SUPERMARKET REFRIGERATION OPTIONS

by

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ABSTRACT

Approximately 2% of the nation's annual electrical consumption can be traced to supermarket refrigeration equipment. Combined with the relatively low COP of refrigeration equipment, even small increases in supermarket refrigeration system performance can significantly affect annual energy consumption. The work documented in this thesis presents the annual results from the simulation of four refrigeration systems in a supermarket. The four refrigeration systems analyzed in this report consisted of a fixed head pressure system, a floating head pressure system, an ambient subcooling system, and a dedicated mechanical subcooling cycle. The fixed head pressure system is a common refrigeration system and was the basis against which all other systems were compared.

The floating head pressure system was found to reduce electrical energy consumption in cool climates due to the decrease in condensing pressure at low ambient temperatures. By removing the fixed head pressure mechanism, the floating head pressure system eliminates the inefficiencies associated with fixed head pressure operation at low ambient temperatures. However, the floating head pressure system did

not reduce electrical demand and achieved only minimal savings against the fixed head pressure system in warm climates

With the addition of a heat exchanger downstream of the condenser that rejected excess heat to the ambient, the ambient subcooling system was able to increase COP over the range of operating conditions common to commercial refrigeration. The increased COP led to reduced energy consumption and demand costs when compared to the fixed head and floating head pressure systems.

The use of a second vapor-compression cycle solely for the purpose of providing subcooling to the main cycle constituted the dedicated mechanical subcooling cycle. The second refrigeration cycle provided around 70 degrees of subcooling to the main cycle at design conditions due mainly to the low temperature sink provided by the evaporation of the subcooling refrigerant. Using the dedicated mechanical subcooling cycle, substantial savings of refrigeration energy and demand were realized over the fixed head pressure system. The dedicated subcooling cycle was found to perform best at high ambient temperatures and low refrigeration temperatures. When the constraint of fixed head pressure was added to the dedicated subcooling cycle, the system still significantly outperformed the standard fixed head pressure system. The optimum temperature for the evaporation of the subcooling refrigerant for the dedicated subcooling cycle was found to be a constant regardless of the system operating parameters. Finally, design guidelines for achieving the optimal COP of the dedicated subcooling cycle as a function of the thermal sizes of the subcooler, main cycle condenser and subcooling cycle condenser were established.

ACKNOWLEDGEMENTS

After six years at this university, and nineteen years of school its THE DAY! Yes its the day that I've been dreading since I opened my very first book; SEE SPOT RUN (by the way, where does SPOT run to anyway?). The day when I have to finally give up the student life. The day when I just can't milk this thesis for any longer. The day when I can't justify staying out till 4 a.m., sleeping till noon, getting up and eating stale Cheetos and warm Pepsi for breakfast. The day when the bonus monthly checks from ma & pa stop coming to the house. The day I can't skip a lecture on something like "impact strength of raisins " and head to the lake for a little skiing with my friends. The day where I might have to finally start acting like an .... an ADULT? What's that ? Yeah you're right - NO CHANCE!!! Whoever said that "the day you grow up is the day you start growing old" hit it right on the head. Seriously though, this whirlwind tour of life and graduate school wouldn't have been possible without a lot of help and support along the way.

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(P.S. SPOT was running away from the horrible stick people DICK and JANE. I think they were adults and SPOT knew it. RUN SPOT RUN!)

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NOMENCLATURE

ROMAN SYMBOLS

Symbol	Definition
AIC	allowable incremental cost
AS	anti-sweat
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
BTU	British Thermal Unit
C	capacitance rate
Cp	specific heat
CC	case credit
COP	coefficient of performance
d	depreciation rate

ROMAN SYMBOLS cont.

Symbol	Definition
def	defrost
dsc	degrees of subcooling
dsh	degrees of superheat
E	initial equipment cost
E	power
EES	Engineering Equation Solver
EPRI	Electric Power Research Institute
F	first year operating cost savings
f	humidity multiplier
h	enthalpy
i	inflation rate
i_{fuel}	fuel inflation rate
kWh	kilo-Watt hour
LCS	life cycle savings
LMTD	log mean temperature difference
len	length of refrigerated case
m	refrigerant flow rate
misc	fans and lights
N	number of years in economic analysis
NTU	number of transfer units

ROMAN SYMBOLS cont.

Symbol	Definition
P	pressure
P1	present worth factor associated with operating costs
P2	present worth factor associated with equipment costs
plf	part-load factor
plr	part-load ratio
psia	pounds per square inch absolute
PWF	present worth factor
Q	heat transfer
RH	relative humidity
s	entropy
T	temperature
t	income tax rate
T_{amb}	ambient temperature
T_{design}	temperature at which the heat exchangers sizes are to be distributed
t_p	property tax rate
ton	ton of refrigeration (12000 BTU/hr)
TMY	typical meteorological year
TRNSYS	transient simulation program
UA	conductance-area product
W	Watt

ROMAN SYMBOLS cont.

Symbol	Definition
W	work
x	quality of vapor

GREEK SYMBOLS

Symbol	Definition
Δ	difference
ε	effectiveness
η	efficiency
ρ	density
ω	humidity ratio

SUBSCRIPTS

Symbol	Definition
amb	ambient
AS	anti-sweat
c	cold
comp	compressor
comp2	subcooling cycle compressor
cond	condenser
cond2	subcooling cycle condenser
ded	dedicated mechanical subcooling
def	defrost
evap	evaporator
fixed	fixed head pressure
float	floating head pressure
h	hot
i	inlet
id	ideal
is	isentropic
lat	latent
low	low temperature cases
main	main refrigeration cycle
max	maximum

SUBSCRIPTS cont.

Symbol	Definition
med	medium temperature cases
min	minimum
ms	multi-shelf cases
o	outlet
out	outlet
rated	rated capacity
ref	refrigerant
ref2	refrigerant in subcooling cycle
sens	sensible
sub	subcooling cycle
tot	total

CHAPTER ONE

INTRODUCTION

According to a recent study, there are approximately 35,000 supermarkets in the United States today [1]. These supermarkets use about 4% of the nation's annual electrical energy [1] with 55 to 60% of that energy being used for refrigeration [2]. In fact, the annual cost of energy can be equal to the store's yearly profit [3]. Besides just total electrical energy use, another important consideration is electrical demand. Supermarkets tend to use the most energy on hot summer afternoons; coinciding with the utilities peak demand [1]. Therefore, reducing refrigeration energy use can be seen to be important to both the store owners and the utilities. For these reasons, this study targets various means of reducing supermarket refrigeration energy costs.

1.1 SCOPE OF STUDY

Refrigeration equipment accounts for a significant portion of the energy consumed in supermarkets today. In addition, the Coefficient of Performance (COP) of refrigeration is quite low. Therefore, even a small change in the system COP can greatly affect the system operating costs. This study investigates the potential for increasing the refrigeration system COP by changing the operating system. A logical procession of system configurations will be explored and compared. The relative advantages and disadvantages of each system will be discussed, and annual results generated.

Since a fixed head pressure vapor compression system represents the most common refrigeration system used in supermarkets today, the discussion starts with this cycle. The system will be defined and described, and the trends that affect system performance investigated.

The next logical step from a fixed head pressure system is to a floating head pressure system. Floating head pressure implies that the condenser pressure is allowed to "float" with the ambient conditions. The advantages and disadvantages of floating head pressure systems will be discussed, and comparisons made to the fixed head pressure system.

Ambient subcooling represents the next step towards improvement of system COP. Ambient subcooling entails the addition of a small heat exchanger downstream of the condenser that allows the ambient air to subcool the refrigerant leaving the condenser.

The performance of the ambient subcooling system will be compared to the two previous systems and conclusions drawn about the effectiveness of ambient subcooling.

The use of an additional vapor compression cycle solely to subcool the refrigerant leaving the condenser is defined as dedicated mechanical subcooling. This next step towards the goal of improved system COP will be investigated. Sensitivity analyses will be run on such parameters as: ambient temperature, evaporator temperature, sub-cycle evaporator temperature, heat exchanger sizes and heat exchanger flow rates. An optimization will be conducted to investigate the effects of heat exchanger size distribution on system performance. From these results some system design criteria will be formulated and explained. Finally, the results will be summarized and compared to the previous systems.

The four systems mentioned above constitute the range of refrigeration systems investigated. Once the systems have been compared and contrasted, the effects of part load ratio on system performance will be evaluated. A short economic analysis will follow the systems portion of this study and will attempt to give dollar values to the energy savings achieved by each system. From the economic results, conclusions will be drawn about the four refrigeration systems studied.

1.2 LITERATURE SEARCH

Although there are literally hundreds of studies on fixed head pressure, floating head pressure and ambient subcooling, there are few published studies on dedicated mechanical subcooling. Most of the work is done by refrigeration contractors and manufacturers [4].

Foster-Miller included dedicated subcooling in their EPRI report titled "Supermarket Refrigeration Modeling and Field Demonstration" [5]. Foster-Miller found that dedicated subcooling was directly responsible for a savings of about 10 to 15% for their system simulations. However, the tests were performed at high ambient temperatures and further testing is needed at lower ambient temperatures before a seasonal evaluation is possible.

A paper by R.J. Couvillion, M.W. Larson. and M.H. Somerville describes the benefits of a mechanical subcooling system for the grocery industry [4]. Their computer model predicted improvements in COP ranging from 6 to 82%, and in capacity from 20 to 170%, for ambient temperatures in the range 80 to 120°F. The effect of different refrigerants was also included in the test. The most notable result was that dedicated subcooling was most effective at the extremes of operation; high ambient temperatures or low evaporator temperatures.

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CHAPTER TWO

SYSTEM MODELLING

In order to evaluate the various refrigeration systems on an equal basis, annual simulations were performed using the TRNSYS [1] simulation program. To better understand the results of these tests, some discussion of the way the systems were modelled must be included. This chapter addresses the reasons for choosing the parameters and inputs for the refrigerated display case and supermarket models to be used in this simulation.

2.1 SUPERMARKET MODEL

To allow the potential savings of the refrigeration systems to be estimated, a store model had to be developed. The store inputs are not representative of a specific store, but are chosen to represent a typical supermarket today.

The supermarket is assumed to be open 24 hours a day, seven days a week. The store set points are maintained at 75°F dry-bulb and 55% relative humidity. Although the dry-bulb set point is not extremely important for the refrigeration concepts, the humidity set point is important and is be discussed in the next section. The refrigerated display cases are divided into three different types; the low-temperature reach-in cases (Low), the medium-temperature single shelf cases (Med), and the medium-temperature multi-shelf cases (MS). The low-temperature cases are assumed to have an evaporator set point temperature of -20°F, and the medium temperature cases are assumed to have a set point of 20°F. For the store model, it is assumed that there are 300 linear feet of low-temperature cases corresponding to a design cooling capacity of 180,000 BTU/hr or 15 tons, 300 linear feet of medium-temperature single shelf cases corresponding to a design cooling capacity of 180,000 BTU/hr or 15 tons, and 210 linear feet of medium-temperature multi-shelf cases corresponding to a design cooling load of 315,000 BTU/hr or 26.25 tons. The design cooling loads are shown graphically in Figure 2.1.

The annual simulations were run for two different cities using Typical Meteorological Year (TMY) weather data [2]. The cities that were chosen for this simulation were Madison, WI and Miami FL. Madison was chosen because of the large temperature extremes that are found between summer and winter, and Miami was chosen for its relatively high ambient temperatures. The Madison annual simulations should provide trends that show how the systems would operate at the seasonal extremes, while the Miami annual simulations should show how the systems would operate near constant system capacity.

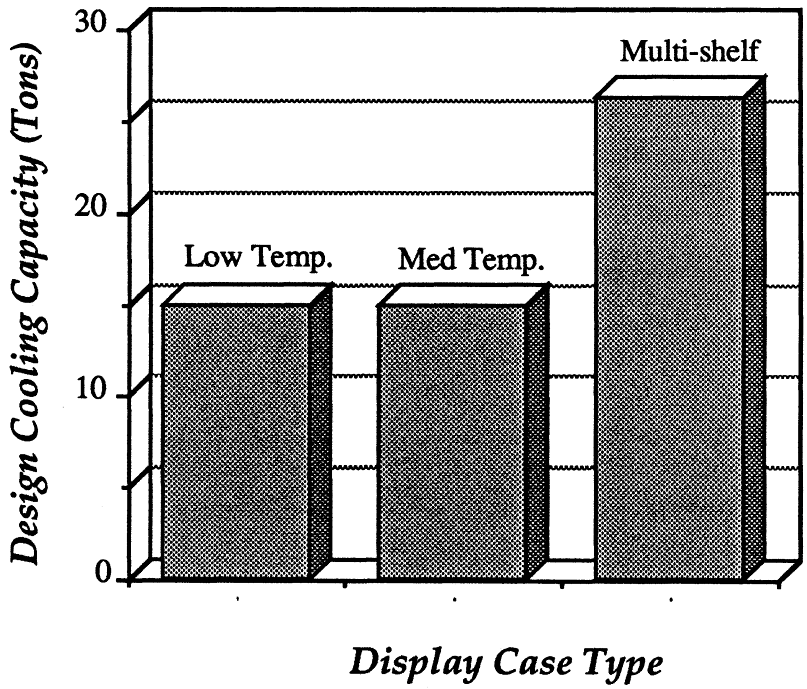

Figure 2.1 Design cooling capacity for the three refrigerated display case types.

To perform the annual simulations, a TRNSYS [1] simulation deck was written. TRNSYS [1] is a modular system simulation program that takes user supplied subroutines and "plugs" them together to form a "deck". The TMY [2] weather data, the store model and the refrigerated case model were combined to form the simulation TRNSYS [1] deck to be used for this study. The deck is contained in Appendix A for reference purposes. A system simulation diagram is included as Figure 2.2.

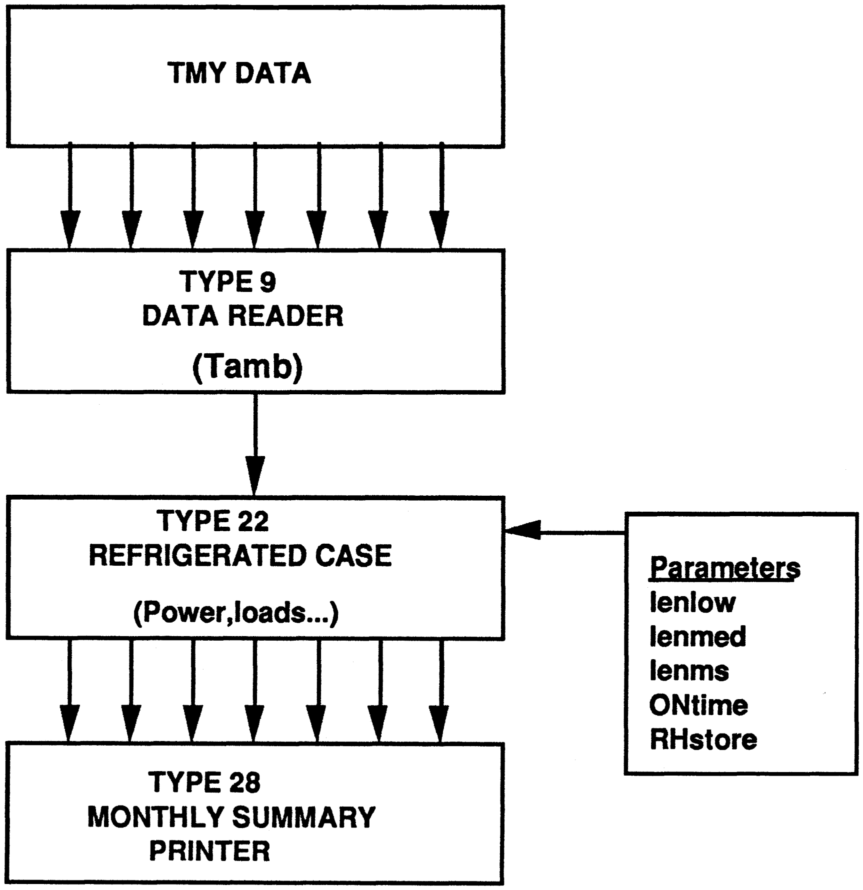

Figure 2.2 System simulation diagram for annual results.

2.2 REFRIGERATED DISPLAY CASES

The refrigerated display cases are responsible for determining the loads on the refrigeration equipment. This section addresses the relations behind the display case models and attempt to describe the variables which affect the system loads.

The refrigerated display cases are broken up into three representative types; the low temperature reach-in cases, the medium temperature single shelf cases, and the medium temperature multi-shelf cases. The discussion that follows, applies for all three case types.

The refrigerated display case models that were used in this study were written as TRNSYS [1] components, and employ a mechanistic model to determine the refrigerated case loads and power requirements. Data were taken from Hussman (Hussman, 1989) to produce representative values for the design cooling capacity and the power consumption due to defrost, anti-sweat heaters, fans and lights. The values that were chosen for typical design cooling capacities, defrost loads, anti-sweat loads, and fan and light loads are listed in Table 2.1 and correspond to values taken from Hussman at 55% relative humidity. Since the data from Hussman did not take into account the relative humidity of the store, humidity correlations were needed for the cooling capacity, defrost load and the anti-sweat load.

The correlation that was used for the cooling capacities and the defrost load was from a detailed case model prepared by EPRI [3]. The EPRI [3] correlation multiplier is shown below as a function of the percent relative humidity. It should be noted that the defrost load only applies to the low temperature display cases, and not to the medium temperature cases.

$$f = -0.1 + 0.02 * RH \qquad (2.1)$$

If the multiplier falls below zero, it is set to zero.

The effect of the relative humidity on the anti-sweat power could not be found directly from the Hussman data, so a simple mechanistic model was developed by Lindell [4]. The model predicts the energy needed to raise the temperature of the outside surface of the case above the dew point temperature of the store air. The results, which were correlated with the Hussman data, are listed in Table 2.2. For the purposes of this report, it was assumed that the anti-sweat heaters would work continuously at the store set points of 75°F dry bulb and 55% relative humidity.

Some of the load on the refrigerated cases is due to the cooling and dehumidification of the store air near the cases. Although the cooling and dehumidification of the air is a load on the cases, it is a benefit or *credit* to the air conditioning system. The cooling of the air is a sensible case credit to the store while the dehumidification of the air is a latent case credit to the store. The latent case credits on the store air were modelled using the Lindell method [4], and were found to be 12% of the cooling capacity (which is a function of relative humidity) for low temperature cases, and 19% of the cooling capacity for medium temperature cases. The sensible case credits are independent of the relative humidity and are defined as:

$$\begin{aligned} \textit{CCsens} = \ & \textit{Design Cooling Capacity - Fan Load - Lighting Load} \\ & \textit{- Anti-Sweat Load at 55\% RH - Defrost Load at 55\% RH} \\ & \textit{- Latent Case Credits at 55\% RH} \end{aligned} \tag{2.2}$$

The case credits are shown as a function of the relative humidity in Figure 2.3.

To determine the loads that the refrigeration equipment must meet, the respective loads are summed. The resulting equation is shown below.

$$\begin{aligned} \textit{Caseload} = \ & \textit{Fan Load + Lighting Load + Anti-Sweat Load + Defrost} \\ & \textit{Load + Latent Case Credits + Sensible Case Credits} \end{aligned} \tag{2.3}$$

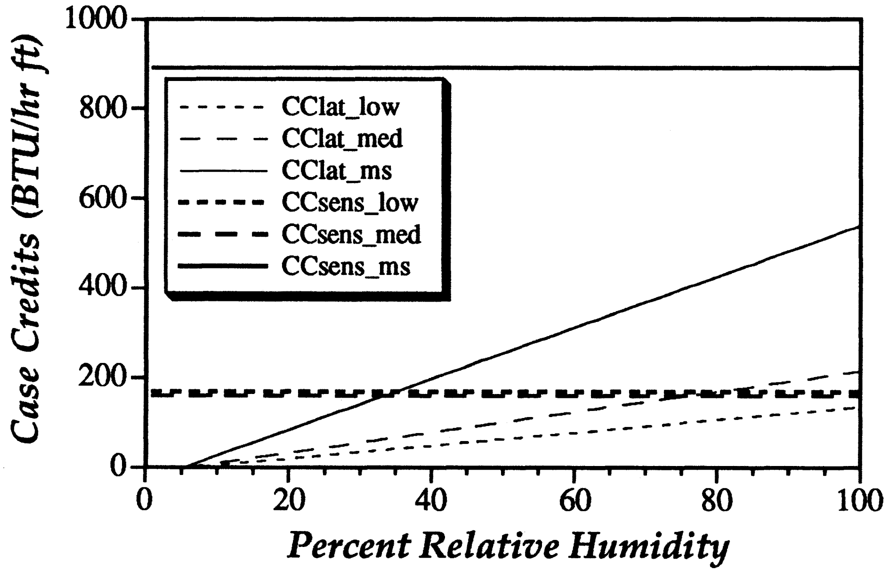

Figure 2.3 Case credits, per linear foot of display case, as a function of relative humidity.

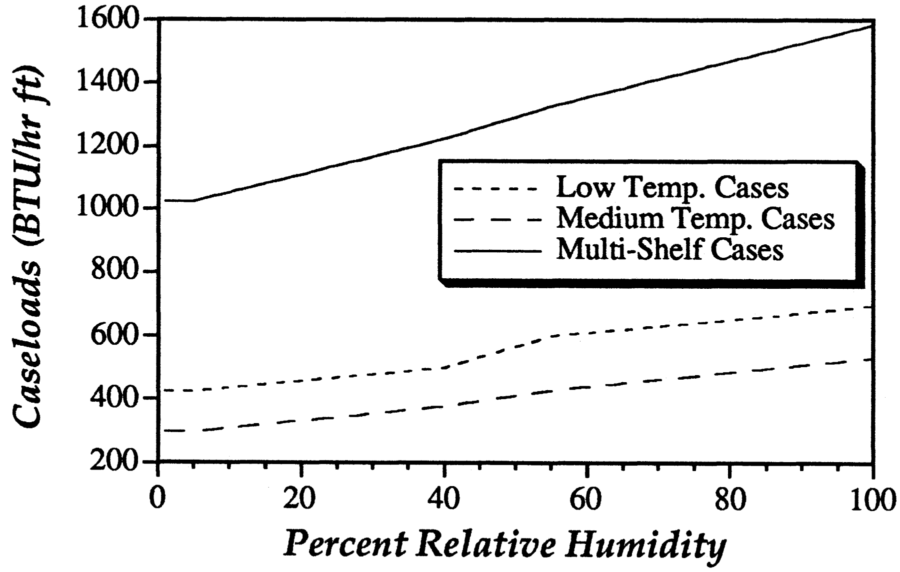

Figure 2.4 Caseloads, per linear foot of display case, as a function of relative humidity.

Figure 2.4 shows the relationships between the caseloads and the relative humidity for the three display case types. The refrigerated case evaporator set point temperature (20°F for low temperature systems and 20°F for medium temperature systems) and the corresponding caseloads, are the inputs for the various type of refrigeration systems to be studied in the following chapters.

The case power consumed by the refrigerated cases is now defined as:

$$\textit{Case Power} = \textit{Caseload} \; / \; \textit{COP} \qquad (2.4)$$

Since the caseload at each instant is be the same for each refrigeration system, the differences in case power can be directly attributed to the different COP's for each system. The COP of the refrigerated case system was found from detailed steady-state models for the refrigeration system in question. The COP was input to the refrigerated case model as performance equations based on the evaporator and ambient temperatures. The steady-state models and the associated COP's for the systems studied are discussed in the following chapters.

Once the case power has been calculated, the total power can be determined.

$$\begin{aligned}\textit{Total Power} = \textit{Case Power} + \textit{Fan Power} + \textit{Lighting Power} + \\ \textit{Anti-Sweat Power} + \textit{Defrost Power} \qquad (2.5)\end{aligned}$$

It is here that an important distinction must be made. The refrigeration system will be judged based on the case power consumed, not the total power consumed. The reason for this is that the power required for the fans, lights, defrost and anti-sweat heaters is independent of the type of refrigeration *system* chosen. Therefore, it's the savings in case power between the different systems that is the relative effect. Referring to Figure

2.5, it can be seen that the case power constitutes about one-half of the total system power for a representative system.

The refrigerated case model is included in Appendix A for reference.

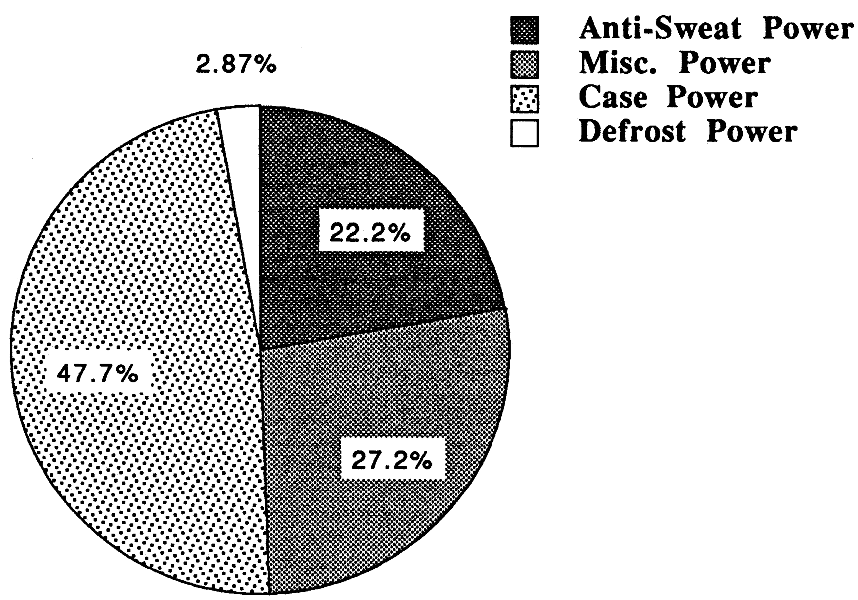

Figure 2.5 Breakdown of total system power for a representative refrigeration system.

2.3 CHAPTER SUMMARY

This chapter dealt exclusively with discussion of the TRNSYS [1] models to be used in the system simulations. The supermarket component was not modelled after a particular store, but rather tries to typify a "common" supermarket. The annual results for this supermarket are based on TMY [2] weather data from two cities, Madison and Miami. The store has 300 feet of low temperature refrigerated cases, 300 feet of medium temperature single-shelf cases, and 210 feet of medium temperature cases for design cooling capacities of 15 tons, 15 tons, and 26.25 tons respectively. The data for the display cases were taken from Hussman catalog data and correlated with multipliers based on the store relative humidity. The COP of the various refrigeration systems was input as performance equations derived from detailed steady-state models. Finally, the various systems to be looked at will be evaluated in terms of case power consumption and not total power consumption due to the similarities in the systems.

PARAMETER	LOW TEMP.	MED. TEMP.	MULTI-SHELF
DESIGN COOLING CAPACITY (BTU/hr/ft)	600	600	1500
FAN POWER (W/ft)	20	20	20
ANTI-SWEAT HEATER POWER (W/ft)	40	10	10
DEFROST POWER (W/ft)	10	0	0
POWER FOR LIGHTS (W/ft)	15	15	15

Table 2.1 Typical refrigerated case values at 55% relative humidity.

MULTIPLIER	55% RH	50% RH	45% RH	40% RH
ANTI-SWEAT HEATER	1.00	0.67	0.33	0.00

Table 2.2 Anti-sweat heater multipliers as a function of relative humidity.

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CHAPTER
THREE

REFRIGERATION CONCEPTS

This chapter includes a quick review of vapor-compression refrigeration thermodynamics. Following the thermodynamics section, the steady-state mechanistic modelling of the refrigeration system components is discussed. The modelling section includes only those components common to each system. Components not common to each system will be discussed in the appropriate chapter. With the component models derived, the integration of these components into a steady-state system model is discussed.

3.1 REVIEW OF REFRIGERATION

Most of the refrigeration systems in existence today, including those which were studied in this paper, are mechanical vapor-compression systems. Figure 3.1 illustrates the typical components in a vapor-compression refrigeration system. The Pressure-Enthalpy diagram for this cycle is also included as Figure 3.2. Refrigeration cycles are characterized by four main processes; condensation, expansion, evaporation and compression. A typical vapor-compression cycle is now described.

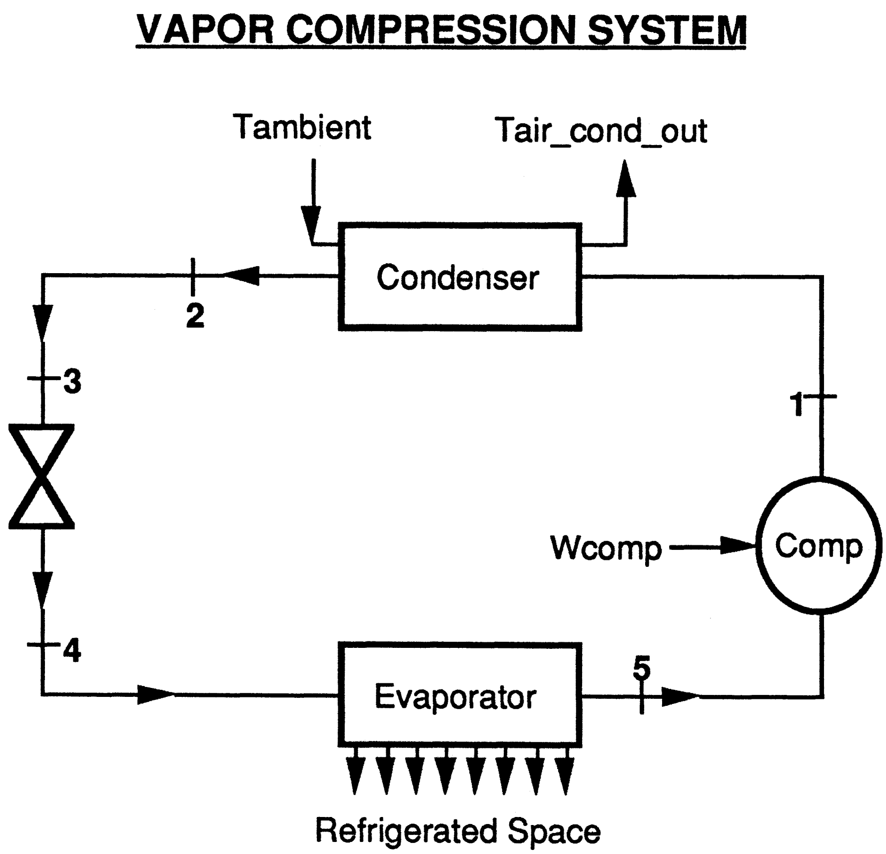

Figure 3.1 Components in a typical vapor-compression cycle.

Refrigerant leaves the evaporator at state 5 as low temperature, low pressure, slightly superheated vapor, and enters the compressor. In the compressor, the refrigerant undergoes an adiabatic compression and exits as high temperature, high pressure superheated vapor at state 1. From here, the vapor enters the condenser where it is first desuperheated and then condensed at constant pressure. Exiting the condenser is a high pressure, medium temperature saturated liquid that corresponds to state 2. This saturated liquid then flows through the expansion valve where it undergoes an adiabatic expansion and enters the evaporator at state 4 as a low temperature, low pressure, low quality vapor. In the evaporator, the low quality refrigerant is evaporated and the process is repeated.

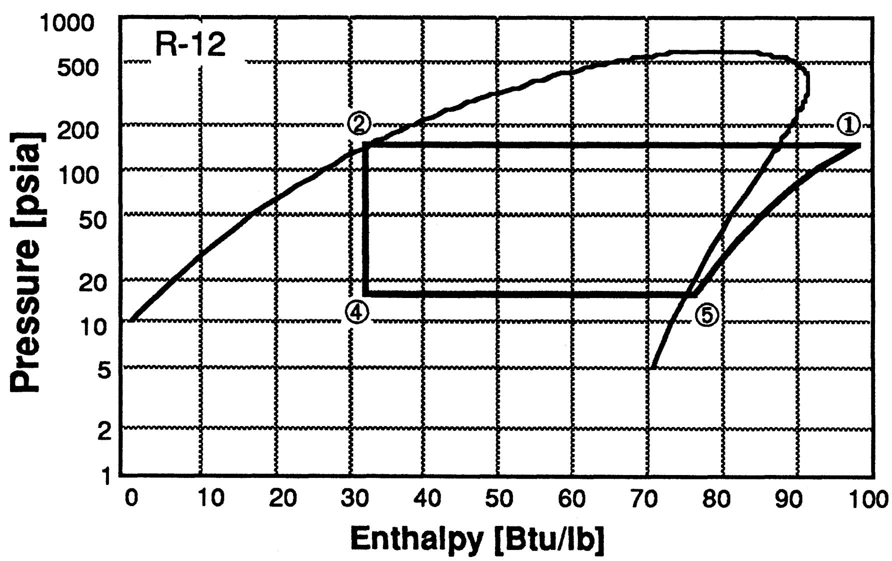

Figure 3.2 Pressure Enthalpy diagram for a typical mechanical vapor compression system.

In practice, the compression and expansion processes are not adiabatic, the heat exchangers have pressure drops, and there are heat and pressure losses in the lines. The components were modelled as discussed above for simplicity; the trends are not expected to depend on these assumptions. Before modelling the steady-state components, a discussion of some of the thermodynamic terms that will be used is included.

