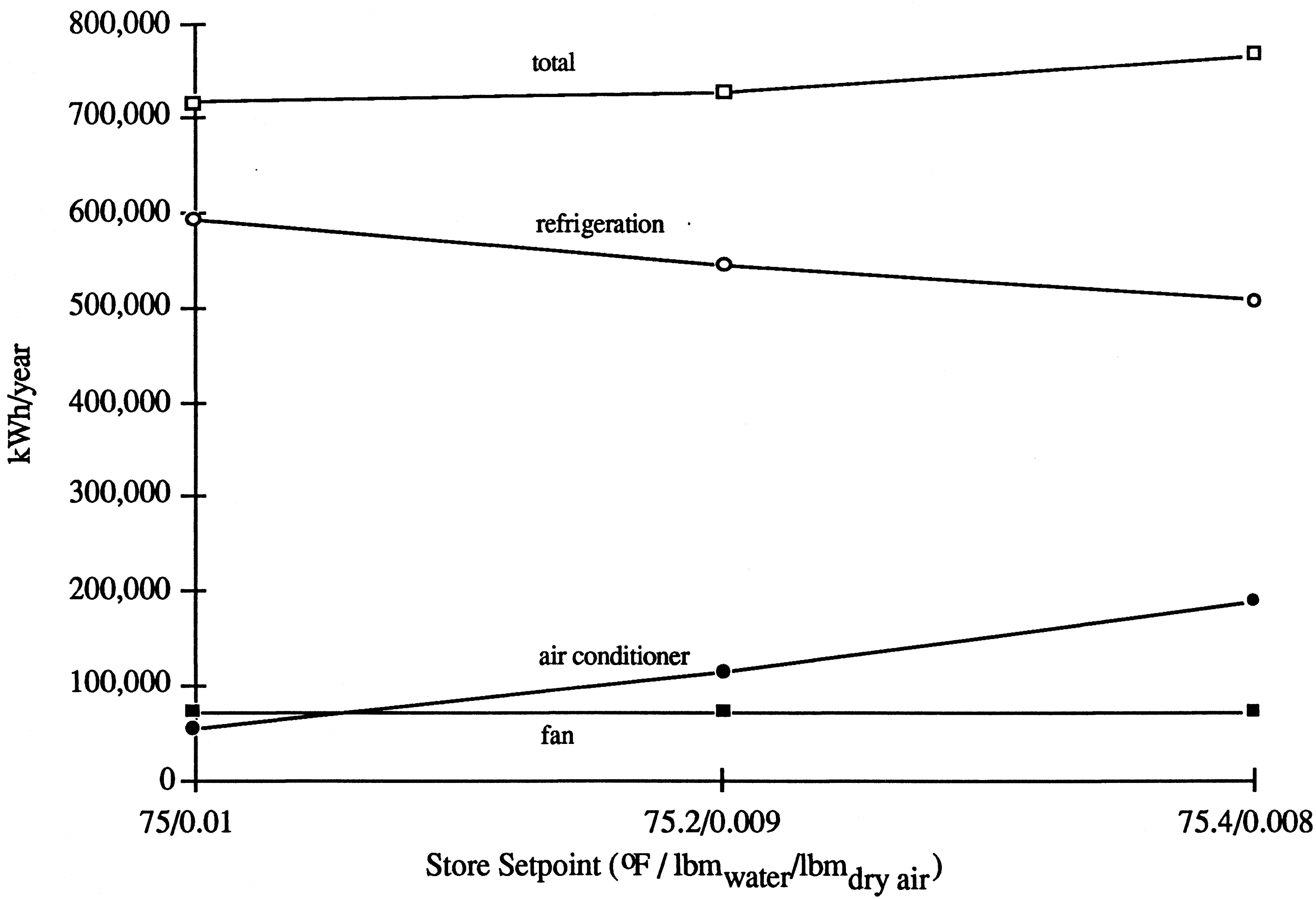

Fig. 4.2.2.1: Store Setpoint Dependence (Conventional System)

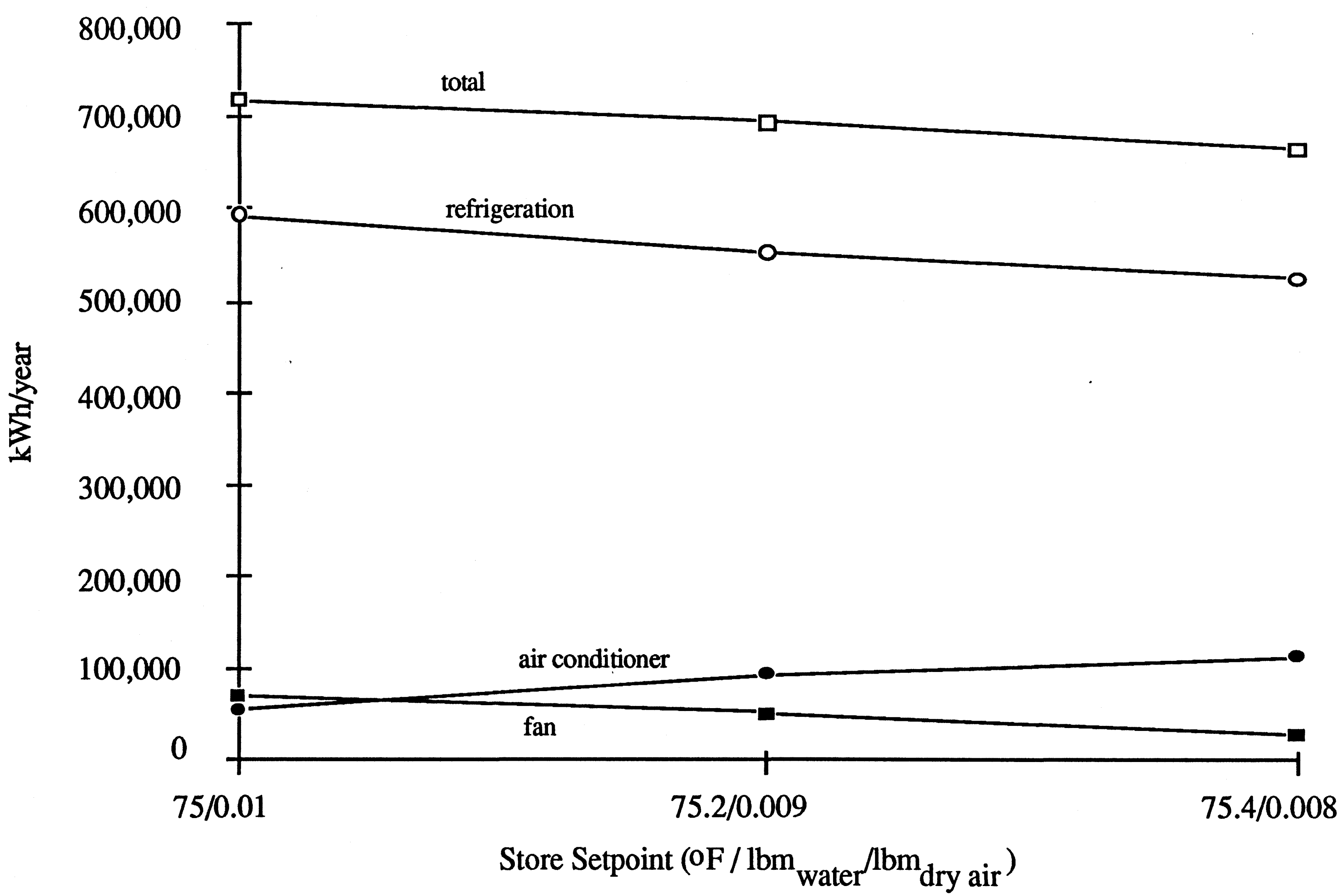

Fig. 4.2.2.2: Store Setpoint Dependence (Resized Conventional System)

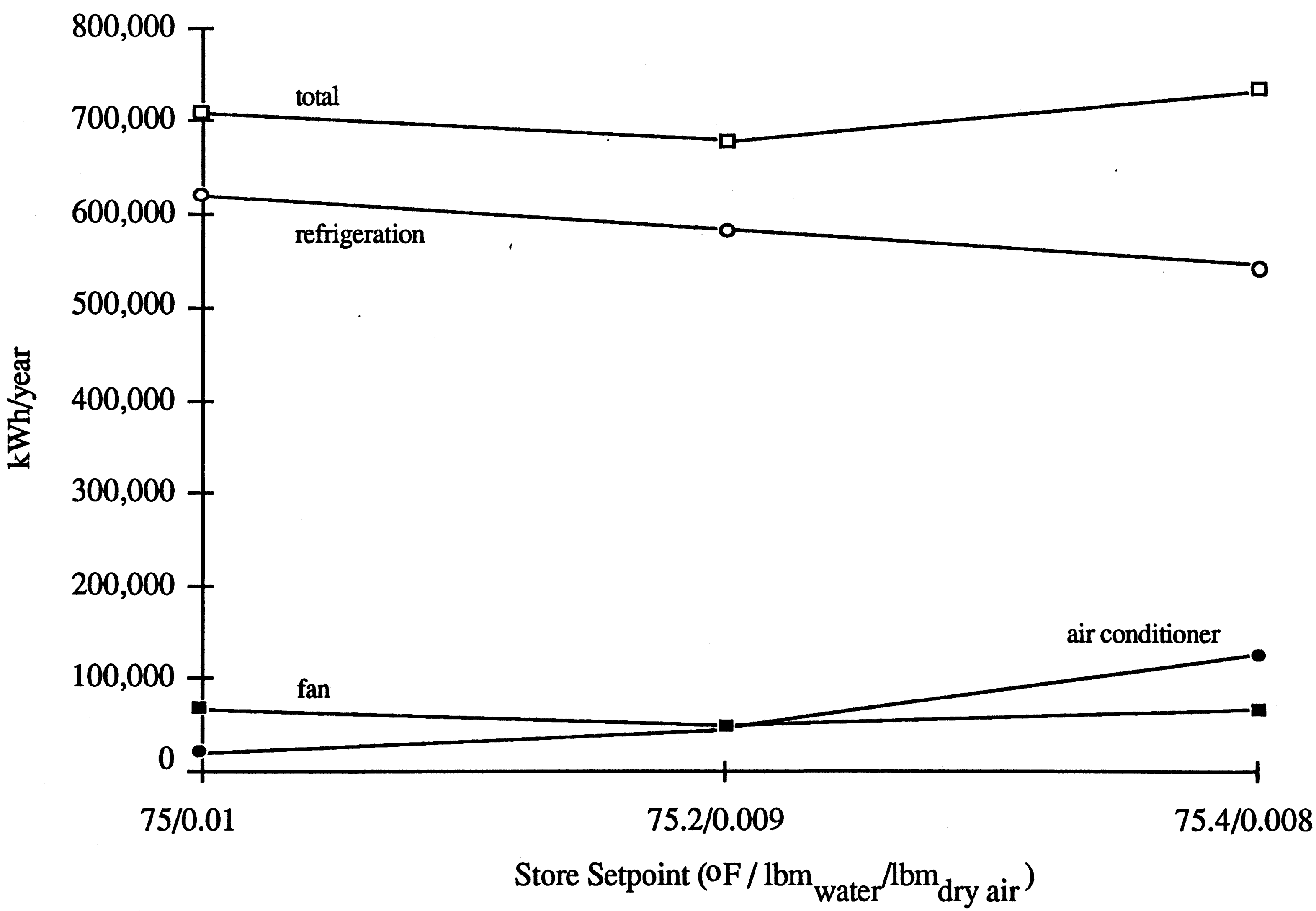

Fig. 4.2.2.3: Store Setpoint Dependence (Resized Humid-Climate System)

level of 0.009 $lbm_{water}/lbm_{dry\ air}$, trial and error showed that the off-peak ventilation rate needed to be 2400 cfm instead of 1350 cfm. For the system with a 0.008 $lbm_{water}/lbm_{dry\ air}$ setpoint, the same trial and error process was used to determine that the peak ventilation flow rate must be 5300 cfm with an off-peak rate of 4400 cfm. Higher flow rates require more energy, making the total energy consumption vs. humidity setpoint relationship more complicated than that for the conventional system. The system sized for a 0.009 $lbm_{water}/lbm_{dry\ air}$ setpoint uses 5% less total energy annually while the system sized for a 0.008 $lbm_{water}/lbm_{dry\ air}$ setpoint uses 4% more.

It should be noted that the secondary air conditioner was used only a few hours of the year in the lower setpoint humidity humid-climate systems. In an actual installation, reprocessing a small amount of return air through the primary air conditioner along with the minimum amount of ventilation air and eliminating the secondary air-conditioner would be more energy efficient than increasing the ventilation flow rate.

For the conventional system, if a lower setpoint humidity is to be used, the circulation flow rate should be reset. This would also hold true for humid climate systems modified as described above. Consideration should be given to the cost of the increased air conditioner capacities necessary to maintain lower humidity levels. The increased capacities are shown in Table 4.2.2.1.

4.2.3: Ventilation Flow Rate

Annual simulations were run for varying ventilation flow rates. Figure 4.2.3.1 displays these results, showing the energy requirements as a function of the peak hours ventilation flow rate. In all cases, the off-peak hours ventilation rate was chosen to be

	Humidity Setpoint ($lbm_{water}/lbm_{dry\ air}$)		
	0.01	0.009	0.008
Air Conditioner Type			
Conventional	495,000*	510,000	555,000
Humid Climate System:			
Primary	235,000	250,000	475,000
Secondary	215,000	200,000	135,000

*all capacities in Btu/hr

Table 4.2.2.1: Air Conditioning Unit Capacities for Humidity Setpoint Tests

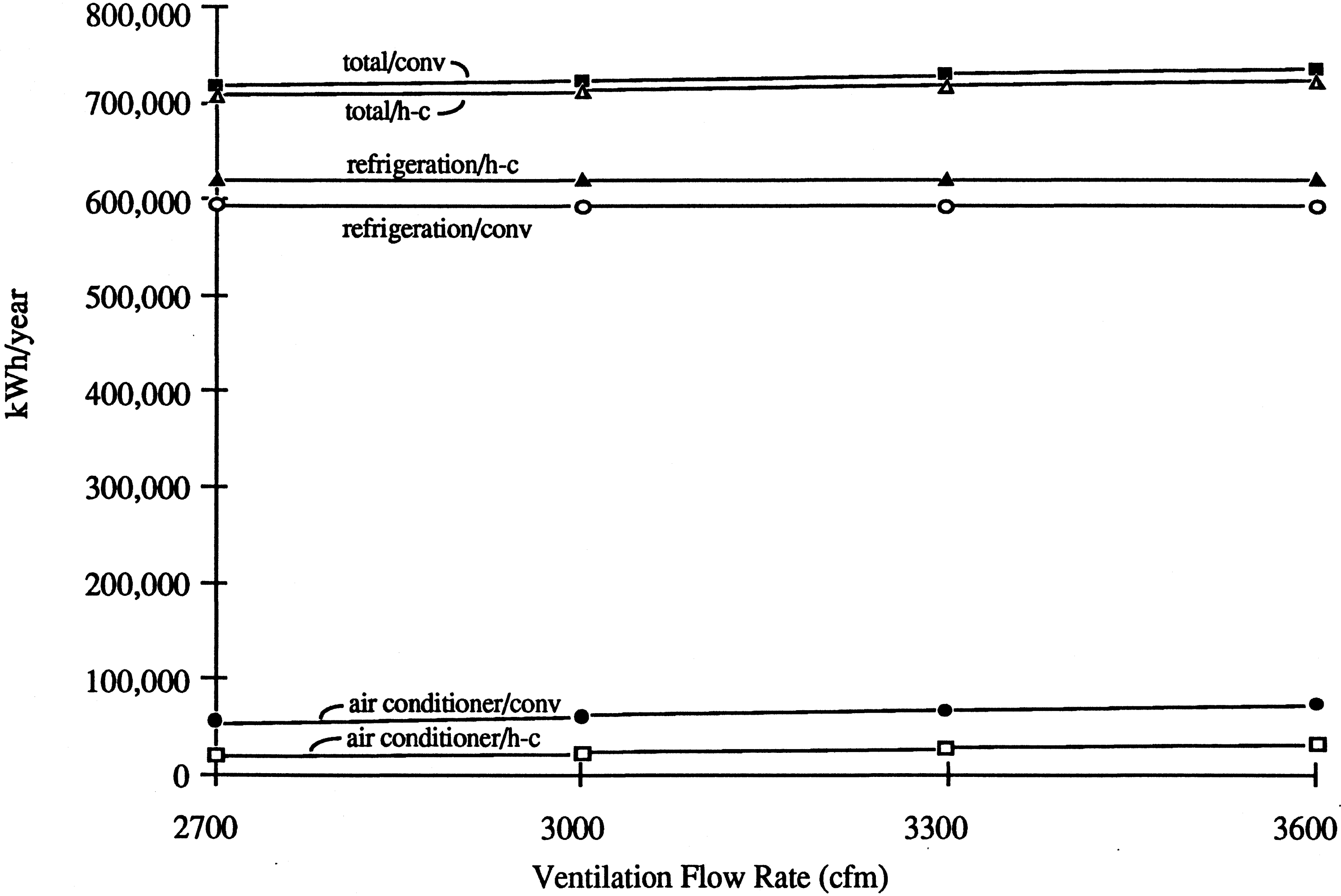

Fig. 4.2.3.1: Ventilation Flow Rate Dependence

one-half that of the peak rate. The fan power in both types of systems increased, as did the air conditioner energy requirements. The refrigeration energy requirements also increased as the result of increased annual average store humidity. For a 33% increase in ventilation flow rate, the total energy consumption increased 3% for the conventional system and 2% for the humid-climate system. The ventilation flow rate does not appear to be a critical factor, although the minimum should be used for maximum energy efficiency with either system.

4.2.4: Circulation Flow Rate

Increasing circulation flow rate has little effect on air conditioner or refrigeration energy consumption for either type of system, however fan power requirements are noticeably increased. These results are shown in Fig. 4.2.4.1. For a 44% increase in circulation flow rate, the annual energy consumption increased 22% for the conventional system and 19% for the humid-climate system. Clearly the circulation flow rate should be set as low as possible, regardless of the system used.