The rate that heat is rejected in the condenser is the product of the refrigerant flow rate and the enthalpy difference across the condenser.

$$Q_{cond} = m_{ref} * (h_1 - h_2) \tag{3.1}$$

The refrigeration capacity is defined as the product of the refrigerant flow rate and the enthalpy difference across the evaporator. The capacity also equals the load on the refrigerated cases.

$$Capacity = caseload = m_{ref} * (h_5 - h_4) \tag{3.2}$$

The power supplied to the compressor is the product of the refrigerant flow rate and the enthalpy difference between evaporator exit and condenser inlet.

$$Power = m_{ref} * (h_1 - h_5) \tag{3.3}$$

The COP, which is the measure of the efficiency of the refrigeration cycle, is defined as the capacity of the refrigeration system system divided by the power supplied to the system. Since each system, at each instant of time, has the same caseload, the COP is the true measure of the system performance.

$$COP = Capacity \,/\, Power = Caseload \,/\, Power \tag{3.4}$$

Two terms need to be defined here:

Suction Pressure - the pressure at the compressor inlet.

Head Pressure - the pressure at the compressor discharge.

With these definitions and concepts in place, discussion can proceed with the actual modelling of the systems.

3.2 COMPONENT MODELLING

This section is devoted to the components common to each system. The components that are specific to a system are discussed in that particular chapter. All the systems that are investigated in this report were modelled using Engineering Equation Solver (EES) [1]. EES [1] is a flexible tool for solving large systems of equations. The program solves systems of non-linear equations by a Newton-like algorithm. Built into this program are the refrigerant thermophysical properties that were needed for the simulations. The program also includes parametric tables and optimization algorithms that were used to evaluate the refrigeration systems over a range of conditions. Using EES [1], the relations for each component were developed and then integrated into the steady-state system model.

3.2.1 HEAT EXCHANGERS

All of the heat exchangers in this report were modelled using the Log Mean Temperature Difference (LMTD) approach. The LMTD approach is developed from heat exchanger energy balances on the hot and cool fluids, and assumes that the total heat transfer is proportional to the *mean* value of the temperature difference [2]. The LMTD for a cross-flow or a counter-flow heat exchanger is expressed as:

$$LMTD = ((T_{h,i} - T_{c,o})-(T_{h,o}-T_{c,i})) \,/\, ln\, ((T_{h,i} - T_{c,o}) \,/\, (T_{h,o} - T_{c,i})) \qquad (3.5)$$

The total heat transfer can then be expressed as:

$$Q = UA * LMTD$$

where U represents the overall heat transfer coefficient, and A represents the heat exchanger area. It is assumed that the pressure drop across the heat exchanger is negligible, there are no potential or kinetic energy losses, and the overall heat transfer coefficient (U) remains constant regardless of the flow conditions. To get some measure of the performance of the heat exchanger, the effectiveness of the heat exchangers was calculated. The effectiveness is a measure of the actual heat transfer in the exchanger to the maximum heat transfer in the exchanger [2].

$$\varepsilon = Q_{actual} \,/\, Q_{max} \qquad (3.6)$$

For a counter-flow or a cross-flow heat exchanger with a change in phase of one of the fluids, the governing equation is:

$$\varepsilon = 1 - exp^{(-NTU)} \qquad (3.7)$$

NTU is defined as the Number of Transfer Units, and can be expressed as:

$$NTU = UA \,/\, C_{min} \qquad (3.8)$$

C_{min} is the minimum capacitance rate of the two fluids in question, and is the product of the flow rate and the specific heat of the fluid.

The condensers are assumed to be air-cooled cross-flow heat exchangers with cooling air flow rates of 3800 pounds of air per hour per ton of refrigeration. This value corresponds to a value of about 900 cfm per ton of refrigeration, and is representative of current practice. Performing an energy balance on the condenser, two equations are derived. The states are defined in Figures 3.1 and 3.2.

$$Q_{cond} = m_{ref} * (h_1 - h_2) \quad (3.9)$$

$$Q_{cond} = m_{air,cond} * Cp_{air,cond} * (T_{air,cond,out} - T_{amb}) \quad (3.10)$$

From the LMTD approach described earlier the condenser heat transfer is also described as:

$$Q_{cond} = UA * LMTD \quad (3.11)$$

Because the desuperheating of the refrigerant entering the condenser is a small part of the total heat transfer in the condenser (approximately 10 to 15%), the superheat is not included in the equations for the LMTD. Therefore, the hot side inlet temperature is T2 instead of T1 (this was necessary due to iteration problems). Since the refrigerant is condensing, its temperature remains constant ($T_{h,i} = T_{h,o} = T_2$). The LMTD then becomes:

$$LMTD_{cond} = (T_{amb} - T_{air,cond,out}) \,/\, \ln((T_2 - T_{air,cond,out}) \,/\, (T_2 - T_{amb})) \quad (3.12)$$

For a condensing refrigerant, the effective specific heat of the refrigerant becomes infinity. Therefore, the C_{min} term needed to evaluate the effectiveness must be the capacitance rate of the cooling air.

$$C_{min} = m_{air,cond} * CP_{air} \tag{3.13}$$

$$NTU = UA \;/\; C_{min} \tag{3.14}$$

$$\varepsilon = 1 - exp^{(-NTU)} \tag{3.15}$$

The refrigerant leaving the condenser is assumed to be liquid at the saturation temperature and pressure. Figure 3.3 illustrates the temperature - distance concepts of the condenser for this simulation. With these equations and parameters describing the condenser relations, the performance of the heat exchanger can be evaluated at any given conditions.

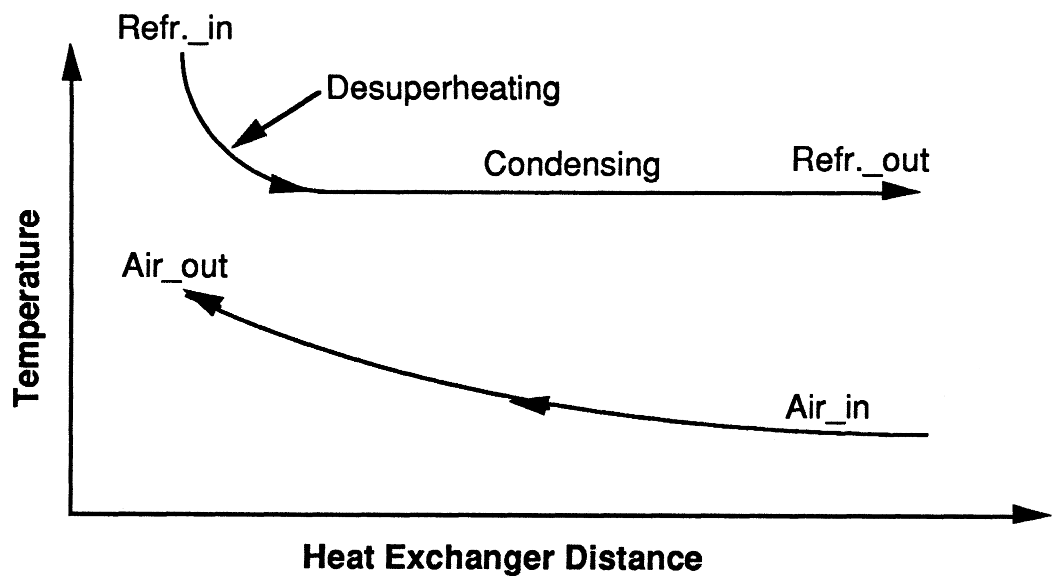

Figure 3.3 Condenser temperature profile as a function of heat exchanger distance.

3.2.2 EVAPORATORS

In most supermarket applications, the refrigerated display cases act as the evaporators for the refrigeration system. The equations describing the refrigerated display cases for the annual simulations were given in chapter 2. However for the steady state model, the only inputs that are needed for the evaporator are the evaporating temperature and the degrees of superheat exiting the evaporator. The evaporator set point temperature is defined by the type of display case being used and was either -20°F for low temperature refrigeration or 20°F for medium temperature refrigeration. The remaining parameter is the degree of superheat. For this simulation, it is assumed that the refrigerant leaves the evaporator with seven degrees of superheat. With these two parameters set, the exit state of the evaporator (state 5) is defined.

T4 = Evaporator temperature

T5 = T4 + Degrees of superheat

P5 = Saturation Pressure at the evaporating temperature (constant pressure evaporation)

Knowing the pressure and temperature of the superheated vapor, the enthalpy and entropy at state 5 can be found.

3.2.3 COMPRESSORS

The steady state compressor models were solved using the concept of isentropic efficiency. It was assumed for this simulation that the compressor was a reciprocating compressor and had negligible heat transfer with the surroundings. For a reciprocating compressor, isentropic efficiency is relatively independent of compressor size for a given refrigerant [3]. To solve for the compressor unknowns, the ideal entropy at state 1 is set equal to the entropy at state 5 (found in previous section). From knowledge of the pressure and ideal entropy at state 1, the ideal enthalpy can be found. The ideal work of compression can then be found.

$$W_{comp,id} = m_{ref} * (h_{1,id} - h_5) \tag{3.16}$$

With the ideal work known, the actual work can be found using the compressor isentropic efficiency.

$$W_{comp} = W_{comp,id} \ / \ \eta_{is} \tag{3.17}$$

The enthalpy at state 1 can now be found.

$$h_1 = h_5 + W_{comp} / m_{ref} \tag{3.18}$$

3.2.4 EXPANSION VALVES

A typical vapor compression refrigeration system contains one expansion device. For this simulation, it was assumed that the expansion device was a thermostatic expansion valve. Thermostatic expansion valves control the refrigerant flow rate in response to the degrees of superheat exiting the evaporator. Its basic function is to control the flow rate so that unevaporated refrigerant is not passed to the compressor [4]. In general, the heat transfer between the valve and the surroundings is quite small, and is neglected. The energy balance on the expansion valve becomes:

$$h_3 = h_4 \tag{3.19}$$

3.3 INTEGRATED MODELS

With the equations derived for the system components, the integrated system model becomes quite easy to formulate. None of the components can be solved for directly because of the interactions between the components. However, when the components are put together, the equations for the entire system can be solved. With this type of integrated format, the impact of changing a system parameter on the system performance can be easily evaluated.

The steady-state models were developed to provide COP performance equations as a function of ambient temperature and evaporator temperature for the annual simulation models. The performance equations that were derived were used in the TRNSYS [5] refrigerated case model to generate the power consumption data.

The steady-state models are based on a design refrigeration load of 15 tons. Even though the results that are discussed in the following chapters are for the 15 ton system, the performance equations generated are independent of the size of the system. For this reason, the performance equations, and hence the TRNSYS [5] refrigerated case model, are quite general in nature and can be used for a wide variety of applications.

3.4 CHAPTER SUMMARY

This chapter describes the modelling of the basic mechanical vapor compression refrigeration system. The components and processes of the typical vapor compression cycle were discussed, and the equations for the component models derived. The condenser was modelled using the Log Mean Temperature Difference approach, and the evaporator was modelled as discussed in Chapter 2. The pressure drop across the evaporator and condenser was assumed to be negligible. The expansion process was assumed to be isenthalpic and the compression process was modelled using isentropic efficiency. Both the expansion and compression processes were assumed to have negligible heat transfer with the surroundings. With the components described, the next step was to describe the system. Even though the unknowns for the components could not be solved singly, as a system they could be solved. With the basic components described, the four refrigeration systems can now be modelled and evaluated. The four chapters to follow describe the steady-state system models used to generate the performance equations for the annual simulations.

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2. Incropera, F.P., and DeWitt, D.P., *Introduction to Heat Transfer*, John Wiley and Sons, New York, 1985

3. Couvillion, R.J., Larson, M.W., and Somerville, M.H., *Analysis of a Vapor-Compression Refrigeration System with Mechanical Subcooling*, ASHRAE Transactions, Vol. 94 Part 2, 1988

4. *ASHRAE Handbook, Equipment Volume, 1983*, American Society of Heating, Refrigeration, and Air-Conditioning Engineers Inc., Atlanta, Georgia

5. Klein, S.A., et al., TRNSYS: *A Transient Simulation Program*, University of Wisconsin - Madison, Engineering Experiment Station Report 38-12, Version 13.1

CHAPTER
FOUR

FIXED HEAD PRESSURE

A fixed head pressure system is commonly employed in commercial refrigeration. Fixed head pressure systems keep the condensing pressure above some minimum set point to ensure adequate system operation. However, by keeping the condensing pressure at a set point, inefficiencies are introduced into the system. This chapter discusses the system operation, explore the reasons for fixed head pressure, and develop trends that to be used for this system and the systems to follow.

4.1 FIXED HEAD SYSTEMS

Fixed head pressure systems are refrigeration systems that employ some means of keeping the condensing pressure or *head* pressure at or above some minimum value. As long as the condensing pressure is above the set point, the system modulates with the ambient conditions. The system modulates with the ambient conditions due to the change in potential heat transfer across the condenser. As the ambient temperature rises, the condensing temperature must rise in order to reject heat to the ambient. When the ambient temperature falls, the condensing temperature can correspondingly fall However when the ambient conditions dictate that the head pressure could fall below the set point, a means of keeping the head pressure at the set point is utilized. The fixed head pressure system diagram and Pressure - Enthalpy diagrams are shown in Figures 4.1 and 4.2 respectively.

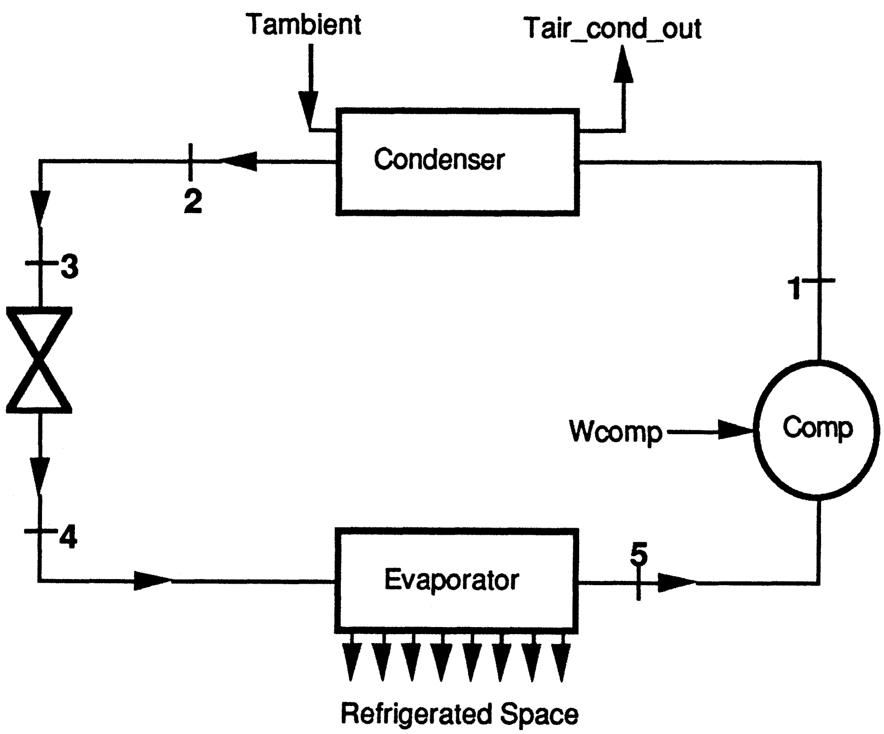

Figure 4.1 Components for a fixed head pressure refrigeration cycle.

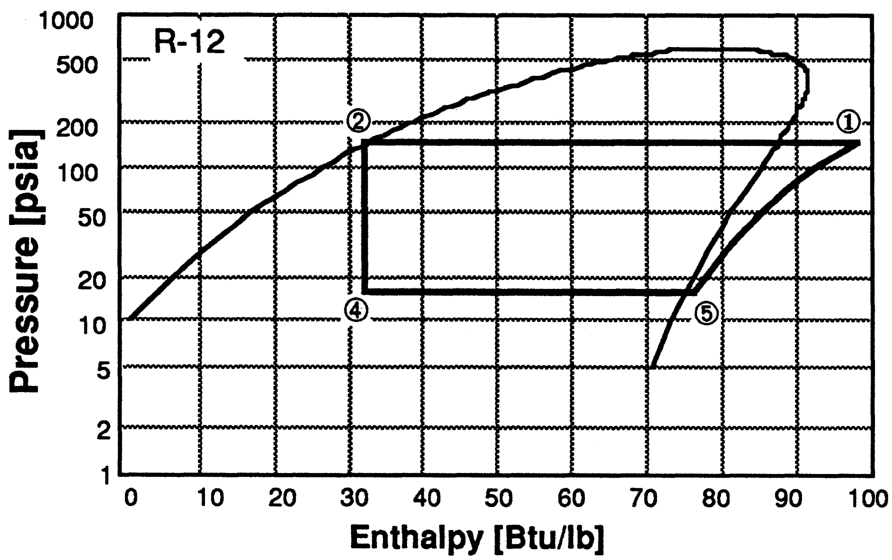

Figure 4.2 Pressure Enthalpy diagram for a fixed head pressure system.

4.2 REASONS FOR FIXED HEAD

There are several reasons why the head pressure must be kept at some minimum value. Fixing the head pressure ensures that the compressor operates properly, and that there is a large enough pressure differential for adequate refrigerant flow through the thermostatic expansion valve. The head pressure is important to the compressor due to the minimum compression ratio needed for proper operation. With a lowered pressure differential across the expansion valve, the refrigerant flow rate is decreased. This decrease in flow rate lowers the available capacity of the evaporator and causes the evaporator pressure to increase [1]. Both these problems are undesirable for commercial refrigeration. Figure 4.3 illustrates the concept of reduced expansion valve pressure differential for low ambient temperatures.

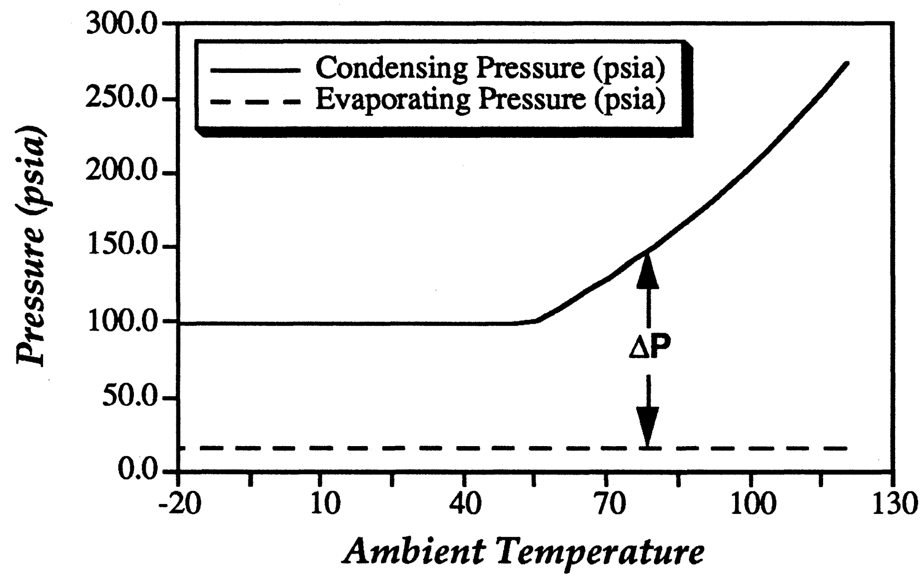

Figure 4.3 Expansion valve pressure differential as a function of ambient temperature.

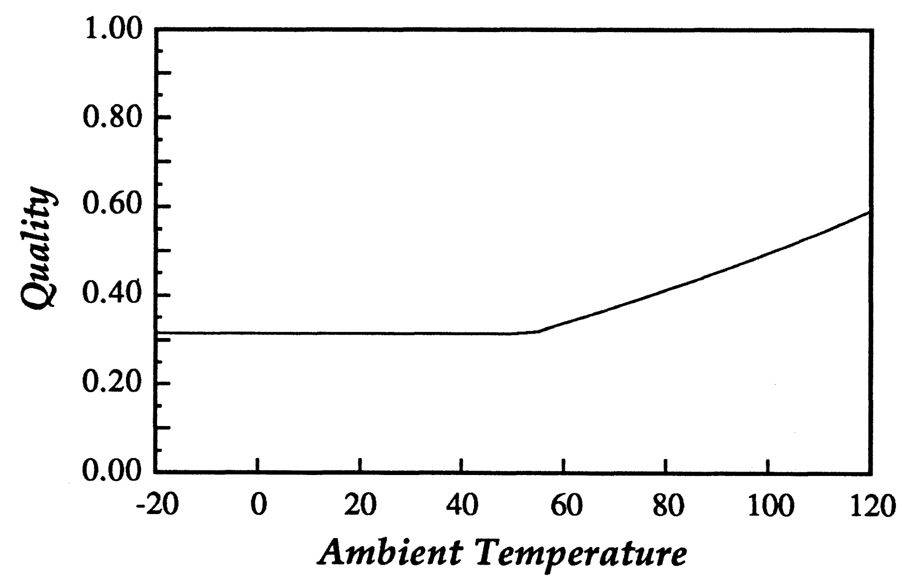

Figure 4.4 Evaporator inlet quality as a function of ambient temperature.

Also, by fixing the head pressure the minimum liquid quality is maintained which is important for proper evaporator and expansion valve performance. If the quality at the evaporator inlet is too low, the heat transfer to the evaporator may not be enough to completely vaporize the refrigerant entering the compressor. Refer to Figure 4.4 for a plot of evaporator inlet quality as a function of ambient temperature.

In commercial refrigeration practice today, there are many ways to control the head pressure. A few of the control methods are described below.

Limiting the condenser cooling air flow rate or temperature reduces the available heat transfer thereby raising the condensing pressure and temperature. Refer to section 3.2.1, heat exchangers, to review the mechanisms behind this concept.

Hot gas bypassing consists of diverting some of the hot gas from the condenser inlet back to the low pressure or suction side. This is a way of loading the compressor artificially to maintain proper system operation.

A pressure amplifier is a device that requires a work input to pressurize the fluid at the head conditions. The pressure amplifier is usually placed between the exit of the condenser and the expansion valve inlet.

Gravity head implies that the condenser is set at some height above the expansion valve in order to utilize gravity as a means of increasing the pressure at the entrance to the expansion valve.

Regardless of the method used to fix the head pressure, inefficiencies are introduced into the system. For this report, the method of achieving fixed head pressure is not specified. It is only assumed that the head pressure is fixed at some minimum set point. For steady-state modelling purposes, the head pressure is fixed at 100 psia corresponding to a minimum condensing temperature of about 82°F for refrigerant-12. This pressure was chosen because it represents a typical value used in refrigeration system practice. Because the head pressure method is not specified, there are assumed to be no additional work inputs to the system. Therefore, the model represents a conservative estimate of the penalties associated with fixed head pressure. The program that models the fixed head pressure system and the solutions for this system are included in Appendix B for reference.

4.3 RESULTS AND TRENDS

Because the refrigerated caseload for the steady-state model is set at a constant 15 tons, the only factor affecting the COP is the amount of work done on the system (equation 3.4). Therefore, the variables that affect the work are important to the performance. The variables that affect compressor work through the change in condenser conditions are the ambient temperature, the size of the condenser and the condenser cooling air flow rate. These three variables affect the heat rejection to the ambient which in turn sets the compressor discharge pressure. The compressor isentropic efficiency directly affects the amount of work done by the compressor (equation 3.17). Finally, the evaporator temperature sets the compressor suction pressure which influences the compressor work.

The cooling air flow rate and the compressor isentropic efficiency parameters have been set (sections 3.2.1 and 3.2.3). The condenser size was set and is discussed in Chapter 7. With these three parameters set, the only variables left that affected the system performance were the ambient temperature and the evaporator temperature. This was the goal of the steady-state modelling, to get system performance equations as a function of the ambient and evaporator temperatures.

4.3.1 AMBIENT TEMPERATURE

In the condenser, the pressure and the temperature change with the ambient conditions. As the ambient temperature falls, the condensing pressure and temperature fall until the minimum set point is reached. At this point, the pressure remains at the set point pressure regardless of the falling ambient temperatures. Above this set point the system "floats" with the ambient conditions. At or below the ambient conditions corresponding to this set point, the system is "fixed". Figure 4.5 shows the dependence of COP on the ambient temperature.

Figure 4.5 shows the two sections of the curve; the "fixed" section and the "floating" section. The point at which the two curves intersect corresponds to the set point. The refrigerant flow rate as a function of ambient temperature is shown in Figure 4.6. Referring to the figure, it can be seen that the flow rate decreases as the ambient temperature falls. The evaporator inlet quality decreases also (Figure 4.4), with a corresponding decrease in the entering evaporator enthalpy. With the capacity (caseload) fixed and the evaporator exit enthalpy fixed (h_5), the decrease in entrance enthalpy

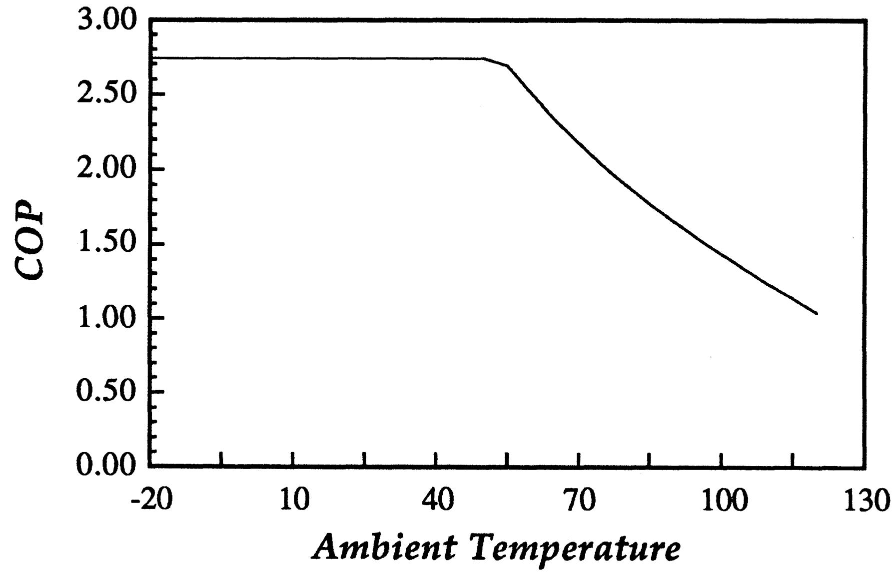

Figure 4.5 Coefficient of Performance of a fixed head pressure system as a function of the ambient temperature.

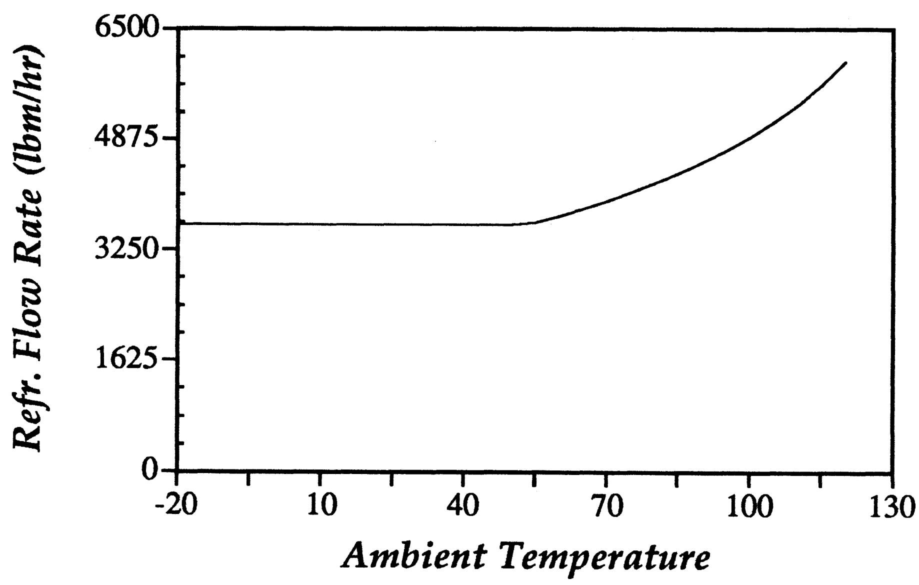

Figure 4.6 Refrigerant flow rate as a function of the ambient temperature for a fixed head pressure system.

corresponds to a decrease in the required flow rate. The results can be seen in Figure 4.6 and the governing equation is shown below.

$$m_{ref} = caseload \;/\; (h_5 - h_4) \qquad (4.1)$$

The compressor work decreases as the ambient temperature falls for two of the reasons mentioned above; the condensing pressure falls (Figure 4.3), and the refrigerant flow rate required to meet the load is decreased (Figure 4.6). The compressor work is shown as a function of the ambient temperature in Figure 4.7. Since the COP is inversely proportional to the compressor work, the COP increases as the compressor work decreases.

Every curve in this section has a characteristic horizontal portion at low ambient temperatures. The condenser is the only component to interact with the ambient conditions. Once this interaction is removed by fixing the pressure, the system operation remains the same regardless of the outdoor conditions.

4.3.2 EVAPORATOR TEMPERATURE

As the evaporating temperature increases for a given head pressure, the pressure differential across the compressor decreases and the work correspondingly decreases. With the capacity held constant, the COP then increases as the evaporator temperature increases. This trend can be seen in Figure 4.8. There is no characteristic horizontal section of this curve because the evaporator conditions do not affect the head pressure. Even if the ambient temperature was below that which fixed the head pressure, the COP curve would retain the same shape.

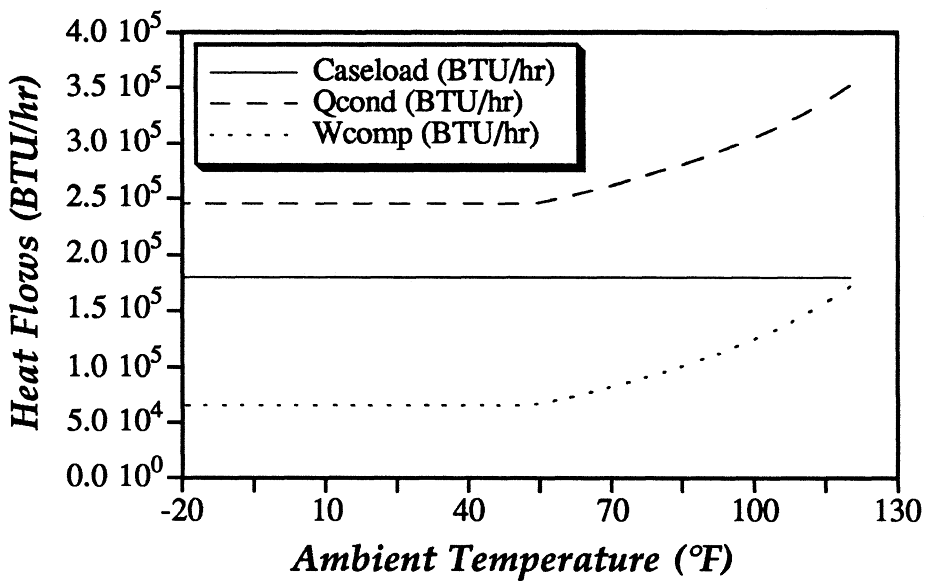

Figure 4.7 Component heat flows as a function of the ambient temperature for the fixed head pressure system.

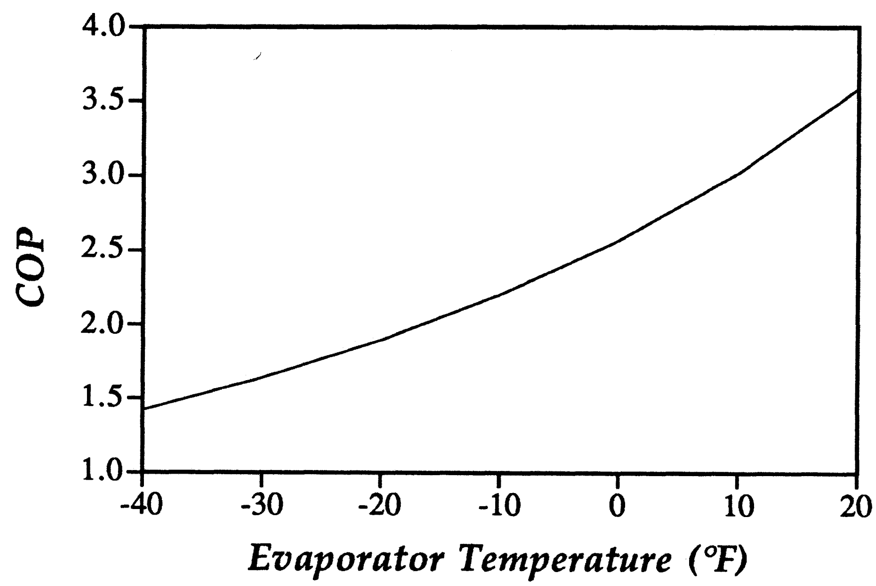

Figure 4.8 COP as a function of evaporator temperature for a fixed head pressure system.

4.3.3 PERFORMANCE CURVES

The refrigeration systems were modelled as steady-state systems since the transient effects are small and are neglected. The steady-state program was used in order to generate performance curves based on ambient and evaporator temperature. These performance curves were then curve-fit to derive the performance equations that were input to the annual simulation program. Because in practice the evaporator temperature is known and the ambient temperature changes continuously, the performance curves were variable in ambient temperature for fixed values of the evaporator temperature. Figure 4.9 shows the performance curves that were curve-fit to solve for the performance equations.