4.2.5: Refrigerated Case Capacity

The latent and sensible credit due to the refrigerated cases account for a substantial portion of the total cooling performed on the supermarket air. The refrigerated case capacity was varied to study the resulting effect on the energy consumption (see Fig. 4.2.5.1). As expected, the refrigeration energy consumption increases as refrigeration capacity increases. The air-conditioning energy consumption decreases for both systems, with the difference between the two systems also decreasing. This occurs because the portion of the supermarket load not met by the refrigerated cases decreases, making the differences between the two systems less

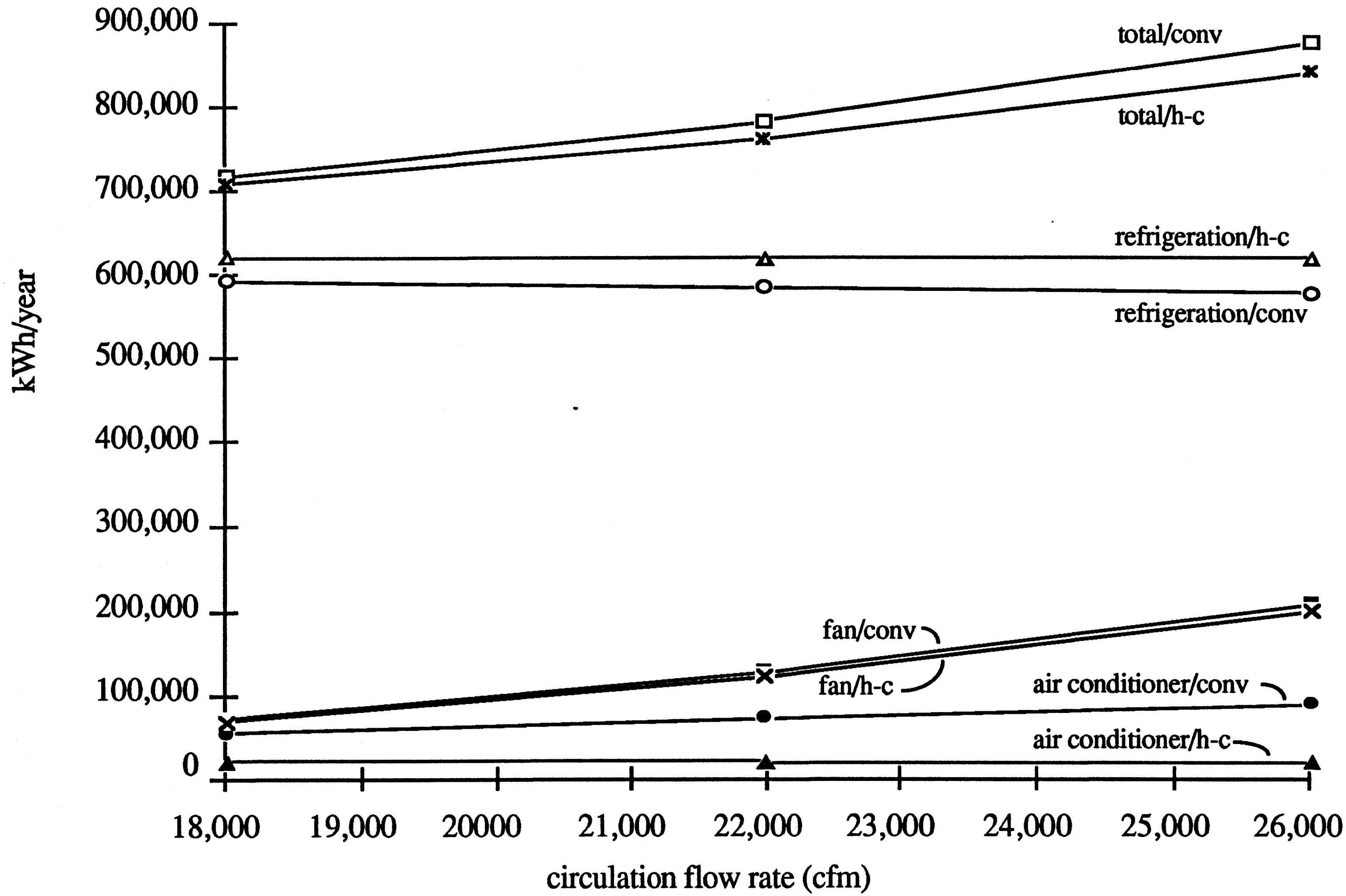

Fig. 4.2.4.1: Circulation Flow Rate Dependence

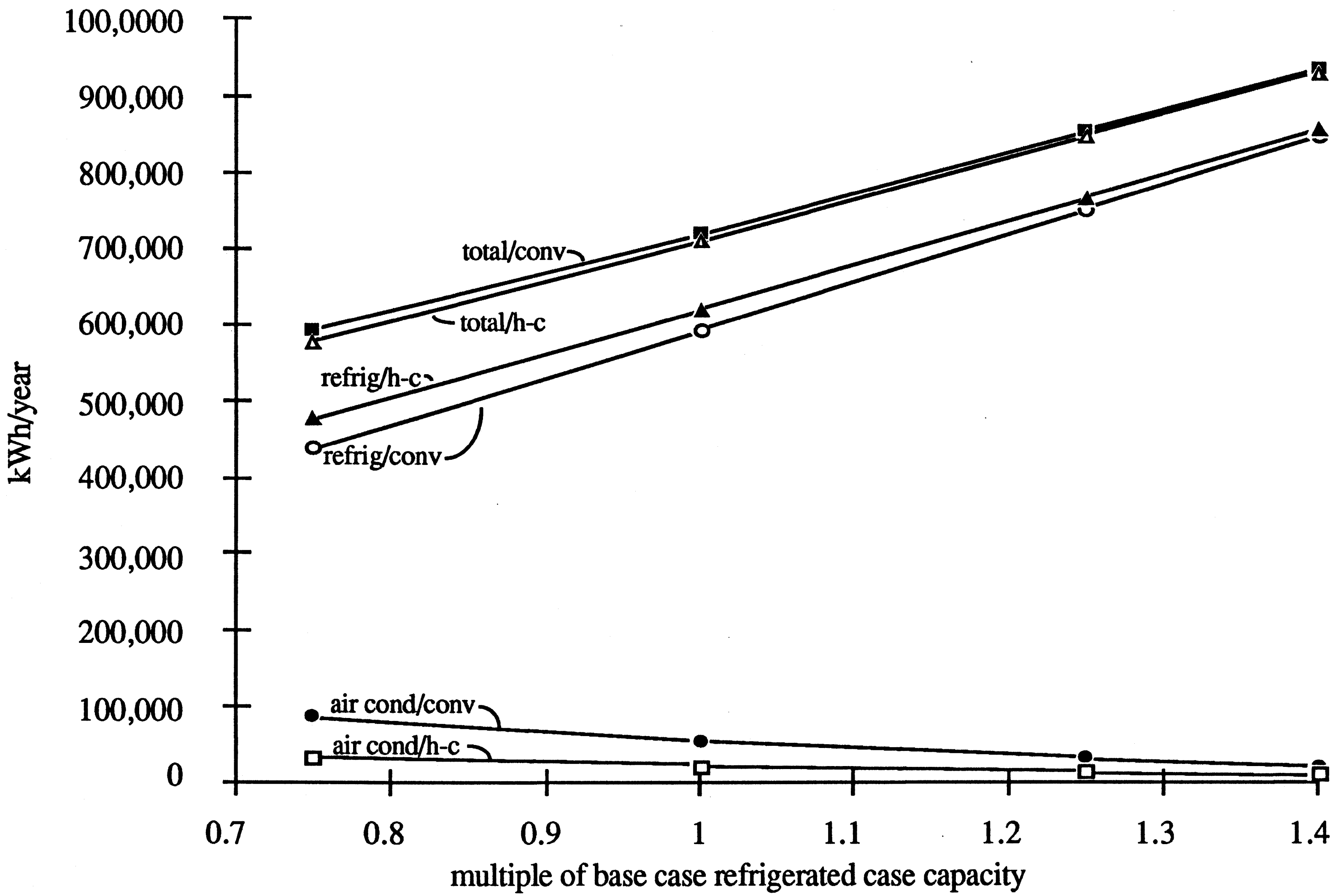

Fig. 4.2.5.1: Refrigerated Case Capacity Dependence

important. This implies that stores with very high refrigeration capacities will benefit less from humid-climate systems.

Given a refrigerated case capacity 0.5 times the base case capacity, it was not possible to cool and dry the ventilation air sufficiently in the humid-climate system to meet the store loads. This suggests that humid-climate systems may not be feasible for stores with low refrigerated case capacities.

4.2.6: Internal Sensible Load

The internal sensible load was varied by changing the lighting and equipment load. This variation also caused the sensible-to-latent-load ratio to change. The results of annual simulations using 0.5, 1 and 1.5 times the standard lighting load are displayed in Fig. 4.2.6.1. The air conditioner energy requirements increase only slightly, as the primary air conditioner is often already over-cooling in order to meet the latent load. The refrigeration energy requirements generally increase very slightly (1 - 2%) with increasing sensible load as less reheat is done (and hence the refrigerated case COP decreases). At 1.5 times the base case internal sensible load, the increase in the humid-climate system refrigeration requirements is sufficiently larger than that for the conventional system to eliminate the energy consumption difference between the systems.

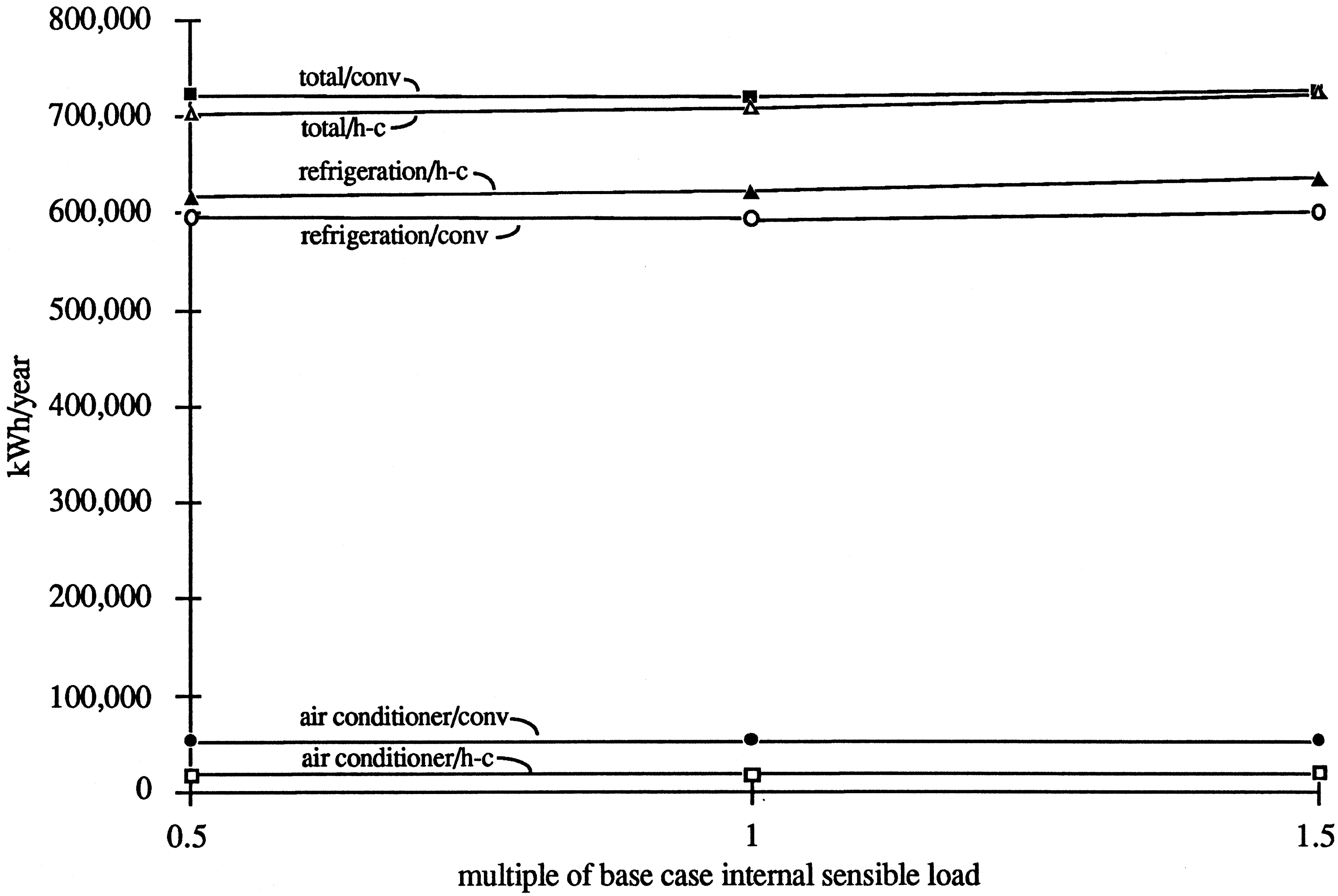

Fig. 4.2.6.1: Internal Sensible Load Dependence

CHAPTER 5: ECONOMIC CONSIDERATIONS

This chapter considers two economic aspects of supermarket air-conditioning systems: the potential for operating cost savings, and the allowable increase in first costs for an improved system given a specific payback period.

5.1: Annual Operating Cost Savings

In Miami, substituting a humid-climate vapor-compression air-conditioning system for a conventional air-conditioning system resulted in a savings of 9,700 kWh (see section 4.1.3). The peak demand (calculated hourly) was reduced by 2.7 kW (121.8 kW to 119.1 kW) or 2%. Assuming a low flat rate electricity schedule (see Table 5.1.1), this represents a \$600 annual operating cost savings. A high flat rate schedule (again see Table 5.1.1) gives a \$1,200 savings.

	Energy Charge (\$/kWh)	Demand Charge (\$/kW)
Low rate	0.04	5.50
High rate	0.076	13.00

Table 5.1.1: Flat Rate Electricity Rate Schedules (Blatt, 1987)

The GRI study reported a total energy savings of \$2,400 to \$4,900 for a hybrid-desiccant system in Miami (see section 1.2) using an electricity price of \$0.0446/kWh with a demand charge of \$6.25/kW and a gas rate of \$0.00483/ft^3. This rate schedule gives an annual savings of $600 for the base case comparison of the current study. This suggests that a hybrid-desiccant system offers a greater potential for energy savings than a humid-climate vapor-compression air-conditioning system, however it is not clear if the GRI study accounted for the effect on the refrigerated case COP of reclaiming heat from the condenser.

Fig. 5.1.1 shows the total energy consumption breakdown by hour of day for both base case systems in Miami. (As an example, the annual sums of the refrigeration, air-conditioning and fan energy consumed each day between midnight and 1 am is represented by the first pair of bars.) As there is very little difference in the energy use profiles for the systems, a time-of-use electricity schedule will not benefit one system to a greater extent than it will the other. Assuming on-peak hours from 8 am to 8 pm, rate schedule A of Table 5.1.2 gives an annual energy cost savings of $900 while rate schedule B gives a savings of $700.

	Energy Charge ($/kWh)	Demand Charge ($/kW)
Schedule A:		
on-peak	0.065	8.00
off-peak	0.05	4.00
Schedule B:		
on-peak	0.08	9.00
off-peak	0.08	0.00

Table 5.1.2: Time-of-Use Electricity Rate Schedules (Blatt, 1987)

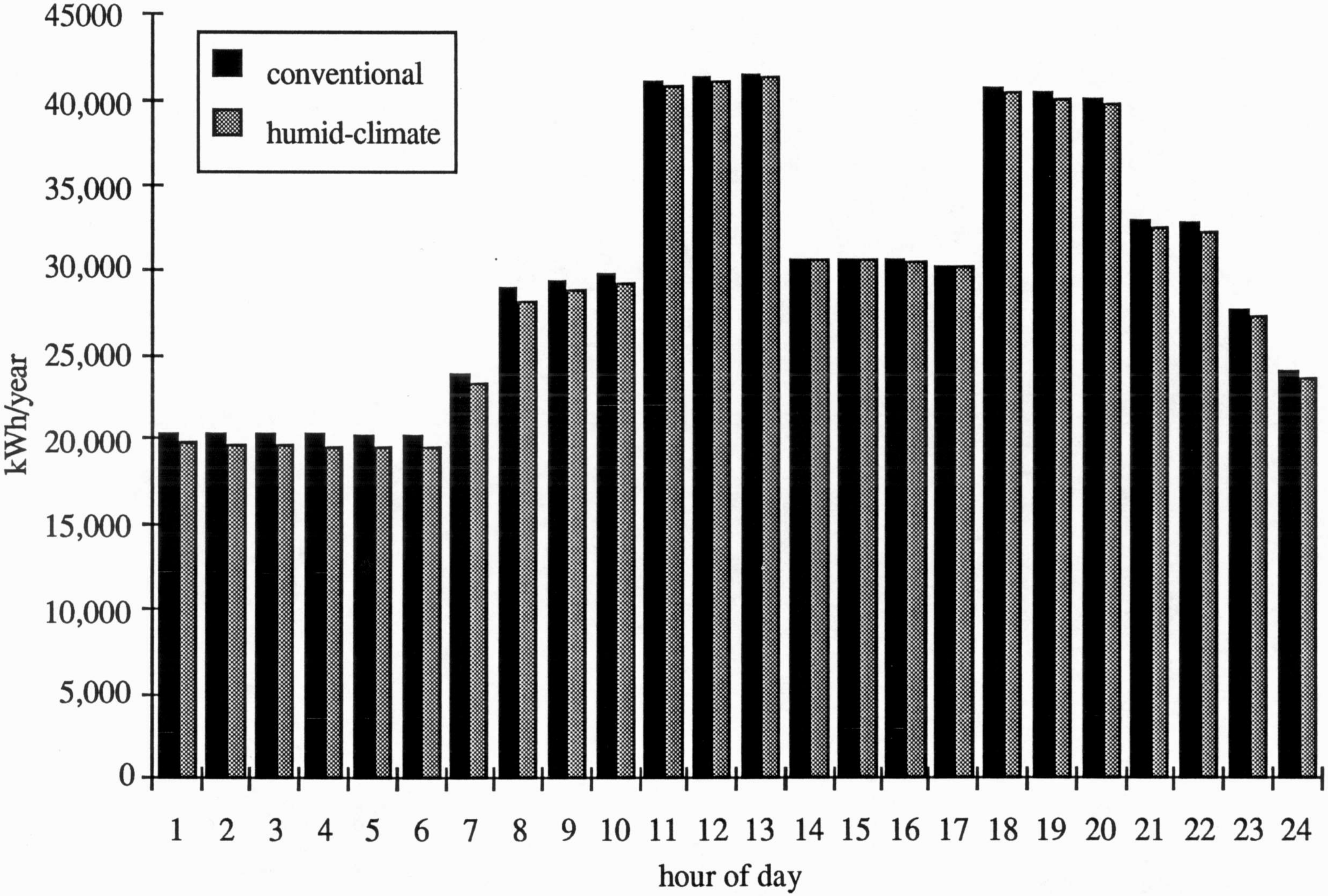

Fig. 5.1.1: Hour-of-Day Energy Consumption Breakdown

Considering only the air conditioner energy requirement (to approximate systems for which the refrigeration energy consumption does not change), the humid-climate system saves 34,200 kWh annually, or 63%. Neglecting demand charges, a low flat rate electricity rate schedule generates an annual savings of $1,400 while a high flat rate schedule produces a $2,600 annual savings. Burns found an annual savings of 50-70% reduction in annual air conditioner energy consumption for a hybrid desiccant system (see section 1.2), suggesting that a humid-climate vapor-compression air-conditioning system may provide the same magnitude of savings as a hybrid-desiccant system.

The humid-climate system in Madison, WI saved a total of 16,900 kWh per year with a 0.5 kW demand reduction. This amounts to a savings of $700 given a low flat rate electricity schedule or $1,400 for a high flat rate electricity schedule (see Table 5.1.1). Note that these savings are slightly higher than those for Miami because there is no refrigeration energy penalty for using less reclaimed heat in Madison. Considering only the air-conditioning energy consumption, and neglecting the demand charge, the annual savings of 1,800 kWh represents a decrease in operating costs of $70 to $140.

A look at the estimated annual savings with respect to the total supermarket electricity bill puts the above calculations in perspective: estimating the principle air conditioner energy consumption as 4% of the total store energy consumption (Whitehead, 1985, 439), a 63% savings is only an overall 2.5% savings.

5.2: Allowable First Costs Increment

If humid-climate air-conditioning systems are to attract the attention of supermarket owners and designers, any increment in first costs must be recuperated by annual operating savings within an acceptably short period of time. The P1, P2 life cycle savings analysis method developed by Brandemuehl and Beckman (Duffie and Beckman, 1980, 376-406) provides a convenient method of determining the allowable increment in first costs. The simplified analysis presented here assumes that no money was borrowed to purchase the system, no miscellaneous costs (such as insurance and maintenance) are involved, and that the system has no resale value after the period of analysis.