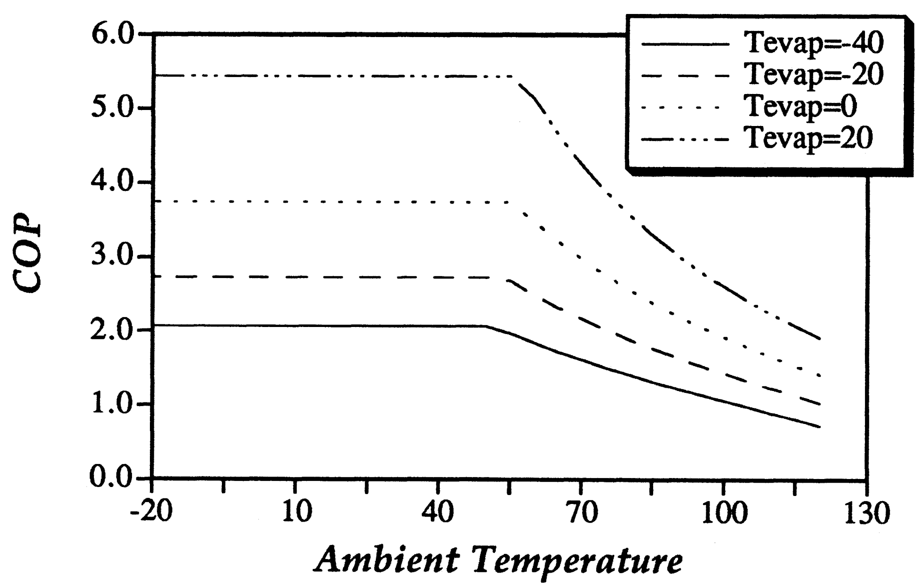

Figure 4.9 Performance curves for the fixed head pressure system.

Appendix B contains the steady-state model, a steady-state solution, and the steady-state performance equations that were input to the fixed head pressure annual simulations.

4.4 CHAPTER SUMMARY

Fixed head pressure systems are systems that artificially keep the condensing pressure at some minimum value as the ambient temperature falls. The reasons for keeping the head pressure at some set point is to ensure proper system operation. The method of fixing the head pressure, and the inefficiencies associated with each method were discussed. Since the steady-state model assumes no work input to maintain the head pressure at the fixed value, it represents a conservative solution. It was shown that the fixed head pressure performance curves are independent of ambient temperature below a certain point. To increase the COP of the fixed head pressure system, the ambient temperature could be lowered, or the evaporator temperature raised. This concept is referred to as decreasing the "thermal" lift of the cycle.

REFERENCES 4

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CHAPTER
FIVE

FLOATING HEAD PRESSURE

Floating head pressure systems provide a performance improvement over fixed head pressure systems at low ambient temperatures. Floating head pressure systems utilize the low condensing pressure associated with low ambient conditions to further improve the COP. In this manner, they eliminate the inefficiencies associated with fixed head systems. This chapter investigates the concept of floating head pressure and compare the performance results to the fixed head pressure system.

5.1 FLOATING HEAD SYSTEMS

The components of the floating head pressure system are shown in Figure 5.1, and are identical to those for the fixed head pressure system. In fact, at ambient temperatures above those which cause the fixed head pressure system to maintain the set point, the two systems act exactly alike. It is only at low ambient conditions that the floating head pressure system becomes effective. The floating head pressure system utilizes expansion valves that allow for greater extremes of operation. The expansion valves have greater capacity and offer a decreasing resistance for a decreasing pressure differential [1]. In this way, an adequate refrigerant flow rate can be maintained at low pressure differentials. For the steady-state model that was developed, it was assumed that the head pressure was allowed to "float" continuously with the ambient conditions without degrading the system performance. The steady-state model, and solutions to this model are included in Appendix C for reference.

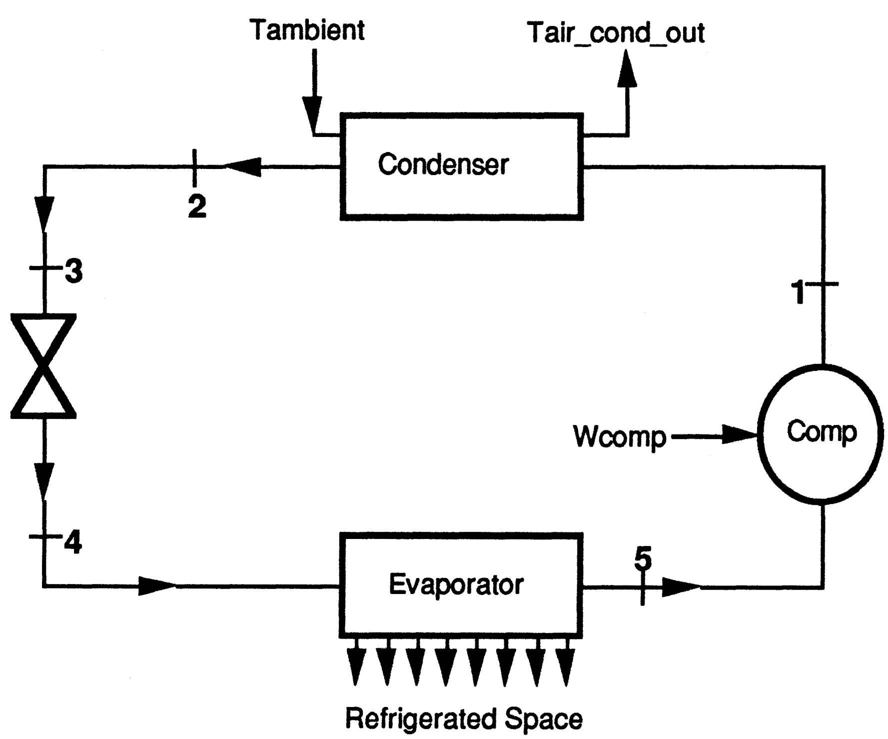

Figure 5.1 System diagram for a floating head pressure system.

5.2 RESULTS AND TRENDS

The reasons the floating head pressure systems outperform the fixed head pressure systems at low ambient temperatures is directly related to the condensing pressure. At low ambient temperatures the fixed head pressure system maintains a constant head pressure while the floating head system allows the condensing pressure to fall. This decrease in pressure affects the compressor in two ways; the discharge pressure is decreased, and the refrigerant flow rate is decreased. Both these factors combine to reduce the compressor work and therefore increase the COP.

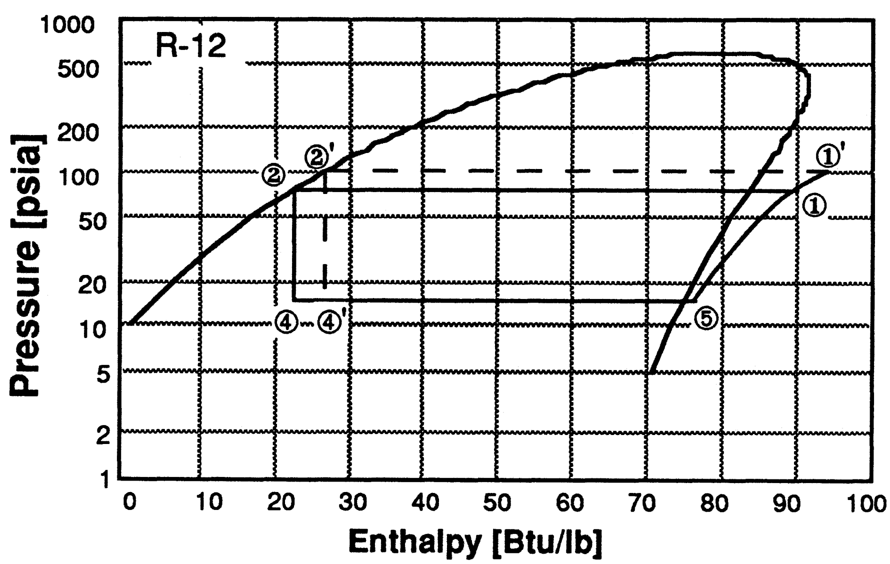

Figure 5.2 Pressure - Enthalpy diagram for a floating head pressure system at low ambient temperatures. The solid lines refer to a floating head pressure system, while the dashed lines represent a fixed head pressure system.

Referring to Figure 5.2, this "double" affect of the lowered condensing pressure can be seen. The compressor discharge pressure is lowered, point 1 as compared to point 1', reducing the compressor work, and there is additional capacity per pound of refrigerant (point 4 to point 4'). The added capacity per pound of refrigerant circulated results in a decreased flow rate for a fixed capacity simulation (capacity is defined as $m_{ref} * (h_5 - h_4)$).

Like the fixed head pressure systems discussed in Chapter 4, the COP of the floating head pressure system only depends on the ambient temperature and the evaporator temperature. The other three parameters (compressor isentropic efficiency, condenser flow rates, and condenser size) that would have affected system performance have been set to the same values as those for the fixed head system. Figure 5.3 shows the relationship between the COP and the ambient temperature for the fixed head and the floating head pressure systems.

Figure 5.4 shows the relationship between the two systems as a function of evaporator temperature for an ambient temperature of 40°F, which is below the critical fixed head pressure value. At higher ambient temperatures (above the set point), the two system COP curves would coincide.

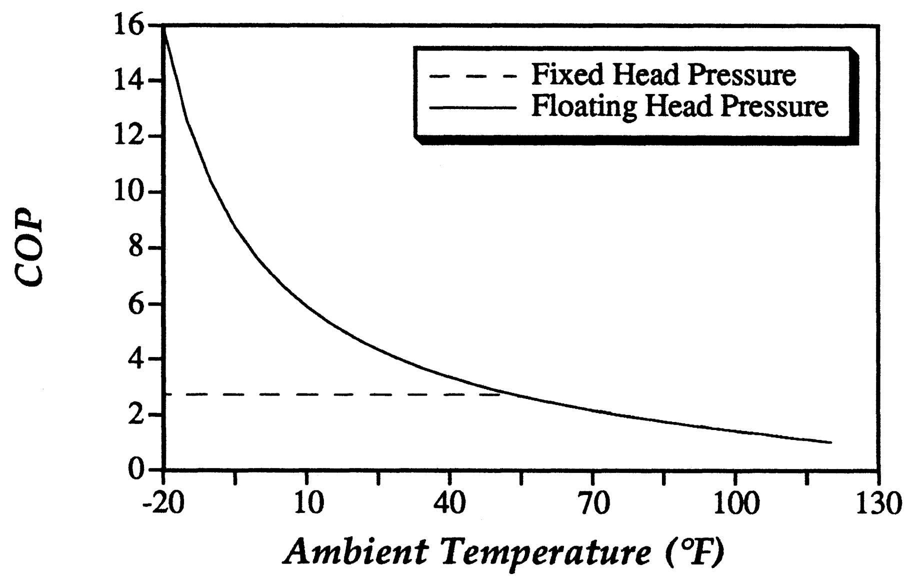

Figure 5.3 COP comparison for the fixed head pressure and floating head pressure systems as a function of the ambient temperature.

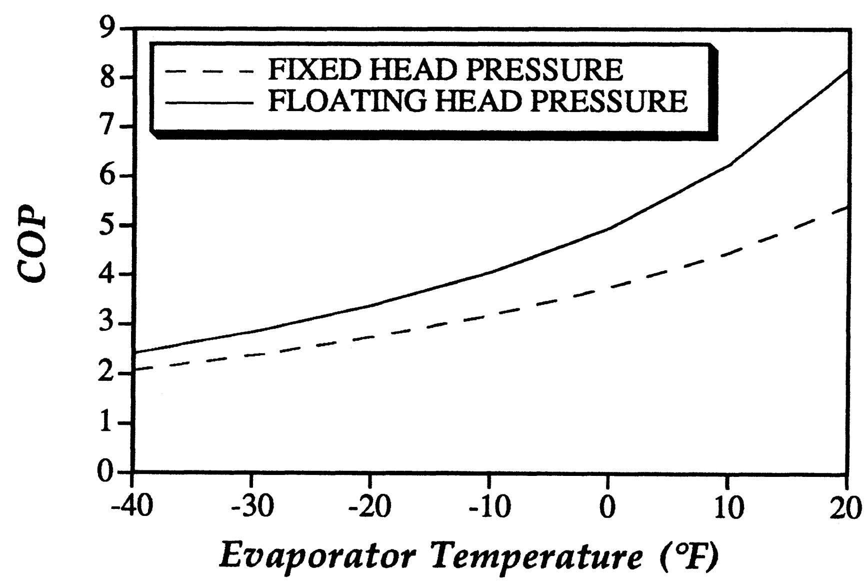

Figure 5.4 COP comparison as a function of the evaporator temperature for low ambient temperatures.

5.3 PERFORMANCE CURVES

With the ambient temperature and evaporator temperature effects derived, the performance curves for the floating head pressure system can be evaluated. Similar to the fixed head pressure system, the performance curves were variable in ambient temperature for fixed values of the evaporator temperature. Figure 5.5 shows the performance curves that were used to generate the performance equations for the annual simulations. The performance equations are listed in Appendix C.

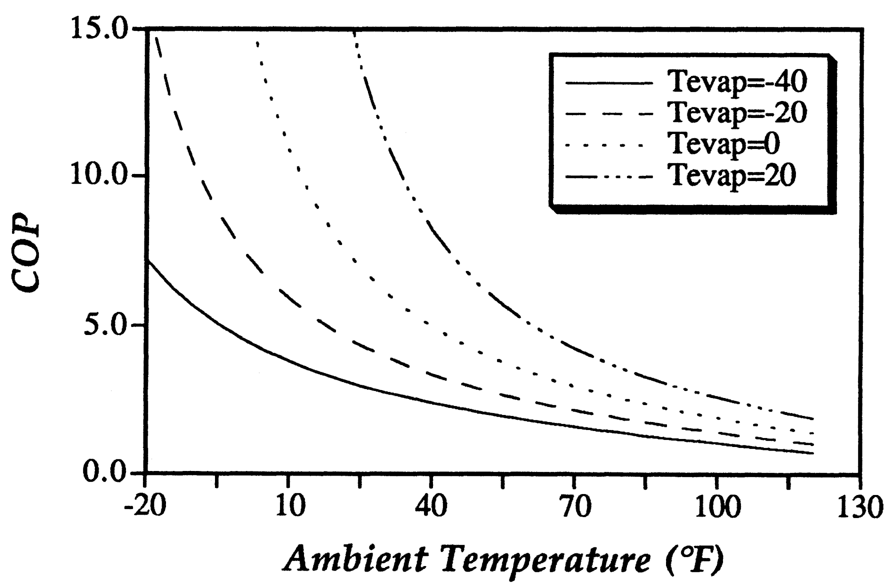

Figure 5.5 Performance equations for the floating head pressure system.

5.4 CHAPTER SUMMARY

Floating head pressure systems represent an improvement over fixed head pressure systems at low ambient temperatures. Through improved components, the floating head pressure systems can modulate with the ambient at temperatures below which cause the fixed head pressure systems to become set. In this manner, floating head pressure systems eliminate the inefficiencies associated with fixed head pressure systems. The trends that affect system performance were discussed and the floating head pressure performance equations were developed from the steady state model.

REFERENCES 5

1. Nakao, M., Ohshima, K. and Uekusa, T., *Control Method of a Cooling Apparatus in Low Outdoor Air Temperatures*, ASHRAE Transactions, Vol. 96 Part 1, 1990

CHAPTER
SIX

AMBIENT SUBCOOLING

Ambient subcooling refrigeration systems utilize a small heat exchanger downstream of the condenser that interacts with the ambient. The small heat exchanger provides additional cooling to the saturated liquid refrigerant exiting the condenser that allows the ambient subcooling system to outperform the floating head and fixed head systems. This chapter investigates the performance of the ambient subcooling systems and discusses the parameters that affect the system performance.

6.1 AMBIENT SUBCOOLING SYSTEMS

Ambient subcooling systems outperform floating head pressure and fixed head pressure systems due to the presence of an additional heat exchanger which allows for increased heat rejection to the ambient. The heat exchanger or *subcooler* subcools the saturated liquid refrigerant leaving the condenser. The method of providing ambient subcooling can be one of two types. The subcooler and condenser can be combined in an oversized condenser, or the subcooler can be a separate heat exchanger located downstream of the condenser. For steady-state modelling, it was assumed that the heat exchanger was a separate unit and consisted of a cross-flow heat exchanger with subcooled liquid refrigerant and ambient air as the two fluids. The cooling air flow rate across the subcooler was assumed to be in proportion to the condenser cooling air flow rate; 3800 lbm air / hr / ton of refrigeration [1]. In practice, the subcooler is usually sized to provide 15 degrees of subcooling at design conditions [2]. The subcooler was modelled using the LMTD approach as discussed in Section 3.2.1. For the steady-state model, the subcooler UA size was chosen so that it provided 15 degrees of subcooling at an ambient temperature of 80°F and an evaporator temperature of -20°F. The subcooler UA was found to be about one-twentieth the size of the condenser for the design parameters chosen. The ambient subcooling system diagram is shown in Figure 6.1.

6.2 RESULTS AND TRENDS

The ambient subcooling system was compared against the two systems previously discussed; fixed head pressure and floating head pressure. Because the head pressure of the ambient subcooling system can either "float" or be "fixed" at low ambient temperatures, two ambient subcooling steady-state models were developed and compared against their equivalent non-subcooled system.

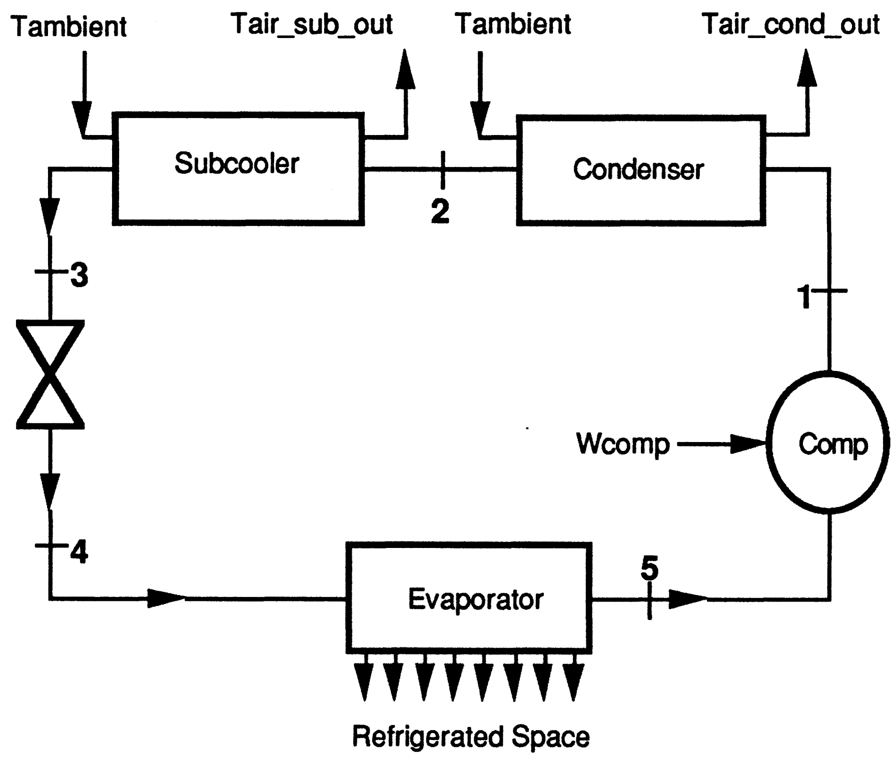

Figure 6.1 System diagram for an ambient subcooling system.

6.2.1 AMBIENT SUBCOOLING WITH FLOATING HEAD PRESSURE

Ambient subcooling systems with floating head pressure outperform standard floating head pressure systems over the range of ambient temperatures studied. The increased performance of the ambient subcooling system is due to the subcooling at the expansion valve inlet. Subcooling allows the refrigerant to enter the evaporator with a lower quality than the systems previously discussed. The lower quality of the refrigerant at the evaporator inlet allows for a higher refrigeration capacity per pound of refrigerant circulated. For a fixed caseload simulation, this implies that the refrigerant flow rate was decreased. The head pressure of the ambient subcooling system was slightly lower than that of the floating head pressure system due to a decrease in the condenser heat transfer (about 10% reduction at design conditions). The combined effects of the lowered head pressure and decreased refrigerant flow rate cause a decrease in the amount of work supplied to the compressor. The ambient subcooling cycle is shown on a Pressure-Enthalpy diagram in Figure 6.2

The increase in refrigeration capacity per pound of refrigerant circulated can be seen as the enthalpy difference between points 4 and 4' in Figure 6.2. This increase in capacity also equals the amount of subcooling. The amount of subcooling is defined as the enthalpy difference between the saturated liquid line at the head pressure and point 3.

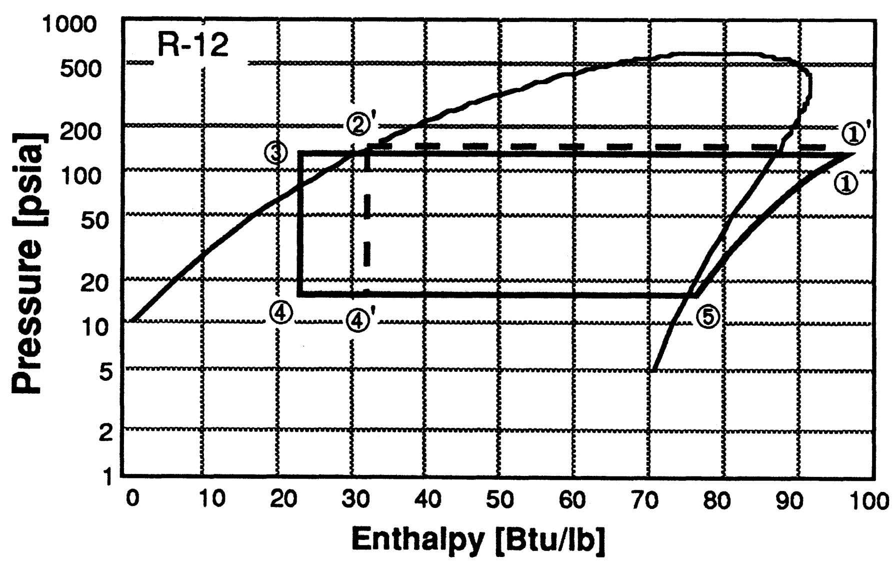

Figure 6.2 Pressure - Enthalpy diagram for an ambient subcooling system. The solid lines refer to the ambient subcooling system, the dashed lines refer to the floating head pressure system.

Because the head pressure of the ambient subcooling system is only slightly lower than that of the floating head pressure system, the increased COP is mainly due to the amount of subcooling. Figure 6.3 shows the amount of subcooling as a function of the ambient temperature. The degrees of subcooling as a function of the ambient temperature is parabolic in nature, but the amount of subcooling does not change much. The reason for the small change in the amount of subcooling is that both the condensing temperature and the subcooled temperature are affected by the ambient conditions as described by the LMTD heat exchanger relations. Because the ambient subcooling system provides an almost constant amount of subcooling regardless of the ambient temperature, the COP of the ambient subcooling system was about 10% higher than that for the floating head pressure system over the range of ambient temperatures.

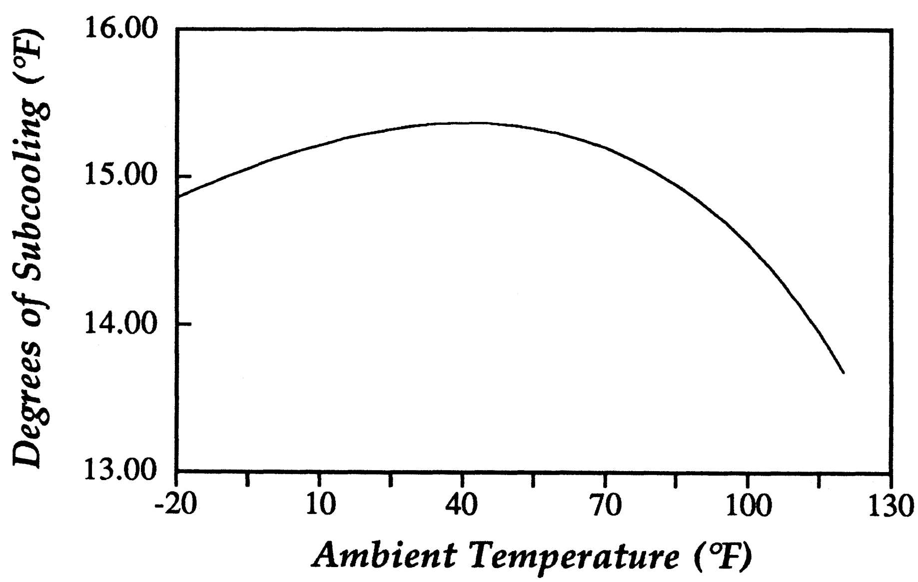

Figure 6.3 Degrees of subcooling as a function of ambient temperature for an ambient subcooling system employing floating head pressure.

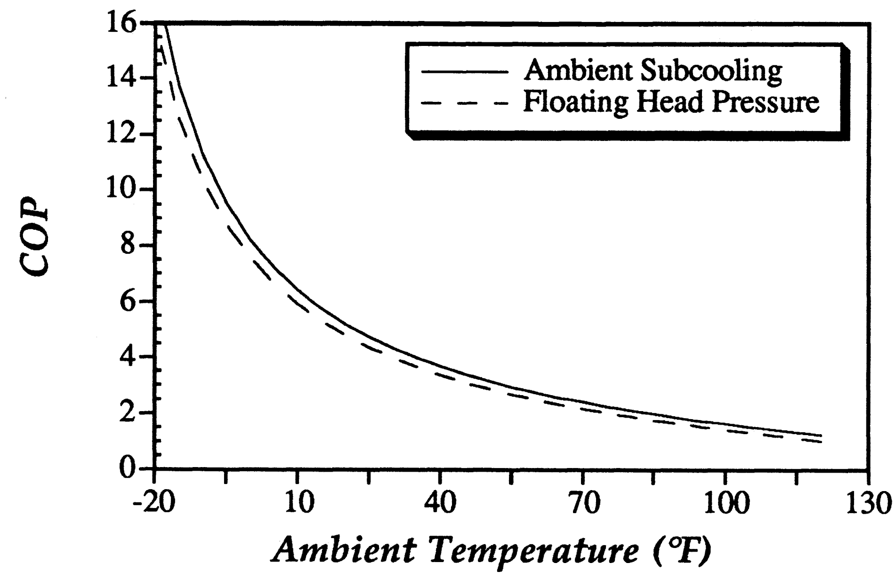

Figure 6.4 COP as a function of ambient temperature for systems employing floating head pressure.

6.2.2 AMBIENT SUBCOOLING WITH FIXED HEAD PRESSURE

Similar to the fixed head pressure systems discussed in Chapter 4, the ambient subcooling system with fixed head pressure "floats" with the ambient conditions as long as the head pressure is above some set value. Below the set point, the condensing pressure and therefore the subcooler pressure, is fixed at the set point. However, because the condensing temperature becomes fixed at low ambient temperatures, the potential for subcooling increases as the ambient temperature falls. The amount of subcooling then increases as the ambient temperature falls below the set point for the fixed head system. This is in direct contrast with the ambient subcooling system employing floating head pressure. The results can be seen in Figure 6.5. It should be noted that the Pressure-Enthalpy diagram for the ambient subcooling system operating under fixed head conditions would look similar to Figure 6.2, except the head pressures of the two systems would be identical.

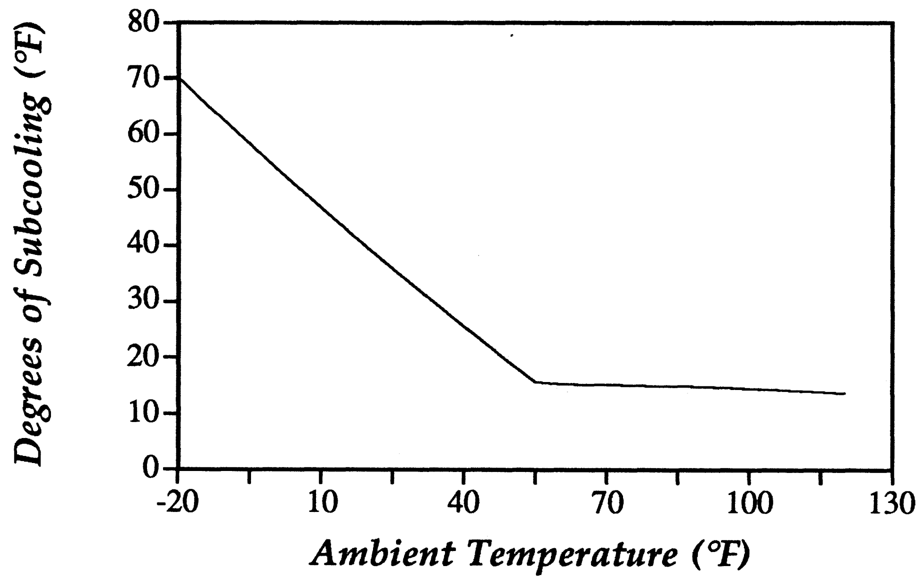

Figure 6.5 The amount of subcooling as a function of the ambient temperature for an ambient subcooling system employing fixed head pressure.

The increase in subcooling lessens the negative impact of the fixed head pressure mechanism. Referring to Figure 6.6, the standard fixed head pressure COP becomes flat as the set point is reached. The COP of the ambient subcooling system with fixed head pressure continues to climb due to the increase in the amount of subcooling. The impact of the fixed head ambient subcooling increases as the ambient temperature decreases. Similar to the fixed head pressure curves discussed in Chapter 4, the ambient subcooling system curves with fixed head pressure has the characteristic change in slope at the set point.

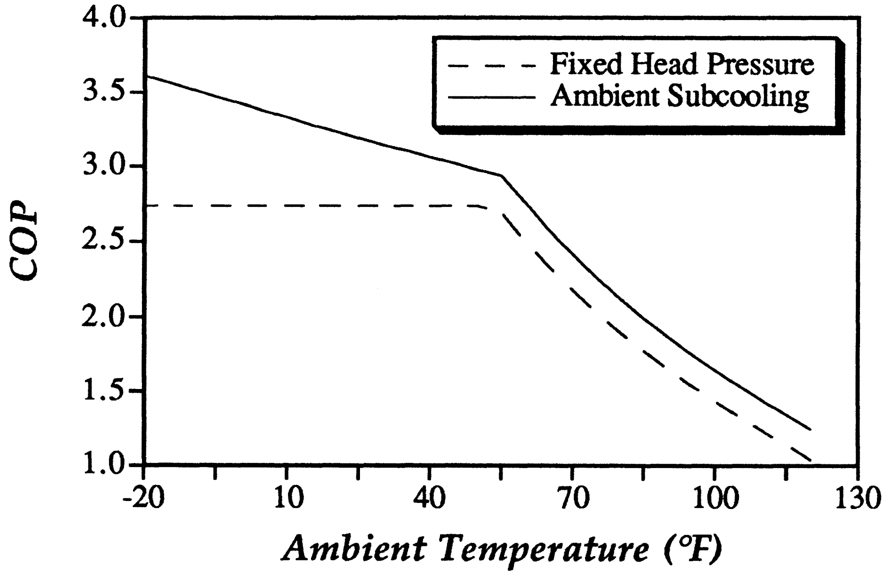

Figure 6.6 COP as a function of ambient temperature for systems employing fixed head pressure.

6.3 PERFORMANCE CURVES

The ambient temperature and evaporator temperature effects on system performance have been shown. Using these trends, the performance equations of the ambient subcooling system can be derived. As discussed in previous chapters, the performance equations were variable in ambient temperature for set values of the evaporator temperature. The fixed head and floating head pressure ambient subcooling performance curves are shown in Figure 6.7. It was these curves that were used to generate the performance equations. The steady-state models, solutions and performance equations are included in Appendix D.

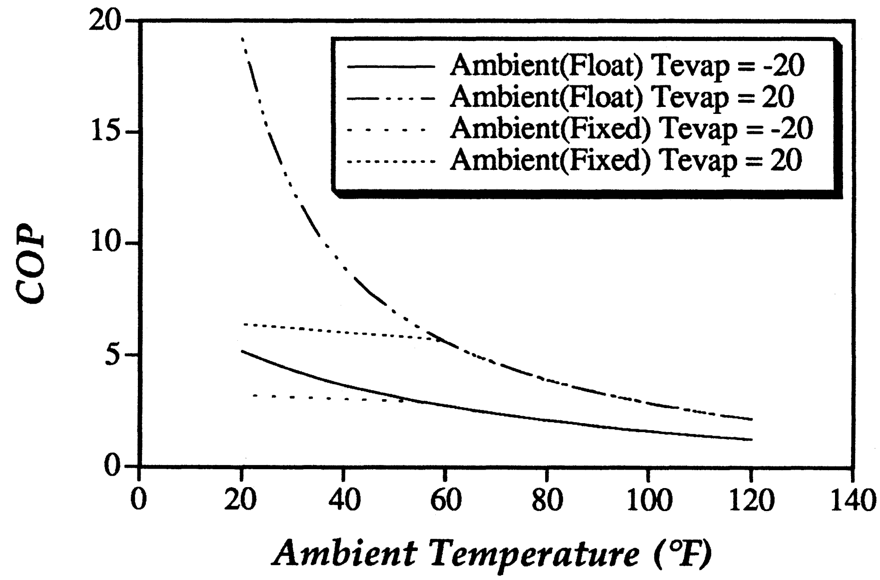

Figure 6.7 Performance curves for the fixed head and floating head pressure ambient subcooling systems

6.4 CHAPTER SUMMARY

Ambient subcooling systems outperform fixed head pressure and floating head pressure systems over the range of ambient temperatures studied. Increased performance is due to the addition of a subcooler located downstream of the condenser. The subcooler rejects heat to the ambient, and provides 15 degrees of subcooling at design conditions. The increased heat rejection to the ambient is the reason for the higher COP. The ambient subcooling steady-state model was developed and discussed. From this model, it was discovered that ambient subcooling with floating head pressure slightly outperforms floating head pressure systems, and that ambient subcooling with fixed head pressure is especially effective at low ambient temperatures as compared to standard fixed head pressure systems. The performance equations for the annual simulations were then developed from curve-fitting the derived performance curves.