The life cycle savings for a given system (S_{LC}) in current dollars is given by the equation:

$$S_{LC} = P_1 * S_{first\ year} - P_2 * C_{equip} \quad (5.2.1)$$

where $S_{first\ year}$ is the first year operating cost savings, and C_{equip} is the initial cost of the equipment. P_1 is the ratio of the life cycle savings due to operating cost savings over the first year operating costs savings, and P_2 is the ratio of the life cycle cost due to the equipment cost over the equipment cost. Both are defined below:

$$P_1 = (1 - t_i) * F_{pw}(N_e, i_F, d) \quad (5.2.2)$$

where t_i is the effective income tax rate, F_{pw} is the present worth factor described below, N_e is the number of years in the economic analysis, i_F is the the fuel inflation rate, and d is the discount rate; and

$$P_2 = 1 + t_p * (1 - t_i) * F_{PW}(N_e, i,d) \quad (5.2.3)$$

where t_p is the property tax rate, and i is the general inflation rate. $F_{pw}(N,i,d)$ is used to calculate the present worth of N future payments given an inflation rate of i and a depreciation rate of d and is computed as:

$$F_{PW}(N,i,d) = \sum_{j=1}^{N} \frac{(1+i)}{(1+d)}^{j-1}$$

Given an acceptable payback period of N years, setting $S_{LC} = 0$ when $N_e = N$ in equ. 5.2.1 allows the calculation of the minimum acceptable increment in first costs. If the above define variables are chosen to have the values defined in Table 5.2.1, P_1 = 1.091 and P_2 = 1.033. If N_e is instead chosen to be 3, P_1 = 1.606 and P_2 = 1.048. For a payback period of 2 years, the allowable increment in first costs for Miami is $630 if the low flat rate electricity schedule of Table 5.1.1 is in effect; $1,260 if the high flat rate electricity schedule is in effect. A payback period of 3 years raises these allowable increments to $920, and $1,840, respectively. A humid-climate system can not initially cost much more than a comparable conventional system, and would preferably cost less, as the annual operating cost savings appear to be small.

Variable	Value
t_i	0.40
N_e	2 years
i_F	0.04
i	0.04
d	0.08
t_P	0.03

Table 5.2.1: Economic Analysis Parameter Values

CHAPTER 6: CONCLUSIONS AND RECOMMENDATIONS FOR FURTHER RESEARCH

A number of conclusions can be drawn from the results of this study, however these conclusions should be verified experimentally. As it was not possible to cover all of the interesting aspects of the modelling of supermarket air-conditioning systems, a number of recommendations will be made for future research. Some of these recommendations will be acted upon during the second phase of this study, taking place in the first part of 1988.

6.1: Conclusions

In Miami, the humid-climate system offered a savings in annual air-conditioning energy consumption of 63% (34,200 kWh). However, the humid-climate system required 5% more refrigeration energy than the conventional system as the result of a decrease in the amount of reheat energy required, and hence a lower average refrigerated case COP (see section 4.1). This increase in refrigeration energy consumption is nearly of the same order of magnitude as the savings in air-conditioning energy consumption. The method by which heat is reclaimed can not be neglected in designing or modelling a supermarket air-conditioning system.

As expected, humid-climate systems offer larger potential air conditioner operating cost savings in climates that have long air-conditioning seasons. However, the penalty in refrigeration energy consumption for using less reclaimed heat is smaller (or non-existent) in colder climates.

Reducing the store humidity level setpoint without redesigning the air-conditioning system raised the energy consumption in the conventional system and made the humid-climate system unusable. Recalculating the necessary circulation flow rates and air conditioner capacities when lowering the humidity setpoint produced a reduction in energy consumption for the conventional system and mixed results for the humid-climate system. For lower humidity setpoints, processing only the minimum amount of ventilation air in the humid-climate system may not be possible. Under these circumstances, consideration should be given to processing a portion of the recirculated air with the primary air conditioner instead of increasing the ventilation flow rate. It may be possible to eliminate the secondary air conditioner of a humid-climate system when store humidity levels are held low. As both types of systems require air conditioners of greater capacity when the store humidity setpoint is lowered, an additional consideration should be the increased equipment costs.

Increasing the ventilation and circulation flow rates had little effect on the required air conditioner energy, however an increased circulation rate increased fan operating costs significantly. The minimum ventilation and circulation flow rates should be used for the greatest energy efficiency. An exception to this rule might occur in cooler, drier climates, where an economizer cycle would be useful.

Increased refrigeration capacity decreased the air conditioner energy requirements, as greater portion of the store load was met by the cases. The energy savings due to a humid-climate system is smaller for stores with large refrigeration capacities, while air conditions in stores with small refrigeration capacities can not be adequately controlled with humid-climate systems.

Increasing the internal sensible load did not significantly increase the air conditioner energy consumption because the processed air is often overcooled to control the humidity. Refrigeration energy consumption increased slightly as the result of less reheat. At high sensible loads, the energy consumption difference between the two kinds of systems appears to be minimal.

For Miami, FL, annual operating cost savings due to a humid-climate system appear to be on the order of $1,000. If a payback period of 2 or 3 years is specified, a humid-climate system cannot cost more than $600 to $1,800 more than a conventional system.

6.2: Recommendations for Further Research

Several features of vapor-compression air-conditioning systems were not modelled in this study. Of these, the most interesting are an economizer cycle, which takes advantage of cool, dry outside air whenever possible, and a variable air volume system, which would circulate only the amount of air necessary to meet building loads. Figures 4.1.2.1 and 4.1.2.2 show that the required fan power is often larger than that of the air-conditioning unit, hence decreasing the volume of circulated air has the potential for significant savings.

This study assumed that heat was reclaimed from the refrigerated case condenser(s) as necessary. During annual simulation runs it was noted that the heat available from the refrigerated cases' condenser(s) was not always sufficient to meet the demand for reheat energy. The remaining reheat could be reclaimed from the air conditioner condenser. As the effect of reclaiming heat on the refrigerated case COP proved to be significant, further studies should adjust the air conditioner COP when reclaiming heat from the air conditioner condenser. Comparing different heat reclaim strategies (ie. reclaiming first from the air conditioner condenser, reclaiming first from the refrigerated cases, etc.) could prove helpful for the future design of supermarket systems.

To prevent the buildup of frost on refrigerated cases, several types of defrost cycles can be used. The refrigerated case model used in this study did not include the extra energy required to run these cycles. As less frost, and hence fewer defrost cycles, are expected with lower store humidities, including defrost cycles in the refrigerated case model may effect the savings obtained by choosing a lower store humidity setpoint. It is recommended that the defrost cycle energy be approximated, and, should it prove to be significant, the refrigerated case model modified appropriately. Another shortcoming of the current refrigerated case model is the lack of distinction between closed and open cases. Closed case performance should be less dependent on store air conditions. It is suggested that the model be modified to handle open and closed cases separately.

At least two alternative supermarket air-conditioning systems were not considered during this study. Modelling a hybrid desiccant and a thermal storage system and comparing the performance of these systems with the vapor-compression systems already modelled is recommended.

A computer simulation can not be more accurate than the experimental data on which it is based. Difficulties arose during the course of this study in obtaining performance data over a wide range of operating conditions for air conditioners and refrigerated cases. Certain store model parameter values, such as the hourly latent gain from sources other than people, were also difficult to determine. Experimental determination of these values and performance characteristics would greatly simplify the task of creating an accurate computer simulation.

Finally, the ratio of annual energy required by the air-conditioning units to that required by the refrigerated cases is 11:1 for the conventional system and 31:1 for the humid-climate system. The potential for significant energy savings through improved refrigerated case design appears to be considerable, and should be the subject of further research.

APPENDIX A: TRNSYS DECKS

-Conventional System Base Case

-Humid-Climate System Base Case

```
*this deck simulates the performance of a conventional HVAC system--1/2/88
NOLIST
SIMULATION 0 8760 1
TOLERANCE 0.000001 0.000001
LIMITS 50 5 45
WIDTH 132
************************************************************************
UNIT 12 TYPE 9 weather data reader
*read Idn, I, T, omega, wind velocity and convert to English units
*  from logical UNIT #20
PARAMETERS 19
*N deltatd ----Idn----  -----I-----  ---T------ --omega---- ----Vw-----
 5    1    -1 0.0881 0  -2 0.0881 0  -3 0.18 32 -4 0.0001 0 -5 2.2369 0
*lur frmt
  20  1
(9X,F4.0,1X,F4.0,1X,F4.0,1X,F6.0,1X,F2.0)
************************************************************************
UNIT 13 TYPE 14 ventilation schedule (cfm)
PARAMETERS 12
*---mid to 7am-  --7am to 10pm--    ---10pm to mid--
 0 1350  7 1350  7 2700  22 2700    22 1350  24 1350
************************************************************************
UNIT 14 TYPE 14 infiltration schedule (cfm)
PARAMETERS 12
*-mid to 7 am-  -7am to 10pm--  -10 pm to mid-
 0 240   7 240  7 990   22 990  22 240  24 240
************************************************************************
UNIT 15 TYPE 14 internal radiative gain (excluding people) (Btu/hr)
PARAMETERS 12
*-mid to 7 am-     -7am to 10pm--      -10 pm to mid-
0 136520 7 136520  7 204780 22 204780  22 136520  24 136520
************************************************************************
UNIT 16 TYPE 14 internal latent gain (excluding people) (lbm water)
PARAMETERS 4
*-all hours
 0 10 24 10
************************************************************************
UNIT 17 TYPE 14 number of people in store
PARAMETERS 38
*-mid to 6 -6 to 7-- -7 to 10-- -10 to 1pm--- --1 to 5--- ---5 to 8----
 0 30 6 30 6 60 7 60 7 90 10 90 10 180 13 180 13 90 17 90 17 180 20 180
*---8 to 10--- -10 to 11-- --11 to mid-----
 20 140 22 140 22 90 23 90 23 60 24 60 24 30
************************************************************************
UNIT 18 TYPE 14 refrigerator on-time (%)
PARAMETERS 38
*---mid to 6-- ----6 to 7----- ----7 to 10----- -10 to 1pm--- ----1 to 5------
 0 0.42 6 0.42 6 0.516 7 0.516 7 0.612 10 0.612 10 0.9 13 0.9 13 0.612 17 0.612
*---5 to 8---  -----8 to 10----- ---10 to 11------ -----------11 to mid-----
17 0.9 20 0.9  20 0.708 22 0.708 22 0.612 23 0.612 23 0.516 24 0.516 24 0.42
************************************************************************
UNIT 31 TYPE 33 pyschometric calculation
*calculate the dry air density of the outside air from the dry bulb temp.
* and the humidity ratio
PARAMETERS 4
4 0 2 1.0
INPUTS 2
*TDB    W
 12,3  12,4
  75  0.01
************************************************************************
```

```
UNIT 34 TYPE 33 pyschometric calculation
*calculate the dry air density of the room air from the dry bulb temp.
* and the humidity ratio
PARAMETERS 4
4 0 2 1.0
INPUTS 2
*TDB  W
 5,1  5,2
  75  0.01
************************************************************************
UNIT 32 TYPE 15 calculation of mass flow rates and diverter ratios
PARAMETERS 16
*1st output: mdotcirc=Vcirc*dens*60
*2nd output: mdotvent=Vvent*dens*60
*3rd output: mdotret=mdotcirc-mdotvent
0 0 -1 60 1 1 -3  0 0 -1 60 1 1 -3   4 -4
INPUTS 4
*Vcirc denscirc   Vvent   densvent
 0,0   34,8       13,1    31,8
 18000 0.0703     1350   0.0703
************************************************************************
UNIT 6 TYPE 22 refrigerator cases
PARAMETERS 17
504000 180000 2.7 3.1 1.0 1.86 1.07 100 140 35 -10 510 300 5 5 0 -40
INPUTS 7
*ontime  Ti  wi   mdot   Tsup Tout reheat
  18,1  5,1 5,2   32,1   28,1 12,3  28,2
  0.42  75  0.010 78840  75   75    0
************************************************************************
UNIT 22 TYPE 15
*calculation of moisture gain due to infiltration and internal loads
*calculation of sensible gain due to infiltration
*50% of latent and sensible refrigerator case credits are subtracted
PARAMETERS 37
*1st output: Vinf * dens * (Wout-Win) * 60 + mdot internal gain -
*            0.5 * latent case credit / 1061
*2nd output: Vinf * dens * (Tout - Tin) * 0.240 * 60 -
*            0.5 * sensible case credit
0 0 1 0 0 4 1  -1 60 1 0 3 -1 0.5 0 1 -1 1061 2 4 -4
-11 -12 1 0 0 4 1 -1 14.4 1 -1 0.5 0 1 4 -4
INPUTS 9
*Vinf dens  omegao omegai mdoti  cclat   Tout   Tin  ccsens
 14,1 31,8  12,4   5,2    16,1   6,5     12,3   5,1  6,4
 240  0.073 0.0164 0.01   10     114000  75     5,1  501600
************************************************************************
UNIT 24 TYPE 16 solar processor
PARAMETERS 7
5 1 199 25.78 433 -5.18 -1
INPUTS 13
*I    Idn  td1  td2 rhog betaN zetaN betaE zetaE betaS zetaS betaW zetaW
12,2 12,1 12,6 12,7  0,0  0,0   0,0   0,0   0,0   0,0   0,0   0,0   0,0
 0    0    0    0   0.2   90    180    90   270    90    0     90    90
************************************************************************
UNIT 7 TYPE 11 diverter box--to ac2
*modify to change control
PARAMETER 1
7
INPUTS 4
*Tref Wref mdotcirc mdot2/mdoti
6,2  6,3    32,1    0,0
75   0.01  78840    0
```

```
**********************************************************************
UNIT 25 TYPE 15 calculation of mdot2/mdoti for unit 2
PARAMETERS 4
0 0 2 -4
INPUTS 2
*mdotret mdotcirc
32,3     32,1
72927    78840
**********************************************************************
UNIT 10 TYPE 11 diverter box--exhaust air, and to ac1
*modify to change control
PARAMETER 1
7
INPUTS 4
*Tref Wref mdotcirc mdot2/mdoti
7,1  7,2    7,3     25,1
75   0.01  78840    0.93
**********************************************************************
UNIT 1 TYPE 11 mixing box--return air and ventilation air
PARAMETER 1
6
INPUTS 6
*Tout Wout mdotvent Tret  Wret  mdotret
12,3  12,4    32,2  10,4  10,5   10,6
 75   0.01    5913  75    0.01   72927
**********************************************************************
UNIT 2 TYPE 7 air-conditioning unit 1--sensible and latent cooling
PARAMETERS 7
0 495000 4 0.05 18000 75 0.01
INPUTS 9
*Tin Win  mdot1  Tcond Tret Wret mdot2 Qsstore Qlstore(con)
 1,1 1,2  1,3    12,3  7,4  7,5  7,6    5,16    5,19
 75  0.01 78840  75    75   0.01 0      0       0
**********************************************************************
UNIT 3 TYPE 11 mixing box--return air and ventilation air
*REVISE IF AC UNIT 2 USED
PARAMETERS 1
6
INPUTS 6
*Tac1 Wac1 mdotac1 Tac2  Wac2  mdotac2
2,1   2,2   2,3     7,4  7,5    7,6
 75   0.01  78840 75    0.01   0
**********************************************************************
UNIT 28 TYPE 15 reheat
*first output--leaving temperature
*Tset-Qsstore/(mdotcirc*0.24) or Tbefore--whichever greater
*second output--q
*(output1 - Tbefore) * 0.24 * mdotcirc
PARAMETERS 19
0 0 0 2 -1 0.24 2 4 0 12 -3
-14 4 -1 0.24 1 -13 1 -4
INPUTS 4
*Tset  Qsstore mdotcirc Tbefore
  0,0    5,16  32,1      3,1
  75     0     78840     75
**********************************************************************
UNIT 5 TYPE 19 building
PARAMETERS 14
 1 2 9600000 0 0 0 780000 7 75 0.01 75 75 0 0.01
INPUTS 11
*Tout omegaout Tv   mv    omegav ml   Npeople activity Qrad   Qint  W
12,3  12,4     28,1 32,1   3,2    22,1 17,1    0,0      15,1   22,2  12,5
```

```
75     0.0164  75   78840  0.01   10.0 30      5        136520 0     0
*north wall
PARAMETERS 27
1 1 2771.3 0.5 0.7 4 1.4556
6 6 5
0.00000 0.00089 0.00558 0.00487 0.00078 0.00002
0.90331 -1.73552 1.06692 -0.23867 0.01628 -0.00017
-1.44430 0.67485 -0.11916 0.00659 -0.00005
INPUTS 1
24,6
*0,0
0
*east wall
PARAMETERS 3
2 -1 2771.3
INPUTS 1
24,11
*0,0
0
*south wall
PARAMETERS 3
3 -1 1939.9
INPUTS 1
24,14
*0,0
0
*west wall
PARAMETERS 3
4 -1 2771.3
INPUTS 1
24,17
*0,0
0
*roof
PARAMETERS 7
6 1 30000 0.5 0.7 1 17
INPUTS 1
24,4
*0,0
0
*floor
PARAMETERS 34
7 3 30000 0.5 0.7 4 1.4556
7 9 8
 0.00000  0.00000  0.00000  0.00000  0.00001  0.00001  0.00001
 0.17219 -0.64834  0.99020 -0.79035  0.35502 -0.09050  0.01271 -0.00092 0.00003
-3.22145  4.10406 -2.64277  0.91566 -0.16939  0.01590 -0.00068  0.00001
INPUTS 3
0,0 0,0 0,0
75  75  0
*south wall window
PARAMETERS 8
5 5 831.4 1 0.8 1.4556 1 7
INPUTS 5
24,14 24,15 0,0  0,0    0,0
*0,0 0,0  0,0  0,0    0,0
   0    0   0.87 14.16   1
*view factors-4 walls, floor, roof
PARAMETERS 11
1 16 173.21 173.21 1 2 3 4 7 6 1
*view factor for window
```

```
PARAMETERS 6
5 3 45 3 10 83.14
******************************************************************************
UNIT 48 TYPE 15 calculate hourly total power consumption (in kW's) for hist.
PARAMETERS 7
0  0 -1 3413 2  3  -4
INPUTS 2
*KWhacl Bturefrig
2,4     6,1
0       0
******************************************************************************
UNIT 46 TYPE 28 monthly summary printer--integrated quantities, unless specified
PARAMETERS 37
-1 0 8760 21 2
0 -4     0 -4    0 -1 3413 2 -4   0 -4   0 -1 3413 2 -4  0 -2 2 -4   0 -2 2 -4
0 -2 2 -4   0 -2 2 -4
INPUTS 9
*Qheater KWhacl Bturefrig KWtot Btuas Toutavg  Woutavg Tstavg Wstavg
28,2     2,4    6,1       48,1  6,6   12,3       12,4    5,1    5,2
LABELS 9
Qheater KWhacl  Kwhrefrig KWtot KWas  Toutavg  Woutavg Tstavg Wstavg
******************************************************************************
UNIT 36 TYPE 2 sums, maximums by hour of day--hours 1 through 8
PARAMETERS 2
1 8
INPUTS 1
48,1
0
******************************************************************************
UNIT 37 TYPE 2 sums, maximums by hour of day--hours 9 through 16
PARAMETERS 2
9 16
INPUTS 1
48,1
0
******************************************************************************
UNIT 38 TYPE 2 sums, maximums by hour of day--hours 17 through 24
PARAMETERS 2
17 24
INPUTS 1
48,1
0
******************************************************************************
*UNIT 30 TYPE 25 printout of hour of day sums and maximums
*PARAMETERS 3
*-1 0 8760
*INPUTS  10
*36,1 36,2 36,3 36,4 36,5 36,6 36,7 36,8 36,9 36,10
*LABELS 10
*1sum 1max 2sum 2max 3sum 3max 4sum 4max 5sum 5max
******************************************************************************
*UNIT 39 TYPE 25 printout of hour of day sums and maximums
*PARAMETERS 3
*-1 0 8760
*INPUTS  10
*36,11 36,12 36,13 36,14 36,15 36,16 37,1 37,2 37,3 37,4
*LABELS 10
*6sum 6max 7sum 7max 8sum 8max 9sum 9max 10sum 10max
******************************************************************************
*UNIT 49 TYPE 25 printout of hour of day sums and maximums
*PARAMETERS 3
*-1 0 8760
```

```
*INPUTS  10
*37,5 37,6 37,7 37,8 37,9 37,10 37,11 37,12 37,13 37,14
*LABELS 10
*11sum 11max 12sum 12max 13sum 13max 14sum 14max 15sum 15max
********************************************************************************
*UNIT 50 TYPE 25 printout of hour of day sums and maximums
*PARAMETERS 3
*-1 0 8760
*INPUTS  10
*37,15 37,16 38,1 38,2 38,3 38,4 38,5 38,6 38,7 38,8
*LABELS 10
*16sum 16max 17sum 17max 18sum 18max 19sum 19max 20sum 20max
********************************************************************************
*UNIT 29 TYPE 25 printout of hour of day sums and maximums
*PARAMETERS 4
*-1 0 8760 21
*INPUTS 8
*38,9 38,10 38,11 38,12 38,13 38,14 38,15 38,16
*LABELS 8
*21sum 21max 22sum 22max 23sum 23max 24sum 24max
********************************************************************************
UNIT 47 TYPE 27  histogram--Qheater, plr1, Qsstore, Qlstore
PARAMETERS 18
1 -12 -12 0 8760
0 1000000 10
0 2 20
-100 100 20
-100 100 20
21
INPUTS 4
*Qheater plr1 Qsstore Qlstore
 28,2    2,7  5,7     5,8
 0       0    0       0
********************************************************************************
*UNIT 40 TYPE 28 summary printer--for debugging
*PARAMETERS 45
**1 5209 5233 21 2
*1 360 384 21 2
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4
*INPUTS 10
**Qs   Ql   Tenst Wenst Tst Wst   Tinacl  Winac Texacl Wexacl
* 5,7  5,8  28,1  3,2   5,1 5,2   1,1     1,2   2,1    2,2
*LABELS 10
*Qs Ql Tenst Wenst Tst Wst Tinacl Winacl Texacl Wexacl
********************************************************************************
*UNIT 41 TYPE 28 summary printer--for debugging
*PARAMETERS 13
**1 5209 5233 21 2
*1 360 384 21 2
*0 -2 2 -4  0 -2 2 -4
*INPUTS 2
**KWhacl  plr1
* 2,4     2,7
*LABELS 2
*KWhacl  plr1
********************************************************************************
*UNIT 42 TYPE 28 summary printer--for debugging
*PARAMETERS 25
**1 5209 5233 21 2
*1 360 384 21 2
```

```
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4
*INPUTS 5
**Tinheat Winheat Toutheat Texref  Wexref
*  3,1     3,2     28,1     6,2    6,3
*LABELS 5
* Tinheat Winheat Toutheat Texref  Wexref
********************************************************************************
*UNIT 44 TYPE 15 calculate energy added by dumping/adding ventilation air
**first output (Tout - Tref) * mdotvent * 0.24
**second output (Wout - Wref) * mdotvent * 1061
*PARAMETERS 18
*0 0 4 0 1 -1 0.24 1 -4
*0 0 4 -13 1 -1 1061 1 -4
*INPUTS 5
**Tout Tref  mdotvent Wout Wref
*12,3  6,2    32,2    12,4 6,3
*  75  75     5913    0.01 0.01
********************************************************************************
*UNIT 45 TYPE 28 summary printer-does energy balances
*PARAMETERS 28
**1 5209 5233 21 2
*1 360 384 21 2
**1 0 24 21 2
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -1 2 2 -4  0 -2 2 -4
*INPUTS 5
**Qsstore Qheater Qsacl scc svent
* 5,16    28,2    2,5   6,4 44,1
*LABELS 5
*Qsstore Qheater Qsacl  scc svent
*CHECK 0.2 1 2 -3 -4 5
********************************************************************************
*UNIT 43 TYPE 28 summary printer-does energy balances
*PARAMETERS 28
**1 5209 5233 21 2
*1 360 384 21 2
**1 0 24 21 2
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -1 2 2 -4  0 -2 2 -4  0 -2 2 -4
*INPUTS 5
**Qlstore Qlacl lcc lvent deltacapw
* 5,17    2,6   6,5 44,2  5,18
*LABELS 5
*Qlstore Qlacl  lcc lvent deltacapw
*CHECK 0.2 1 -2 -3  4  -5
********************************************************************************
END
```

```
*this deck simulates the performance of a humid-climate HVAC system--1/2/88
NOLIST
SIMULATION 0 8760 1
TOLERANCE 0.000001 0.000001
LIMITS 50 5 45
WIDTH 132
*************************************************************************
UNIT 12 TYPE 9 weather data reader
*read Idn, I, T, omega, wind velocity and convert to English units
*  from logical UNIT #20
PARAMETERS 19
*N deltatd ----Idn----  -----I-----  ---T------ --omega---- ----Vw-----
 5    1    -1 0.0881 0  -2 0.0881 0  -3 0.18 32 -4 0.0001 0 -5 2.2369 0
*lur frmt
  20  1
(9X,F4.0,1X,F4.0,1X,F4.0,1X,F6.0,1X,F2.0)
*************************************************************************
UNIT 13 TYPE 14 ventilation schedule (cfm)
PARAMETERS 12
*---mid to 7am-  --7am to 10pm--    ---10pm to mid--
 0 1350  7 1350  7 2700  22 2700    22 1350  24 1350
*************************************************************************
UNIT 14 TYPE 14 infiltration schedule (cfm)
PARAMETERS 12
*-mid to 7 am-  -7am to 10pm--  -10 pm to mid-
 0 240   7 240  7 990   22 990  22 240  24 240
*************************************************************************
UNIT 15 TYPE 14 internal radiative gain (excluding people) (Btu/hr)
PARAMETERS 12
*---mid to 7 am---  ---7am to 10pm----   ----10 pm to mid---
 0 136520 7 136520  7 204780 22 204780   22 136520 24 136520
*************************************************************************
UNIT 16 TYPE 14 internal latent gain (excluding people) (lbm water)
PARAMETERS 4
*-all hours
 0 10 24 10
*************************************************************************
UNIT 17 TYPE 14 number of people in store
PARAMETERS 38
*-mid to 6 -6 to 7-- -7 to 10-- -10 to 1pm--- --1 to 5--- ---5 to 8----
 0 30 6 30 6 60 7 60 7 90 10 90 10 180 13 180 13 90 17 90 17 180 20 180
*---8 to 10--- -10 to 11-- --11 to mid-----
 20 140 22 140 22 90 23 90 23 60 24 60 24 30
*************************************************************************
UNIT 18 TYPE 14 refrigerator on-time (%)
PARAMETERS 38
*---mid to 6-- ----6 to 7----- ----7 to 10----- -10 to 1pm--- ----1 to 5-------
 0 0.42 6 0.42 6 0.516 7 0.516 7 0.612 10 0.612 10 0.9 13 0.9 13 0.612 17 0.612
*---5 to 8---  -----8 to 10----- ---10 to 11------ -----------11 to mid-----
17 0.9 20 0.9  20 0.708 22 0.708 22 0.612 23 0.612 23 0.516 24 0.516 24 0.42
*************************************************************************
UNIT 31 TYPE 33 pyschometric calculation
*calculate the dry air density of the outside air from the dry bulb temp.
* and the humidity ratio
PARAMETERS 4
4 0 2 1.0
INPUTS 2
*TDB    W
 12,3  12,4
  75  0.01
*************************************************************************
```

```
UNIT 34 TYPE 33 pyschometric calculation
*calculate the dry air density of the room air from the dry bulb temp.
* and the humidity ratio
PARAMETERS 4
4 0 2 1.0
INPUTS 2
*TDB  W
 5,1  5,2
  75  0.01
***********************************************************************
UNIT 32 TYPE 15 calculation of mass flow-rates
PARAMETERS 16
*1st output:  mdotcirc=Vcirc*dens*60
*2nd output:  mdotvent=Vvent*dens*60
*3rd output:  mdotret=mdotcirc-mdotvent
0 0 -1 60 1 1 -3    0 0 -1 60 1 1 -3    4 -4
INPUTS 4
*Vcirc  denscirc  Vvent  densvent
 0,0    34,8      13,1   31,8
 18000  0.0703    1350   0.0703
***********************************************************************
UNIT 6 TYPE 22 refrigerator cases
PARAMETERS 17
504000 180000 2.7 3.1 1.0 1.86 1.07 100 140 35 -10 510 300 5 5 0 -40
INPUTS 7
*ontime  Ti   wi    mdot    Ts    Tout  Qr
  18,1   5,1  5,2   32,1    28,1 12,3  28,2
  0.42   75   0.010 78840   75    75    0
************************************************************************
UNIT 22 TYPE 15
*calculation of moisture gain due to infiltration and internal loads
*calculation of sensible gain due to infiltration
*50% of latent and sensible refrigerator case credits are subtracted
PARAMETERS 37
*1st output: Vinf * dens * (Wout-Win) * 60 + mdot internal gain -
*            0.5 * latent case credit / 1061
*2nd output: Vinf * dens * (Tout - Tin) * 0.240 * 60 -
*            0.5 * sensible case credit
0 0 1 0 0 4 1  -1 60 1 0 3 -1 0.5 0 1 -1 1061 2 4 -4
-11 -12 1 0 0 4 1 -1 14.4 1 -1 0.5 0 1 4 -4
INPUTS 9
*Vinf dens  omegao omegai mdoti  cclat   Tout   Tin   ccsens
 14,1 31,8  12,4   5,2    16,1   6,5     12,3   5,1   6,4
 240  0.073 0.0164 0.01   10     114000  75     5,1   501600
***********************************************************************
UNIT 24 TYPE 16 solar processor
PARAMETERS 7
5 1 199 25.78 433 -5.18 -1
INPUTS 13
*I    Idn  td1  td2 rhog betaN zetaN betaE zetaE betaS zetaS betaW zetaW
12,2 12,1 12,6 12,7  0,0  0,0   0,0   0,0   0,0   0,0   0,0   0,0   0,0
 0     0   0     0  0.2   90    180    90   270    90     0    90    90
***********************************************************************
UNIT 25 TYPE 15 calculation of mdot2/mdoti for unit 7
PARAMETERS 4
0 0 2 -4
INPUTS 2
*mdotret mdotcirc
32,3     32,1
72927    78840
************************************************************************
```

```
UNIT 7 TYPE 11 diverter box--to ac2
*modify to change control
PARAMETER 1
7
INPUTS 4
*Tref Wref mdotcirc  mdot2/mdoti
6,2  6,3   32,1      25,1
75   0.01  78840    0.93
*************************************************************************
UNIT 1 TYPE 11 mixing box--return air and ventilation air
PARAMETER 1
6
INPUTS 6
*Tout Wout mdotvent Tret  Wret  mdotret
12,3  12,4    32,2   0,0  0,0   0,0
 75   0.01    5913   75   0.01  0
*************************************************************************
UNIT 2 TYPE 7 air-conditioning unit 1--sensible and latent cooling
PARAMETERS 7
1 235000 4 0.05 2700 75 0.01
INPUTS 9
*Tin Win  mdot1  Tcond Tret Wret mdot2 Qsstore Qlstore
 1,1 1,2  1,3    12,3  7,4  7,5  7,6    5,16    5,19
 75  0.01 5913   75    75   0.01 72927  0       0
*************************************************************************
UNIT 8 TYPE 10 air-conditioning unit 2--sensible cooling
PARAMETERS 5
215000 4 0.05 15300 75
INPUTS 7
*Tin Win  mdot2   Tcond Tac1ex mdot1 Qsstore
 7,4 7,5  7,6     12,3  2,1    2,3   5,16
 75  0.01 72927   75    75     5913  0
*************************************************************************
UNIT 3 TYPE 11 mixing box--return air and ventilation air
PARAMETERS 1
6
INPUTS 6
*Tac1 Wac1 mdotac1 Tac2  Wac2  mdotac2
2,1   2,2   2,3    8,1  8,2    8,3
 75   0.01  5913   75   0.01   72927
*************************************************************************
UNIT 28 TYPE 15 reheat
*first output--leaving temperature
*Tset-Qsstore/(mdotcirc*0.24) or Tbefore--whichever greater
*second output--q
*(output1 - Tbefore) * 0.24 * mdotcirc
PARAMETERS 19
0 0 0 2 -1 0.24 2 4 0 12 -3
-14 4 -1 0.24 1 -13 1 -4
INPUTS 4
*Tset  Qsstore mdotcirc Tbefore
  0,0    5,16  32,1     3,1
  75     0     78840    75
*************************************************************************
UNIT 5 TYPE 19 building
PARAMETERS 14
 1 2 9600000 0 0 0 780000 7 75 0.01 75 75 0 0.01
INPUTS 11
*Tout omegaout Tv   mv    omegav  ml   Npeople activity Qrad   Qint  W
12,3  12,4     28,1 32,1   3,2    22,1 17,1    0,0      15,1   22,2  12,5
75     0.0164  75   78840  0.01   10.0 30      5        136520 0     0
*north wall
```

```
PARAMETERS 27
1 1 2771.3 0.5 0.7 4 1.4556
6 6 5
0.00000 0.00089 0.00558 0.00487 0.00078 0.00002
0.90331 -1.73552 1.06692 -0.23867 0.01628 -0.00017
-1.44430 0.67485 -0.11916 0.00659 -0.00005
INPUTS 1
24,6
*0,0
0
*east wall
PARAMETERS 3
2 -1 2771.3
INPUTS 1
24,11
*0,0
0
*south wall
PARAMETERS 3
3 -1 1939.9
INPUTS 1
24,14
*0,0
0
*west wall
PARAMETERS 3
4 -1 2771.3
INPUTS 1
24,17
*0,0
0
*roof
PARAMETERS 7
6 1 30000 0.5 0.7 1 17
INPUTS 1
24,4
*0,0
0
*floor
PARAMETERS 34
7 3 30000 0.5 0.7 4 1.4556
7 9 8
 0.00000  0.00000  0.00000  0.00000  0.00001  0.00001  0.00001
 0.17219 -0.64834  0.99020 -0.79035  0.35502 -0.09050  0.01271 -0.00092 0.00003
-3.22145  4.10406 -2.64277  0.91566 -0.16939  0.01590 -0.00068  0.00001
INPUTS 3
0,0 0,0 0,0
75  75  0
*south wall window
PARAMETERS 8
5 5 831.4 1 0.8 1.4556 1 7
INPUTS 5
24,14 24,15 0,0  0,0    0,0
*0,0 0,0  0,0  0,0    0,0
   0    0   0.87 14.16   1
*view factors-4 walls, floor, roof
PARAMETERS 11
1 16 173.21 173.21 1 2 3 4 7 6 1
*view factor for window
PARAMETERS 6
5 3 45 3 10 83.14
```

```
******************************************************************************
UNIT 48 TYPE 15 calculate hourly total power consumption (in kW's) for hist.
PARAMETERS 9
0  0 -1 3413 2  3  0 3  -4
INPUTS 3
*KWhac1 Bturefrig KWhac2
2,4     6,1       8,4
0       0         0
******************************************************************************
UNIT 46 TYPE 28 monthly summary printer--integrated quantities, unless specified
PARAMETERS 39
-1 0 8760 21 2
0 -4     0 -4   0 -4    0 -1 3413 2 -4    0 -4   0 -1 3413 2 -4    0 -2 2 -4
0 -2 2 -4  0 -2 2 -4   0 -2 2 -4
INPUTS 10
*Qheater KWhac1 KWhac2  Bturefrig KWtot Btuas Toutavg Woutavg Tstavg Wstavg
28,2     2,4    8,4     6,1       48,1 6,6   12,3    12,4    5,1    5,2
LABELS 10
Qheater KWhac1  KWhac2  Kwhrefrig KWtot KWas Toutavg Woutavg Tstavg Wstavg
******************************************************************************
UNIT 36 TYPE 2 sums, maximums by hour of day--hours 1 through 8
PARAMETERS 2
1 8
INPUTS 1
48,1
0
******************************************************************************
UNIT 37 TYPE 2 sums, maximums by hour of day--hours 9 through 16
PARAMETERS 2
9 16
INPUTS 1
48,1
0
******************************************************************************
UNIT 38 TYPE 2 sums, maximums by hour of day--hours 17 through 24
PARAMETERS 2
17 24
INPUTS 1
48,1
0
******************************************************************************
*UNIT 30 TYPE 25 printout of hour of day sums and maximums
*PARAMETERS 4
*-1 0 8760 21
*INPUTS  10
*36,1 36,2 36,3 36,4 36,5 36,6 36,7 36,8 36,9 36,10
*LABELS 10
*1sum 1max 2sum 2max 3sum 3max 4sum 4max 5sum 5max
******************************************************************************
*UNIT 39 TYPE 25 printout of hour of day sums and maximums
*PARAMETERS 4
*-1 0 8760 21
*INPUTS  10
*36,11 36,12 36,13 36,14 36,15 36,16 37,1 37,2 37,3 37,4
*LABELS 10
*6sum 6max 7sum 7max 8sum 8max 9sum 9max 10sum 10max
******************************************************************************
*UNIT 49 TYPE 25 printout of hour of day sums and maximums
*PARAMETERS 4
*-1 0 8760 21
*INPUTS  10
*37,5 37,6 37,7 37,8 37,9 37,10 37,11 37,12 37,13 37,14
```

```
*LABELS 10
*11sum 11max 12sum 12max 13sum 13max 14sum 14max 15sum 15max
********************************************************************************
*UNIT 50 TYPE 25 printout of hour of day sums and maximums
*PARAMETERS 4
*-1 0 8760 21
*INPUTS  10
*37,15 37,16 38,1 38,2 38,3 38,4 38,5 38,6 38,7 38,8
*LABELS 10
*16sum 16max 17sum 17max 18sum 18max 19sum 19max 20sum 20max
********************************************************************************
*UNIT 29 TYPE 25 printout of hour of day sums and maximums
*PARAMETERS 4
*-1 0 8760 21
*INPUTS 8
*38,9 38,10 38,11 38,12 38,13 38,14 38,15 38,16
*LABELS 8
*21sum 21max 22sum 22max 23sum 23max 24sum 24max
********************************************************************************
UNIT 47 TYPE 27 histogram--Qheater, plr1, plr2, Qsstore, Qlstore
PARAMETERS 21
1 -12 -12 0 8760
0 1000000 10
0 1 20
0 1 20
-100 100 20
-100 100 20
21
INPUTS 5
*Qheater plr1 plr2 Qsstore Qlstore
28,2     2,7  8,7   5,7      5,8
0        0    0     0        0
********************************************************************************
*UNIT 40 TYPE 28 summary printer--for debugging
*PARAMETERS 45
**1 5209 5233 21 2
*1 360 384 21 2
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4
*INPUTS 10
**Qs   Ql   Tenst Wenst Tst Wst   Tinac1  Winac Texac1 Wexac1
* 5,7  5,8  28,1  3,2   5,1 5,2   1,1     1,2   2,1    2,2
*LABELS 10
*Qs Ql Tenst Wenst Tst Wst Tinac1 Winac1 Texac1 Wexac1
********************************************************************************
*UNIT 41 TYPE 28 summary printer--for debugging
*PARAMETERS 37
**1 5209 5233 21 2
*1 360 384 21 2
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -4
*INPUTS 8
**KWhac1  plr1    Tinac2  Winac2  Texac2  Wexac2  KWhac2 plr2
* 2,4     2,7     7,4     7,5      8,1    8,2     8,4    8,7
*LABELS 8
* KWhac1  plr1    Tinac2  Winac2  Texac2  Wexac2  KWhac2 plr2
********************************************************************************
*UNIT 42 TYPE 28 summary printer--for debugging
*PARAMETERS 25
**1 5209 5233 21 2
*1 360 384 21 2
```

```
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4
*INPUTS 5
**Tinheat Winheat Toutheat Texref  Wexref
*  3,1     3,2     28,1     6,2    6,3
*LABELS 5
*Tinheat Winheat Toutheat Texref  Wexref
********************************************************************************
*UNIT 44 TYPE 15 calculate energy added by dumping/adding ventilation air
**first output (Tout - Tref) * mdotvent * 0.24
**second output (Wout - Wref) * mdotvent * 1061
*PARAMETERS 18
*0 0 4 0 1 -1 0.24 1 -4
*0 0 4 -13 1 -1 1061 1 -4
*INPUTS 5
**Tout Tstore mdotvent Wout Wstore
*12,3  6,2    32,2     12,4 6,3
*  75  75     5203     0.01 0.01
********************************************************************************
*UNIT 45 TYPE 28 summary printer-does energy balances
*PARAMETERS 32
**1 5209 5233 21 2
*1 360 384 21 2
**1 0 24 21 2
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -1 2 2 -4  0 -2 2 -4
*INPUTS 6
**Qsstore Qheater Qsac1 Qsac2 scc svent
* 5,16    28,2    2,5   8,5   6,4 44,1
*LABELS 6
*Qsstore Qheater Qsac1 Qsac2 scc svent
*CHECK 0.2 1 2 -3 -4 -5 6
********************************************************************************
*UNIT 43 TYPE 28 summary printer-does energy balances
*PARAMETERS 32
**1 5209 5233 21 2
*1 360 384 21 2
**1 0 24 21 2
*0 -2 2 -4  0 -2 2 -4  0 -2 2 -4  0 -2 2 -1 2 2 -4  0 -2 2 -4  0 -2 2 -4
*INPUTS 6
**Qlstore Qlac1 Qlac2 lcc lvent deltacapw
* 5,17    2,6   8,6   6,5 44,2  5,18
*LABELS 6
*Qlstore Qlac1  Qlac2 lcc lvent deltacapw
*CHECK 0.2 1 -2 -3 -4 5 -6
********************************************************************************
END .
```