REFERENCES 6

1. ASHRAE Handbook, Equipment Volume, 1983, American Society of Heating, Refrigeration, and Air-Conditioning Engineers Inc., Atlanta, Georgia

2. Supermarket Refrigeration Modeling and Field Demonstration, EPRI Project 2569-2, March 1989, Prepared by Foster-Miller Inc.,Waltham, Massachusetts

CHAPTER
SEVEN

DEDICATED MECHANICAL SUBCOOLING

The COP of refrigeration can be increased beyond that which is possible through ambient subcooling by a process called mechanical subcooling. Mechanical subcooling cycles employ a second vapor compression cycle that is coupled to the first cycle through a subcooler located at the exit of the condenser. The subcooler provides approximately 70 degrees of subcooling to the main cycle condensed liquid at design conditions and acts as the evaporator for the second cycle [1]. Because the second cycle provides a lower temperature sink for the subcooler heat transfer than the ambient, the mechanical subcooling system is especially effective at high ambient temperatures and low evaporator temperatures. The second cycle components are a fraction of the size of the main cycle components and act through a much smaller temperature extreme. For these reasons, the COP of the second cycle is quite high. With the small second cycle equipment sizes and high COP, the overall COP of the cycle is increased. This chapter discusses in detail the mechanical subcooling cycle, performance, and the parameters that affect system performance. From the trends that are developed, recommendations and design guidelines for the mechanical subcooling system are developed.

7.1 DEDICATED MECHANICAL SUBCOOLING SYSTEMS

Mechanical subcooling has been used in refrigeration practice for approximately 25 years, and is used primarily in low temperature applications [1]. Mechanical subcooling can be accomplished by using some of the capacity of a medium temperature system or by the use of a separate cycle. Dedicated mechanical subcooling implies that a second vapor compression cycle is used solely for the purpose of providing subcooling to the main cycle. This chapter deals exclusively with dedicated mechanical subcooling. The dedicated mechanical subcooling system diagram is included as Figure 7.1.

The second cycle, or the subcooling cycle, is modelled like the first cycle. The compressor is solved using the concept of isentropic efficiency, the air-cooled condenser is modelled using the LMTD approach, and the refrigerant in the valve is assumed to undergo an isenthalpic expansion. The only difference between the two cycles is the presence of the subcooler. The compressor isentropic efficiencies are assumed to be identical, and the condenser cooling air flow rates have the same ratio. The cooling air flow rates were set to 3800 lbm air / hr / ton of refrigeration [2]. The tons of refrigeration referred to is the evaporator load in tons. For the subcooling condenser, the subcooler heat transfer in tons was used for the determination of the cooling air flow rate. The subcooler heat transfer is used for determination of the subcooling condenser cooling air flow rates because the subcooler represents the evaporator for the subcooling cycle. For the steady-state model, the subcooler was assumed to be a concentric-tube, counter flow heat exchanger. The equation for the LMTD of this heat exchanger is the same as that for the condenser. Since the subcooling cycle refrigerant is evaporating in the subcooler, its specific heat is effectively infinite. Therefore, the minimum capacitance rate for the heat exchanger relations was that of the main cycle subcooled refrigerant.

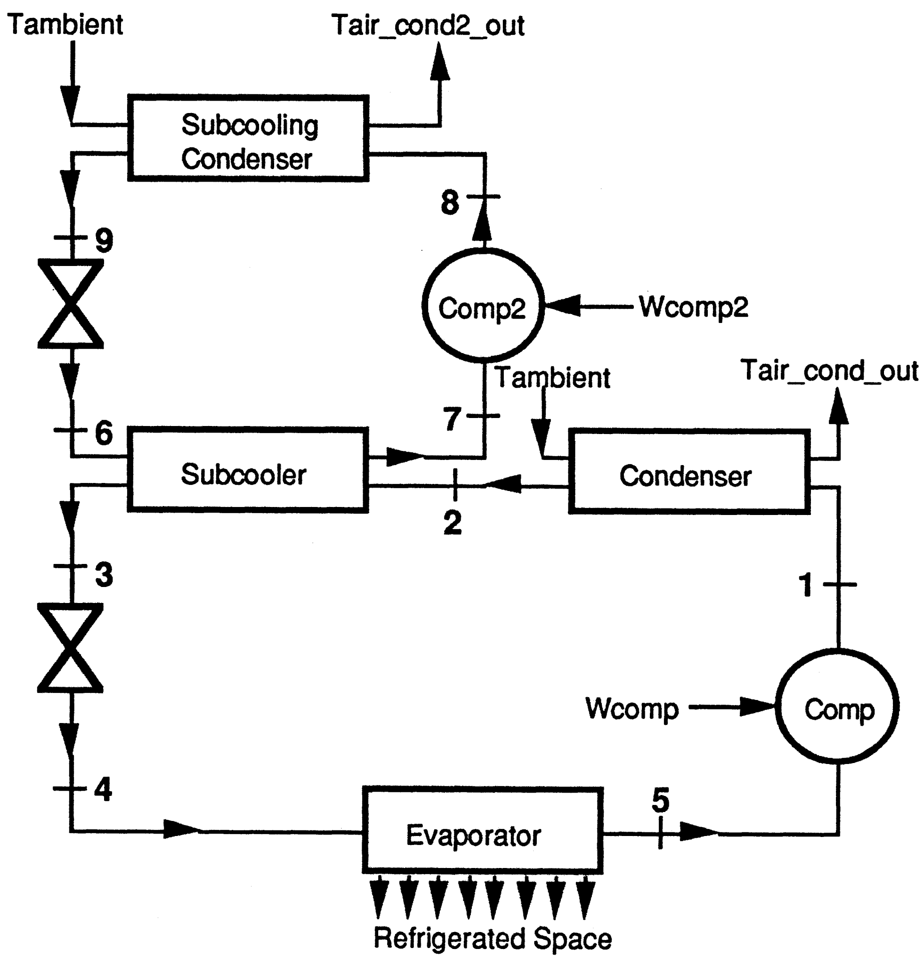

Figure 7.1 System diagram for a dedicated mechanical subcooling refrigeration system.

In Chapter 6, the effect of subcooling on performance was discussed. The COP was found to increase with subcooling due to the lower quality at the evaporator inlet. The lower quality at the evaporator inlet decreases the refrigerant flow rate for fixed caseload simulations. Dedicated subcooling provides for about 70 degrees of subcooling at design conditions. For this simulation, the subcooler was sized to provide 70 degrees of subcooling at an ambient temperature of 80°F and an evaporator temperature of -20°F. This represents an increase in subcooling of approximately 55 degrees over an ambient subcooling system for the same ambient and evaporator temperatures. The relatively large amount of subcooling performed for the dedicated system corresponds to a large increase in the capacity per pound of refrigerant circulated. However, the increase in subcooling is not without cost. To provide the extra subcooling, a small subcooling cycle compressor must be run. Therefore, there is a tradeoff between the extra subcooling provided by the second cycle and the work that must be supplied to the compressor. The COP's of the dedicated cycle can now be defined as:

$$COP_{system} = Capacity \;/\; (W_{comp,main} + W_{comp,sub}) \tag{7.1}$$

$$COP_{main} = Capacity \;/\; W_{comp,main} \tag{7.2}$$

$$COP_{sub} = Q_{subcooler} \;/\; W_{comp,sub} \tag{7.3}$$

The subcooling cycle evaporator temperature must effectively lie between the main cycle evaporating temperature and the main cycle condensing temperature. In practice, the subcooling cycle evaporator temperature is approximately 50 degrees higher than the main cycle evaporating temperature [1]. The amount of subcooling in the subcooler depends on the heat rejection temperature.

For the dedicated system, the subcooling cycle evaporator temperature represents the temperature to which heat is rejected. Since the subcooling cycle evaporator temperature is usually much lower than the ambient temperature, it is possible to subcool to a greater extent with the dedicated subcooling system than with the ambient subcooling system.

The COP of the subcooling cycle (Eq. 7.3) is appreciably higher than that of the main cycle due to the lower temperature extremes. The Pressure-Enthalpy diagram for the dedicated subcooling system is shown in Figure 7.2. The main cycle must operate between the evaporator temperature and the condensing temperature, while the subcooling cycle must operate between the subcooling cycle evaporator temperature (which is approximately 50 degrees higher than the main cycle evaporating temperature) and the subcooling cycle condensing temperature. Referring to Figure 7.2, the difference between the thermal lift of the two cycles can be viewed. The condensing temperatures of the two cycles are close because both condensers interact with the same ambient conditions. Combining the high COP with the fact that the subcooling cycle components are about 1/5 the size of the main cycle components, the work supplied to the subcooling compressor becomes only a small part of the total work. Therefore, the COP of the combined system is increased due to the large amount of subcooling done and the relatively small amount of additional work. Refer to Figure 7.3 for a comparison of the component heat flows as a function of the ambient temperature.

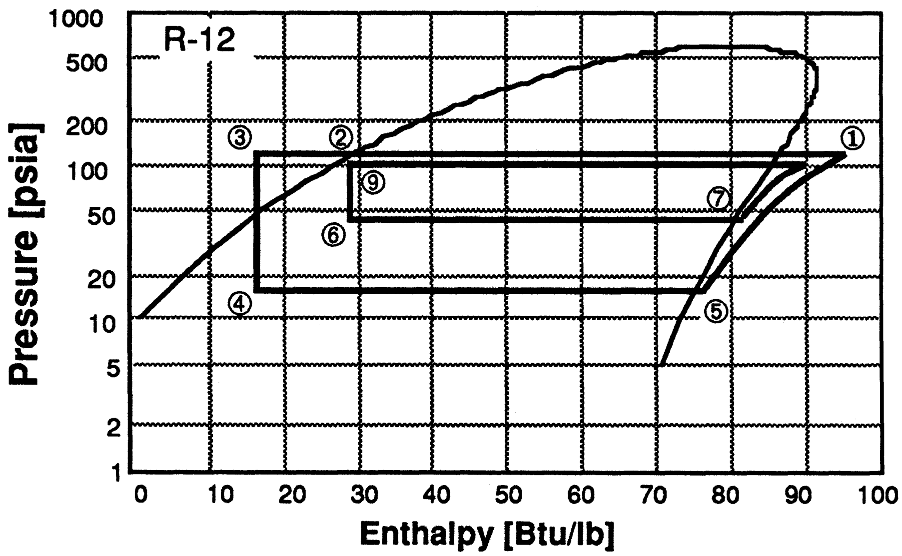

Figure 7.2 System diagram for a dedicated mechanical subcooling system.

It should be noted that the head pressure of the subcooling condenser may be higher, lower, or equal to the main cycle condensing pressure. For the steady-state simulations to follow, the head pressure of the condensers in the dedicated subcooling system are allowed to float continuously with the ambient conditions. In a later section, the dedicated subcooling system with fixed head pressure is explored.

Because the head pressure is allowed to float with the ambient conditions, the main cycle condensing temperature decreases with the ambient temperature. However, the subcooler evaporating temperature is independent of ambient temperature. Therefore, the amount of subcooling decreases with the ambient temperature. This is opposite of what happens with ambient subcooling. Figure 7.4 shows the dependence of the amount of subcooling on the ambient temperature. Because of the large amount of subcooling provided, the dedicated subcooling system outperforms all other systems studied at high ambient temperatures. At low ambient temperatures, there is little subcooling done and therefore, little work done by the subcooling compressor. The COP of the main cycle is then approximately the COP of the system. (Eq's 7.1 to 7.3). Figure 7.5 illustrates the ambient temperature effect on the COP. The COP of the main cycle is higher than the system COP because the main cycle COP includes the effect of the subcooling without the cost of the work associated with the subcooling.

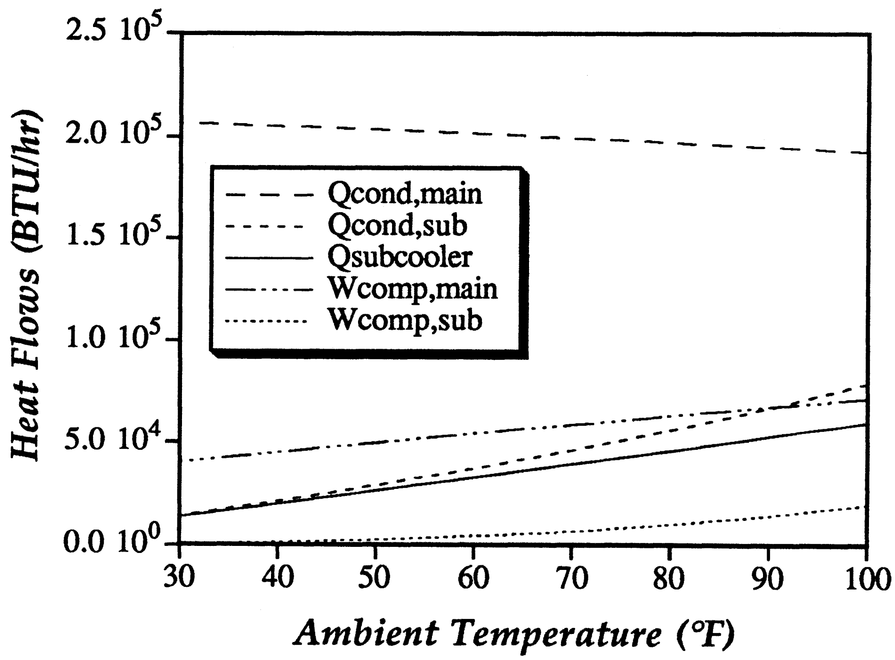

Figure 7.3 Component heat flows for a dedicated mechanical subcooling system as a function of the ambient temperature.

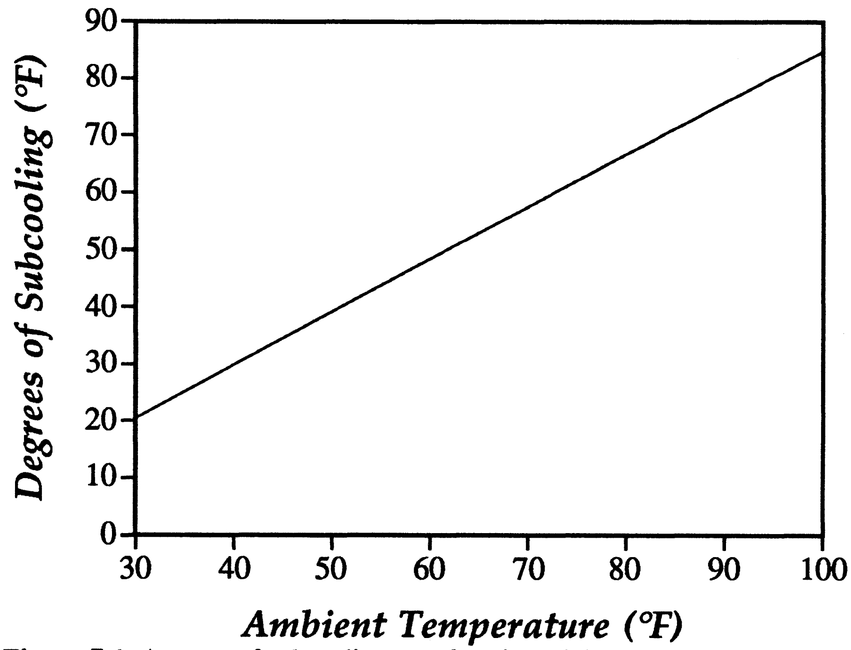

Figure 7.4 Amount of subcooling as a function of the ambient temperature.

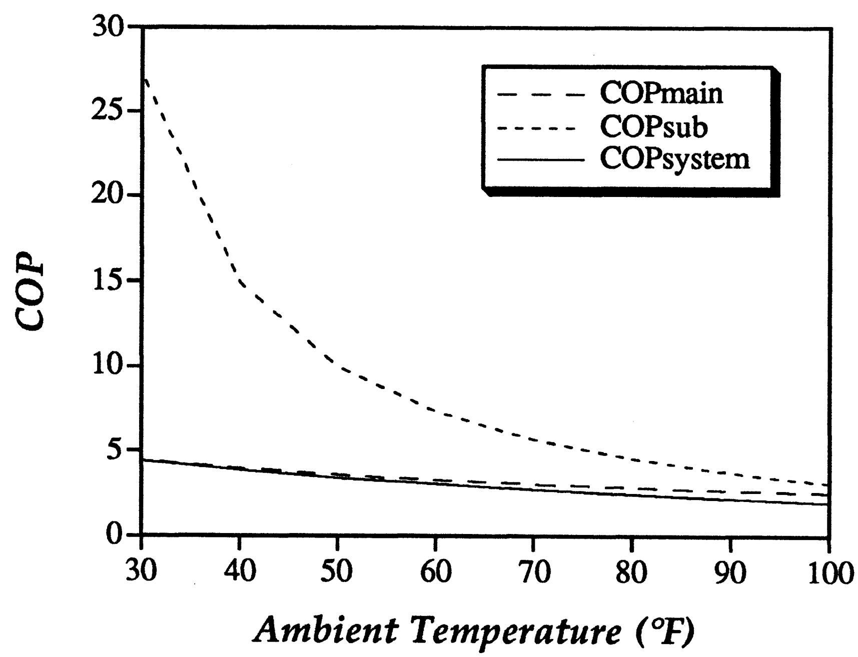

Figure 7.5 COP as a function of the ambient temperature.

7.2 PARAMETERS THAT AFFECT SYSTEM PERFORMANCE

There were eight variables, listed below, that significantly affected the dedicated subcooling system performance. The values in parentheses represent the default values to be used in the study of the other seven parameters. In order to solve the steady-state dedicated subcooling model, the values of these eight parameters must be specified. This section and the subsequent subsections give insight into the values that were chosen for these parameters. The values for these parameters were used in the dedicated subcooling model, and all other steady-state models that were developed where applicable.

- *Main cycle evaporator temperature (-20 °F)*
- *Ambient temperature (80 °F)*
- *Subcooling cycle evaporator temperature (30 °F)*
- *Compressor isentropic efficiency (0.8)*
- *Condenser cooling air flow rate (3800 lbm / hr / ton)*
- *Main cycle condenser size (UA = 15000 BTU / hr °F)*
- *Subcooling cycle condenser size (UA = 5000 BTU / hr °F)*
- *Subcooler size (UA = 2000 BTU / hr °F)*

These eight variables were broken up into three groups; parameters that were assumed constant and were set for ease of calculation, parameters studied and optimized, and parameters varied for performance calculations. The parameters that were set for ease of calculation were the compressor isentropic efficiency, the condenser cooling air flow rates, and the subcooling cycle evaporator temperature. The parameters that were optimized were the subcooler size, the main cycle condenser size, and the subcooling cycle condenser size. The remaining two variables, the ambient and evaporator temperatures, were used for the performance calculations once the values of the other six parameters were set.

7.2.1 CONDENSER COOLING AIR FLOW RATES

The eight unknown parameters must be resolved in order to solve the dedicated subcooling system. By setting the condenser cooling air flow rates to a value that is commonly used in practice, one of the set parameters is found. The value that was chosen for the cooling air flow rates for this system and all other systems studied was 3800 lbm air / hr / ton of refrigeration. This value falls within the range of values recommended by the ASHRAE 1983 Equipment volume for an air-cooled condenser. For the main cycle condenser, the tons of refrigeration was based on the evaporator capacity. For the subcooling cycle condenser, the tons of refrigeration was referenced to the subcooler heat transfer since the subcooler acts as the evaporator for the second cycle.

From viewing Figures 7.6 and 7.7, the choice of the cooling air flow rate meets two separate design criteria. The degrees of subcooling at the set cooling air flow rate for the default parameters is seen to be approximately 67°F, which is in agreement with existing practice for which mechanical subcooling systems provide around 70°F of subcooling at design conditions [1]. The second design criteria that is met is the concept of "picking the knee of the curve". Referring to Figure 7.7, the set cooling air flow rate corresponds to a value near the bend or "knee" of the curve. Choosing a value below the knee of the

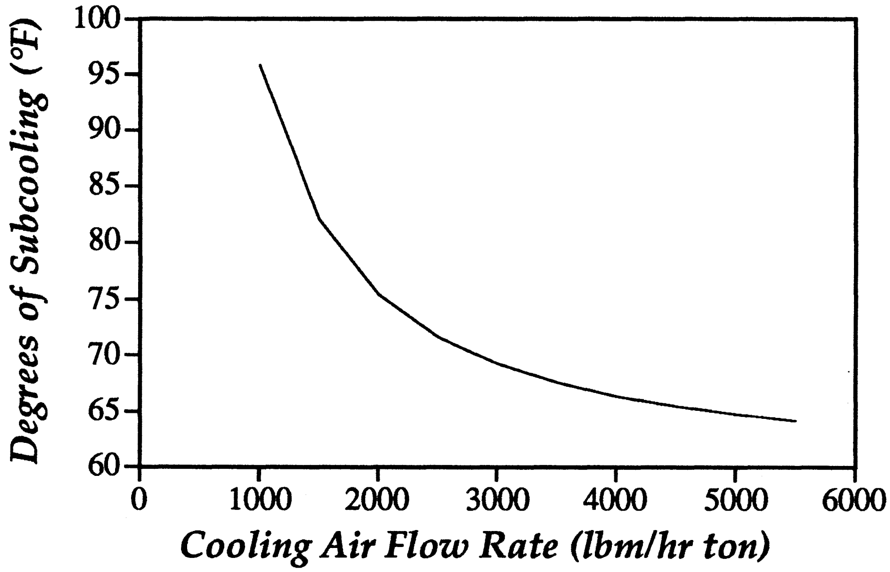

Figure 7.6 Amount of subcooling as a function of the condenser cooling air flow rate.

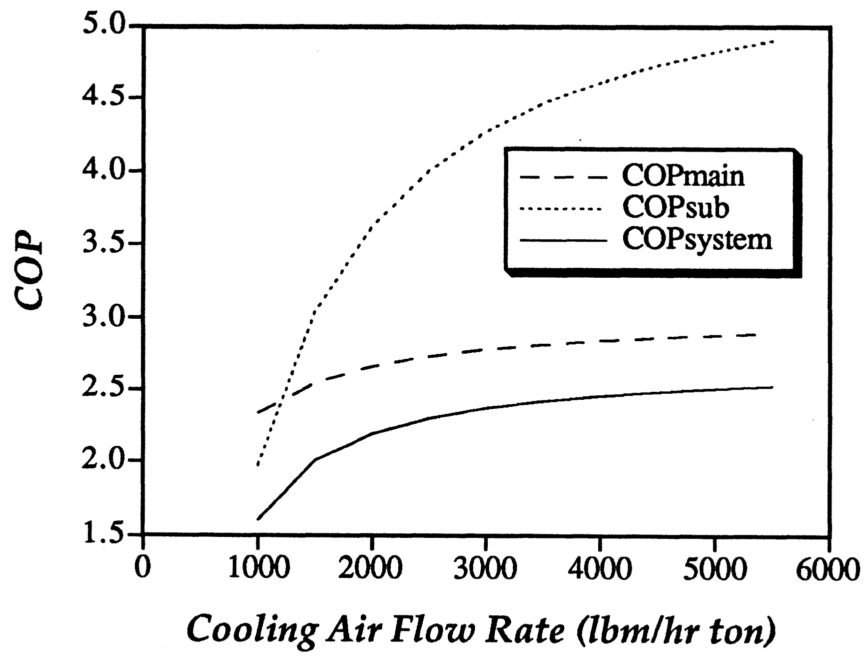

Figure 7.7 COP as a function of the condenser cooling air flow rate.

curve corresponds to not utilizing the possible improvements to the COP. Selecting a value past the knee of the curve is wasteful since not much increase in COP is gained at the cost of increased fan power. Although the knee of the curve may not be the economically optimum point, it is still a reasonable design criteria.

7.2.2 COMPRESSOR ISENTROPIC EFFICIENCY

The dedicated mechanical subcooling system utilizes two hermetically sealed reciprocating compressors. As discussed in Section 3.2.3, the compressors were modelled using the concept of isentropic efficiency. For the steady-state models developed, an isentropic efficiency of 0.8 was used. The performance results for each system were generated using an isentropic efficiency of 0.8. However, a sensitivity analysis on the COP was performed using an isentropic efficiency of 0.6. The magnitude of the results was changed, but the relative performance of each system was unchanged. For example, the ambient subcooling system still outperformed the floating head pressure system over the entire range of ambient temperatures. The change in isentropic efficiency just scales the performance curves up or down.

Figures 7.8 and 7.9 show the compressor work and COP curves as a function of the isentropic efficiency. Although it may seem unusual that the COP curves are linear while the work curves are not, there is a reasonable explanation. The COP is defined as the caseload divided by the total work. Since the caseload is fixed, the COP is inversely proportional to the work. If the quantity 1/work was plotted as a function of the isentropic efficiency, the result turns out to be a straight line.

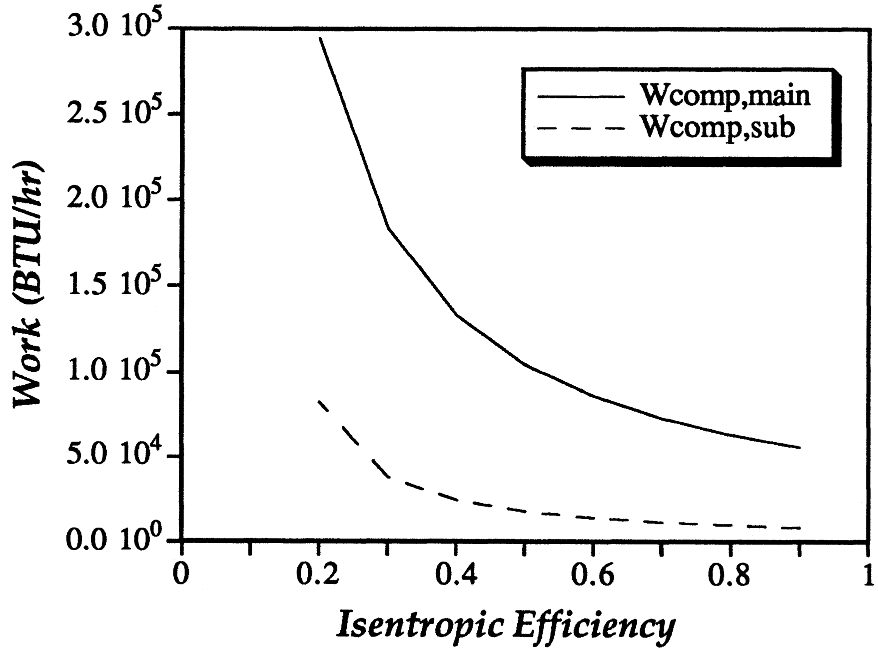

Figure 7.8 Compressor work as a function of the isentropic efficiency.

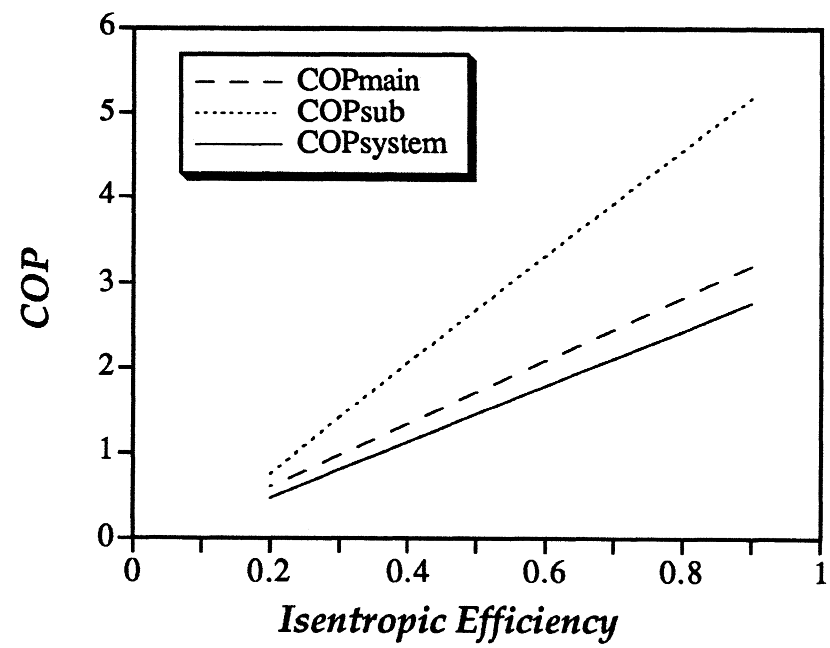

Figure 7.9 COP as a function of the isentropic efficiency.

Figure 7.9 shows the linear relationship between the isentropic efficiency and the system COP for a dedicated subcooling system. If the ambient subcooling system COP was added to Figure 7.9, the curve would lie below the dedicated subcooling curve with approximately, but not exactly, the same slope. Because the slopes of the COP curves as a function of the isentropic efficiency for each system are similar, the relative effect between the systems is unchanged when the isentropic efficiency is varied. With the choice of 0.8 for the compressor isentropic efficiency, another of the unknown parameters is set.

7.2.3 SUBCOOLING CYCLE EVAPORATOR TEMPERATURE

The temperature at which the subcooling cycle evaporates represented an interesting thermodynamic compromise. The subcooling cycle evaporator temperature was set at a fixed value in order to solve the steady-state model. However in order to set the subcooling cycle evaporator temperature at some reasonable value, insight into the problem had to be gained. The choice of a proper subcooling cycle evaporator temperature is important since it affects the performance of both cycles in the dedicated subcooling system. The subcooling cycle evaporator temperature is effectively bounded by the main cycle evaporator temperature and the main cycle condenser temperature, and is represented by point 6 on Figure 7.1.

If the subcooling evaporator temperature is located at the upper extreme, the main cycle condensing temperature, the combined system effectively becomes one system. The subcooling cycle provides no subcooling to the main cycle because there is no temperature difference in the subcooler between the refrigerant flows. With no heat transfer in the subcooler, there is no work supplied to the subcooling cycle compressor. In fact, if the subcooling cycle evaporator temperature was higher than the main cycle condensing temperature, the subcooler would provide heating to the main cycle; opposite of what is desired. Therefore, if the subcooling cycle evaporator temperature is at the upper extreme, the system acts like one system; a floating head pressure system with no subcooling, operating between the same temperature extremes as the dedicated subcooling system. The COP of this system is identical to the COP of the standard floating head pressure system operating between the same extremes.

If the subcooler evaporator temperature is located at the lower extreme, the main cycle evaporator temperature, the dedicated subcooling system effectively becomes two systems. Both cycles now operate between the ambient temperature and the evaporator temperature. The reason that this operation degrades the system performance is due to the effectiveness of the subcooling. As discussed earlier, the increase in subcooling equals the increase in capacity. Because the two systems are operating over the same thermal lift, both systems could provide the extra capacity at approximately the same COP. The main cycle could provide the extra capacity by increasing the flow rate, while the subcooling cycle could provide the extra capacity as subcooling supplied to the main cycle. In this way, the dedicated cycle is acting like two floating head pressure systems operating between the evaporator temperature and the ambient conditions. In fact, the COP of the dedicated system with the subcooling evaporator temperature at the main cycle evaporator temperature is within 5% of the COP for the floating head pressure system operating over the same thermal lift. The 5% difference is due to the small condensing temperature differences between the main cycle and the subcooling cycle.

The COP of the dedicated subcooling system with the evaporator temperature at either extreme is approximately the same. Therefore if there is a optimum point for the subcooling cycle evaporator temperature, it must lie between the two extremes.

Since the optimum point must lie between the two extremes, the range of interest for the evaporator temperature is fixed. There are two phenomena that affect the choice of the subcooling cycle evaporator temperature. If the subcooling evaporator temperature is lowered, the amount of subcooling to the main cycle is increased due to the larger temperature differences in the subcooler. The increase in the amount of subcooling supplied to the main cycle enhances the COP of the main cycle. This trend can be seen in Figures 7.10 and 7.11. However with the lowered evaporator temperature, the subcooling cycle must now operate with a greater thermal lift. As discussed in Chapter 4, raising the thermal lift of the subcooling cycle causes the COP to deteriorate. The COP as a function of the subcooling cycle evaporator temperature is shown in Figure 7.11.

The optimal point at which to set the subcooling cycle evaporator temperature must balance the two effects previously mentioned; the increased subcooling to the main cycle and the greater subcooling cycle thermal lift. A plot of the system COP as a function of the subcooling cycle evaporating temperature is shown in Figure 7.12.

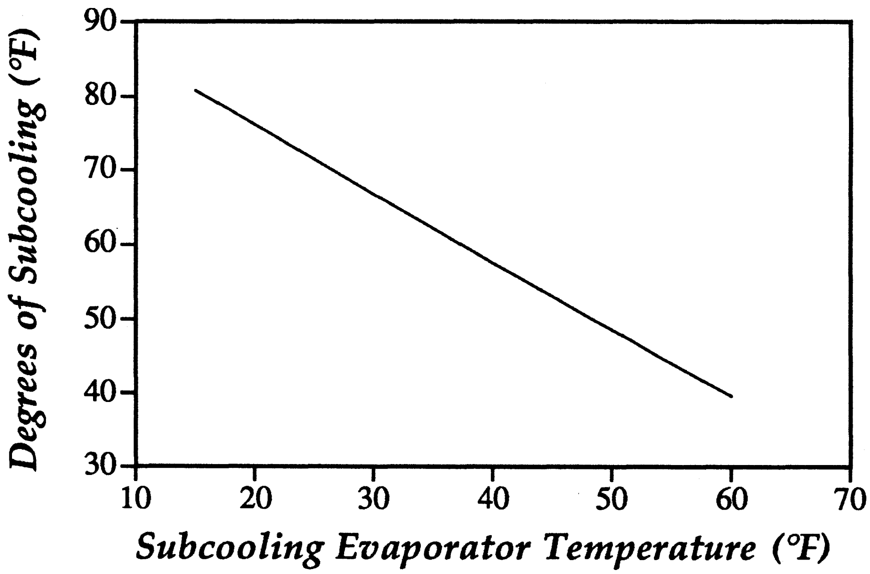

Figure 7.10 Amount of subcooling as a function of the subcooling evaporator temperature.