APPENDIX B: FORTRAN CODE

-TYPE 2: Hour-of-day Sum and Maximum Finder

-TYPE 7: Air Conditioner Unit (Sensible and Latent) Model

-TYPE 10: Air Conditioner Unit (Sensible-only) Model

-TYPE 11: Flow Diverter, Mixer or T-Piece Model (Improved)

-TYPE 19: Single Zone Model (Extracts)

-TYPE 22: Refrigerated Case Model

-TYPE 33: Psychrometric Calculator (Improved)

```
      SUBROUTINE TYPE2(TIME,XIN,OUT,T,DTDT,PAR,INFO)
c***************************************************************************
c                                                                          *
c     This subroutine looks at a given variable each specified hour of the *
c     day (up to 8 continuous hours) keeping track of the maximum and      *
c     calculating the monthly sum.  This information is then available to *
c     be printed out by TYPE25.  All maximums and sums are cleared monthly. *
c     This routine was designed to allow the calculation of electricity    *
c     cost given demand charges and/or time-of-use rate schedules.         *
c     (REU -- January 1988)                                                *
c                                                                          *
c***************************************************************************

      implicit none          !don't allow default definitions

c     define variables used in this component
      integer hourstart      !first hour of day to consider
      integer hourend        !last hour of day to consider
      integer numhours       !number of hours of day being considered
      integer acthour        !current hour of the day
      integer runmempnt      !points to memory spot at beginning
                             ! of memory devoted to running sums and maximums
      integer temmempnt      !points to memory spot devoted to temporary
                             ! storage of input (input stored here until last
                             ! iteration of time step performed)
      integer i              !loop counter
      integer offset         !memory offset for given hour

c     set up the usual TRNSYS variables, including storage space
      real time,dtdt,t
      real xin(10),out(20),par(15),s
      integer info(10),nstore,iav
      common/store/nstore,iav,s(5000)

c     grab the beginning and ending hours, and calculate the number of
c       hours being considered
      hourstart = par(1)
      hourend = par(2)
      numhours = hourend - hourstart + 1

c     Do the usual TRNSYS stuff if this is the first call:
c       Force an error message if more than 8 hours requested or if
c       the last hour is less than the first hour.
c       Also request two storage locations for each hour to be considered
c       plus a temporary storage location; check the number of inputs,
c       parameters specified in the TRNS" deck; and specity the number
c       of outputs.
      if (info(7) .le. -1) then
        if ((hourend .lt. hourstart) .or. (numhours .gt. 8))
     +     call typeck(4,info,0,0,0)
        info(6) = numhours * 2
        info(10) = numhours * 2 + 1
        call typeck(1,info,1,2,0)
      endif

c     Calculate the current hour of the day, and then, if then, if the current
c       hour is within the range of hours being checked, start the main part
c       of this routine.

      acthour = mod(int(time),24)
      if (acthour .eq. 0) acthour = 24
```

```
      if ((acthour .ge. hourstart) .and. (acthour .le. hourend)) then

c       Get pointers to memory.
        runmempnt = info(10)
        temmempnt = info(10) + numhours * 2

c       If it's the first call for this time step, add last time step's values
c         to the running totals and put last time's maximums in the running
c         maximum holders if necessary.
        if (info(7) .eq. 0) then

          if (acthour .ne. hourstart) then
            offset = 2 * (acthour - hourstart - 1)
          else
            offset = 2 * (numhours - 1)
          endif

          s(runmempnt+offset) = s(runmempnt+offset) + s(temmempnt)
          if (s(temmempnt) .gt. s(runmempnt + offset + 1)) then
            s(runmempnt + offset + 1) = s(temmempnt)
          endif


c         If it's the beginning of the month, reset the running totals/maximums.
c           This negates what was done above, but that's ok.
          if ((time .eq. 0 + hourstart) .or.
     +        (time .eq. 744 + hourstart) .or.
     +        (time .eq. 1416 + hourstart) .or.
     +        (time .eq. 2160 + hourstart) .or.
     +        (time .eq. 2880 + hourstart) .or.
     +        (time .eq. 3624 + hourstart) .or.
     +        (time .eq. 4344 + hourstart) .or.
     +        (time .eq. 5088 + hourstart) .or.
     +        (time .eq. 5832 + hourstart) .or.
     +        (time .eq. 6552 + hourstart) .or.
     +        (time .eq. 7296 + hourstart) .or.
     +        (time .eq. 8016 + hourstart) .or.
     +        (time .eq. 8760 + hourstart)) then

            do i = 0,numhours* 2 - 1
              s(runmempnt + i) = 0
            end do

          endif

        endif

c       Dump current input into temporary variable regardless of which
c         iteration this is and output the running sums and maximums.
        s(temmempnt) = xin(1)
        do i=0,numhours*2-1
          out(i + 1) = s(runmempnt + i)
        end do

c       The curent hour's outputs are special because the most recent input
c         must be temporarily added to them.
        i = 2 * (acthour - hourstart) + 1
        out(i) = out(i) + s(temmempnt)
        out(i + 1) = amax1(out(i + 1),s(temmempnt))
```

```
endif

return
end
```

```
      subroutine type7(time,xin,out,t,dtdt,par,info)
c*****************************************************************************
c
c     This subroutine simulates the performance of an air-conditioning unit.
c     Default curves from the DOE2 system are currently used to describe how
c     the capacity, EIR (1/COP) and bypass factor are modified by off-rated
c     evaporator entering air conditions, condenser entering conditions and
c     flow rates.  Rated values are those at ARI conditions: 67 (F) evaporator
c     entering wet bulb temperature, 95 (F) condenser entering dry bulb
c     temperature and 37.5 cfm/1000 Btu/h cooling capacity.
c
c     All quantities are to be expressed in English units.
c     Control in this unit assumes this is AC unit 1--doing both sensible and
c     latent cooling.
c
c     (REU--December 1987)
c*****************************************************************************

      implicit      none                !disallow default variable definitions

c     define the standard TRNSYS variables:
      real          xin(9),out(7),par(7),dtdt(1),t,time
      integer       info(10)

c     define variables needed for this routine:

      integer       ac2on               !flag: 1--ac2 is in system
                                        !      0--ac2 is not in system
      real*8        CAPrated            !capacity at ARI conditions
      real*8        EIRrated            !1/COP at ARI conditions
      real*8        BFrated             !by-pass factor at ARI conditions
      real*8        CFMrated            !vol. flow-rate at ARI conditions
      real*8        Tset                !store temperature set point
      real*8        Wset                !store humidity ratio set point
      real*8        Tin                 !dry bulb temperature of incoming air
      real*8        Win                 !humidity ratio of incoming air
      real*8        mdot1               !mass flow-rate of incoming air
      real*8        Tcond               !dry bulb temp. of air entering cond.
      real*8        Tret                !dry-bulb temp. of air returning from
                                        !  under refrigerator cases
      real*8        Wret                !humidity ratio of air returning from
                                        !  under refrigerator cases
      real*8        mdot2               !mass flow-rate of return air not
                                        !  going through ac1
      real*8        Qsstore             !store sensible load, neglecting ventil.
      real*8        Qlstore             !store latent load, neglecting ventil.

      real*8        mdottot             !total circulation mass flow-rate:
                                        !  (mdot1 + mdot2)
      real*8        CAP                 !current maximum capacity
      real*8        power               !current required compressor power
      real*8        BF                  !current by-pass factor
      real*8        CFM                 !current volume flow-rate
      real*8        Tsurf               !temperature of surface of cooling coil
      real*8        Wsurf               !humidity ratio of air at Tsurf
      real*8        Texit               !temperature of exiting air
      real*8        Wexit               !humidity ratio of exiting air
      real*8        Tdes                !desired temperature of exiting air
      real*8        Wdes                !desired humidity ratio of exiting air
      real*8        ewb                 !wet-bulb temp. of air entering evap.
      real*8        frctcfm             !CFMactual/CFMrated
```

```
      real*8       plr                  !actual load/maximum capacity
      real*8       Qsens                !sensible cooling actually done by ac
      real*8       Qlat                 !latent cooling actually done by ac

      real*8       Fcap_t               !function which modifies rated capacity
                                        !  for off-rated entering condenser Tdb
                                        !  and entering evaporator Twb
      real*8       Fcap_cfm             !function which modifies rated capacity
                                        !  for off-rated volume flow-rates
      real*8       Feir_t               !function which modifies rated EIR
                                        !  for off-rated entering condenser Tdb
                                        !  and entering evaporator Twb
      real*8       Feir_cfm             !function which modifies rated EIR
                                        !  for off-rated volume flow-rates
      real*8       Feir_plr             !function which modifies rated EIR
                                        !  for partial loads
      real*8       Fbf_cfm              !function which modifies rated by-pass
                                        !  factor for off-rated flow-rates
      real*8       TfromW               !correlates T at saturation from W

      parameter    on = 1

c     define variables necessary for psych subroutine:
      parameter    iunits=2             !unit flag to psych subroutine: 2=English
      integer      wbmode               !wet bulb mode flag to psych subroutine:
                                        !     1--calculate Twb
                                        !     0--don't calculate Twb
      integer      mode                 !mode flag to psych subroutine:
                                        !     1--Tdb and Twb input
                                        !     2--Tdb and rh input
                                        !     4--Tdb and W input
      real         psydat(9)            !array of psychometric data:
                                        !     (1)--Patm (atm)
                                        !     (2)--Tdb (F)
                                        !     (3)--Twb (F)
                                        !     (4)--relative humidity fraction
                                        !     (5)--Tdp (F)
                                        !     (6)--humidity ratio (lbm water/
                                        !          lbm dry air
                                        !     (7)--enthalpy (Btu/lbm dry air)
                                        !     (8)--dens. of air-water mixture
                                        !          (lbm/ft**3)
                                        !     (9)--density of air in mixture
                                        !          (blm/ft**3)

c     define statement functions--these are currently DOE2 default curves:
      Fcap_t(ewb,Tcond) = 0.418934 + 0.017421 * ewb - 0.00617 * Tcond
      Fcap_cfm(frctcfm) = 0.69717199 + 0.39555 * frctcfm
     +                    - 0.092727 * frctcfm**2
      Feir_t(ewb,Tcond) = 0.282094 - 0.005832 * ewb + 0.01167 * Tcond
      Feir_cfm(frctcfm) = 1.13318 - 0.13318 * frctcfm
      Feir_plr(plr) = 0.29333333 + 0.70666667 * plr
      Fbf_cfm(frctcfm) = frctcfm
      TfromW(Wsurf) = 6.8123 + 1.1679 * 7000 * Wsurf - 0.007985 *
     +               (7000 * Wsurf)**2 + 0.000023332 * (7000 * Wsurf)**3


c     if this is the first call to this unit, do the usual TRNSYS stuff:
      if (info(7) .eq. -1) then
        call typeck(1,info,9,7,0)       !check no. of inputs, parameters
        info(6) = 7                     !seven outputs
```

```
        info(9) = 0                      !outputs depend only on inputs, not time
      endif

c     get parameters, inputs from arrays:
      ac2on = par(1)
      CAPrated = par(2)
      EIRrated = 1/par(3)
      BFrated = par(4)
      CFMrated = par(5)
      Tset = par(6)
      Wset = par(7)
      Tin = xin(1)
      Win = xin(2)
      mdot1 = xin(3)
      Tcond = xin(4)
      Tret = xin(5)
      Wret = xin(6)
      mdot2 = xin(7)
      Qsstore = xin(8)
      Qlstore = xin(9)

c     call psych to get complete information on incoming air:
      psydat(1) = 1
      psydat(2) = Tin
      psydat(6) = Win
      mode = 4
      wbmode = 1
      call psych(info,iunits,mode,wbmode,psydat)
      Win = psydat(6)                   !this is a fix in case the incoming
                                        !  humidity is physically impossible--
                                        !  at least one of the weather files
                                        !  has mistakes

c     get evaporator entering wet bulb temp. and calculate volume flow-rate:
      ewb = psydat(3)
      CFM = (mdot1 + mdot1 * Win) / (psydat(8) * 60.0)

c     calculate current bypass fraction:
      frctcfm = CFM/CFMrated
      BF = BFrated * Fbf_cfm(frctcfm)

c     calculate total mass flow, as this quantity will often be used:
      mdottot = mdot1 + mdot2

c     calculate desired exit state (assuming ac2 unit off):
      Wdes = ( (Wset - Qlstore/(mdottot * 1061.0)) * mdottot
     +        - mdot2 * Wret ) / mdot1
      Tdes = ( (Tset - Qsstore/(mdottot * 0.24)) * mdottot
     +        - mdot2 * Tret ) / mdot1

c     set exit state to input state, with the expectation that most of the
c       the time the exit conditions will be reassigned:

      Texit = Tin
      Wexit = Win

c     if ac unit 2 isn't on, try to meet the sensible load first
c       (unless the incoming temp is already low enough):
      if ((Texit .gt. Tdes) .and. (ac2on .ne. on)) then

c       calculate the desired air temperature at the surface of the coil,
c         then calculate the corresponding humidity
```

```
          Tsurf = (Tdes - (BF * Tin) ) / (1 - BF)
          psydat(2) = Tsurf
          psydat(4) = 1
          mode = 2
          wbmode = 0
          call psych(info,iunits,mode,wbmode,psydat)
          Wsurf = psydat(6)

c         if condensation has occured, recalcuate Wexit
          if (Win .gt. Wsurf) then
            Wexit = Win * BF + (1-BF) * Wsurf
          endif

c         at any rate, assign Texit to be the desired temperature
          Texit = Tdes
        endif

c       if ac unit 2 is on and dehumidification is needed, or humidity is not
c         controlled from above, meet the latent load, then calculate the
c         temperature.  this calculation depends on a regression equation
c         relating Tsurf to Wsurf, and an error will flagged if Wsurf is
c         out of bounds for the equation.
        if (Wexit .gt. Wdes) then

          Wsurf = (Wdes - (BF * Win) ) / (1 -BF)
          if ( (Wsurf .gt. 0.0187) .or. (Wsurf .lt. 0.00394)) then
            write(21,100)Wsurf
100         format('**CAUTION** Wdes = ',D13.5, ' out of range for',
     +        ' Tdb-at-saturation correlation')
          endif
          Tsurf = 11.308 + 0.9229 * (7000 * Wsurf)
     +           - 0.0037528 * (7000 * Wsurf)**2

          Texit = Tin * BF + (1-BF) * Tsurf
          Wexit = Wdes
        endif

c       calculate actual needed capacity, maximum capacity, and then
c         required power (note that the partial-load-ratio is set to 1
c         if the needed capacity exceeds the maximum capacity):

        Qsens = mdot1 * 0.24 * (Tin - Texit)
        Qlat = mdot1 * 1061 * (Win - Wexit)
        CAP = CAPrated * Fcap_t(ewb,Tcond) * Fcap_cfm(frctcfm)
        plr = (Qsens + Qlat) / CAP
        if (plr .gt. 1.0) plr = 1.0
        power = (Qsens + Qlat) * (EIRrated) * Feir_t(ewb,Tcond) *
     +          Feir_cfm(frctcfm) * Feir_plr(plr)
        power = power/3413.0              !convert to kW's

c       put out output
        out(1) = Texit
        out(2) = Wexit
        out(3) = mdot1
        out(4) = power
        out(5) = Qsens
        out(6) = Qlat
        out(7) = plr

        return
        end
```

```
      subroutine type10(time,xin,out,t,dtdt,par,info)
c*****************************************************************************
c
c     This subroutine simulates the performance of an air-conditioning unit.
c     Default curves from the DOE2 system are currently used to describe how
c     the capacity, EIR (1/COP) and bypass factor are modified by off-rated