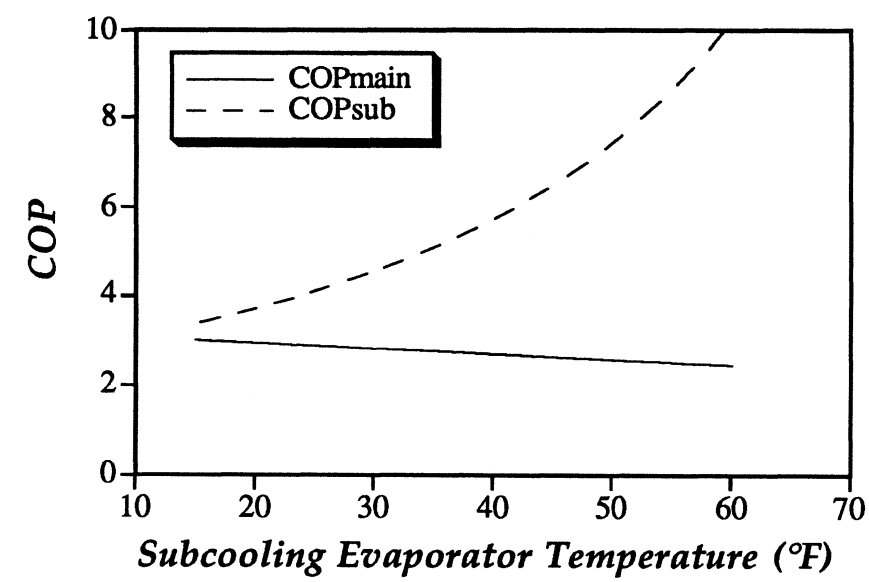

Figure 7.11 COP for the main and subcooling cycles as a function of the subcooling evaporator temperature.

The maximum COP is seen to occur at a temperature of approximately 30 to 35 degrees for the default values of the other seven parameters. With the choice of 30 degrees for the subcooling evaporator temperature, another of the set parameters was fixed. Although the value of 30 degrees represents a maximum COP for the default parameters listed, it was possible that this was just a condition of the default values chosen. To make sure that the 30 degree set parameter was valid over the range of conditions studied in this report, a sensitivity analysis was performed.

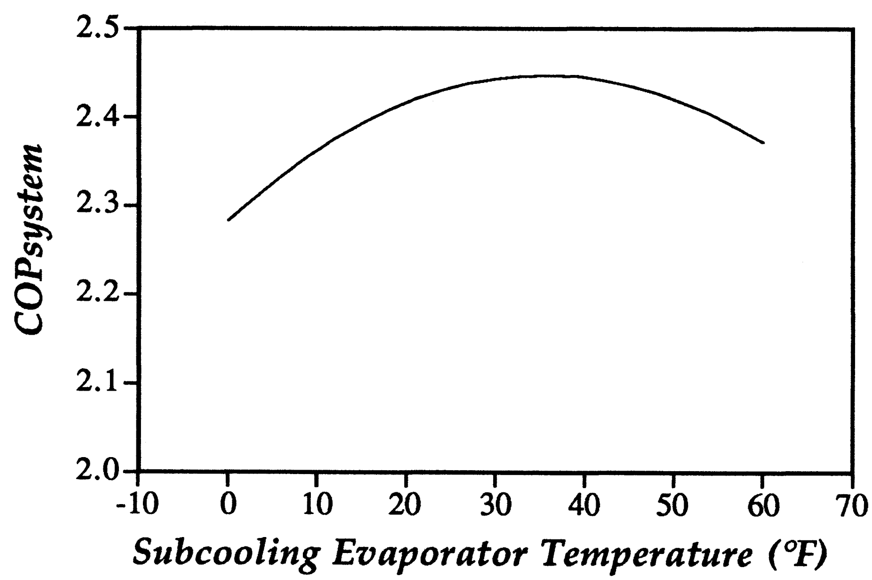

Figure 7.12 COP of the dedicated subcooling system as a function of the subcooling evaporator temperature.

To justify the choice of 30 degrees for the subcooling evaporator temperature, the sensitivity analysis explored the effect of each of the seven parameters on the 30 degree set point.

Ambient Temperature

The ambient temperature effect on the maximum COP point is shown in Figure 7.13. The increasing ambient temperature sends the relative COP down as expected, and shifts the maximum COP point slightly towards higher subcooling evaporator temperatures. However, the curves are very flat and the choice of 30 degrees for any of the curves is an acceptable estimate. The trend towards higher subcooling evaporator temperatures as the ambient temperature rises implies that the system is trying to reduce the subcooling cycle thermal lift rather than increasing the amount of subcooling supplied to the main cycle. One reason for this trend may be the added potential for subcooling due to the increase in the ambient temperature. As the ambient temperature rises, the condensing temperature rises which increases the subcooling potential.

Main Cycle Evaporator Temperature

The effect on the maximum COP point due to varying the main cycle evaporator temperature is shown in Figure 7.14. As expected, increasing the main cycle evaporator temperature sends the relative COP curves up due to the decrease in thermal lift. The maximum COP point is also seen to shift slightly towards higher subcooling cycle evaporator temperatures. The COP curves as a function of the main cycle evaporator temperature are, however, extremely flat and again 30 degrees represents an acceptable estimate to the maximum operating point.

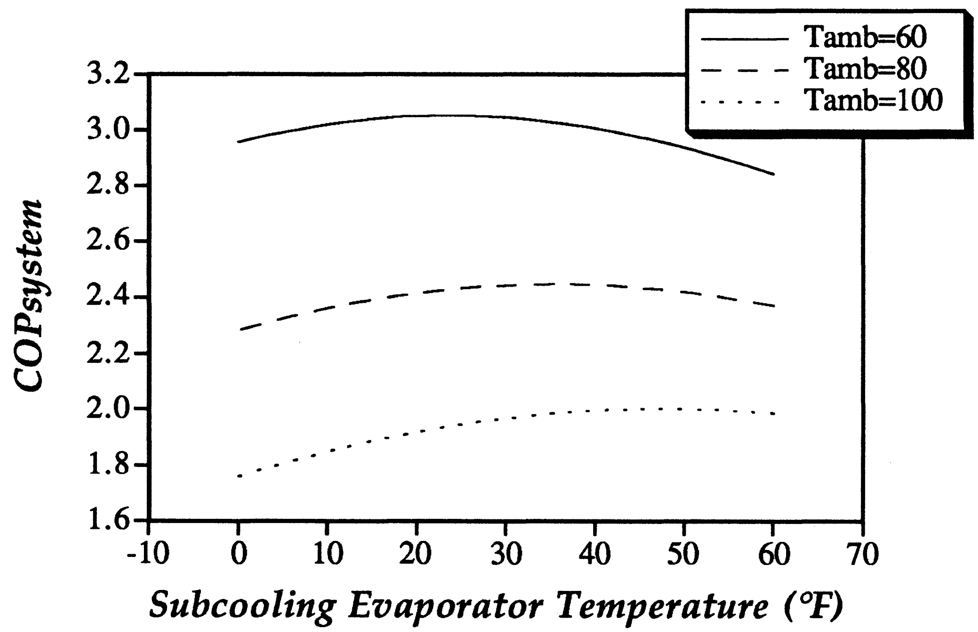

Figure 7.13 COP as a function of the subcooling evaporator temperature for set values of the ambient temperature.

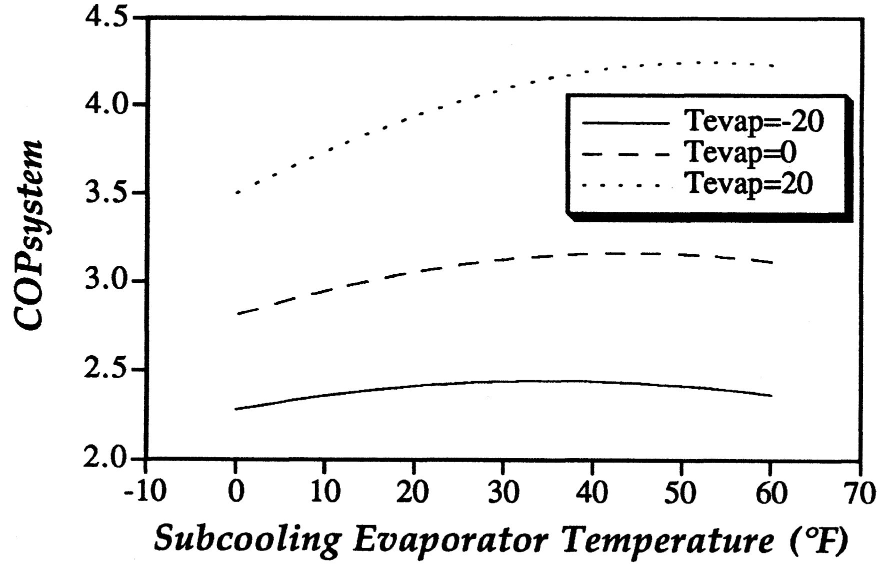

Figure 7.14 COP as a function of the subcooling evaporator temperature for set values of the main cycle evaporator temperature.

Heat Exchangers

The sensitivity of the optimal subcooling evaporator temperature to the heat exchanger sizes was explored. Changing the UA size of a heat exchanger not only affects the heat transfer in that component, but ultimately affects the performance of the entire system.

The default values that were chosen for the three heat exchangers sizes are listed below. They are referred to as the standard UA values for this section.

UA Main Cycle Condenser	= 15000 BTU/hr °F
UA Subcooling Cycle Condenser	= 5000 BTU/hr °F
UA Subcooler	= 2000 BTU/hr °F

Effect of Total UA

The effect of changing the sizes of all three heat exchangers by the same amount was investigated. In this case, the UA product of all three heat exchangers was multiplied by a constant. By multiplying the UA products by a constant, the ratio of each heat exchanger size to the total remained constant regardless of the multiplier. In this way, the effect of the total heat exchanger size on the maximum COP point was studied. The heat exchanger UA products were increased and decreased by 33% for this study. The results are shown in Figure 7.15. The COP curves are seen to increase with the total UA size as expected. With larger UA's the heat exchangers are allowed to transfer more heat to the ambient thereby increasing the COP. The most important point for this section though, is that the sizes of the heat exchangers do not affect the maximum COP point if the ratio of the heat exchanger sizes to the total remains constant.

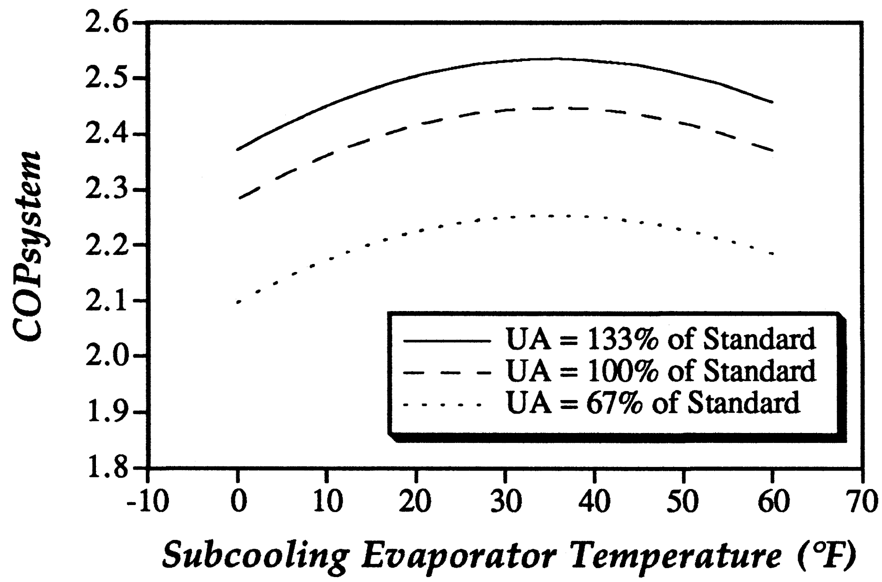

Figure 7.15 COP as a function of the subcooling evaporator temperature for set values of the total heat exchanger UA product.

Effect of Relative Condenser UA

The next problem that needed study was whether the relative sizes of the heat exchangers affect the maximum COP point. To test this concept, the UA products for the main cycle and subcooling cycle condensers were switched. Therefore, the subcooling condenser UA was now three times larger than the main cycle condenser UA. The subcooler UA product was left unchanged initially. The results for this test are shown in Figure 7.16. The inverted UA labelled on the graphs represents the switch from the standard condenser UA's. With the main cycle condenser now being only one-third the size of the subcooling cycle condenser, the COP curves were shifted down by approximately 20%. However, the subcooling evaporator temperature at the maximum COP point was left virtually unchanged. This implies that the maximum COP point is not a strong function of the relative sizes of the condensers either.

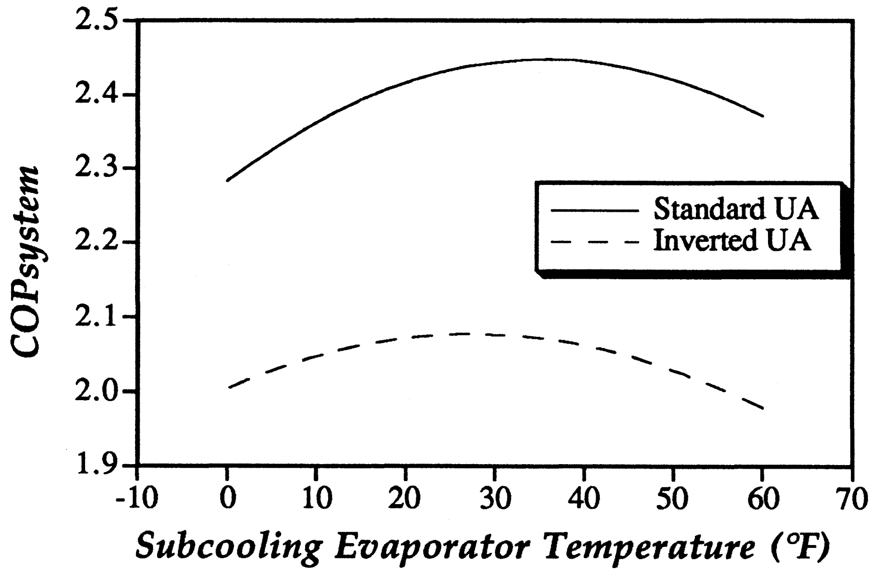

Figure 7.16 COP as a function of the subcooling evaporator temperature for set values of the condenser UA products.

Effect of Subcooler UA

The only heat exchanger effect left to test is the subcooler size effect. The condenser UA products were returned to their normal values and the subcooler UA size was varied while leaving the condenser thermal sizes alone. Figure 7.17 shows the impact of these tests. The COP curves increase with the UA size of the subcooler. As the subcooler size increases, the amount of subcooling supplied to the main cycle increases, which in turn raises the system COP. The subcooling evaporator temperature at the maximum COP point also drifts up as the subcooler size is increased. This implies that the subcooling evaporator temperature is increasing to alleviate some of the thermal lift in the subcooling cycle. One reason for this trend might be the additional subcooling provided by the increased subcooler UA product. Regardless of the reason, the important point is that the maximum COP point drift is slight. Therefore, the 30 degree set point for the subcooling cycle evaporator temperature is again a viable optimum point for the subcooler sizes studied.

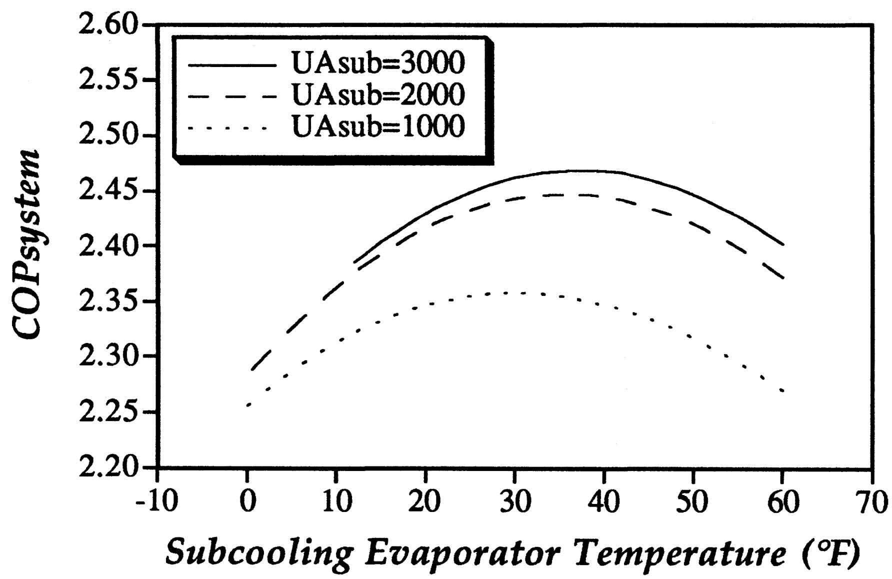

Figure 7.17 COP as a function of the subcooling evaporator temperature for set values of the subcooler UA product.

SUBCOOLING EVAPORATOR CONCLUSIONS

This section dealt with the effect of the other seven parameters on the subcooling cycle evaporator temperature set point. Although some of the studies caused some slight drift of the maximum COP point, it was discovered that a subcooling evaporator temperature of 30 °F provided a near optimum condition for the performance regardless of the other parameters. By setting the subcooling evaporator temperature to 30 °F, one of the remaining unknowns is fixed without sacrificing performance.

7.2.4 HEAT EXCHANGER UA PRODUCT

With the subcooling evaporator temperature found, the parameters that were assumed constant for this system have all been found. These *set* parameters made the choice of the heat exchanger sizes much easier since the heat exchanger sizes now depend on fewer variables. The methods of choosing the UA for the condenser, the subcooler, and the subcooling condenser are now discussed. The results that were found were used to fix the sizes of the three heat exchangers, thereby eliminating three more unknown parameters.

Increasing the UA product for a heat exchanger increases the potential heat transfer for that component. The method of choosing the UA product of the main cycle condenser is the same method that was used to choose the UA product for the subcooler and the subcooling cycle condenser.

To solve for the UA product of one of the heat exchangers, the effectiveness of the other two heat exchangers was set to 1.0. As discussed in section 3.2.1, there are two ways to increase the effectiveness of a heat exchanger subject to one fluid undergoing a phase change; increasing the UA product or lowering the minimum fluid capacitance rate. Since the minimum fluid capacitance rate is affected by the system operating conditions, the effectiveness was set to 1.0 by making the UA product of the heat exchangers very large. By setting the effectiveness of the heat exchangers to 1.0, the relative sizes of the two heat exchangers becomes unimportant. The two heat exchangers are now considered "perfect" heat exchangers. Therefore, any change in the size of the heat exchanger in question directly affects the performance of the entire system. To choose the UA size for the heat exchanger in question, the system performance was plotted against the UA product of the heat exchanger. The concept of "picking the knee of the curve" was again used as a design method. This method was first discussed in section 7.2.1. Choosing a UA product that is too small cuts down on the potential performance of the entire system, while choosing a UA product that is too large is considered wasteful. The cost of a heat exchanger is proportional to the UA product of the heat exchanger.

MAIN CYCLE CONDENSER

With the effectiveness of the subcooler and subcooling condenser set to 1.0, the UA product of the main cycle condenser was determined using the method described above. Increasing the UA product of the condenser has two significant effects; the amount of subcooling decreases, and the head pressure of the system decreases.

With the effectiveness of the subcooler being 1.0, the maximum possible heat transfer occurs when the temperature of the subcooled refrigerant leaving the subcooler is at the temperature of the refrigerant evaporating in the subcooler. This temperature was investigated in section 7.2.3 and was set to 30°F. Because the effectiveness of the subcooler is 1.0, the temperature of the subcooled refrigerant exiting the subcooler is 30°F; regardless of the main cycle condensing temperature of the fluid. The main cycle condensing temperature decreases as the UA product of the condenser increases due to the greater heat transfer to the ambient. The amount of subcooling, which is defined for this system as the temperature difference between the main cycle condensing temperature and the temperature of the subcooled refrigerant exiting the subcooler, then decreases as the UA product for the main cycle condenser increases. This trend is shown in Figure 7.18.

Another important fact about increasing the UA product of the main cycle condenser is that the head pressure of the main cycle is decreased. The decrease in head pressure results in a decrease in the amount of work supplied to the main cycle compressor.

As discussed, increasing the UA size for the condenser decreases the amount of subcooling. In previous sections, decreasing the amount of subcooling supplied to the main cycle was seen to hurt the system performance. However for this discussion, the decrease in subcooling actually helps the system performance. The exit state of the subcooled refrigerant is fixed at 30°F by the "perfect" condition of the subcooler. The fixed exit state of the subcooled refrigerant implies that the state of the refrigerant entering the main cycle evaporator is fixed.

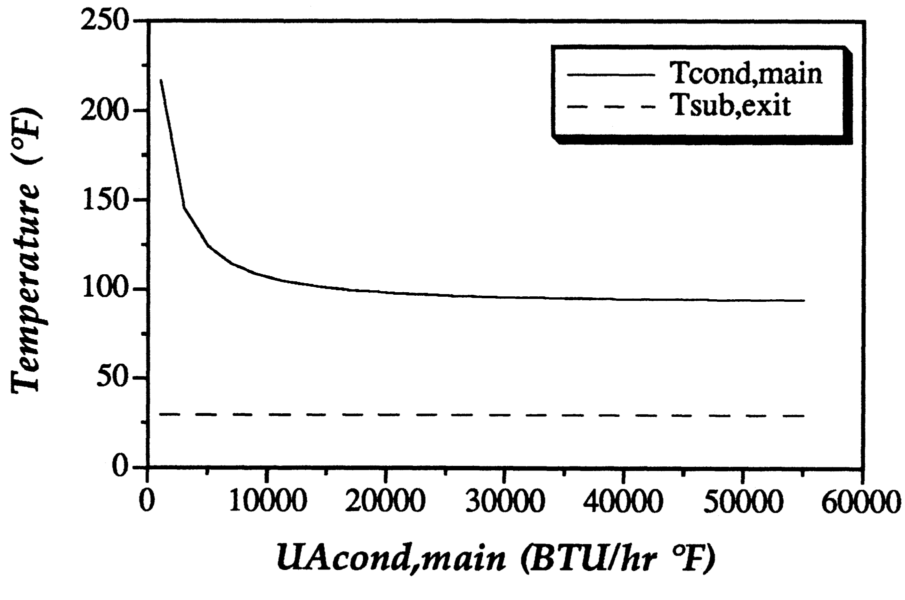

Figure 7.18 Temperature profile for the main cycle condenser and the subcooler exit as a function of the main cycle condenser UA product.

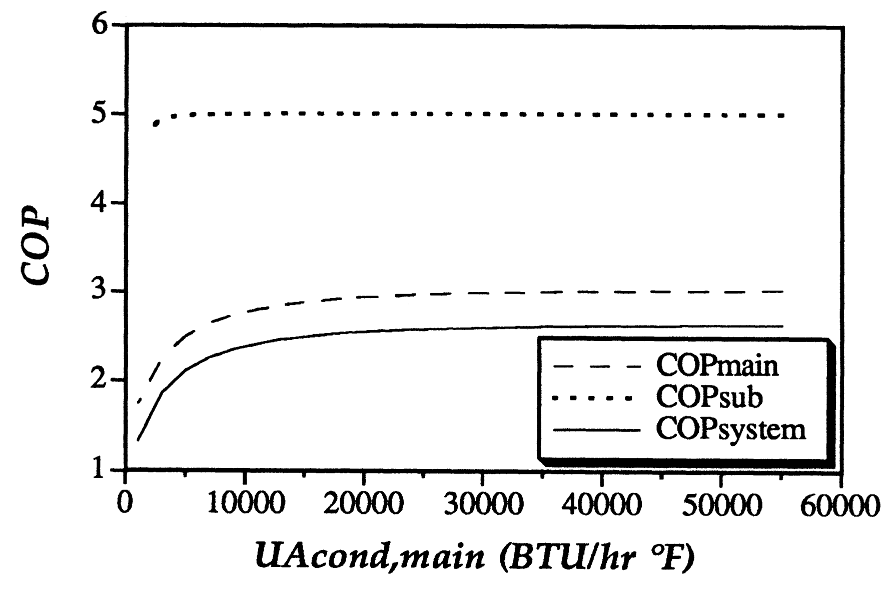

Figure 7.19 System performance as a function of the main cycle condenser UA product.

For the dedicated subcooling system, it was the change of the main cycle evaporator inlet towards saturated liquid that increased the main cycle COP. Since this inlet evaporator condition is fixed, changing the amount of subcooling only affects the load on the subcooling cycle. The greater the load on the subcooling cycle, the greater the work of the subcooling compressor. By increasing the UA product of the main cycle condenser, a greater proportion of the head side heat transfer occurs to the ambient and not to the subcooling cycle. This concept may be summed up as follows and is typical of dedicated subcooling systems: reject as much heat as possible to the ambient as conditions allow, then use the subcooling cycle, with its lower temperature sink, to reject more heat in the form of subcooling. Remember, rejecting heat to the ambient is "free" (with the exception of fan power) while rejecting heat to the subcooling cycle is at cost due to the power supplied to the subcooling cycle compressor. However, the heat rejection to the subcooling cycle is better than no heat rejection at all for most ranges of operating conditions.

The COP of the system increases as the main cycle condenser UA product increases due to the lowered head pressure and the shift in heat transfer from the subcooling cycle to the ambient. Figure 7.19 shows the relationship between the main cycle condenser UA and the system performance.

A value of 15000 BTU/hr °F was chosen for the size of the main cycle condenser. The UA choice represents a value near the top of the "knee" for Figure 7.19. The UA value of 15000 BTU/hr °F was used as the main cycle condenser size for this system and all other systems studied.

An interesting note is that at very large condenser sizes (i.e. all three heat exchangers are "perfect") the system COP is approximately 2.7 for an ambient temperature of 80°F and an evaporator temperature of -20°F. This value represents about one-half the Carnot COP for the same conditions.

SUBCOOLING CYCLE CONDENSER

The subcooling condenser UA product was chosen using the same method as the main cycle condenser. The value that was chosen for the subcooling condenser UA product was 5000 BTU/hr °F. The effect of the subcooling condenser size on system performance is explained below.

The head pressure of the subcooling cycle is lowered as the UA product increases due to the increase in heat transfer to the ambient. The lowered head pressure decreases the compressor work and increases the subcooling cycle capacity per pound of refrigerant circulated. Therefore the subcooling cycle refrigerant flow rate decreases as the subcooling condenser size increases.

Since the effectiveness of the subcooler and main cycle condenser is 1.0, the inlet and exit temperatures of the subcooler are fixed. Therefore, the amount of subcooling is not influenced by the size of the subcooling condenser. With a constant amount of subcooling, the COP of the main cycle and the heat transfer in the subcooler remain constant. With this condition, the COP of the system is only influenced by the amount of work supplied to the subcooling condenser.

The effect of varying the subcooling condenser size is shown in Figure 7.20. An important note is that at the value of 5000 BTU/hr °F chosen for the subcooling condenser, the COP of the subcooling cycle is still rising while the system COP has almost reached its asymptotic value. Since the system is designed based upon system COP, this is the only important factor in choosing a subcooling condenser size.

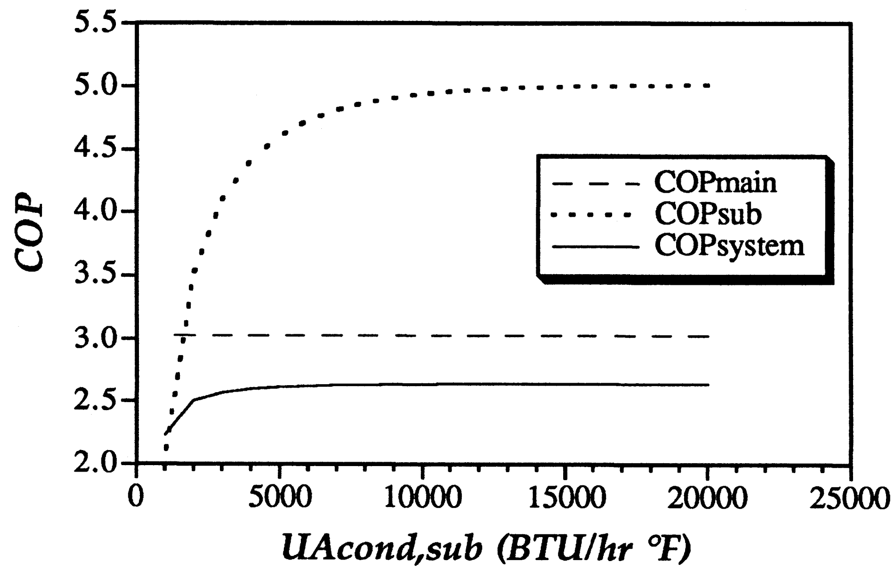

Figure 7.20 System performance as a function of the subcooling cycle condenser UA product.

SUBCOOLER

Setting the effectiveness of the main cycle and subcooling cycle condensers to 1.0, the UA product of the subcooler was found to be 2000 BTU/hr °F.

The subcooler size affects the amount of subcooling supplied to the main cycle. Increasing the amount of subcooling increases the COP of the main cycle due to the increase in capacity per pound of refrigerant circulated. In the main cycle condenser section, it was found that shifting the amount of head side heat rejection towards the ambient increased the COP. However for the subcooler UA case, the main cycle condenser is rejecting as much heat as possible (main cycle condenser effectiveness=1.0). Therefore, no shift is possible. The COP of the subcooling cycle is independent of the subcooler UA since the head conditions and evaporator conditions remain fixed. The subcooling cycle head conditions are set by the effectiveness of the subcooling condenser being 1.0. The subcooling cycle evaporator conditions are set by the choice of 30°F for the subcooling evaporating temperature (section 7.2.3).

The UA product of the subcooler only affects the amount of subcooling performed on the main cycle. Since the COP of the subcooling cycle is higher than the COP of the main cycle due to a less extreme thermal lift (remember - this is the reason for using dedicated subcooling), the greater the amount of subcooling performed, the greater the system COP. The limit on the subcooler exit temperature is 30°F due to the setting of the subcooling evaporator temperature in the previous section. To increase the amount of subcooling a greater subcooler UA product is needed. The amount of subcooling and the COP as a function of the subcooler size are included as Figures 7.21 and 7.22 respectively. At the value of 2000 BTU/hr °F chosen for the subcooler, the effectiveness of the subcooler was near 1.0.

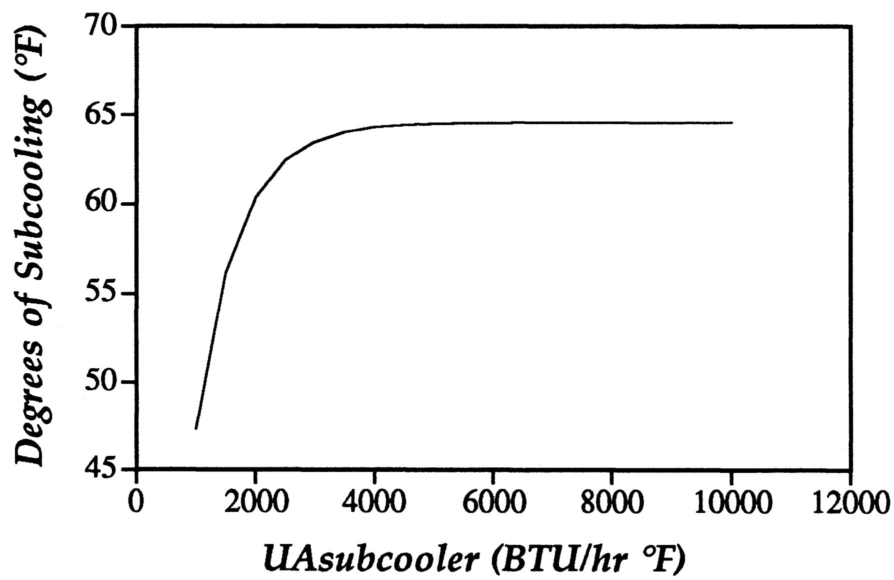

Figure 7.21 Amount of subcooling as a function of the subcooler UA product.