c     evaporator entering air conditions, condenser entering conditions and
c     flow rates.  Rated values are those at ARI conditions: 67 (F) evaporator
c     entering wet bulb temperature, 95 (F) condenser entering dry bulb
c     temperature and 37.5 cfm/1000 Btu/h cooling capacity.
c
c     All quantities are assumed to be expressed in English units.
c     Control equations assume this is AC unit 2--sensible coil
c
c     (REU--December, 1987)
c
c*****************************************************************************

      implicit     none                !disallow default variable definitions

c     define the standard TRNS" variables:
      real         xin(7),out(7),par(5),dtdt(1),t,time
      integer      info(10)

c     define variables needed for this routine:

      real*8       CAPrated            !capacity at ARI conditions
      real*8       EIRrated            !1/COP at ARI conditions
      real*8       BFrated             !by-pass factor at ARI conditions
      real*8       CFMrated            !vol. flow-rate at ARI conditions
      real*8       Tset                !store temperature set point
      real*8       Wset                !store humidity ratio set point
      real*8       Tin                 !dry bulb temperature of incoming air
      real*8       Win                 !humidity ratio of incoming air
      real*8       mdot2               !mass flow-rate of incoming air
      real*8       Tcond               !dry bulb temp. of air entering cond.
      real*8       Taclex              !dry-bulb temp. of air exiting from
                                       !  air conditioner 1
      real*8       mdot1               !mass flow-rate of air through ac1
      real*8       Qsstore             !store sensible load, neglecting ventil.
      real*8       mdottot             !total circulation mass flow-rate:
                                       !  (mdot1 + mdot2)
      real*8       CAP                 !current maximum capacity
      real*8       power               !current required compressor power
      real*8       BF                  !current by-pass factor
      real*8       CFM                 !current volume flow-rate
      real*8       Tsurf               !temperature of surface of cooling coil
      real*8       Wsurf               !humidity ratio of air at Tsurf
      real*8       Texit               !temperature of exiting air
      real*8       Wexit               !humidity ratio of exiting air
      real*8       Tdes                !desired temperature of exiting air
      real*8       ewb                 !wet-bulb temp. of air entering evap.
      real*8       frctcfm             !CFMactual/CFMrated
      real*8       plr                 !actual load/maximum capacity
      real*8       Qsens               !sensible cooling actually done by ac
      real*8       Qlat                !latent cooling actually done by ac

      real*8       Fcap_t              !function which modifies rated capacity
                                       !  for off-rated entering condenser Tdb
                                       !  and entering evaporator Twb
      real*8       Fcap_cfm            !function which modifies rated capacity
```

```
                                        !  for off-rated volume flow-rates
      real*8        Feir_t              !function which modifies rated EIR
                                        !  for off-rated entering condenser Tdb
                                        !  and entering evaporator Twb
      real*8        Feir_cfm            !function which modifies rated EIR
                                        !  for off-rated volume flow-rates
      real*8        Feir_plr            !function which modifies rated EIR
                                        !  for partial loads
      real*8        Fbf_cfm             !function which modifies rated by-pass
                                        !  factor for off-rated flow-rates

c     define variables necessary for psych subroutine:
      parameter     iunits=2            !unit flag to psych subroutine: 2=English
      integer       wbmode              !wet bulb mode flag to psych subroutine:
                                        !     1--calculate Twb
                                        !     0--don't calculate Twb
      integer       mode                !mode flag to psych subroutine:
                                        !     1--Tdb and Twb input
                                        !     2--Tdb and rh input
                                        !     4--Tdb and W input
      real          psydat(9)           !array of psychometric data:
                                        !     (1)--Patm (atm)
                                        !     (2)--Tdb (F)
                                        !     (3)--Twb (F)
                                        !     (4)--relative humidity fraction
                                        !     (5)--Tdp (F)
                                        !     (6)--humidity ratio (lbm water/
                                        !          lbm dry air
                                        !     (7)--enthalpy (Btu/lbm dry air)
                                        !     (8)--dens. of air-water mixture
                                        !          (lbm/ft**3)
                                        !     (9)--density of air in mixture
                                        !          (blm/ft**3)

c     define statement functions--these are currently DOE2 default curves:
      Fcap_t(ewb,Tcond) = 0.418934 + 0.017421 * ewb - 0.00617 * Tcond
      Fcap_cfm(frctcfm) = 0.69717199 + 0.39555 * frctcfm
     +                    - 0.092727 * frctcfm**2
c     Fshc_t(ewb,Tcond) = 4.4222 - 0.0464365 * ewb - 0.0032733 * Tcond
c     Fshc_cfm(frctcfm) = 0.37188 + 0.73956 * frctcfm -
c    +                    0.11144 * frctcfm**2
      Feir_t(ewb,Tcond) = 0.282094 - 0.005832 * ewb + 0.01167 * Tcond
      Feir_cfm(frctcfm) = 1.13318 - 0.13318 * frctcfm
      Feir_plr(plr) = 0.29333333 + 0.70666667 * plr
      Fbf_cfm(frctcfm) = frctcfm


c     if this is the first call to this unit, do the usual TRNSYS stuff:
      if (info(7) .eq. -1) then
        call typeck(1,info,7,5,0)       !check no. of inputs, parameters
        info(6) = 7                     !signal seven outputs
        info(9) = 0                     !outputs depend only on inputs, not time
      endif

c     get parameters, inputs from arrays:
      CAPrated = par(1)
      EIRrated = 1/par(2)
      BFrated = par(3)
      CFMrated = par(4)
      Tset = par(5)
      Tin = xin(1)
```

```
      Win = xin(2)
      mdot2 = xin(3)
      Tcond = xin(4)
      Taclex = xin(5)
      mdot1 = xin(6)
      Qsstore = xin(7)

c     call psych to get complete information on incoming air:
      psydat(1) = 1
      psydat(2) = Tin
      psydat(6) = Win
      mode = 4
      wbmode = 1
      call psych(info,iunits,mode,wbmode,psydat)

c     get evaporator entering wet bulb temp. and calculate volume flow-rate:
      ewb = psydat(3)
      CFM = (mdot2 + mdot2 * Win) / (psydat(8) * 60.0)

c     calculate current bypass fraction:
      frctcfm = CFM/CFMrated
      BF = BFrated * Fbf_cfm(frctcfm)

c     calculate total mass flow, as this quantity will often be used:
      mdottot = mdot1 + mdot2

c     calculate desired exit state:
      Tdes = ( (Tset - Qsstore/(mdottot * 0.24)) * mdottot
     +        - mdot1 * Taclex ) / mdot2

c     set exit state to input state, with the expectation that most of the
c       the time the exit conditions will be reassigned:

      Texit = Tin
      Wexit = Win

c     try to meet the part of the sensible load not met by ac1
c       (unless the incoming temp is already low enough)
      if (Texit .gt. Tdes) then

        Tsurf = (Tdes - (BF * Tin) ) / (1 - BF)
        psydat(2) = Tsurf
        psydat(4) = 1
        mode = 2
        wbmode = 0
        call psych(info,iunits,mode,wbmode,psydat)
        Wsurf = psydat(6)

c       if condensation has occured, recalculate the exit humidity
        if (Win .gt. Wsurf) then
          Wexit = Win * BF + (1-BF) * Wsurf
        endif

        Texit = Tdes
      endif

c     calculate actual needed capacity, maximum capacity, and then
c       required power.  (Note that the partial-load-ratio is set to
c       1 if needed capacity exceeds maximum capacity.

      Qsens = mdot2 * 0.24 * (Tin - Texit)
      Qlat = mdot2 * 1061 * (Win - Wexit)
```

```
      CAP = CAPrated * Fcap_t(ewb,Tcond) * Fcap_cfm(frctcfm)
      plr = (Qsens + Qlat) / CAP
      if (plr .gt. 1.0) plr = 1.0
      power = (Qsens + Qlat) * (EIRrated) * Feir_t(ewb,Tcond) *
     +        Feir_cfm(frctcfm) * Feir_plr(plr)
      power = power/3413.0             !convert to kW's

c     put out output
      out(1) = Texit
      out(2) = Wexit
      out(3) = mdot2
      out(4) = power
      out(5) = Qsens
      out(6) = Qlat
      out(7) = plr

      return
      end
```

```
C  MODIFIED AUG. 87 TO MIX/DIVERT TEMPERATURE AND HUMIDITY   REU
C
C  NOTE CHANGES MADE BETWEEN LINES 73 AND 74 JPK 6-27-86
      SUBROUTINE TYPE11(TIME,XIN,OUT,T,DTDT,PAR,INFO)                   TYP110001
C  THIS SUBROUTINE SIMULATES A FLOW DIVERTER, MIXER, OR T-PIECE.        TYP110002
C  MODES 1-5 MIX/DIVERT A FLUID WITH 1 PROPERTY (USUALLY TEMPERATURE)
C  MODES 6-10 MIX/DIVERT A FLUID WITH 2 PROPERTIES (USUALLY TEMP. AND HUMIDITY
C    IN MODES 1 AND 6, A MIXING T-PIECE IS SIMULATED
C    IN MODES 2 AND 7, THE FLOW DIVERTER HAS 1 INLET AND 2 POSSIBLE OUTLETS
C    IN MODES 3 AND 8, THE FLOW MIXER HAS 2 POSSIBLE INLETS AND 1 OUTLET
C    IN MODES 4 AND 9, THE COMPONENT FUNCTIONS AS A FLOW DIVERTOR WITH INTERNAL
C   CONTROL DERIVED FROM INPUTS 1,3 AND 4.                              TYP110007
C    IN MODES 5 AND 10, THE COMPONENT FUNCTIONS AS IN MODE 4 EXCEPT THAT ALL
C   FLOW IS DIVERTED WHEN THE SOURCE TEMPERATURE IS GREATER THAN THE    TYP110009
C   HEATED FLOWSTREAM TEMPERATURE (TYPICALLY THE TANK TEMPERATURE).     TYP110010
      DIMENSION XIN(10),OUT(20),PAR(15),INFO(10)                        TYP110011

C     DETERMINE THE MODE OF OPERATION
      MODE=IFIX(PAR(1)+0.5)                                             TYP110012
C     PRINT OUT A WARNING IF MODE IS CLEARLY AN ERROR
      IF (MODE .LT. 1 .OR. MODE .GT. 10) THEN
        WRITE(LUW,999)
999     FORMAT('MODE NOT BETWEEN 1 AND 10 INCLUSIVE--DEFAULTING'
     +         'TO MODE 1')
      ENDIF

C     DETERMINE IF THIS IS THE FIRST CALL OF SIMULATION
      IF (INFO(7) .LE. -1) THEN
C       FIRST CALL OF SIMULATION                                        TYP110015
        INFO(9)=0                                                       TYP110014

C       CASE ON (MODE)
        GO TO (21,22,23,24,24,26,27,28,29,29),MODE
C       MODE 1
21        INFO(6)=2                                                     TYP110017
          CALL TYPECK(1,INFO,4,1,0)                                     TYP110018
        GO TO 90                                                        TYP110019
C       MODE 2
22        INFO(6)=4                                                     TYP110020
          CALL TYPECK(1,INFO,3,1,0)                                     TYP110021
        GO TO 90                                                        TYP110022
C       MODE 3
23        INFO(6)=2                                                     TYP110023
          CALL TYPECK(1,INFO,5,1,0)                                     TYP110024
          GO TO 90                                                      TYP110025
C       MODES 4 AND 5
24        INFO(6)=5                                                     TYP110026
          INFO(9)=1                                                     TYP110027
          CALL TYPECK(1,INFO,4,2,0)                                     TYP110028
          GO TO 90
C       MODE 6
26        INFO(6)=3
          CALL TYPECK(1,INFO,6,1,0)
          GO TO 90
C       MODE 7
27        INFO(6)=6
          CALL TYPECK(1,INFO,4,1,0)
          GO TO 90
C       MODE 8
28        INFO(6)=3
          CALL TYPECK(1,INFO,7,1,0)
```

```
          GO TO 90
C       MODE 9 AND 10
29        INFO(6)=7
          CALL TYPECK(1,INFO,5,2,0)
          GO TO 90
C       END CASE
90      CONTINUE
      END IF

C     BRANCH ACCORDING TO MODE  (CASE)                                  TYP110030
      GO TO (100,200,300,400,400,600,700,800,900,900), MODE
C     MODE 1:  T-PIECE                                                  TYP110032 -
100     T1=XIN(1)                                                       TYP110033
        FLOW1=XIN(2)                                                    TYP110034
        T2=XIN(3)                                                       TYP110035
        FLOW2=XIN(4)                                                    TYP110036
        FLOWO=FLOW1+FLOW2                                               TYP110037
        IF(FLOWO.GT.0.0) THEN                                           TYP110038
          TO=(FLOW1*T1+FLOW2*T2)/FLOWO                                  TYP110039
          OUT(1)=TO                                                     TYP110040
        END IF
        OUT(2)=FLOWO                                                    TYP110041
      RETURN                                                            TYP110042
C     MODE 2:  FLOW DIVERTER                                            TYP110043
200     TIN=XIN(1)                                                      TYP110044
        FLOW=XIN(2)                                                     TYP110045
        GAM=XIN(3)                                                      TYP110046
        FLOW2 = FLOW*GAM                                                TYP110047
        FLOW1 = FLOW - FLOW2                                            TYP110048
        IF (FLOW1 .GT. 0.) OUT(1) = TIN                                 TYP110049
        OUT(2) = FLOW1                                                  TYP110050
        IF (FLOW2 .GT. 0.) OUT(3) = TIN                                 TYP110051
        OUT(4) = FLOW2                                                  TYP110052
      RETURN                                                            TYP110053
C     MODE 3:  FLOW MIXER                                               TYP110054
300     T1=XIN(1)                                                       TYP110055
        FLOW1=XIN(2)                                                    TYP110056
        T2=XIN(3)                                                       TYP110057
        FLOW2=XIN(4)                                                    TYP110058
        GAM=XIN(5)                                                      TYP110059
        FLOWO=FLOW1*(1.-GAM)+FLOW2*GAM                                  TYP110060
        IF (FLOWO .GT. 0.0) THEN                                        TYP110061
           OUT(1)=(T1*FLOW1*(1.-GAM)+T2*FLOW2*GAM)/FLOWO                TYP110062
        END IF
        OUT(2)=FLOWO                                                    TYP110063
      RETURN                                                            TYP110064
C     MODES 4 AND 5:  TEMPERATURE CONTROLLED FLOW DIVERTER              TYP110065
400     NSTK=IFIX(PAR(2)+0.5)                                           TYP110066
        IF(INFO(7).GT.NSTK)RETURN                                       TYP110067
        TSOURC=XIN(1)                                                   TYP110068
        FLOW=XIN(2)                                                     TYP110069
        THOT=XIN(3)                                                     TYP110070
        TSET=XIN(4)                                                     TYP110071
        GAM = 1.0                                                       TYP110072
        IF (THOT .GT. TSET) GAM = (TSET-TSOURC)/(THOT-TSOURC)           TYP110073
C  *** NOTE CHANGE
        IF (THOT.LT.TSOURC .AND. MODE.EQ.4) GAM=1.0
        IF (THOT.LT.TSOURC .AND. MODE.EQ.5) GAM = 0.0                   TYP110074
C       MODE 4 AND 5 OUTPUTS                                            TYP110075
        OUT(1)=TSOURC                                                   TYP110076
        OUT(2)=GAM*FLOW                                                 TYP110077
```

```
         OUT(3)=TSOURC                                            TYP110078
         OUT(4)=FLOW-OUT(2)                                       TYP110079
         OUT(5)=GAM                                               TYP110080
       RETURN                                                     TYP110081
C      MODE 6:  T-PIECE
600      T1=XIN(1)                                                TYP110033
         OMEGA1=XIN(2)
         FLOW1=XIN(3)                                             TYP110034
         T2=XIN(4)                                                TYP110035
         OMEGA2=XIN(5)
         FLOW2=XIN(6)                                             TYP110036
         FLOWO=FLOW1+FLOW2                                        TYP110037
         IF(FLOWO.GT.0.0) THEN                                    TYP110038
           TO=(FLOW1*T1+FLOW2*T2)/FLOWO                           TYP110039
           OUT(1)=TO                                              TYP110040
           OMEGAO=(FLOW1*OMEGA1+FLOW2*OMEGA2)/FLOWO
           OUT(2)=OMEGAO
         END IF
         OUT(3)=FLOWO                                             TYP110041
       RETURN                                                     TYP110042
C      MODE 7:  FLOW DIVERTER                                     TYP110043