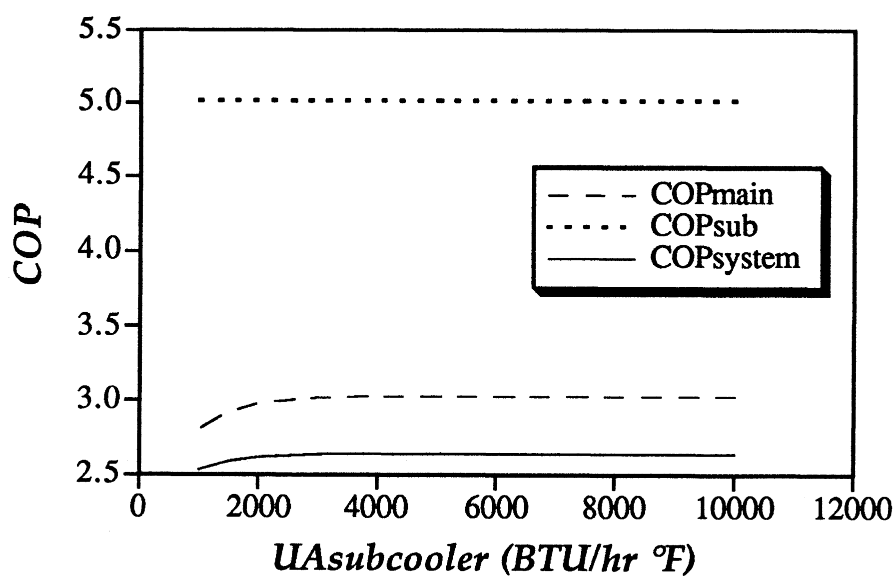

Figure 7.22 COP as a function of the subcooler UA product.

7.3 SYSTEM COMPARISONS

With the setting of the three heat exchanger UA products, only two unknowns remain; the ambient temperature and the main cycle evaporator temperature. For this steady-state model, the assumption was made that both condensers were allowed to float with the ambient conditions. In Figure 7.5, the COP as a function of the ambient temperature was shown. When this curve is compared against the same curves for the two systems previously studied, Figure 7.23 is generated. Dedicated subcooling outperforms ambient subcooling and floating head pressure in the range of ambient temperatures from 30°F and up.

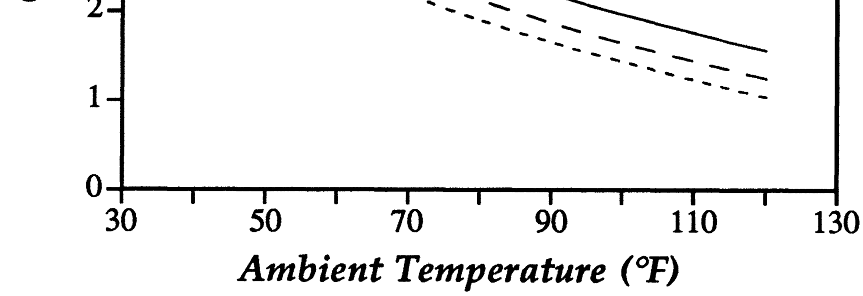

Figure 7.23 COP as a function of the ambient temperature for floating head pressure systems.

The reason that the curves only go down to an ambient temperature of 30°F is due to the temperature constraints on the subcooling cycle. When the ambient temperature falls below approximately 20°F, the condensing temperature of the subcooling cycle falls to thirty degrees. With the evaporator temperature and condensing temperature of the subcooling cycle at 30 degrees, there is no thermal lift between the two cycles. With no thermal lift, the subcooling cycle will not operate. Because the refrigeration system was investigated in Madison, WI where the ambient temperature falls way below 20°F, something had to be done about this subcooling cycle problem. To accommodate this problem, the subcooling cycle was shut off when the ambient temperature fell below 30°F. Past this shut-off point, the dedicated subcooling system acts like a floating head pressure system since there is no subcooling provided to the main cycle. At this shut-off point, the dedicated subcooling system and the floating head pressure system operated in a very similar manner. Table 7.1 shows the system operating characteristics at the shut-off point.

DEDICATED SUBCOOLING SYSTEM

PARAMETER	JUST ABOVE SHUT-OFF	JUST BELOW SHUT-OFF
Condensing Temp.	52.7 °F	54.8 °F
Condensing Pressure	64.3 PSIA	66.5 PSIA
Degrees of Subcooling	20.6 °F	0.0 °F
Compressor Work	40,335 BTU/hr	45,098 BTU/hr

Table 7.1 Operating characteristics of the 15 ton dedicated subcooling system at the shut-off point.

The entire dedicated performance curve can now be plotted as a function of the ambient temperature. The results are shown in Figure 7.24. The jump from dedicated subcooling operation to standard floating head pressure operation can be seen at the shut-off temperature of 30°F. When the ambient temperature falls below the shut-off temperature, the ambient subcooling system outperforms the dedicated mechanical subcooling system. This is the reason that the dedicated subcooling system is known as a warm-weather system and that the ambient subcooling system is known as a cool-weather system.

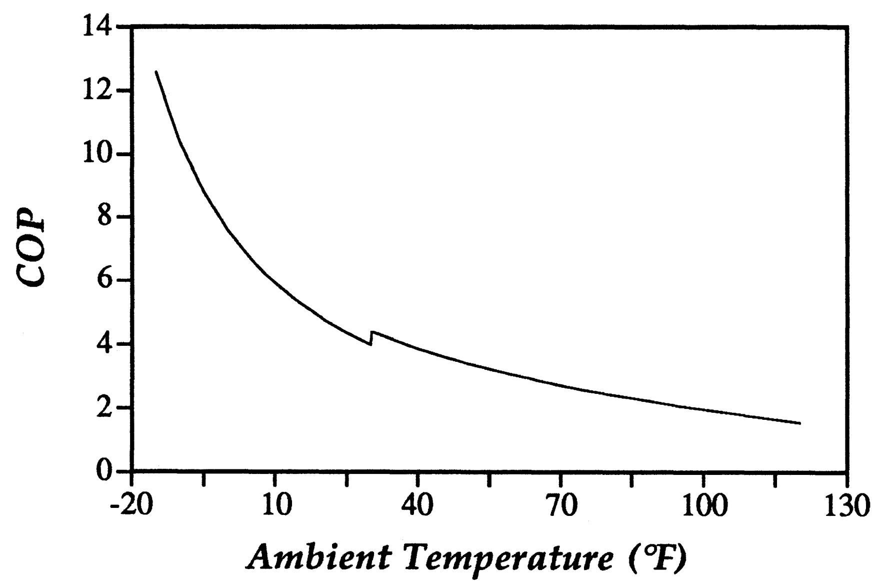

Figure 7.24 COP as a function of the ambient temperature for the dedicated subcooling system.

As with the previous systems studied, the ambient temperature and evaporator temperature were used to generate performance curves. The performance curves were variable in ambient temperature for fixed values of the evaporator temperature. The performance equations are included in Appendix E.

7.4 DEDICATED SUBCOOLING WITH FIXED HEAD PRESSURE

If the dedicated mechanical subcooling system is subject to the constraints of fixed head pressure, it still outperforms the other fixed head pressure systems studied at high ambient temperatures. For the steady-state model, the set point for the main cycle and subcooling cycle condenser head pressures was assumed to be 100 psia. This value is the same as that for the fixed head pressure system discussed in Chapter 4. As with the other fixed head systems studied, the dedicated subcooling system with fixed head shows the characteristic change in slope at the set point. The results can be seen in Figure 7.25.

At low ambient temperatures, the ambient subcooling system outperforms the dedicated system due to the increase in the amount of subcooling with the falling ambient temperature for the ambient subcooling system. The amount of subcooling for both systems is compared in Figure 7.26. The COP of the ambient subcooling and dedicated subcooling systems are equal at approximately 45°F. However when viewing the amount of subcooling performed at 45°F, there is a disparity. The disparity is easily explained; the benefits of the additional subcooling provided by the dedicated subcooling cycle over the ambient subcooling system match the detriment on the dedicated cycle due to the subcooling compressor work at this temperature.

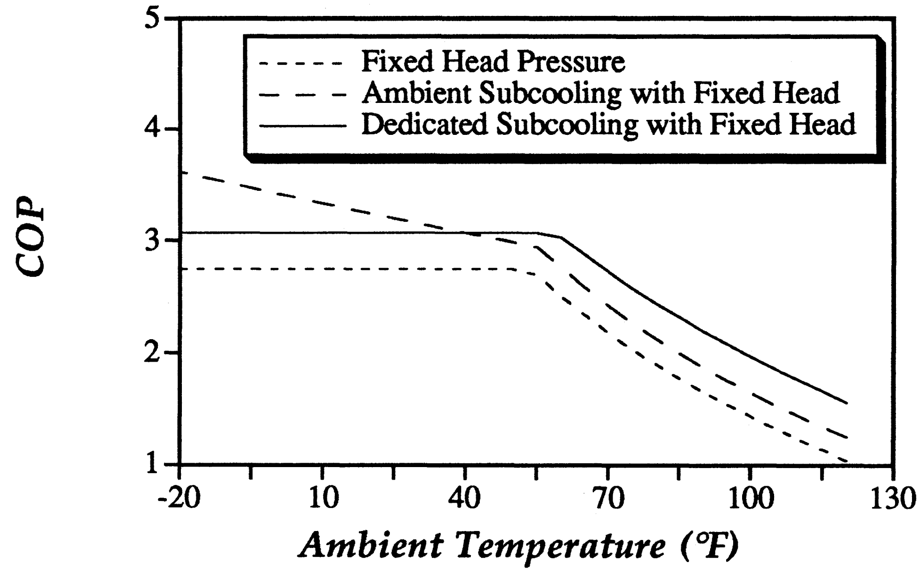

Figure 7.25 COP comparison as a function of the ambient temperature for the fixed head systems.

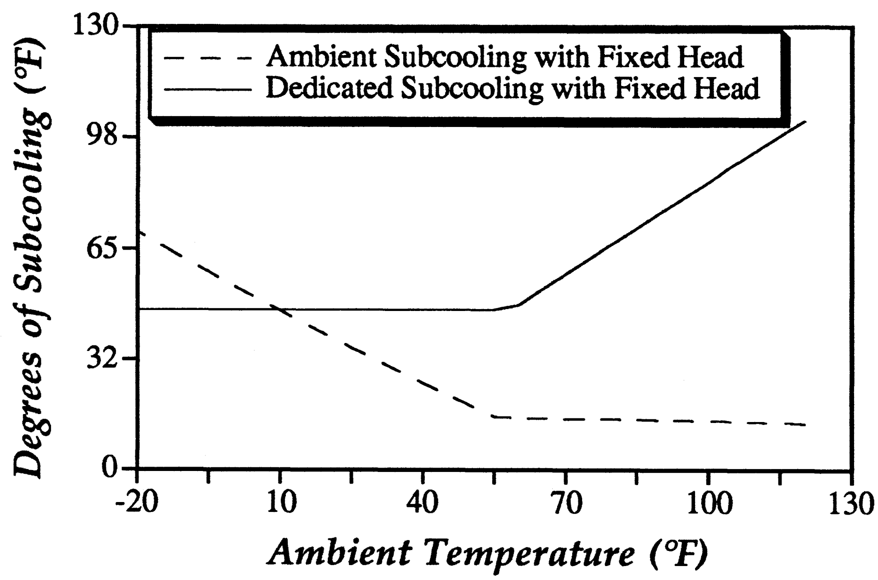

Figure 7.26 Subcooling comparison as a function of the ambient temperature for the fixed head systems.

7.5 PERFORMANCE CURVES

Performance curves were developed for the fixed head and floating head pressure dedicated subcooling systems. The performance equations that were input to the annual simulations were then developed from the performance curves using curve-fitting techniques. The dedicated subcooling with floating head pressure performance curves are shown in Figures 7.27. The curves were developed using the design parameters of the heat exchanger sizes, the subcooling evaporator temperature, the compressor isentropic efficiency, and the condenser cooling air flow rates chosen in the previous sections. Below the critical ambient temperature that causes the subcooling cycle to shut down, the dedicated subcooling system uses the performance equations from the floating head pressure system since both systems act identically. The dedicated subcooling with fixed head pressure performance curves are not shown. If shown, they would exhibit the characteristic horizontal section at the head pressure set point. The dedicated subcooling with fixed head pressure performance equations used the dedicated subcooling with floating head pressure performance equations and the COP at the set point.

7.6 UA OPTIMIZATION

In section 7.2.4, the UA products for the heat exchanger were developed based on the design concept of "picking the knee of the curve". The values chosen for the main cycle condenser, subcooling condenser, and subcooler were 15000, 5000, and 2000 BTU/hr °F respectively.

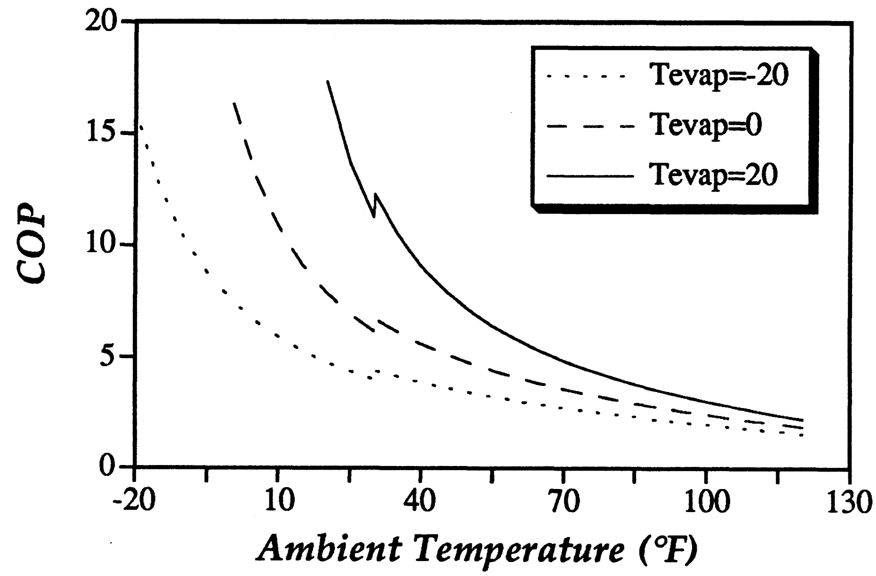

Figure 7.27 Performance curves for the dedicated subcooling system with floating head pressure. The jumps in COP at 30°F represent the change from dedicated subcooling to a standard floating head pressure system.

The values chosen represent a typical design for a dedicated subcooling system; a small subcooler and a subcooling condenser that is a fraction of the size of the main cycle condenser. With these values, the performance of the dedicated subcooling cycle was evaluated. However, it is possible to improve this system COP by redistributing the total heat exchanger UA product for the three heat exchangers. This section investigates the optimum UA distribution, develop some design guidelines, and evaluate the optimum conditions over the entire range of operating conditions.

The sum of the heat exchanger UA products chosen for the dedicated subcooling system is 22,000 BTU/hr °F. This total UA product remained constant and was a constraint on the optimization. With three heat exchanger sizes unspecified and one constraint, the problem became a two-variable optimization. In order to solve this two-variable optimization problem, two separate methods were utilized. These methods involved solving the two-variable problem using a direct search methodology, or a one variable parametric table optimization using the Golden Section search methodology. The parametric approach was used to generate trends, while the two-variable solution was used to find the exact solution. The parametric table solution involved varying the UA of the subcooler, and then maximizing the system COP based on the size of the main cycle condenser. In this way all three unknowns are solved; the UA for the subcooler is set, the UA of the main cycle condenser corresponding to maximum COP is found. The UA of the subcooling condenser is then the sum of the subcooler and main cycle condenser UA's subtracted from the total UA.

Before explaining the results, the "ratio" between the cycles must be defined. The ratio is defined as the amount of any variable in the main cycle divided by the sum of this variable in the subcooling cycle and the main cycle. For example, the refrigerant flow rate ratio is:

$$m_{ref}ratio = m_{ref,main} / (m_{ref,main} + m_{ref,sub}) \tag{7.4}$$

The optimization was based on standard conditions; an ambient temperature of 80°F and an evaporator temperature of -20°F. The optimized COP as a function of the subcooler UA is shown in Figure 7.28. At the optimum point, the subcooler size represents about 10% of the total UA product and corresponds to an effectiveness near 1.0.To generate an exact solution to the optimization problem, a two-variable optimization was performed.

The results are listed in Table 7.2. From the results, the initial choice of the main cycle condenser was near optimal, while the choice of the UA values for the subcooler and subcooling condenser were only slightly off. This explains the small difference between the optimized COP and the COP found earlier for the standard conditions (about 1% at default conditions). The optimization would be expected to provide more noticeable results if the total allocated UA product was somewhat smaller.

PARAMETER	DEFAULT CONDITION	OPTIMIZED VALUE
Main Cycle Condenser	15000 BTU/hr °F	14967 BTU/hr °F
Subcooling Condenser	5000 BTU/hr °F	4528 BTU/hr °F
Subcooler	2000 BTU/hr °F	2505 BTU/hr °F
COP	2.44	2.45

Table 7.2 Component UA comparison between the default and optimized values.

The relative size of the condensers represents an important solution to the optimization problem. From viewing Figure 7.29, the optimized size of the main cycle condenser falls as the subcooler size increases. The subcooling condenser size remains virtually constant with the increasing subcooler. This implies that the condenser UAratio falls as the subcooler UA increases.

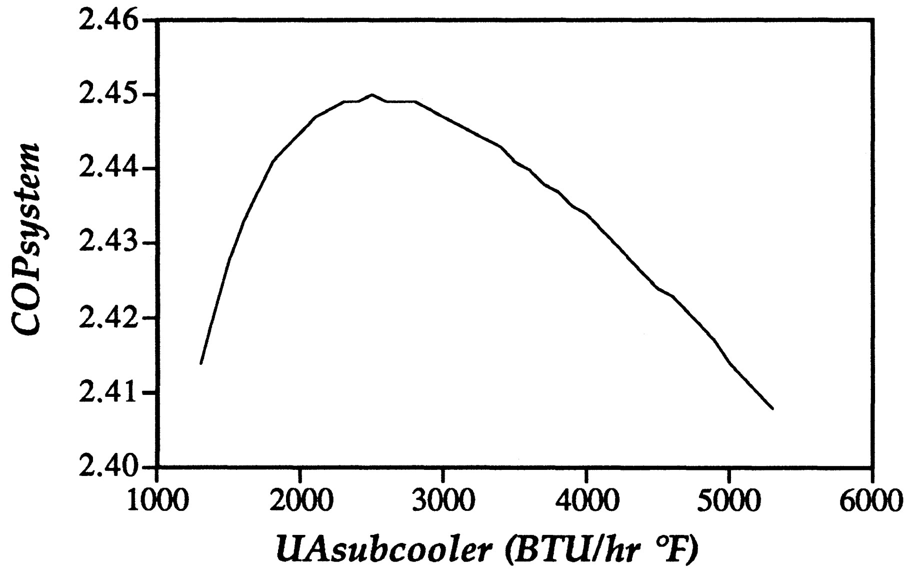

Figure 7.28 Standard conditions optimized COP as a function of the subcooler UA product.

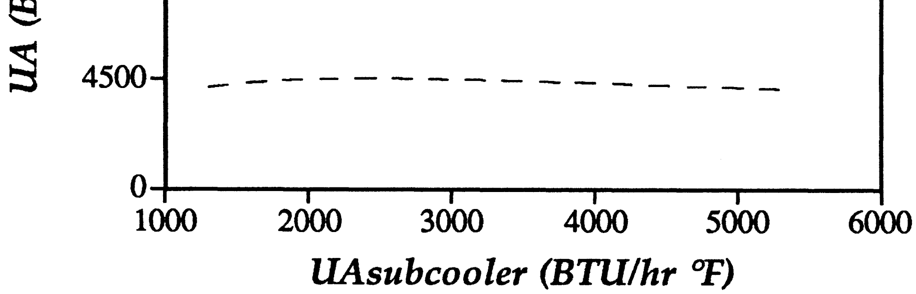

Figure 7.29 Standard conditions optimized condenser sizes as a function of the subcooler UA product.

An important trend is revealed when the refrigerant flow rate ratio and the UA ratio are plotted as functions of the subcooler UA. Referring to Figure 7.30, the two curves are almost exactly alike. This implies that to maximize the COP, the UA sizes of the condensers should be distributed in the same ratio as the refrigerant flow rates.

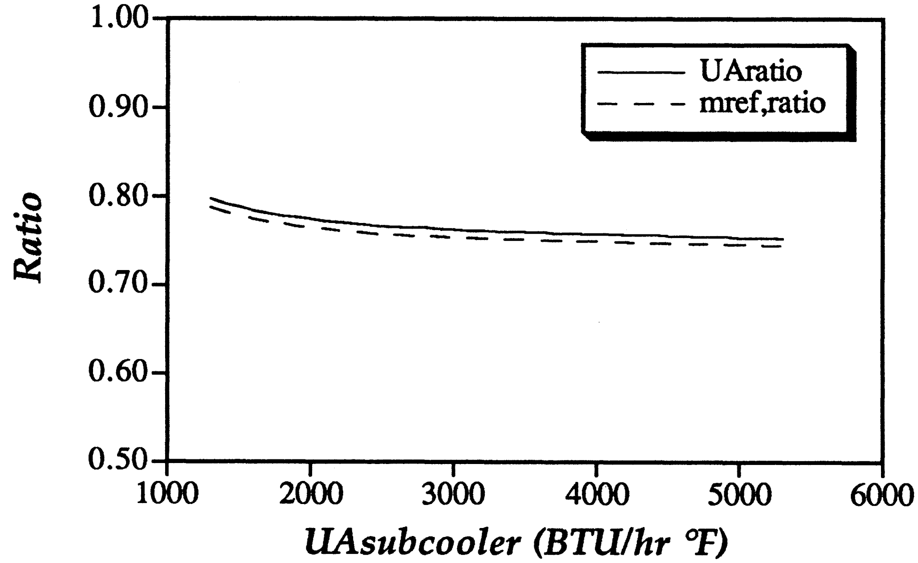

Figure 7.30 Standard conditions optimized ratios as a function of the subcooler UA product.

The two trends developed above gave rise to a design guideline for the dedicated subcooling system. Choose a UA size for the subcooler that meets the requirements of the system. Distribute the remaining UA product according to the expected ratios of the refrigerant flow rate at the design conditions. In this section, the design ambient temperature does not correspond to the temperature at which sizing of the compressors is done, it is simply the temperature at which the system will most likely operate. The optimized COP as a function of the subcooler size was found to be very flat, so the UA subcooler size choice is relatively unimportant to the optimum COP. To make sure that this design criteria holds over all ranges of ambient and evaporator temperatures, a sensitivity analysis was performed.

7.6.1 AMBIENT TEMPERATURE EFFECTS

To perform the ambient temperature sensitivity analysis, a three-variable optimization was used. When the optimized values of the heat exchanger sizes are plotted versus the ambient temperature, three trends stand out; the optimized main cycle condenser UA rises as the ambient temperature decreases, the optimized subcooling condenser UA falls as the ambient temperature decreases, and the subcooler optimized UA remains virtually constant. These trends are shown in Figure 7.31. If the system were to be designed for low ambient temperature operation, the main cycle condenser should be much larger than the subcooling condenser (about nine times as large at an ambient design temperature of 40°F). If the system were to be designed for a high ambient temperature application, the main cycle condenser should be only slightly larger than the subcooling condenser (about 2.3 times as large for an ambient design temperature of 100°F).

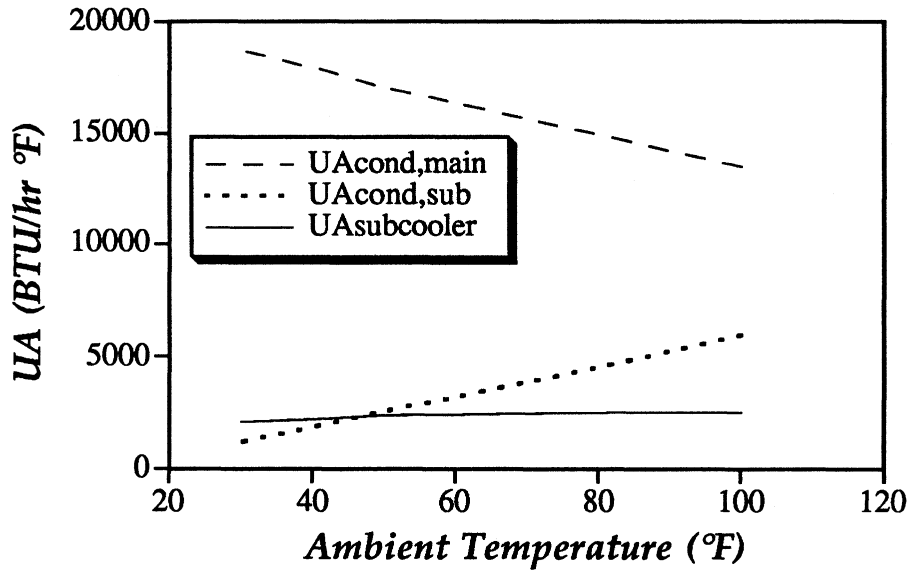

Figure 7.31 Optimized component UA products as a function of the design ambient temperature.

The reason for the ambient temperature effects is due to the optimum amount of subcooling performed. Since the optimized subcooler UA product remains almost constant for the entire range of ambient temperatures, the amount of subcooling decreases as the ambient temperature falls. With the decreasing amount of subcooling done, the subcooling cycle condenser should become smaller since the load on the system is decreasing. The optimal amount of subcooling is shown as a function of the ambient temperature in Figure 7.32. Regardless of the design ambient temperature, the subcooler should represent about 10 to 15% of the total UA product available for distribution.

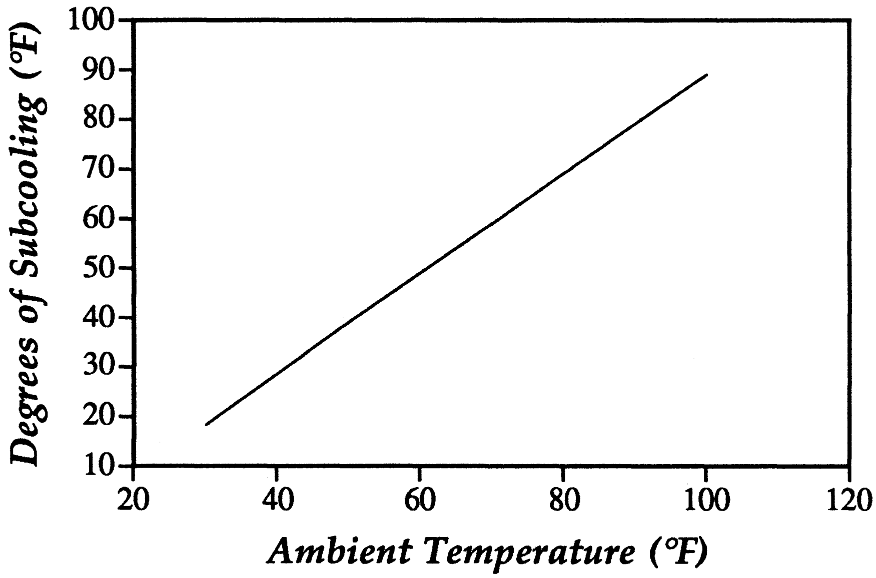

Figure 7.32 Optimized amount of subcooling as a function of the design ambient temperature.

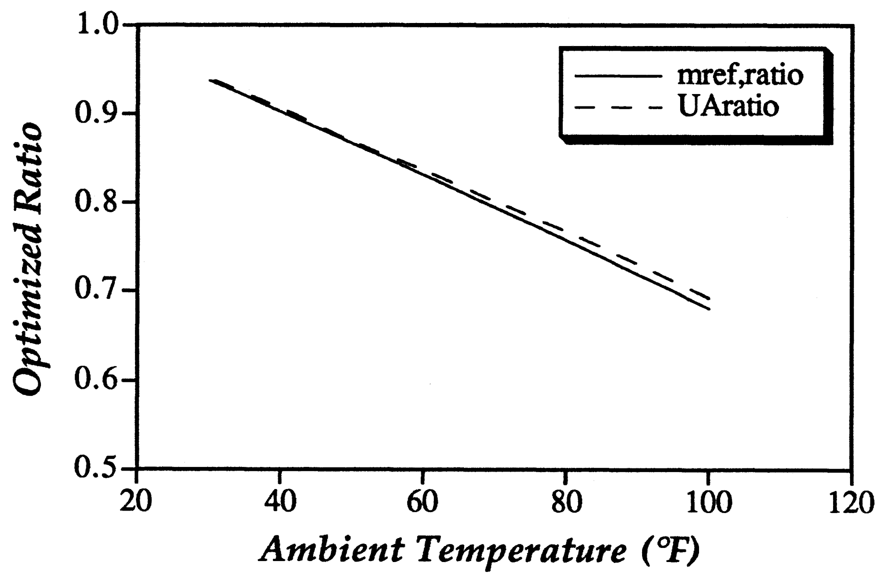

Figure 7.33 Optimized system ratios as a function of the design ambient temperature.

For the standard conditions, optimization of the refrigerant flow rate ratio and the UA ratio were found to be practically identical. If these two optimized ratios are plotted as a function of the ambient design temperature, the same trend holds true. Therefore regardless of the ambient temperature selected for distributing the UA product, the condenser UA's should be distributed in the same ratio as the flow rate ratios between the systems. These results can be seen in Figure 7.33.

7.6.2 EVAPORATOR TEMPERATURE EFFECTS

Like the ambient temperature sensitivity analysis, the evaporator temperature sensitivity analysis relied on a three-variable optimization. When the optimized component UA's are plotted as a function of the evaporator temperature, Figure 7.34 is generated. From the figure, the component UA's are not heavily affected by the design evaporator temperature. The main cycle condenser UA drifts slightly up with increasing evaporator temperature while the subcooler UA decreases slightly with increasing evaporator temperature.

The reason for the slight drift of the UA's is due to the amount of subcooling. Figure 7.35 shows the optimal amount of subcooling as a function of the evaporator temperature. The optimal amount of subcooling can be seen to decrease with increasing evaporator temperature. The lower amount of subcooling is needed because the thermal lift of the main cycle is decreasing as the evaporator temperature increases.

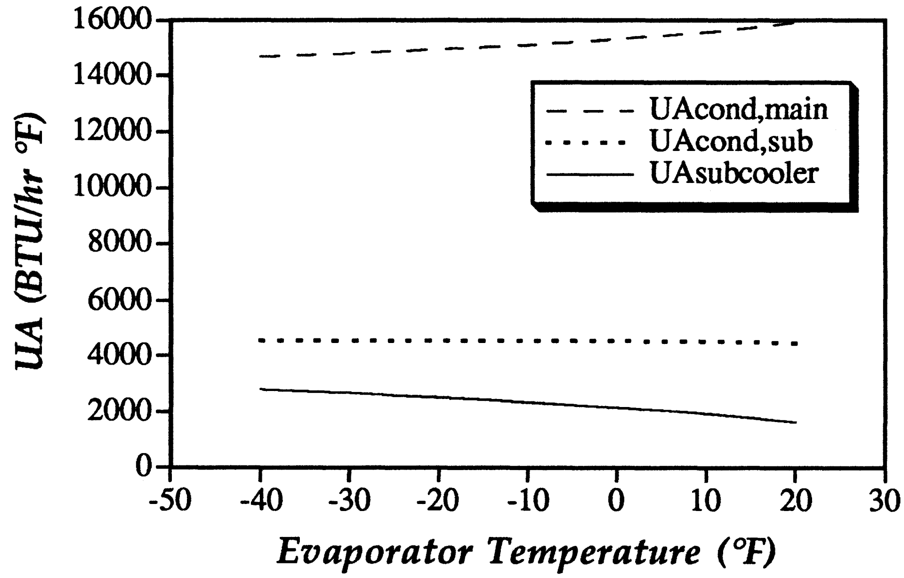

Figure 7.34 Optimized component UA products as a function of the design evaporator temperature.

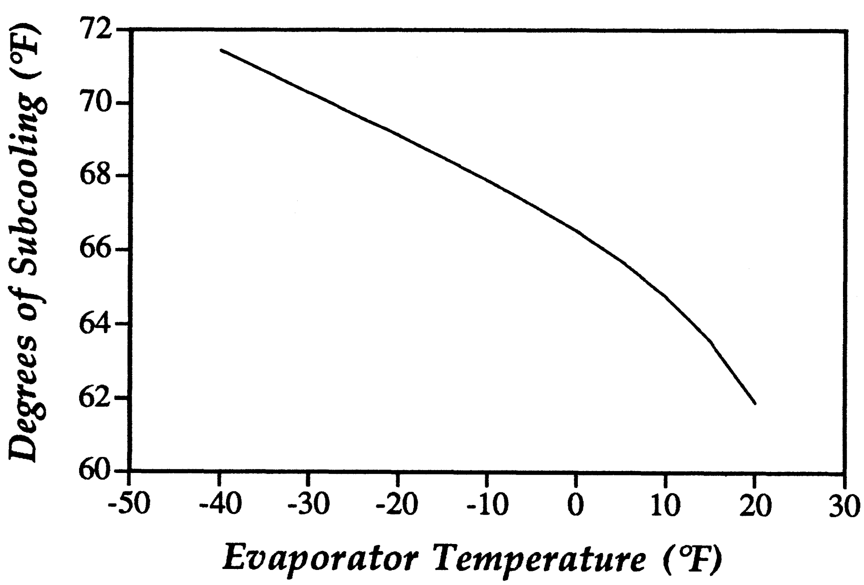

Figure 7.35 Optimized amount of subcooling as a function of the design evaporator temperature.

To make sure that the design guideline proposed in Section 7.5 is still valid, the optimized ratios were plotted as a function of the evaporator temperature. The refrigerant flow rate ratio and the UA ratio are still very close over the entire range of main cycle evaporator temperatures. The results are shown in Figure 7.36.

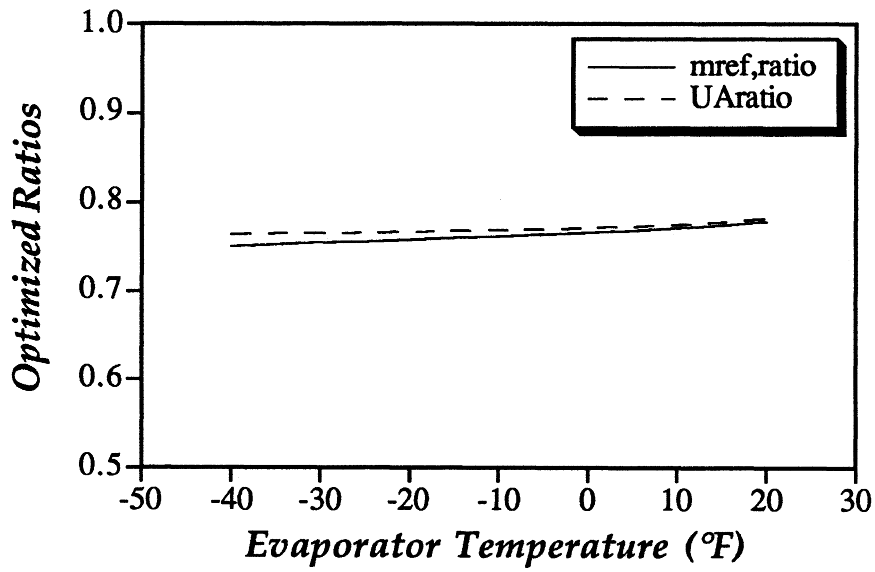

Figure 7.36 Optimized system ratios as a function of the design evaporator temperature.

7.6.3 EFFECT OF DESIGN TEMPERATURE

Questions have arisen at to what happens to the COP if the system is designed at 100°F and the ambient temperature is 40°F. Figure 7.37 and 7.38 should help answer some of these questions. The curves represent the effect of the ambient temperature on four different systems. These four system were designed using the previous guidelines and design ambient temperatures of 40, 60, 80, and 100°F. The high design temperature system outperforms the other three at high ambient temperatures and the low temperature system outperforms the others at low ambient temperature as expected. The only problem that was encountered was the operation of the 40°F system at ambient temperatures above 100°F. The conclusion that can be drawn from this is that to guarantee proper system operation, design the UA distribution at temperatures that ensure that the system will work at high ambient temperatures. The UA sizes of the components for the four design temperatures are listed in Appendix E for reference.

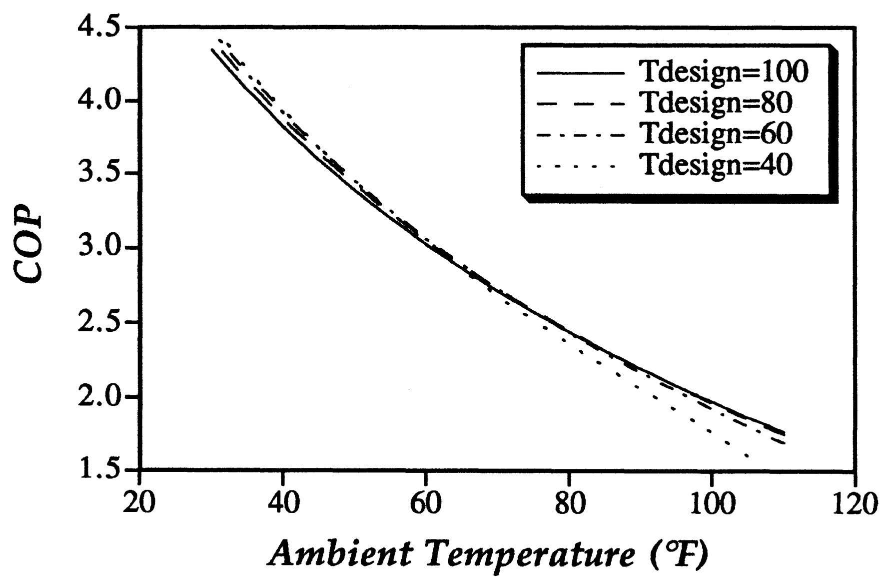

Figure 7.37 COP as a function of the ambient temperature for four systems using the design criteria.

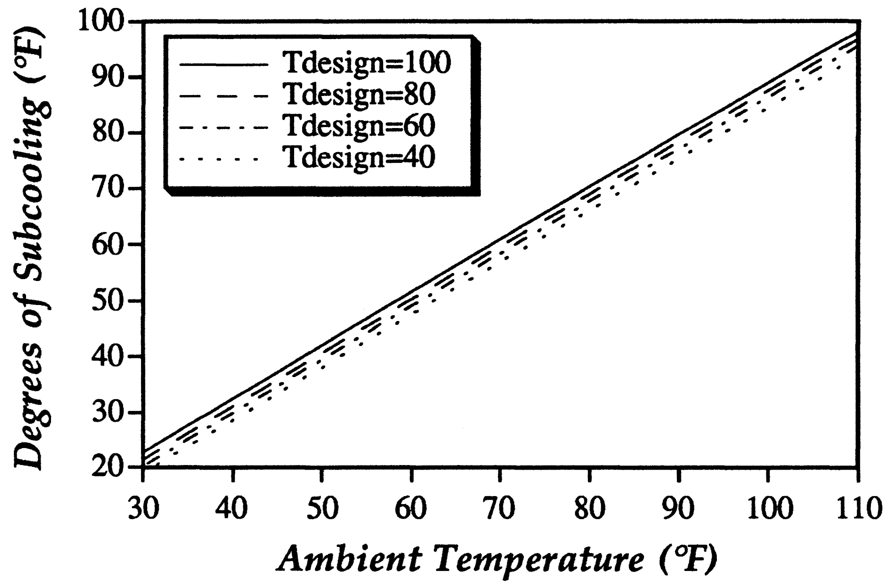

Figure 7.38 Amount of subcooling as a function of the ambient temperature for four systems using the design criteria.

7.6.4 DESIGN GUIDELINE

Regardless of the design temperatures, the following represents a design guideline:

1) Decide on the total UA product available for the dedicated subcooling system based on economics.

2) Choose the UA of the subcooler based on the design evaporator temperature - refer to Figure 7.34 for the optimal choice. The UA of the subcooler is nearly independent of the ambient design temperature so the evaporator design *temperature is the most important. The subcooler UA should represent about 10% of the total UA and correspond to an effectiveness near 1.0.*

3) Choose the design temperature - this is not the temperature used to size the equipment. This temperature should be the temperature that the system will operate at most of the time.

4) Distribute the remaining UA product according to the expected refrigerant flow rates of the two cycles at the design temperature.

7.7 CHAPTER SUMMARY

This chapter explored the concept of dedicated mechanical subcooling. Dedicated mechanical subcooling involves the use of a second vapor compression cycle solely for the purpose of providing subcooling to the main cycle. The cycles are coupled together through the use of a subcooler. The subcooler provides subcooling to the main cycle and acts as the evaporator for the subcooling cycle. The amount of subcooling for a dedicated system is greater than that for a corresponding ambient system because the subcooling evaporator usually provides a lower temperature sink than the ambient conditions. The subcooling cycle acts through a smaller temperature extreme than the main cycle and therefore has a higher COP. With the large amounts of subcooling provided at a high

COP, the COP of the dedicated system is increased beyond that of the systems previously studied at high ambient temperatures.

There were eight variables that were investigated that significantly affected the system performance; the ambient temperature, the evaporator temperature, the subcooling evaporator temperature, the condenser cooling air flow rates, the compressor isentropic efficiency and the three heat exchanger UA products. Each of these parameters was investigated and values were chosen in order to solve the steady-state model. The values that were chosen for these variables were used in all systems studied where applicable.

The compressor isentropic efficiency was set to 0.8 and it was shown that the choice of this value only affected the magnitude of the results. The relative effects of this variable between the systems was unnoticeable.

The condenser cooling air flow rates were set to 3800 lbm air / hr / ton. This value falls within the guidelines established for the type of condenser studied in this report.

The subcooling evaporator temperature provided an interesting result. Raising the subcooling evaporator temperature lowered the thermal lift of the subcooling cycle, but also cut into the amount of subcooling provided to the main cycle. In order to find the optimum point for this temperature, a balance between the amount of subcooling and the subcooling cycle thermal lift had to be reached. The optimum point was seen to be approximately 30°F regardless of the choice of the other seven parameters. This value of

30°F represents a value about half-way between the main cycle evaporator temperature and the main cycle condensing temperature.

The size of the heat exchangers was chosen based on the concept of "picking the knee" of the curve. To solve for the heat exchanger size in question, the effectiveness of the other two heat exchangers was set to 1.0. Setting the other two heat exchangers "perfect" eliminated the relative size effects of the heat exchangers and allowed the effect of the heat exchanger size in question to be seen. The UA sizes chosen for the condenser, subcooling condenser, and subcooler were 15000, 5000, and 2000 BTU/hr °F respectively.

With the values chosen for the eight parameters, the dedicated subcooling system was compared against the systems previously studied. It was found that regardless of the head pressure system (fixed or floating), the dedicated system outperformed all systems studied at high ambient temperatures. At low ambient temperatures, the ambient subcooling system outperformed the dedicated subcooling system due to a large amount of ambient subcooling. This trend held true regardless of the head pressure system employed.

The COP of the dedicated system could be improved further by redistributing the UA for the three heat exchangers. It was discovered that the optimum way to distribute the UA was to choose a value for the subcooler that ensured proper system operation and then distribute the remaining UA according to the refrigerant flow rates at the design temperature. Four systems were created using the design criteria and then studied over the range of ambient temperatures. As expected, the systems designed for high ambient temperatures fared the best at higher temperatures and the worst at lower ambient temperatures.

Using all the results developed in this chapter,with the exception of the optimization, the performance curves and performance equations were derived. The performance equations were then used for the annual simulations.

REFERENCES 7

1. Couvillion, R.J., Larson, M.W., and Somerville, M.H., *Analysis of a Vapor-Compression Refrigeration System with Mechanical Subcooling*, ASHRAE Transactions, Vol. 94 Part 2, 1988

2. ASHRAE Handbook, Equipment Volume, 1983, American Society of Heating, Refrigeration, and Air-Conditioning Engineers Inc., Atlanta, Georgia

CHAPTER
EIGHT

ANNUAL RESULTS

The four systems studied have been compared in terms of instantaneous performance over the range of operating temperatures common to commercial refrigeration. However, it is the annual performance and associated economics that determines the benefits of one system over another. The systems were evaluated on an annual basis at two sites; Madison, WI and Miami Fl. The two sites were chosen for their differences in climate. The Madison data should provide results that show how the systems perform for large swings in ambient temperature. The Miami data should provide results that illustrate how the systems operate near constant full load conditions. This section discusses the annual results and draws some conclusions based on these results for the refrigeration systems studied.

8.1 ANNUAL SIMULATIONS

The steady-state models developed in the past four chapters were used to generate system performance equations for COP as a function of the ambient and evaporator temperatures. The performance equations were then input to the TRNSYS [1] refrigerated case model and combined with the supermarket model and the Typical Meteorological Year weather data to form a TRNSYS [1] deck. Annual simulations were run for Madison, WI and Miami Fl. For reference, a graph of the system COP's as a function of the ambient temperature for the four systems studied is shown in Figures 8.1 and 8.2.

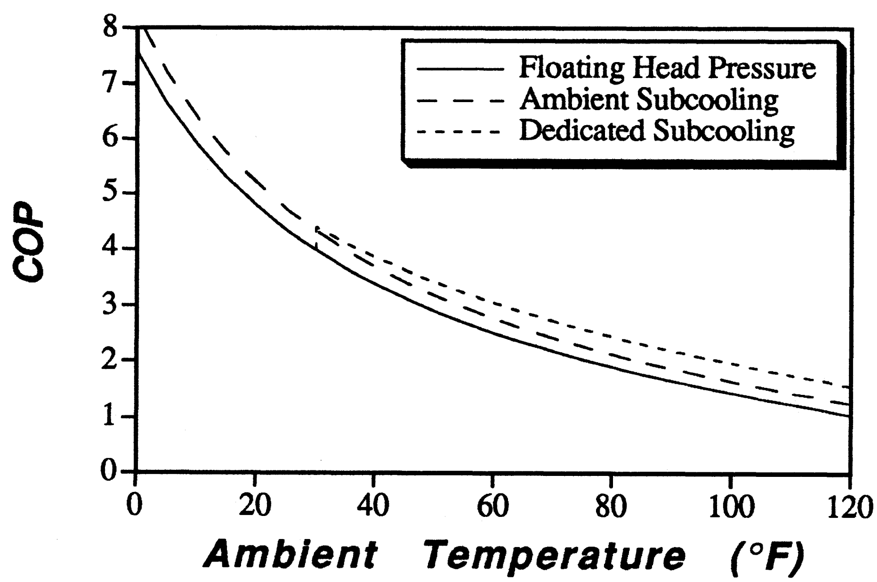

Figure 8.1 Performance comparisons for the systems studied that employed floating head pressure.

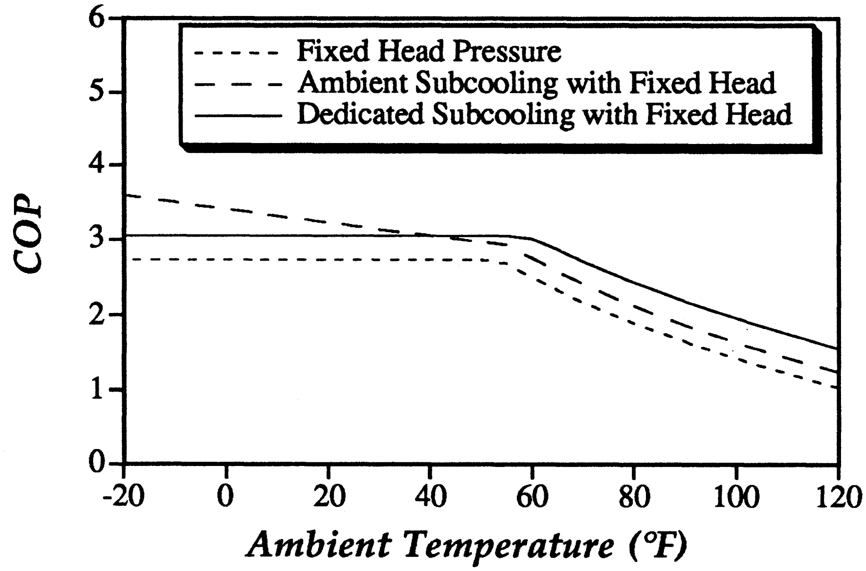

Figure 8.2 Performance comparisons for the systems studied that employed fixed head pressure.

The output from the annual simulation program was the refrigerated case power consumption profile for the refrigeration systems studied. The power consumption profile of the refrigerated cases consisted of the anti-sweat power, the defrost power, the power for the fans and lights, the power required for the actual refrigeration process (case power), and the total refrigerated case power. Since the anti-sweat power, the defrost power, and the power for the fans and lights is independent of the systems studied, the refrigerated cases were evaluated on the basis of the *case* power. The case power for each system is just the total caseload divided by the respective COP. The annual results that are included in later sections are for just the case power, not the total refrigeration power. A power profile for the fixed head pressure system is included as Figure 8.3. The percentage of case power to total power is seen to be approximately 50%.

Figure 8.4 shows the power breakdown as a function of the refrigerated case type. The figure shown is for the fixed head pressure system. The low temperature refrigerated cases are seen to be the large consumers of case power. Approximately one-half of all case power consumed is due to the low-temperature cases.

8.2 SYSTEM COMPARISONS

The ambient temperature in Madison, WI varies greatly throughout the year, much more so than Miami Fl. Using Madison weather data also shows both extremes of operating conditions; the very-high and very-low ambient temperatures. For these reasons, the Madison annual simulations are discussed and explained. The Miami annual simulations were used to generate annual comparisons for warm climates and are discussed with the Madison annual comparisons in Section 8.3.

8.2.1 FIXED HEAD PRESSURE SYSTEMS

The case power of the fixed head pressure system for the twelve months studied is shown in Figure 8.5. The case power varies over the twelve months with a relative maximum occurring in July as expected. The power is relatively constant over the year due to the presence of the fixed head pressure mechanism that keeps the condensing pressure from falling below the set point. Refer to Figure 8.2 for a plot of the fixed head pressure performance as a function of the ambient temperature.

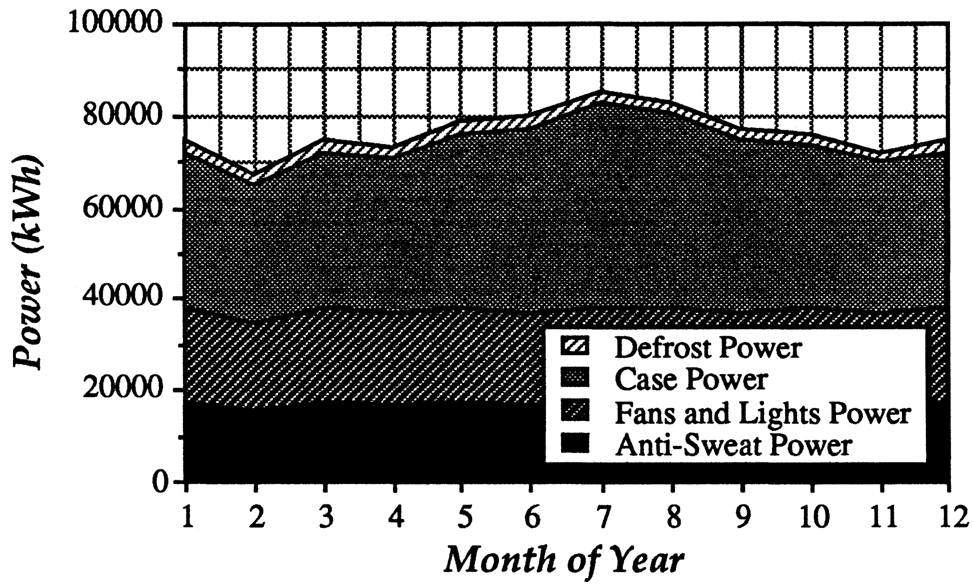

Figure 8.3 Power profile for a fixed head pressure refrigeration system operating in Madison, WI.

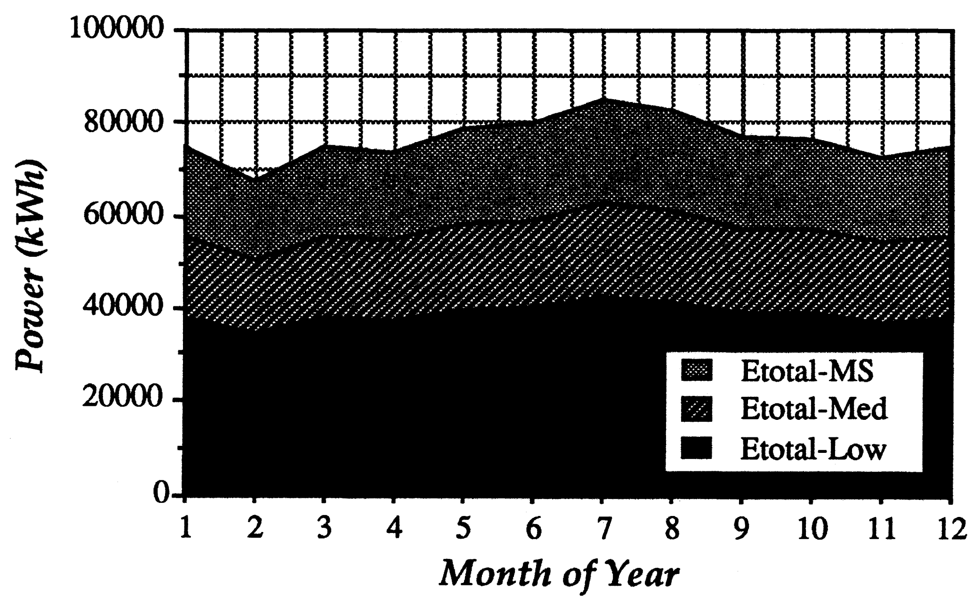

Figure 8.4 Power breakdown for the three refrigerated case types in a fixed head pressure refrigeration system operating in Madison, WI.

The fixed head pressure system is a common refrigeration system and was the basis for the system comparisons to follow. The refrigeration systems were compared in terms of the case power, and for this reason the graphs to follow show the *case* power as a function of time.

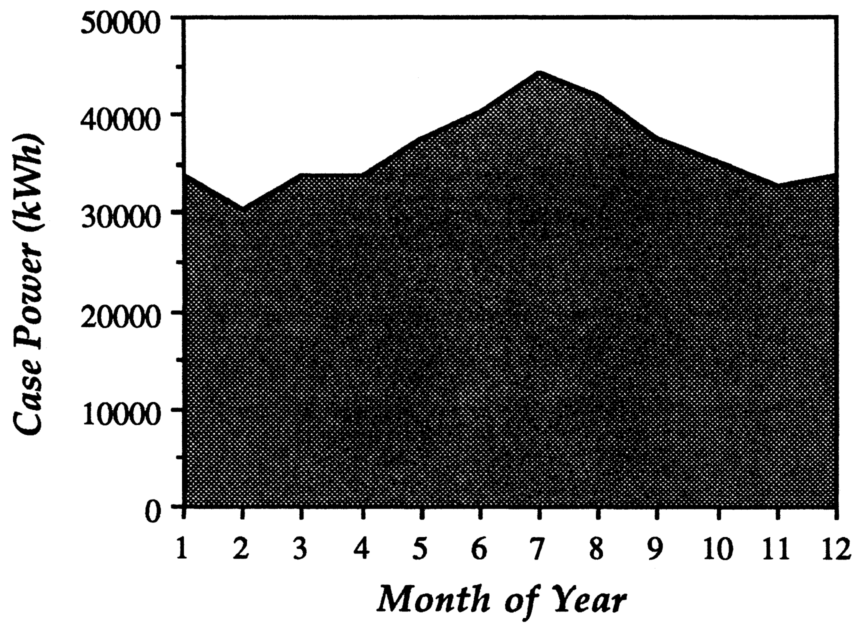

Figure 8.5 Fixed head pressure refrigerated case power profile for the Madison annual simulation.

8.2.2 FLOATING HEAD PRESSURE

The floating head pressure system is compared to the fixed head pressure system in Figure 8.6. The darker area represents the savings of the floating head pressure system over the fixed head pressure system. The energy savings are seen to be the greatest during the winter months and practically zero during the summer months. In the summer, there are few times when the ambient temperature falls low enough for the fixed head pressure mechanism to affect the performance. Therefore, both systems act like floating head pressure systems and there is no relative energy savings. During the winter, there are many times when the fixed head system's condensing pressure is kept at the set point while the floating system's condensing pressure is allowed to fall. The difference in head pressure at these times is the reason for the energy savings of the floating head pressure system over the fixed head pressure system. Refer to Figures 8.1 and 8.2 for plots of the system performance as a function of the ambient temperature.

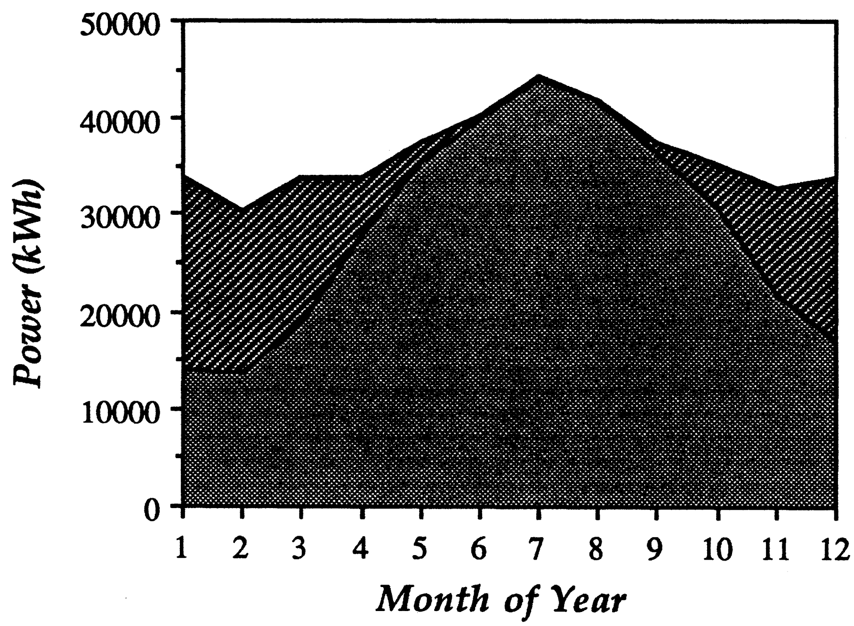

Figure 8.6 Fixed head and floating head pressure power comparisons for Madison, WI. The light area represents the power profile for the floating head pressure system.

Although the floating head pressure system saves energy over the fixed head pressure system, there is no associated demand reduction. The demand charges are based on an ambient temperature of 120°F. At this temperature, both the floating head and fixed head pressure systems act identically.

8.2.3 AMBIENT SUBCOOLING SYSTEMS

In Chapter 6, it was shown that the ambient subcooling system outperformed the floating head pressure system and the fixed head pressure systems due to the increase in heat rejection to the ambient. For this simulation, the head pressure of the ambient subcooling system was allowed to float with the ambient conditions. When the ambient subcooling system is compared to the fixed head pressure system, Figure 8.7 is generated. The ambient subcooling system outperforms the fixed head pressure system due to the subcooling provided to the cycle by the ambient conditions and the floating head pressure effects. Because the head pressure of both systems is allowed to float, the benefits of the increased heat rejection to the ambient becomes apparent when the ambient subcooling system is compared to the floating head pressure system (Figure 8.8). For both graphs, the darker area represents the energy savings associated with the ambient subcooling system.

Unlike the floating head pressure system, the ambient subcooling system saves demand charges over the fixed head pressure system due to the higher COP and lower amounts of work of the ambient subcooling system at high ambient temperatures.

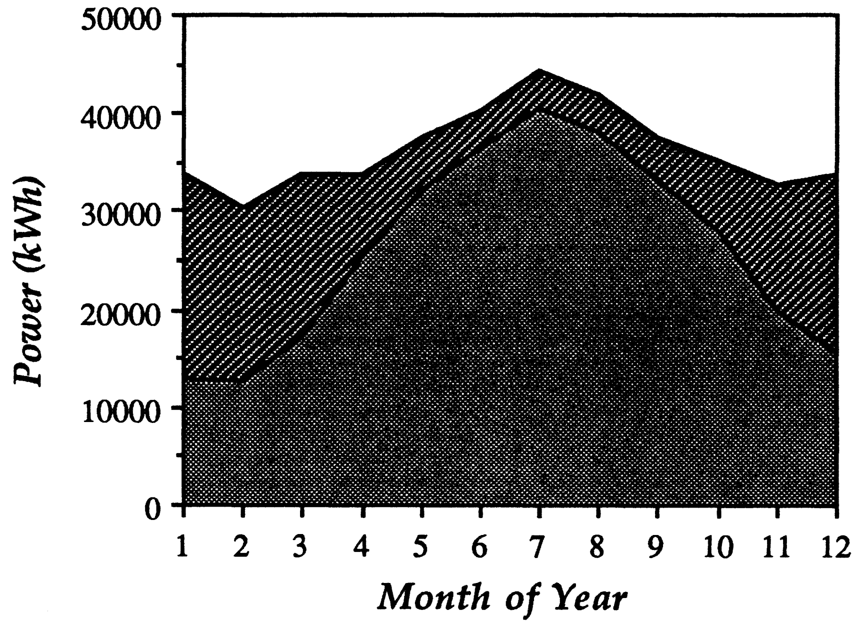

Figure 8.7 Fixed head pressure and ambient subcooling power comparisons for Madison, WI. The light area represents the power profile for the ambient subcooling system.

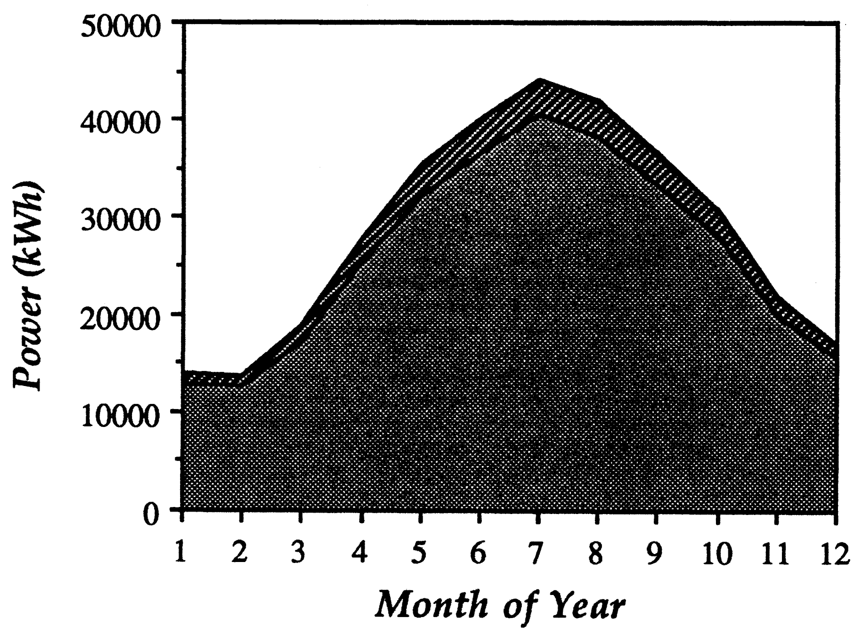

Figure 8.8 Floating head pressure and ambient subcooling power comparisons for Madison, WI. The light area represents the power profile for the ambient subcooling system.

A crude estimate of the demand savings may be obtained from Figure 8.7. The percent demand savings are *approximately* the percent reduction in electricity consumption for July in Madison. The demand savings are seen to be approximately 5 to 10% for the ambient subcooling system. An economic analysis, including the demand savings, is included in a later section.

8.2.4 DEDICATED MECHANICAL SUBCOOLING SYSTEMS

Dedicated mechanical subcooling was found to outperform all other systems studied at high ambient temperatures. The increased performance was due to the operation of a second cycle solely for the purpose of providing subcooling to the main cycle. The dedicated system is compared to a fixed head pressure system in Figure 8.9. The savings of the dedicated system are due to the subcooling performed and the floating head pressure of the dedicated system. To estimate the savings due just to the subcooling, the dedicated system employing fixed head pressure was compared to the standard fixed head pressure system. The results are shown in Figure 8.10.

To better appreciate the savings due to the addition of a subcooling cycle, the dedicated system was compared to the floating head pressure system (Figure 8.11) and the ambient subcooling system employing floating head pressure (Figure 8.12). The energy savings associated with the dedicated subcooling systems for each comparison is represented as the darker areas on the graphs. As expected, the dedicated subcooling system performs the best during the warm summer months when compared to the floating head systems.

The demand savings associated with the dedicated subcooling system are seen to be approximately 15% when compared to fixed head and floating head pressure systems, and 5% when compared to ambient subcooling (Figures 8.9 to 8.12).

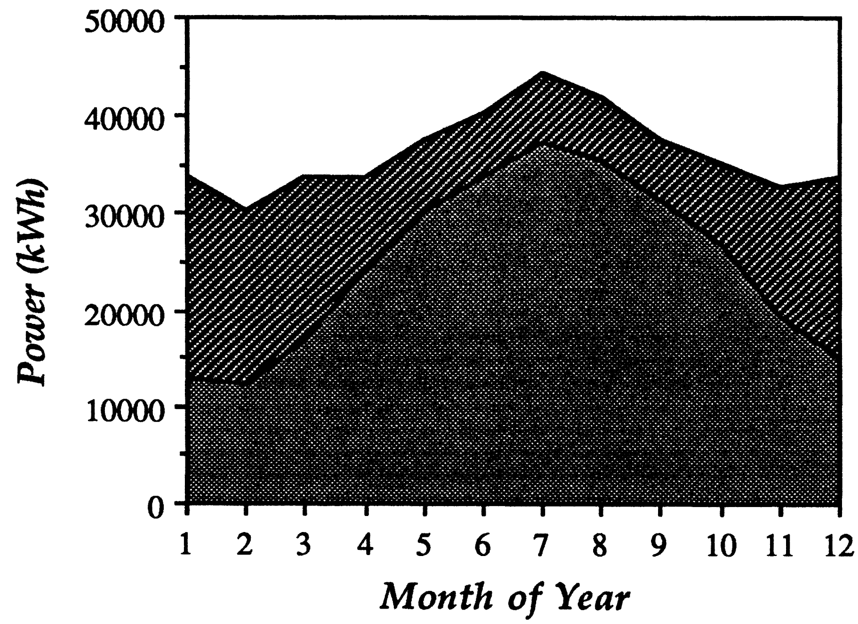

Figure 8.9 Fixed head pressure and dedicated subcooling system employing floating head pressure power comparisons for Madison, WI. The light area represents the power profile for the dedicated subcooling system.