700      TIN=XIN(1)                                               TYP110044
         OMEGAIN=XIN(2)
         FLOW=XIN(3)                                              TYP110045
         GAM=XIN(4)                                               TYP110046
         FLOW2 = FLOW*GAM                                         TYP110047
         FLOW1 = FLOW - FLOW2                                     TYP110048
         IF (FLOW1 .GT. 0.) THEN
           OUT(1) = TIN                                           TYP110049
           OUT(2) = OMEGAIN
         END IF
         OUT(3) = FLOW1                                           TYP110050
         IF (FLOW2 .GT. 0.) THEN
           OUT(4) = TIN                                           TYP110051
           OUT(5) = OMEGAIN
         END IF
         OUT(6) = FLOW2                                           TYP110052
       RETURN                                                     TYP110053
C      MODE 8:  FLOW MIXER                                        TYP110054
800      T1=XIN(1)                                                TYP110055
         OMEGA1=XIN(2)
         FLOW1=XIN(3)                                             TYP110056
         T2=XIN(4)                                                TYP110057
         OMEGA2=XIN(5)
         FLOW2=XIN(6)                                             TYP110058
         GAM=XIN(5)                                               TYP110059
         FLOWO=FLOW1*(1.-GAM)+FLOW2*GAM                           TYP110060
         IF (FLOWO .GT. 0.0) THEN                                 TYP110061
            OUT(1)=(T1*FLOW1*(1.-GAM)+T2*FLOW2*GAM)/FLOWO         TYP110062
            OUT(2)=(OMEGA1*FLOW1*(1.-GAM)+OMEGA2*FLOW2*GAM)/FLOWO
         END IF
         OUT(3)=FLOWO                                             TYP110063
       RETURN                                                     TYP110064
C      MODES 9 AND 10:  TEMPERATURE CONTROLLED FLOW DIVERTER      TYP110065
900      NSTK=IFIX(PAR(2)+0.5)                                    TYP110066
         IF(INFO(7).GT.NSTK)RETURN                                TYP110067
         TSOURC=XIN(1)                                            TYP110068
         OMEGAS=XIN(2)
         FLOW=XIN(3)                                              TYP110069
         THOT=XIN(4)                                              TYP110070
         TSET=XIN(5)                                              TYP110071
         GAM = 1.0                                                TYP110072
```

```
         IF (THOT .GT. TSET) GAM = (TSET-TSOURC)/(THOT-TSOURC)          TYP110073
C   *** NOTE CHANGE
         IF (THOT.LT.TSOURC .AND. MODE.EQ.9) GAM=1.0
         IF (THOT.LT.TSOURC .AND. MODE.EQ.10) GAM = 0.0
C        MODE 9 AND 10 OUTPUTS                                          TYP110075
         OUT(1)=TSOURC                                                  TYP110076
         OUT(2)=OMEGAS
         OUT(3)=GAM*FLOW                                                TYP110077
         OUT(4)=TSOURC                                                  TYP110078
         OUT(5)=OMEGAS
         OUT(6)=FLOW-OUT(2)                                             TYP110079
         OUT(7)=GAM                                                     TYP110080
       RETURN                                                           TYP110081
       END                                                              TYP110082
```

```
c     additional outputs added--see un-numbered lines-----December 1987-----REU
      SUBROUTINE TYPE19(TIME,XIN,OUT,DUM1,DUM2,PAR,INFO)                TY190001

                        (extracts showing changes only)

C  TYPECK CALL, STORE INFORMATION IN S ARRAY                            TY190276
      NSTOR=2*NS**2+(NS+1)**2+NTC*NSHFT+NS+4                            TY190277
      IF(MODE.EQ.1) NSTOR=NSTOR+(NS+1)**2                               TY190278
      IF(INFO(4).NE.IP) IP=IP+2*NOUTS+1                                 TY190279
C     INFO(6)=10+NOUTS                                                  TY190280
c     change number of outputs
      INFO(6)=19
      INFO(10)=NSTOR                                                    TY190281
      CALL TYPECK(1,INFO,IN,IP,0)                                       TY190282
      ISTORE=INFO(10)-1                                                 TY190283
      TR=T0                                                             TY190284
      WR=W0                                                             TY190285
                                    .
                                    .
                                    .
      IF(MODE.EQ.2) GO TO 360                                           TY190654
      OUT(7)=QSENS                                                      TY190655
      OUT(8)=QLAT                                                       TY190656
      IF(QSENS.GT.0.) OUT(9)=AMAX1(QSENS+AMAX1(QLAT,0.),OUT(9))         TY190657
      IF(QSENS.LT.0.) OUT(10)=AMIN1(QSENS+AMIN1(QLAT,0.),OUT(10))       TY190658
c     add additional outputs:
     OUT(11)=QLPEPL
     OUT(12)=QLINF
     OUT(13)=QLVENT
     OUT(14)=QLGEN
     OUT(15)=QINT
     OUT(16) = QSENS - QSVENT             !for energy balance and ac control
     OUT(17) = QLINF + QLGEN + QLPEPL     !for energy balance
     OUT(18) = MAIR * (WRF - WRLAST) * HVAP(IU)
     OUT(19) = (FLWI * (WAMB - WMAX) - MAIR * (WMAX - WRLAST) ) *
    +           HVAP(IU) + QLGEN + QLPEPL
360   IF(NOUTS.EQ.0) RETURN                                             TY190659
C                                                                       TY190660
C  OPTIONAL OUTPUTS                                                     TY190661
C                                                                       TY190662
      DO 450 L=1,NOUTS                                                  TY190663
```

```
      subroutine type22(time,xin,out,t,dtdt,par,info)
c***************************************************************************
c
c     This subroutine simulates refrigerator and freezer cases.
c     In addition to calculating the power consumed by the cases, this
c     routine also calculates the exit state of the air flowing under
c     the cases.
c     At the moment, open and closed cases are treated identically and
c     English units are used.
c
c     (REU--January, 1987)
c
c***************************************************************************

      implicit none          !don't allow implicit definitions

c     define the usual TRNSYS arrays:
      real    time,T,Dtdt
      real    xin(10),out(20),par(17)
      integer info(10)

c     define parameter variables:
      real    capmed         !rated capacity of medium-temp cases
      real    caplow         !rated capacity of low-temp cases
      real    Rcasemed       !resistance of med-temp cases
      real    Rcaselow       !resistance of low-temp cases
      real    Rair           !case-air resistance
      real    COPdesmed      !design condition COP for med-temp cases
      real    COPdeslow      !design condition COP for low-temp cases
      real    deltatmed      !(Tcond-Tevap) at design cond. for med-temp cases
      real    deltatlow      !(Tcond-Tevap) at design cond. for low-temp cases
      real    Tcasemed       !temperature of med-temp cases
      real    Tcaselow       !temperature of low-temp cases
      real    caslenmed      !linear length of med-temp cases
      real    caslenlow      !linear length of low-temp cases
      real    cashgtmed      !avg. height of med-temp cases--must be value used
                             !  in calculating Rcasemed
      real    cashgtlow      !avg. height of low-temp cases--must be value used
                             !  in calculating Rcaselow
      real    Temed          !evap. temp. for med-temp. cases
      real    Telow          !evap. temp. for low-temp. cases

c     define input variables:
      real    ontime         !average on-time for cases
      real    Ti             !indoor dry-bulb temp.
      real    wi             !indoor humidity ratio
      real    mdot           !(dry air) mass flow
      real    Ts             !store supply dry-bulb temp.
      real    To             !outdoor dry-bulb temp.
      real    Qr             !required heat/reheat

c     define output and internal variables:
      real    Tic            !conditioned air dry-bulb temp.
      real    wic            !conditioned air humidity ratio
      real    ltr            !latent to total case credit ratio
      real    ccmed          !case credit for med-temp cases
      real    lccmed         !latent case credit for med-temp cases
      real    sccmed         !sensible case credit for med-temp cases
      real    cclow          !case credit for low-temp cases
      real    lcclow         !latent case credit for low-temp cases
      real    scclow         !sensible case credit for low-temp cases
```

```
      real    cctotal         !total case capacity
      real    pwrasmed        !power needed by anti-sweat heaters for med. cases
      real    pwraslow        !power needed by anti-sweat heaters for low. cases
      real    powermed        !power needed by med-temp cases--vapor-comp. cycle
      real    powerlow        !power needed by low-temp cases--vapor-comp. cycle
      real    Tdwpt           !dew-point temperature of incoming return air
      real    COPmedout       !actual COP for med-temp cases--no reheat
      real    COPlowout       !actual COP for low-temp cases--no reheat
      real    COPmedreh       !actual COP for med-temp cases--reheat
      real    COPlowreh       !actual COP for low-temp cases--reheat
      real    COPmedavg       !actual COP for med-temp cases--avg
      real    COPlowavg       !actual COP for low-tmp cases--avg
      real    reheatperc      !percentage of time reheat/heat on

c     define variables for psych subroutine
      parameter iunits=2      !unit flag to psych subroutine:  2-English
      integer mode            !mode flag to psych subroutine:
                              !    5--Tdb and enthalpy dry bulb inputs
                              !    4--Tdb and humidity ratio inputs
      integer wbmode          !wet bulb temp. flag to psych subroutine:
                              !    0--don't calculate Twb
                              !    1--calculate Twb
      real psydat(9)          !array of psychometeric data:
                              !    (1)--Patm: system pressure in atms.
                              !    (2)--Tdb (F)
                              !    (3)--Twb (F)
                              !    (4)--relative humidity fraction
                              !    (5)--Tdp (F)
                              !    (6)--humidity ratio (lbm water/lbm dry air)
                              !    (7)--enthalpy (Btuh/lbm dry air)
                              !    (8)--dens. of air-water mixture (lbm/ft**3)
                              !    (9)--density of air in mixture (lbm/ft**3)

c     if this is the first call of the simulation, check number of inputs and
c       parameters, then set up TRNSYS
      if (info(7) .le. -7) then
        call typeck(1,info,7,17,0)
        info(6) = 6             !six outputs
        info(9) = 0             !outputs depend only on inputs
      endif

c     get inputs and parameters out of their respective arrays:
      ontime = xin(1)
      Ti = xin(2)
      wi = xin(3)
      mdot = xin(4)
      Ts = xin(5)
      To = xin(6)
      Qr = xin(7)

      capmed = par(1)
      caplow = par(2)
      Rcasemed = par(3)
      Rcaselow = par(4)
      Rair = par(5)
      COPdesmed = par(6)
      COPdeslow = par(7)
      deltatmed = par(8)
      deltatlow = par(9)
      Tcasemed = par(10)
      Tcaselow = par(11)
```

```
      caslenmed = par(12)
      caslenlow = par(13)
      cashgtmed = par(14)
      cashgtlow = par(15)
      Temed = par(16)
      Telow = par(17)

c     calculate case credits:
      ltr = 0.016 * exp(245.0*wi)
      ccmed = capmed * ontime * 7.986 * (wi**0.452)
      lccmed = ccmed * ltr
      sccmed = ccmed - lccmed
      cclow = caplow * ontime * 3.635 * (wi**0.281)
      lcclow = cclow * ltr
      scclow = cclow - lcclow
      cctotal = ccmed + cclow

c     calculate exiting air state (using the subroutine psych to calculate
c       the humidity ratio from the temp. and enthalpy)--this routine
c       assumes that 50% of the credits go to the return air, and 50%
c       of the credits go to the store air):
      Tic = Ti - 0.5 * (sccmed + scclow)/(mdot * 0.24)
      wic = wi - 0.5 * (lccmed + lcclow)/(mdot * 1061)

c     calculate the inside air's dewpoint temp with the psych subroutine
      mode = 4
      wbmode = 0
      psydat(1) = 1.0
      psydat(2) = Ti
      psydat(6) = wi
      call psych(info,iunits,mode,wbmode,psydat)
      Tdwpt = psydat(5)

c     calculate the power necessary for the anti-sweat heaters:
      pwrasmed = ((Tdwpt - Ti)/Rair + (Tdwpt - Tcasemed)/Rcasemed) *
     +             caslenmed * cashgtmed
      if (pwrasmed .lt. 0) then
        pwrasmed = 0
      endif
      pwraslow = ((Tdwpt - Ti)/Rair + (Tdwpt - Tcaselow)/Rcaselow) *
     +             caslenlow * cashgtlow
      if (pwraslow .lt. 0) then
        pwraslow = 0
      endif

c     calculate the COP when reheat not done

      COPmedout = COPdesmed * deltaTmed / (To + 15 - Temed)
      COPlowout = COPdeslow * deltaTlow / (To + 15 - Telow)

c     if necessary, calculate the COP when reheat is done

      if (Qr .gt. 0) then
        if (Qr .gt. cctotal) then
c           write(21,100)Qr, cctotal, time
c100        format(' Warning--reheat required is ', F10.0, ', while ',
c     +            ' reheat available is ', F10.0, ' at time ', F7.2)
          reheatperc = 1
        else
          reheatperc = Qr / cctotal
        endif
```

```
          COPmedreh = COPdesmed * deltaTmed / (Ts - Temed)
          COPlowreh = COPdeslow * deltaTlow / (Ts - Telow)
        else
          reheatperc = 0
        endif

c       take a weighted average of the reheat/no reheat COP's
        COPmedavg = reheatperc * COPmedreh + (1 - reheatperc) * COPmedout
        COPlowavg = reheatperc * COPlowreh + (1 - reheatperc) * COPlowout

c       calculate power required for the vapor-compression cycle
        powermed = ccmed / COPmedavg
        powerlow = cclow / COPlowavg

c       stuff the answers into the output array
        out(1) = powermed + powerlow + pwrasmed + pwraslow
        out(2) = Tic
        out(3) = Wic
        out(4) = scclow + sccmed
        out(5) = lcclow + lccmed
        out(6) = pwrasmed + pwraslow

        return
        end
```

```
C  THIS IS THE NEW AND IMPROVED TYPE33--JPK'S PSYCH SUBROUTINE PUT
C   INTO TRNSYS FORM (REU--NOVEMBER 1987)

      SUBROUTINE TYPE33(TIME,XIN,OUT,DUM1,DUM2,PAR,INFO)
      COMMON /LUNITS/LUR,LUW,IFORM
      INTEGER WBMD,WBMODE
      DIMENSION XIN(2),OUT(8),PAR(3),INFO(10),PSYDAT(9)
      DATA RA/287.055/, LIMIT/100/, IWARNS/0/,PATMOLD/0./
C
C  MODES 1-5:
C  THESE MODES TAKE AS INPUT:  PATM (IN ATMOSPHERES), A DRY BULB
C  TEMP., AND ONE OTHER PROPERTY: WET BULB TEMP., REL.HUMIDITY FRACTION,
C  DEW PT.TEMP.,HUMIDITY RATIO, OR ENTHALPY DEPENDING ON MODE.
C  OUTPUTS ARE HUMIDITY RATIO (OR REL.HUMIDITY IN MODE 4), WET BULB TEMP.,
C  ENTHALPY(OR REL.HUMIDITY IN MODE 5), MIXTURE DENSITY AND DRY
C  AIR DENSITY
C
C  MODE 6:
C  MODE 6 TAKES AS AN INPUT PATM, HUMIDITY RATIO AND ENTHALPY AND
C  RETURNS ALL OF THE OTHER PROPERTIES.
C
C  THE WET BULB TEMPERATURE IS ONLY CALCULATED IF WBMODE EQUALS ONE.
C
C  TEMPERATURES ARE IN CELSIUS (IUNITS=1) OR FAHRENHEIT (IUNITS=2).
C  ENTHALPY IS IN KJ/KG (IUNITS=1) OR BTU/LBM (IUNITS=2), AND
C  DENSITY IS IN KG/M**3 (IUNITS=1) OR LBM/FT**3 (IUNITS=2).
C  THE REFERENCE STATES FOR ENTHALPIES ARE:
C         HAIR=0.0 AT 0. DEG C AND 0. DEG F
C         HW(LIQUID)=0.0 AT 0. DEG C AND 32. DEG F
C
C  THE INFO ARRAY CONTAINS THE TRNSYS  UNIT AND TYPE NUMBER FROM THE
C  COMPONENT WHICH CALLS THE PSYCH ROUTINE.
C  THE PSYDAT ARRAY CONTAINS THE MOIST AIR PROPERTIES.
C
C  DO THE USUAL TRNSYS STUFF IF IT'S THE FIRST CALL OF THE SIMULATION
      IF (INFO(7) .LE. -1) THEN
        CALL TYPECK(1,INFO,2,4,0)
        INFO(6) = 8              !8 OUTPUTS
        INFO(9) = 0              !OUTPUTS DEPEND ON INPUTS, NOT TIME
      ENDIF

C  ASSIGN LOCAL VARIABLES:
      MODE = PAR(1)
C     CASE ON (MODE)
      GOTO (101,102,103,104,105,106),MODE
C     MODE 1--TDB AND TWB INPUT
101     TDB = XIN(1)
        TWB = XIN(2)
      GOTO 107
C     MODE 2--TDB AND RH INPUT
102     TDB = XIN(1)
        RH = XIN(2)
      GOTO 107
C     MODE 3--TDB AND TDP INPUT
103     TDB = XIN(1)
        TDP = XIN(2)
      GOTO 107
C     MODE 4--TDB AND W INPUT
104     TDB = XIN(1)
        W = XIN(2)
      GOTO 107
```

```
C     MODE 5--TDB AND H INPUT
105     TDB = XIN(1)
        H = XIN(2)
      GOTO 107
C     MODE 6--W AND H INPUT
106     W = XIN(1)
        H = XIN(2)
      GOTO 107
C     END CASE
107   CONTINUE

      WBMODE = PAR(2)
      IUNITS = PAR(3)
      PATM = PAR(4)

      TWBOLD=-9.999E20
C
C  UNIT CONVERSIONS
C
      IF(IUNITS .EQ. 2) THEN
        TDB = (TDB - 32.)/1.8
        TWB = (TWB - 32.)/1.8
        TDP = (TDP - 32.)/1.8
        H   = (H - 7.687)/0.43002
      ENDIF
C-- CHECK THAT THE TOTAL PRESSURE IS WITHIN THE IDEAL GAS RANGE.
      IF (PATM .GT. 5.0) THEN
         WRITE(LUW,500) INFO(1),INFO(2)
         IWARNS = IWARNS + 1
      END IF
      IF (PATM .LE. 0.0) THEN
         WRITE(LUW,250) INFO(1),INFO(2)
         STOP
      END IF
C
C   FOR MODE 6, CHECK THAT THE ENTHALPY IS GREATER THAN THE
C   SATURATION ENTHALPY (MINIMUM) FOR THE GIVEN HUMIDITY RATIO
C   AND THAT THE HUMIDITY RATIO IS GREATER THAN 0.
C
      IF (MODE .EQ. 6) THEN
            IF (W .LT. 0.) THEN
               WRITE(LUW,1500) INFO(1),INFO(2)
               IWARNS = IWARNS + 1
               W = 0.
               GO TO 99
            ELSE IF (W .EQ. 0.) THEN
               GO TO 99
            END IF
         CALL DEWPT(INFO,PATM,W,TDP,IWARNS)
         HDP = TDP + W*(2501. + 1.805*TDP)
         HMIN = HDP
           IF (H .LT. HMIN) THEN
               WRITE(LUW,1000) INFO(1),INFO(2)
               IWARNS = IWARNS + 1
               H = HMIN
           END IF
99       TDB = (H - 2501.*W)/(1. + 1.805*W)
       END IF
C
C  FIND SATURATION PRESSURE OF WATER AT WET BULB, DRY BULB, OR
C  DEW POINT TEMPERATURE.
```

```
C
      CALL SAT(INFO,TDB,PSATDB,IWARNS)
      GOTO (1,2,3,2,2,2) ,MODE
C-- CHECK FOR IMPOSSIBLE WET BULB TEMPERATURES AND CORRECT THEM
C   IF POSSIBLE.
1     IF (TWB .GT. TDB .AND. PSATDB .GE. PATM) THEN
         WRITE(LUW,1750) INFO(1),INFO(2)
         STOP
      ELSE IF (TWB .GT. TDB) THEN
         WRITE(LUW,2000) INFO(1),INFO(2)
         IWARNS = IWARNS + 1
         TWB = TDB
         PSAT = PSATDB
         GOTO 5
      END IF
      CALL SAT (INFO,TWB,PSAT,IWARNS)
C-- ERROR: IF PSATWB IS GREATER THAN PATM.
         IF (PSAT .GE. PATM) THEN
             WRITE(LUW,1750) INFO(1),INFO(2)
             STOP
         END IF
      GOTO 5
2     PSAT = PSATDB
      GOTO 5
C-- CHECK FOR IMPOSSIBLE DEW POINT TEMPERATURES AND
C   CORRECT THEM IF POSSIBLE.
3     IF (TDP .GT. TDB .AND. PSATDB .GE. PATM) THEN
         WRITE(LUW,2250) INFO(1),INFO(2)
         STOP
      ELSE IF (TDP .GT. TDB) THEN
         WRITE(LUW,2500) INFO(1),INFO(2)
         IWARNS = IWARNS + 1
         TDP=TDB
      END IF
      CALL SAT(INFO,TDP,PSAT,IWARNS)
C-- ERROR: IF PSATDP IS GREATER THAN PATM.
      IF (PSAT .GE. PATM) THEN
         WRITE(LUW,2250) INFO(1),INFO(2)
         STOP
      END IF
5     CONTINUE
C
C   CALCULATE HUMIDITY RATIO AND WET BULB TEMPERATURE
C
      GO TO (10,20,30,40,50,60), MODE
C
C  MODE 1 -- DRY BULB AND WET BULB SUPPLIED
C
10    IF (TWB .LE. 0.) THEN
        P = PSAT - 5.704E-4*(TDB-TWB)*PATM
        W = .62198 * P/(PATM-P)
      ELSE
        WSAT = .62198 * PSAT/(PATM-PSAT)
        W = WSAT - (TDB-TWB)*(0.24 + .441*WSAT)/(597.31
     .       + 0.441*TDB - TWB)
      ENDIF
      IF (W .LT. 0.0) THEN
         WRITE(LUW,3000) INFO(1),INFO(2)
         IWARNS = IWARNS + 1
         W = 0.0
         H = TDB
         TWBOLD = TWB
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```
            WBMODE = 1
            PSAT = PSATDB
            GO TO 60
         END IF
         H = TDB + W*(2501. + 1.805*TDB)
         GO TO 100
C
C  MODE 2 -- DRY BULB AND RELATIVE HUMIDITY SUPPLIED
C
20       IF (RH .LT. 0.) THEN
             WRITE(LUW,3500) INFO(1),INFO(2)
             IWARNS = IWARNS + 1
             RH=0.0
         ELSE IF (PSAT .GE. PATM) THEN
             RHMAX = PATM/PSAT
             IF (RH .GE. (.99*RHMAX)) THEN
                WRITE(LUW,3750) INFO(1),INFO(2)
                STOP
             END IF
         ELSE IF (RH .GT. 1.) THEN
             WRITE(LUW,4000) INFO(1),INFO(2)
             IWARNS = IWARNS + 1
             RH=1.0
         END IF
         W = .62198 * PSAT*RH/(PATM-PSAT*RH)
         GO TO 40
C
C  MODE 3 -- DRY BULB AND DEW POINT SUPPLIED
C
30       W = .62198 * PSAT/(PATM-PSAT)
C
C  FIND ENTHALPY FOR MODES 2 - 4
C
40       IF (PSATDB .LT. PATM) THEN
            WMAX = .62198 * PSATDB/(PATM-PSATDB)
            IF (W .GT. WMAX) THEN
                if (INFO(1) .ne. 31) then
                  WRITE(LUW,4500) INFO(1),INFO(2)
                  IWARNS = IWARNS + 1
                endif
              W = WMAX
            END IF
         END IF
         IF (W .LT. 0.0) THEN
             WRITE(LUW,1500) INFO(1),INFO(2)
             IWARNS = IWARNS + 1
             W = 0.
         END IF
         H = TDB + W*(2501. + 1.805*TDB)
         GO TO 60
C
C  MODE 5 -- DRY BULB AND ENTHALPY SUPPLIED
C
50       IF (PSATDB .LT. PATM) THEN
            WMAX = .62198 * PSAT/(PATM-PSAT)
            HMAX = TDB + WMAX*(2501. + 1.805*TDB)
            IF (H .GT. HMAX) THEN
                WRITE(LUW,5000) INFO(1),INFO(2)
                IWARNS = IWARNS + 1
                H = HMAX
            END IF
```

```
      END IF
      HMIN = TDB
      IF (H .LT. HMIN) THEN
          WRITE(LUW,5500) INFO(1),INFO(2)
          IWARNS = IWARNS + 1
          H = HMIN
      END IF
        W = (H-TDB)/(2501.+1.805*TDB)
C
C  FIND WET BULB TEMPERATURE FOR MODES 2 - 6 IF WBMODE EQUALS 1.
C
60    IF (WBMODE .NE. 1) THEN
         TWB=TDB
         GO TO 100
      END IF
      DPRESS = ABS(1.-PATM)
C  THE FOLLOWING CORRELATION IS FOR 1 ATMOSPHERES TOTAL PRESSURE.
C  IF OUTSIDE THE CORRELATION RANGE, THE CORRELATION IS USED FOR
C  THE INITIAL GUESS IN THE ITERATIVE METHOD.
      IF (H .GT. 0. .AND. H .LT. 2000.) THEN
          Y = ALOG(H*.43002+7.687)
          IF (H .LE. 9.473) THEN
             TWB=-17.4422+1.9356*Y+.7556*Y**2+.5406*Y**3
          ELSE IF (H .GT. 9.473) THEN
             TWB=-.6008-22.04556*Y+11.4356*Y**2-.97667*Y**3
          END IF
       ELSE
          TWB = 9.99999E25
       END IF
C  USE A NEWTON'S ITERATIVE METHOD TO FIND THE WET BULB
C  TEMPERATURE.
       IF (DPRESS.GT..001 .OR. H.LE.0. .OR. H.GT.275.) THEN
          ITEST=0
          IF (ABS(PATM-PATMOLD) .GT. 1.0E-10) THEN
          CALL BOIL(PATM,TBOIL)
          PATMOLD = PATM
          END IF
C-- INITIAL GUESS
          TWBNEW=AMIN1(TWB,(TBOIL-0.1),TDB)
70        IF (TWBNEW .GE. (TBOIL-0.09)) TWBNEW=TBOIL-0.1
          CALL SAT(INFO,TWBNEW,PSAT,IWARNS)
          WSSTAR=.62198*PSAT/(PATM-PSAT)
          IF(MODE .EQ. 5) THEN
            W = (H-TDB)/(2501.+1.805*TDB)
          END IF
          WNEW=((2501.-2.381*TWBNEW)*WSSTAR-(TDB-TWBNEW))/
     .         (2501.+1.805*TDB-4.186*TWBNEW)
          ERR = W - WNEW
          IF (ABS(ERR) .LE. (.01*W)) GO TO 75
          IF (W .EQ. 0.) THEN
             IF (ABS(ERR) .LE. .0001) GO TO 75
          END IF
          ITEST = ITEST + 1
          IF (ITEST .GE. 25) GO TO 75
C--FIND THE SLOPE OF THE ERROR FUNCTION
          TSLOPE = 0.999*TWBNEW
          IF (TWBNEW .EQ. 0.) TSLOPE = -.005
          CALL SAT(INFO,TSLOPE,PSLOPE,IWARNS)
          WSSLP=.62198*PSLOPE/(PATM-PSLOPE)
          WSLOPE=((2501.-2.381*TSLOPE)*WSSLP-(TDB-TSLOPE))/
     .         (2501.+1.805*TDB-4.186*TSLOPE)
          ERRSLP = W - WSLOPE
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          DERRDT = (ERRSLP - ERR)/(TSLOPE - TWBNEW)
          TWBNEW = TWBNEW - ERR/DERRDT
          GO TO 70
75        TWB = TWBNEW
      END IF
      IF (TWB .LT. TWBOLD) TWB = TWBOLD
100   CONTINUE
C
C  FIND RELATIVE HUMIDITY DEW POINT, MIXTURE DENSITY
C  AND DRY AIR DENSITY
C
      PV  = PATM*W/(.62198+W)
      IF(MODE .NE. 2) RH = PV/PSATDB
      IF(MODE .NE. 3 .AND. PV .GT. 0.) THEN
        CALL DEWPT(INFO,PATM,W,TDP,IWARNS)
      ELSE IF (MODE .NE. 3 .AND. PV .LE. 0) THEN
C  FOR DRY AIR, THERE IS NO DEW POINT TEMPERATURE
        TDP=-9.99999E25
      ENDIF
      SPCVOL = RA*(TDB+273.15)/(PATM*101325)*(1+1.6078*W)
      RHOWA = 1/SPCVOL
      RHOWM = RHOWA*(1+W)
C
C  CONVERT OUTPUTS TO APPROPRIATE UNITS
C
      IF(IUNITS .EQ. 2) THEN
        H   = H*0.43002+7.68
        TDB = 1.8*TDB + 32.
        TWB = 1.8*TWB + 32.
        IF (TDP .GT. -9.99999E24) THEN
         TDP = 1.8*TDP + 32.
        END IF
        RHOWM = RHOWM/16.02
        RHOWA = RHOWA/16.02
      ENDIF
C SET OUTPUTS
      OUT(1) = TDB
      OUT(2) = TWB
      OUT(3) = RH
      OUT(4) = TDP
      OUT(5) = W
      OUT(6) = H
      OUT(7) = RHOWM
      OUT(8) = RHOWA
C
      IF (IWARNS .GE. LIMIT) THEN
         WRITE(LUW,6000) IWARNS
         STOP
      END IF
      RETURN
C
250   FORMAT(/20X,'*** ERROR IN UNIT',1X,I2,1X,'T-E',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS
     . SUBROUTINE WAS CALLED WITH AN ATMOSPHERIC PRESSURE',/6X,
     .'LESS THAN OR EQUAL TO 0.')
500   FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'T-E',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS SUBROUTINE WAS CALLED
     . WITH A TOTAL PRESSURE',/6X,'GREATER THAN 5 ATMOSPHERES.
     .  BEWARE THAT THE IDEAL GAS RELATIONS USED',/6X,'IN THE
     . PSYCHROMETRICS ARE NOT ACCURATE AT HIGHER PRESSURES.')
1000   FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'T-E',1X,I2,
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     .1X,'***',/6X,'THE PS√HROMETRICS SUBROUTINE WAS CALLED
     . WITH AN ENTHALPY LESS',/6X,'THAN POSSIBLE FOR THE GIVEN
     . HUMIDITY RATIO.  THE AIR WAS ASSUMED TO',/6X,'BE
     . SATURATED AT THE GIVEN HUMIDITY RATIO AND THE ENTHALPY
     . SET FOR',/6X,'THIS CONDITION.')
1500 FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'TYPE',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS SUBROUTINE WAS CALLED
     . WITH A HUMIDITY RATIO LESS',/6X,'THAN 0.0.  DRY AIR WAS
     . ASSUMED AND THE HUMIDITY RATIO SET TO 0.0.')
1750 FORMAT(/20X,'*** ERROR IN UNIT',1X,I2,1X,'TYPE',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS
     . SUBROUTINE WAS CALLED WITH A WET BULB TEMPERATURE',/6X,
     .'ABOVE THE MAXIMUM FOR THE GIVEN ATMOSPHERIC PRESSURE.')
2000  FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'TYPE',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS
     . SUBROUTINE WAS CALLED WITH A WET BULB TEMPERATURE',/6X,
     .'GREATER THAN THE DRY BULB TEMPERATURE.  SATURATED AIR
     . WAS ASSUMED',/6X,'AND THE WET BULB TEMPERATURE SET TO THE
     . DRY BULB TEMPERATURE.')
2250 FORMAT(/20X,'*** ERROR IN UNIT',1X,I2,1X,'T-E',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS
     . SUBROUTINE WAS CALLED WITH A DEW POINT TEMPERATURE',/6X,
     .'ABOVE THE MAXIMUM FOR THE GIVEN ATMOSPHERIC PRESSURE.')
2500 FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'TYPE',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS SUBROUTINE WAS CALLED
     . WITH A DEW POINT TEMPERATURE',/6X,
     .'GREATER THEN THE DRY BULB TEMPERATURE.  THE DEW
     . POINT WAS SET EQUAL',/6X,'TO THE DRY BULB TEMPERATURE.')
3000  FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'TYPE',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS
     . SUBROUTINE WAS CALLED WITH A WET BULB TEMPERATURE',/6X,
     .'BELOW THAT FOR DRY AIR AT THE GIVEN DRY BULB TEMPERATURE.
     .  DRY AIR',/6X,'WAS ASSUMED AND A NEW WET BULB TEMPERATURE
     . FOUND.')
3500 FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'T-E',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS
     . SUBROUTINE WAS CALLED WITH A RELATIVE HUMIDITY',/6X,
     .'LESS THAN 0.  THE RELATIVE HUMIDITY WAS SET TO 0.0.')
3750 FORMAT(/20X,'*** ERROR IN UNIT',1X,I2,1X,'T-E',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS
     . SUBROUTINE WAS CALLED WITH A RELATIVE HUMIDITY',/6X,
     .'GREATER THAN POSSIBLE AT THE GIVEN DRY BULB TEMPERATURE.')
4000 FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'T-E',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS
     . SUBROUTINE WAS CALLED WITH A RELATIVE HUMIDITY',/6X,
     .'GREATER THEN 1.0.  THE RELATIVE HUMIDITY WAS SET TO 1.0.')
4500 FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'TYPE',1X,I2,
     .1X,'***',/6X,'THE PSYCHROMETRICS
     . SUBROUTINE WAS CALLED WITH A HUMIDITY RATIO ABOVE',/6X,
     .'THE SATURATION HUMIDITY RATIO FOR THE GIVEN DRY BULB
     . TEMPERATURE.',/6X,'THE AIR WAS ASSUMED SATURATED AT THE
     . GIVEN DRY BULB TEMPERATURE',/6X,' AND THE HUMIDITY RATIO
     . SET FOR THIS CONDITION.')
5000 FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'TYPE',1X,I2,
     .1X,' ***',/6X,'THE PSYCHROMETERICS SUBROUTINE WAS CALLED
     . WITH AN ENTHALPY GREATER',/6X,'THEN THAT FOR SATURATED
     . AIR AT THE GIVEN DRY BULB TEMPERATURE.  THE',/6X,'AIR
     . WAS ASSUMED SATURATED AND THE ENTHALPY SET FOR THIS
     . CONDITION.')
5500 FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'T-E',1X,I2,
     .1X,' ***',/6X,'THE PSYCHROMETERICS SUBROUTINE WAS CALLED
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     . WITH AN ENTHALPY LESS',/6X,'THEN THAT FOR DRY'
     . AIR AT THE GIVEN DRY BULB TEMPERATURE.  DRY',/6X,'AIR
     . WAS ASSUMED AND THE ENTHALPY SET FOR THIS CONDITION.')
6000  FORMAT(/30X,'*** ERROR ***',/6X,'THE SIMULATION WAS HALTED
     . BECAUSE THERE WERE',1X,I3,1X,'WARNINGS FROM THE',/6X,
     .'PSYCHROMETRICS SUBROUTINE.  CHECK FOR PROPER USE OF
     . THIS SUBROUTINE.')
      END
C
C***********************************************************************
C
C  SUBROUTINE FOR FINDING SATURATION PRESSURE OF WATER AT A GIVEN
C  TEMPERATURE
C
      SUBROUTINE SAT(INFO,TIN,PSAT,IWARNS)
      DIMENSION INFO(10)
      COMMON/LUNITS/LUR,LUW,IFORM
C
C  THE FOLLOWING CORRELATION FOR THE SATURATION PRESSURE OF
C  WATER VAPOR (IN PASCALS) AS A FUNCTION OF TEMPERATURE IS
C  TAKEN FROM THE 1985 ASHRAE FUNDAMENTALS HANDBOOK (SI).
C
      DATA C1/-5674.5359/,C2/6.3925247/,C3/-0.9677843E-2/
      DATA C4/0.62215701E-6/,C5/0.20747825E-8/,C6/-0.9484024E-12/
      DATA C7/4.1635019/,C8/-5800.2206/,C9/1.3914993/
      DATA C10/-0.048640239/,C11/0.41764768E-4/,C12/-0.14452093E-7/
      DATA C13/6.5459673/
C
      T = TIN + 273.15
      IF (T .LE. 0.) THEN
         WRITE(LUW,200) INFO(1),INFO(2)
         STOP
      END IF
C
C  SATURATION PRESSURE OVER ICE (-100 C TO 0 C)
C
      IF (T .LT. 273.15) THEN
         PSAT=EXP(C1/T+C2+C3*T+C4*T**2+C5*T**3+C6*T**4+C7*ALOG(T))
C
C  SATURATION PRESSURE OVER LIQUID WATER (0 C TO 200 C)
C
      ELSE IF (T .GE. 273.15) THEN
         PSAT=EXP(C8/T+C9+C10*T+C11*T**2+C12*T**3+C13*ALOG(T))
      END IF
C
C  TEMPERATURE OUT OF THE RANGE USED FOR THE CORRELATION
C
      IF (T .LT. 173.15 .OR. T .GT. 473.15) THEN
          WRITE(LUW,100) INFO(1),INFO(2)
          IWARNS = IWARNS + 1
      END IF
C  CONVERT PRESSURE FROM PASCALS TO ATMOSPHERES
      PSAT = PSAT/101325
      RETURN
C
100   FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'T-E',1X,I2,
     .1X,'***',/6X,'THE CORRELATION, FROM THE
     . 1985 ASHRAE FUNDAMENTALS HANDBOOK, USED',/6X,'TO CALCULATE
     . THE WATER VAPOR SATURATION PRESSURE WAS INTENDED FOR',
     ./6X,'THE TEMPERATURE RANGE OF -100 C TO 200 C.  THE
     . PSYCHROMETRICS',/6X,'SUBROUTINE WAS CALLED WITH CONDITIONS
     . OUTSIDE THIS RANGE.')
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200   FORMAT(/20X,'*** ERROR IN UNIT',1X,I2,1X,'T-E',1X,I2,
     .1X,'***',/6X,'THE "SAT" SUBROUTINE OF THE PSYCHROMETRICS
     . SUBROUTINE WAS CALLED',/6X,'WITH A TEMPERATURE BELOW -273 C.')
      END
C
C******************************************************************
C
C  SUBROUTINE FOR FINDING THE DEW POINT TEMPERATURE GIVEN
C  THE HUMIDITY RATIO.  THE CORRELATION IS FROM THE 1981
C  ASHRAE FUNDAMETALS HANDBOOK.  THE DEW POINT TEMPERATURE
C  IS IN DEGREES C AND ATMOSPHERIC PRESSURE IN PASCALS.
C
      SUBROUTINE DEWPT(INFO,PATM,W,TDP,IWARNS)
      DIMENSION INFO(10)
      COMMON /LUNITS/ LUR,LUW,IFORM
C
      PV  = PATM*W/(.62198+W)
      Y = ALOG(1.013E05*PV)
      TDP = -35.957 - 1.8726*Y + 1.1689*Y*Y
      IF(TDP .LT. 0.) TDP = -60.45 + 7.0322*Y + 0.3700*Y*Y
C
      IF (TDP.GT.70. .OR. TDP.LT.-60.) THEN
         WRITE(LUW,100) INFO(1),INFO(2)
         IWARNS = IWARNS + 1
      END IF
      RETURN
C
100   FORMAT(/20X,'*** WARNING IN UNIT',1X,I2,1X,'TE',1X,I2,
     .1X,'***',/6X,'THE CORRELATION, FROM THE
     . 1981 ASHRAE FUNDAMENTALS HANDBOOK, USED',/6X,'TO CALCULATE
     . THE DEW POINT TEMPERATURE, IS INTENDED FOR THE',
     ./6X,'TEMPERATURE RANGE OF -60 C TO 70 C.  THE PSYCHROMETRICS
     . SUBROUTINE',/6X,'WAS CALLED WITH CONDITIONS OUTSIDE THIS
     . RANGE.')
      END
C
C******************************************************************
C
C SUBROUTINE FOR FINDING THE BOILING TEMPERATURE OF WATER GIVEN
C THE TOTAL PRESSURE.  A NEWTON'S METHOD IS USED WITH THE
C SATURATED WATER VAPOR PRESSURE CORRELATION FROM THE 1985 ASHRAE
C FUNDAMENTALS HANDBOOK.
C
      SUBROUTINE BOIL(PATM,TBOIL)
C
      DATA C1/-5674.5359/,C2/6.3925247/,C3/-0.9677843E-2/
      DATA C4/0.62215701E-6/,C5/0.20747825E-8/,C6/-0.9484024E-12/
      DATA C7/4.1635019/,C8/-5800.2206/,C9/1.3914993/
      DATA C10/-0.048640239/,C11/0.41764768E-4/,C12/-0.14452093E-7/
      DATA C13/6.5459673/
C
      PBOIL = PATM*101325
      ITEST = 0
      IF (PBOIL .LT. 611.21) GO TO 100
C--USING AHRAE CORRELATION FOR 0 C TO 200 C.
C  FIRST GUESS TBOIL EQUALS 100 C.
      T1 = 100 + 273.15
10    ZZ=C8/T1+C9+C10*T1+C11*T1**2+C12*T1**3+C13*ALOG(T1)
      P1=EXP(ZZ)
      ERR = PBOIL - P1
      IF (ABS(ERR) .LE. (.01*PBOIL)) THEN
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            TBOIL = T1 - 273.15
            RETURN
        END IF
        ITEST = ITEST +1
        IF (ITEST .GT. 100) STOP
        DERRDT=-(P1)*(-C8/T1**2+C10+2*C11*T1+3*C12*T1**2+C13/T1)
        TOLD = T1
        T1 = T1 - ERR/DERRDT
        GO TO 10
C
100     CONTINUE
C--USING AHRAE CORRELATION FOR -100 C TO 0 C.
C  FIRST GUESS TBOIL EQUALS 0 C.
        T1 = 273.15
110     ZZ=C1/T1+C2+C3*T1+C4*T1**2+C5*T1**3+C6*T1**4+C7*ALOG(T1)
        P1=EXP(ZZ)
        ERR = PBOIL - P1
        IF (ABS(ERR) .LE. (.01*PBOIL)) THEN
            TBOIL = T1 - 273.15
            RETURN
        END IF
        ITEST = ITEST +1
        DERRDT=-P1*(-C1/T1**2+C3+2*C4*T1+3*C5*T1**2+4*C6*T1**3+C7/T1)
        TOLD = T1
        T1 = T1 - ERR/DERRDT
        GO TO 110
        END
```

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