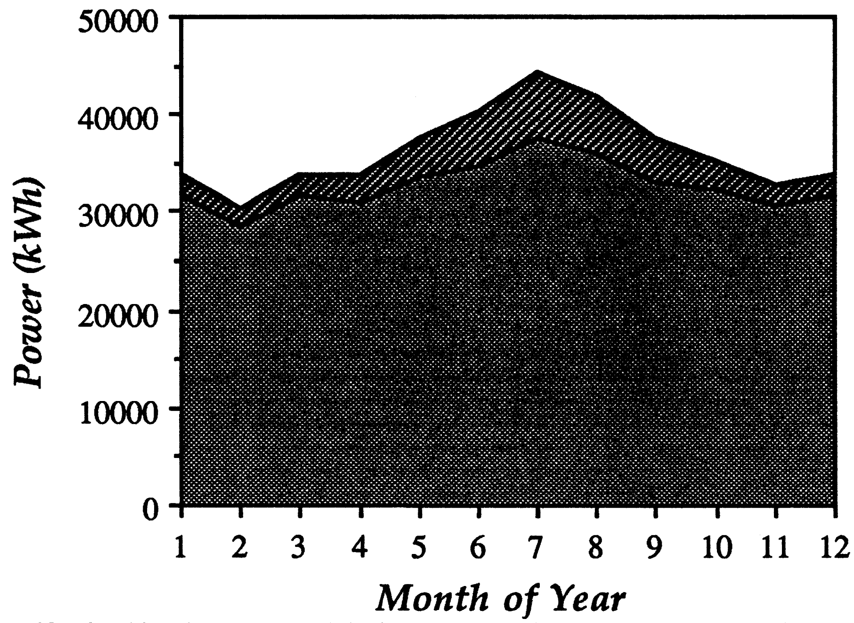

Figure 8.10 Fixed head pressure and dedicated subcooling system employing fixed head pressure power comparisons for Madison, WI. The light area represents the power profile for the dedicated subcooling system.

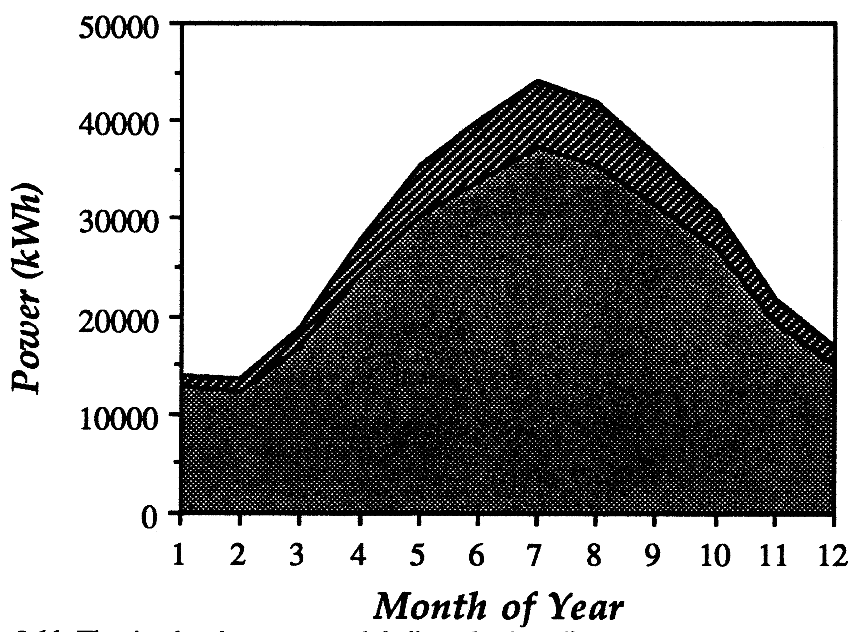

Figure 8.11 Floating head pressure and dedicated subcooling system employing floating head pressure power comparisons for Madison, WI. The light area represents the power profile for the dedicated subcooling system.

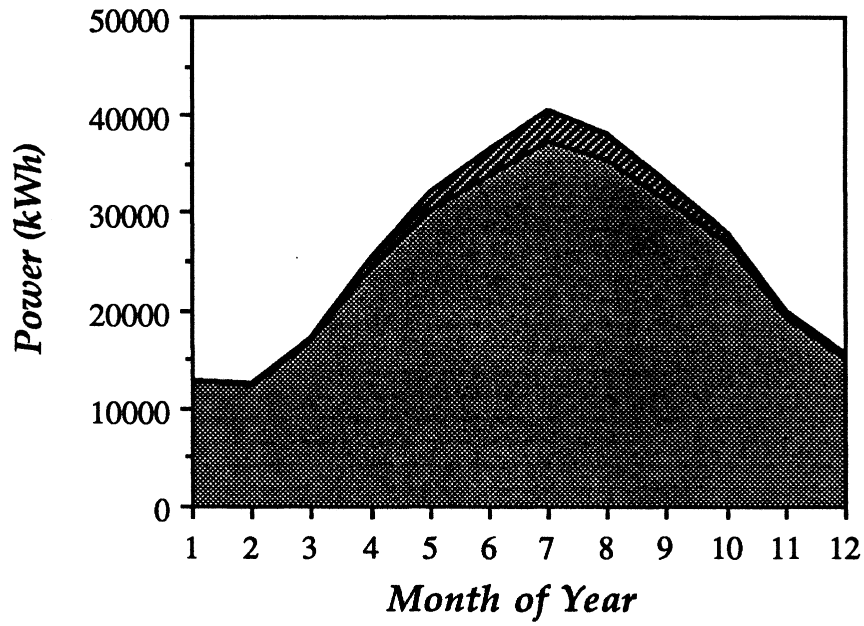

Figure 8.12 Ambient subcooling and dedicated subcooling system employing floating head pressure power comparisons for Madison, WI. The light area represents the power profile for the dedicated subcooling system.

8.3 ANNUAL RESULTS

One of the reasons for this study was to determine the energy savings associated with the four refrigeration system types. Summing the power consumption over the entire year for both Miami and Madison, Figures 8.13 and 8.14 are generated. For both cities, the dedicated subcooling system outperformed the other three systems, implying that for a cool climate like Madison, the savings of the dedicated system over the ambient subcooling system during the summer months outweigh the savings of the ambient subcooling system over the dedicated subcooling system during the winter months.

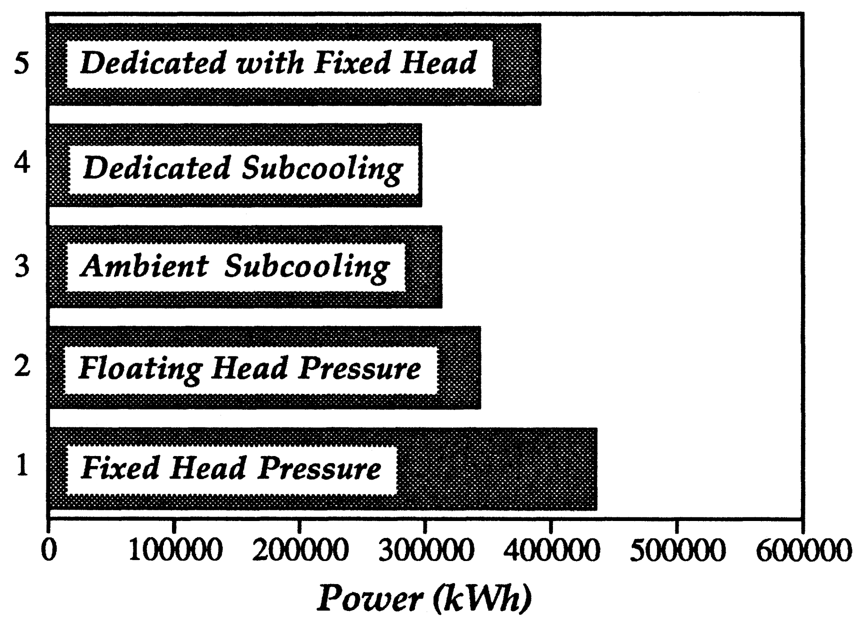

Figure 8.13 Annual case power for Madison WI.

The energy use associated with each system is shown in Table 8.1. The dedicated subcooling system with floating head pressure outperforms all other systems studied in both cities. If the constraint of fixed head pressure is added to the dedicated subcooling system, the choice of cities becomes very important. The dedicated system with fixed head pressure outperforms the floating head pressure and the ambient subcooling system with floating head pressure systems in Miami. However, the ambient subcooling system and the floating head pressure system perform better than the dedicated system with fixed head pressure in Madison due to the extended periods of time when the head pressure is fixed at the set point.

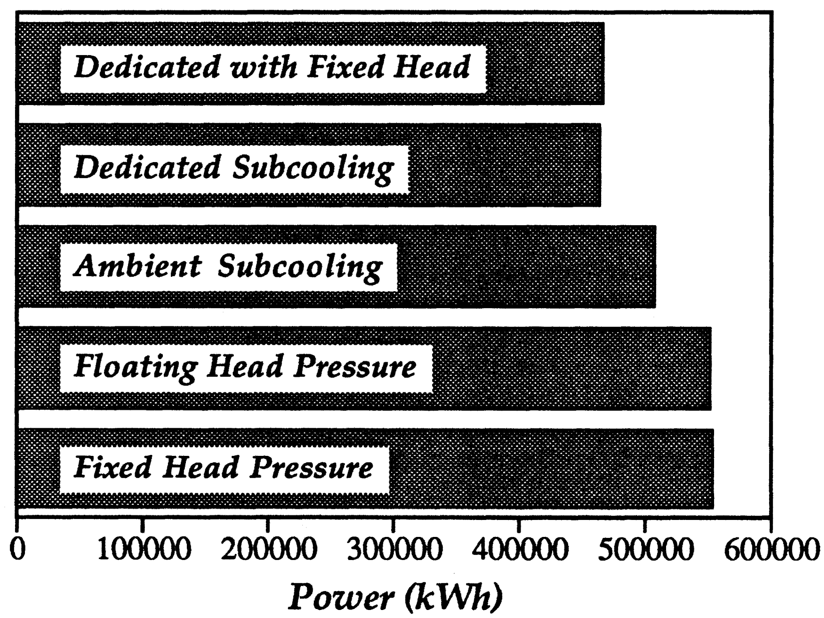

Figure 8.14 Annual case power for Miami FL.

	Annual Energy Consumption for Madison (kWh)	Annual Energy Consumption for Miami (kWh)	Maximum Power (kW)
Fixed Head Pressure	435,687	553,472	126
Floating Head Pressure	343,099	551,714	126
Ambient Subcooling	312,628	508,057	109
Dedicated Subcooling	296,806	463,814	99
Dedicated Subcooling w/ Fixed Head	392,030	466,158	99

Table 8.1 Energy consumption for the refrigerated cases.

The energy savings of the systems studied is important, but the dollar savings associated with each system is the figure that is most important to the purchasers of refrigeration equipment. The associated dollar savings for each system, as compared to the fixed head pressure system for two prices of electricity, are shown in Table 8.2. The prices of electricity for this study were assumed to be \$0.07/kWh with an annual demand charge of \$70/kW, and \$0.11/kWh with an annual demand charge of \$100/kW. These prices reflect the range of costs common to commercial electricity.

	Annual Savings for Madison $0.07/kWh & $70/kW	Annual Savings for Madison $0.11/kWh & $100/kW	Annual Savings for Miami $0.07/kWh & $70/kW	Annual Savings for Miami $0.11/kWh & $100/kW
Fixed Head Pressure	0	0	0	0
Floating Head Pressure	$6,481	$10,184	$123	$193
Ambient Subcooling	$9,804	$15,266	$4,369	$6696
Dedicated Subcooling	$11,612	$17,977	$8,166	$12,562
Dedicated Subcooling w/ Fixed Head	$4,946	$7,502	$8,002	$12,304

Table 8.2 Dollar savings against the standard fixed head pressure system.

8.4 COMPRESSOR SIZING

Another benefit of subcooling that hasn't been mentioned is the actual compressor size reduction. A smaller compressor can be purchased if subcooling is included in the system design. For this study, the compressors were sized to meet the design cooling load for the refrigerated cases based on an ambient temperature of 120°F and evaporator temperatures of -20°F and 20°F. The compressor sizes are listed in Table 8.3 in terms of BTU/hr at these sizing conditions. Using ambient subcooling as an example, the compressor size can be reduced by approximately 2.4 tons for the low temperature cases and 2.5 tons for the medium temperature cases. If a dedicated mechanical subcooling system is operated, the total compressor size can be reduced by approximately 4.8 tons for the low temperature cases and 2.9 tons for the medium temperature cases. Although the compressor size reductions are not included in the economics of this report, they are significant. The compressor size reductions were based on design cooling loads of 15 tons for the low temperature cases, 15 tons for the medium temperature cases, and 26.25 tons for the medium temperature, multi-shelf cases.

8.5 PART LOAD RATIO EFFECTS

One of the major complaints against the floating head pressure systems is the part-load ratio effects on the compressor. Systems that utilize floating head pressure typically have low compressor loads at low ambient temperatures.

	Compressor Capacity (BTU/hr) for Low-temperature cases	Compressor Capacity (BTU/hr) for Medium-temperature cases
FIXED HEAD PRESSURE SYSTEM	172,531	257,595
FLOATING HEAD PRESSURE SYSTEM	172,531	257,595
AMBIENT SUBCOOLING	143,887	227,507
DEDICATED SUBCOOLING	80,060 (Main) 35,307 (Subcooling)	135,927 (Main) 86,199 (Subcooling)

Table 8.3 Reciprocating compressor capacity comparisons.

These low loads usually lead to decreased performance due to compressor modulating processes such as cylinder unloading and cycling. The effect of part-load on the system performance was evaluated to see whether the annual savings of the systems was changed drastically. The part-load factor (plf) as a function of the part-load ratio (plr) that was used for this report is shown in Figure 8.15. The part load ratio is defined as:

$$plr = W_{comp} \,/\, W_{comp,rated} \tag{8.5}$$

The work of the compressor is then found from equation 8.6.

$$W_{comp} = W_{comp} \,/\, plf \tag{8.6}$$

Similar to the systems previously studied, four steady-state models with the part-load effects were developed. The performance equations were then curve-fit from the derived performance curves and input to the annual simulations. Only the final economic results are shown. A steady-state model, the performance equations and the annual energy consumption data with the part load effects are included in Appendix F for reference.

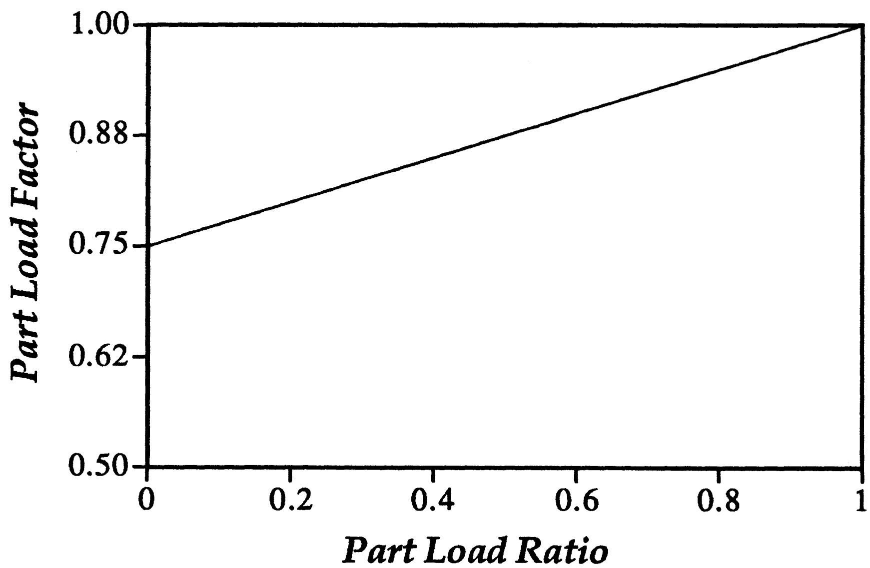

Figure 8.15 Part load factor as a function of the part load ratio for the hermetically sealed, reciprocating compressors modelled in this report.

	Annual Savings for Madison $0.07/kWh & $70/kW	Annual Savings for Madison $0.11/kWh & $100/kW	Annual Savings for Miami $0.07/kWh & $70/kW	Annual Savings for Miami $0.11/kWh & $100/kW
Fixed Head Pressure (plr)	0	0	0	0
Floating Head Pressure (plr)	$6,993	$10,990	$143	$225
Ambient Subcooling (plr)	$10,952	$17,041	$5,353	$8,242
Dedicated Subcooling (plr)	$13,519	$20,974	$10,278	$15,881

Table 8.4 Dollar savings against the standard fixed head pressure system when part-load ratio effects are included.

The saving of the the systems are slightly higher when the part-load ratio effects are considered due to the increase in the amount of work performed. The greater the work, the higher the head pressure, the greater the benefits of floating head pressure and subcooling. The added savings, on average, are around 10 to 20%. The COP as a function of the ambient temperature is shown in Figure 8.16 for the four systems studied with part-load effects. The relative performance of the systems is unchanged by the addition of the part-load ratio effects. Since the part-load effects did not have an effect on system choice, the economics section that follows was not based on the annual savings with part-load ratio effects.

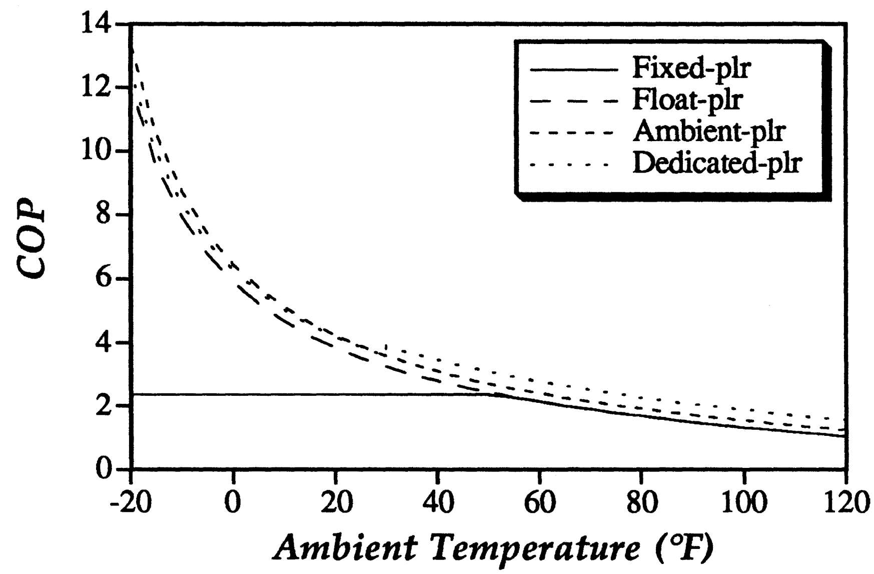

Figure 8.16 COP as a function of the four systems studied with part load effects included.

8.6 SYSTEM CONCLUSIONS

If the refrigeration systems are to be attractive to supermarket owners, the increment in first cost of the new system must be recovered with annual savings in an acceptable amount of time. To determine the allowable first cost of the systems studied, a simplified economic analysis was performed. The P1, P2 life cycle cost method [2] was used to determine the allowable first cost increment of the systems studied. For this study, it was assumed that the equipment has no resale value, there are no miscellaneous costs involved with the operation of one system over the other, and that no money was borrowed to purchase the systems. The allowable first costs were based on a payback period of three years.

The life cycle savings (LCS) associated with a given system in current dollars is given by:

$$LCS = P1 * F \; - \; P2 * E \tag{8.1}$$

where P1 = Present Worth Factor associated with operating costs

P2 = Present Worth Factor associated with equipment costs

F = First year operating cost savings associated with system

E = Initial cost of refrigeration system

The present worth factors are now developed;

$$P1 = (1 - t) * PWF(N,i_{fuel},d) \tag{8.2}$$

$$P2 = 1 + t_p * (1 - t) * PWF(N,i,d) \tag{8.3}$$

where t = income tax rate - assumed to be 40%

N = # years in analysis - assumed to be three years

i_{fuel} = fuel inflation rate - assumed to be 4%

t_p = property tax rate - assumed to be 3%

d = depreciation rate - assumed to be 8%

i = inflation rate - assumed to be 4%

PWF = Present Worth Factor

the Present Worth Factor is defined as:

$$PWF(N,i,d) = \sum_{j=1}^{N} \frac{(1+i)^{j-1}}{(1+d)} \tag{8.4}$$

With the acceptable payback period set to three years and the values given above, P1 was found to be 1.606 and P2 was found to be 1.048. To solve for the allowable incremental cost (AIC) of a system, the LCS in Eq. 8.1 is set to zero. The equation is then solved for E, the allowable incremental cost. The results are given in Table 8.5.

8.6.1 FLOATING HEAD PRESSURE

The floating head pressure system for Madison Wi, seems to be a strong economic candidate. If the incremental cost between the floating head pressure system and the fixed head pressure system is less than $9,931 for an electric rate of $0.07/kWh than the floating head pressures show paybacks of less than three years. If the floating head pressure system is to be run in Miami, the systems must have approximately the same cost or no payback is seen.

	Allowable Incremental Cost for Madison at $0.07/kWh & $70/kW	Allowable Incremental Cost for Madison at $0.11/kWh & $100/kW	Allowable Incremental Cost for Miami at $0.07/kWh & $70/kW	Allowable Incremental Cost for Miami at $0.11/kWh & $100/kW
Fixed Head Pressure	0	0	0	0
Floating Head Pressure	$9,932	$15,606	$188	$296
Ambient Subcooling	$15,024	$23,394	$6,695	$10,261
Dedicated Subcooling	$17,795	$27,548	$12,514	$19,250
Dedicated Subcooling w/ Fixed Head	$7,579	$11,496	$12,263	$18,855

Table 8.5 Allowable incremental costs against the standard fixed head pressure system.

8.6.2 AMBIENT SUBCOOLING

The only major difference between the ambient subcooling system and the fixed head pressure system is the additional heat exchangers needed for the three refrigerated case types. If the total costs of these heat exchangers can be kept under the AIC's listed in Table 8.5, the ambient subcooling system will have payback periods under three years.

8.6.3 DEDICATED MECHANICAL SUBCOOLING

The difference between dedicated subcooling systems and standard systems is the addition of a second vapor-compression cycle to subcool the main cycle. The dedicated subcooling systems were simulated for the three refrigerated case types. For the supermarket studied, the additional subcooling cycles totalled approximately thirteen tons of refrigeration. Typically refrigeration system cost about \$1000 per ton, so the dedicated subcooling system would show paybacks of less than three years for AIC's above approximately \$13,000. For dedicated subcooling systems with floating head pressure, the AIC's range from \$12,500 to \$27,500. Therefore, it is likely that the dedicated subcooling system would have payback periods under three years. An important note is that these results are for all three refrigerated case types. Dedicated subcooling systems are usually used on just low-temperature refrigerated cases. If the analysis was based just on the low temperature cases, the dedicated systems would probably show paybacks under three years for any reasonable electric rate. If the dedicated system is subject to fixed head pressure constraints, the allowable incremental costs range from \$7600 in Madison, to \$18,800 in Miami. The choice of a dedicated system with fixed head pressure may or may not be a wise economic choice. The decision is based on the climate and the electric rates. A warm climate with high electric rates would favor the dedicated subcooling system with fixed head pressure. The dedicated subcooling AIC's for Madison fall below zero against the *floating* head pressure system, so the choice of a dedicated subcooling system with fixed head pressure over a floating head pressure system is not advised for cooler climates like Madison.

8.7 CHAPTER SUMMARY

The economics of the refrigeration systems studied were evaluated using the results from the annual simulations. The allowable incremental cost (AIC) of each system was computed using the life cycle savings method for two cities; Madison, WI. and Miami, FL and for two electric rates; $0.07 and $0.11 per kWh with demand charges of $70/kW and $100/kW respectively..

The floating head pressure system exhibited AIC's ranging from $9900 to $15,600 for Madison, WI and $188 to $296 for Miami FL. The large differences between the two cities is due to the extended periods of time the head pressure must be fixed in Madison, WI for the fixed head pressure system. The floating head pressure system does not save on demand charges.

The ambient subcooling system showed AIC's ranging from $6700 to $23,400, The lower range is for Miami, and the higher AIC's are for Madison. Since an ambient subcooling system just entails the addition of a small heat exchanger downstream of the condenser, it is probable that the ambient subcooling system will exhibit payback periods of less than three years.

The dedicated subcooling system showed AIC's ranging from $12,500 to $27,500 compared to the fixed head pressure system. When the dedicated subcooling system is subject to fixed head pressure, the AIC's ranged from $7600 to $18,800 against the standard fixed head pressure system. An initial estimate of the cost of the dedicated subcooling incremental cost is $13,000. Therefore, the choice of a dedicated mechanical subcooling cycle over a fixed head pressure system will most likely show paybacks of under three years. The choice of cities was found to be very important when comparing dedicated subcooling systems employing fixed head pressure to floating head pressure systems. The dedicated subcooling system economics could be improved by just running the systems on the low temperature refrigeration systems; the medium temperature applications don't offer the potential savings of the low temperature applications.

REFERENCES 8

1. Klein, S.A., et al., TRNSYS: *A Transient Simulation Program*, University of Wisconsin - Madison, Engineering Experiment Station Report 38-12, Version 13.1

2. Mitchell, J.W., *Energy Engineering*, John Wiley and Sons, New York, 1983

CHAPTER NINE

CONCLUSIONS & RECOMMENDATIONS

This chapter presents the results of this thesis and offers recommendations for future research. The results presented are for the four refrigeration systems studied and were based on annual simulations in Madison,WI and Miami, FL.

9.1 CONCLUSIONS

FIXED HEAD PRESSURE SYSTEMS

Fixed head pressure refrigeration represents a common means of producing commercial refrigeration and was a basis against which all other systems were compared.

FLOATING HEAD PRESSURE

In warm climates such as Miami, the floating head pressure system achieved only minimal savings over the fixed head pressure system. However in cooler climates such as Madison, the floating head pressure system significantly outperformed the fixed head pressure system. Allowable incremental costs for the floating head pressure over the fixed head pressure system in Madison ranged from $9900 to $15,600. Floating head pressure systems are strong candidates for cool climate applications.

AMBIENT SUBCOOLING

The 15 degrees of subcooling provided by the ambient subcooler accounted for annual savings for the ambient subcooling system ranging from $2100 to $4800 over the floating head pressure system. When compared to the fixed head pressure system, the subcooling and the effects of the floating head pressure combined to give annual savings ranging from $3200 to $13,600. The magnitude of the annual savings indicate that the addition of a subcooling heat exchanger would most likely be a wise economic choice. With the constraint of fixed head pressure on the ambient subcooling system, the amount of subcooling increased dramatically as the ambient temperature fell. The ambient subcooling system was found to be extremely effective if installed on a fixed head pressure system designed for low ambient temperature applications.

DEDICATED MECHANICAL SUBCOOLING

Dedicated mechanical subcooling systems utilize a second refrigeration cycle to provide subcooling to the main cycle. The amount of subcooling was found to be approximately 70 degrees to the main cycle at an ambient temperature of 80°F and an evaporator temperature of -20°F. The amount of subcooling is greater than that for the ambient subcooling system due to the lower temperature sink provided by the subcooling evaporator. The COP of the subcooling cycle was found to be higher than the main cycle, and the components of the subcooling cycle were found to be a fraction of the size of the main cycle components. With the large amounts of subcooling provided to the main cycle at a relatively high COP, the performance of the entire system was increased. Dedicated subcooling systems were found to be extremely effective at high ambient temperatures and low evaporator temperatures.

The optimal temperature at which the subcooling cycle evaporated was found to be near 30°F regardless of the ambient temperature, the evaporator temperature, or the thermal sizes of the heat exchangers. The distribution of the heat exchangers was optimized based on the total system performance and the following trends were revealed.

- For optimal COP, the ratio of the condenser UA's should be in the same ratio as the refrigerant flow rates regardless of the ambient or evaporator temperatures.

- The optimum value of the subcooler was found to be approximately 10% of the total allocated UA product. The optimal value of the subcooler corresponded to a subcooler effectiveness near 1.0.

The dedicated mechanical subcooling system with floating head pressure outperformed all other systems studied. The allowable incremental cost of the dedicated subcooling system ranged from $27,500 in Madison to $12,500 in Miami when compared to the fixed head pressure system. When the constraint of fixed head pressure is added to the dedicated subcooling system, the dedicated system showed allowable incremental costs ranging from $7600 to $18,800. With these results, strong thought should be given to dedicated subcooling in the following areas:

Dedicated subcooling with floating head pressure:

- Versus any system in either climate. Especially effective for high ambient temperatures and low evaporator temperature.

Dedicated subcooling with fixed head pressure.

- Versus any type of fixed head pressure system - an application might be to retrofit the subcooling cycle to an existing fixed head pressure system.

- Versus any floating head pressure system in a warm climate. - In fact, the dedicated system with fixed head pressure outperformed the standard floating head pressure system in Miami.

9.2 RECOMMENDATIONS FOR FUTURE WORK

By compiling this study of refrigeration options for supermarkets, some areas deserving of future work need to be mentioned.

- Investigation of alternate refrigerants for the dedicated mechanical subcooling cycle. The possibility also exists of using ammonia as a refrigerant for this cycle.

- Integration of the refrigeration system with an air-conditioning system for the supermarket. In this way, the effect of lowered store humidity on both the refrigeration and air-conditioning systems could be evaluated.

- Use of some of the capacity of the medium temperature systems for subcooling of the low-temperature systems. The air-conditioning system could also be used to provide a lower temperature sink than the ambient conditions for condensing purposes.

- Use of a second cycle solely for the purpose of providing a low temperature sink for the condenser.

- Use of a second cycle to provide subcooling to the main cycle <u>and</u> provide a lower temperature sink for the condenser.

- Use of an ice-storage system for subcooling of the refrigeration systems.

Although numerous possible future studies exist using the ideas generated from dedicated mechanical subcooling, the greatest opportunity may be in the field of thermal storage. The optimal subcooling evaporator temperature for the dedicated subcooling was found to be approximately 30°F regardless of the operating conditions. This temperature corresponds to outlet temperatures common to ice-storage systems. Therefore, using an ice-storage system designed for subcooling purposes could become the next logical step. Currently, ice-storage systems are used for some commercial air-conditioning applications. These ice-storage systems are on the verge of becoming a valid economic alternative. If the refrigeration and air-conditioning concepts could be linked together, the potential savings could make ice storage an effective economic option.

The possibility of using some of the capacity of an ice-storage system designed for air-conditioning to subcool the refrigeration systems is currently being explored. Although the ice storage system for subcooling would not be expected to reduce electrical energy consumption beyond that of the dedicated subcooling system, it would allow energy use to be shifted to off-peak hours thereby saving on demand costs. It is these demand savings that usually make ice-storage systems competitive with standard systems.

Using a simplistic chiller model, initial results for the ice subcooling system are favorable. The COP of the ice subcooling system was found to lie between the dedicated and ambient subcooling systems over the range of operating temperatures common to commercial refrigeration. Using the calculated performance, initial annual savings estimates of $4500 to $12,500 for the two cities studied were found. These savings are without the associated savings due to electrical demand. With the magnitude of savings for just the subcooling being so high, the likelihood of an ice-storage system designed for air-conditioning and subcooling purposes becoming economically attractive is good. Although the initial results are favorable, work must be completed in the ice-storage and air-conditioning modelling phases before any conclusions may be drawn.

APPENDIX A

TRNSYS PROGRAMS

TRNSYS DECK

```
* fixed deck
* 16-april 1991
SIMULATION 0 8760 1
TOLERANCE 0.0001 0.0001
LIMITS 50 5 45
WIDTH 132
***********************************************************************
UNIT 1 TYPE 22 specific type of refrigerated cases
PARAMETERS 4
*lenlow lenmed lenms ontime
 300 310 200 1.0
INPUTS 2
* Tamb RHstore
4,5 0,0
 70 55
***********************************************************************
UNIT 4 TYPE 9 data reader
PARAMETERS 22
* #values time interval logical unit format
 6 1 1 1 0 2 1 0 3 1 0 4 0.18 32 5 0.18 32 6 0.0001 0 20 -1
***********************************************************************
*UNIT 5 TYPE 25 printer of hourly values
*PARAMETERS 4
* deltat ton toff Lunit
*1 0 8760 -6
*INPUTS 7
** COPlow To plr Qcond RHstore Wo Qsstore
* 4,5 1,25 1,26 1,27 1,13 1,18 1,23
* To COPlow COPmed COPms caseloadlow caseloadmed caseloadms
***********************************************************************
UNIT 6 TYPE 28 printer of integrated values
* dt ton toff lunit
PARAMETERS 21
 -1 0 8760 8 2 0 -4 0 -4 0 -4 0 -4 0 -4 0 -4 0 -4 0 -4
INPUTS 8
1,1 1,24 1,3 1,4 1,5 1,6 1,7 1,8
LABELS 8
Etot CCsenstot CClattot powastot miscloadtot Ecases defload caseloadtot
***********************************************************************
UNIT 7 TYPE 28 printer of integrated value
* dt ton toff lunit
PARAMETERS 23
 -1 0 8760 9 2 0 -4 0 -4 0 -4 0 -4 0 -4 0 -4 0 -4 0 -4 0 -4
INPUTS 9
1,9 1,10 1,13 1,14 1,15 1,18 1,19 1,20 1,23
```

```
LABELS 9
Etotlow Ecaslow caselow Etotmed Ecasmed casemed Etotms Ecasms casems
************************************************************************
end
```

REFRIGERATED CASE MODEL

```
    subroutine type22(time,xin,out,t,dtdt,par,info)

c***********************************************************************
**********
c
c   *** Fixed Head Pressure System **
c
c   This subroutine calculates the system performance based on
c   a 100 psi minimum set point temperature.
c
c   This subroutine simulates refrigerator and freezer cases.
c
c   The data is taken from a FAX to EPRI (Mukesh Khattar) dated
c   1 November 1989, written by Prof. Mitchell directly following a
c   conference on the 30th of October 1989 in Palo Alto, and from
c   subsequent phone calls.
c
c   This unit breaks cases into three groups:
c      1) reach in, low-temp cases
c      2) multi-shelf, medium-temp cases
c      3) single-shelf, medium-temp cases
c
c   English units are used.
c
c   (CJL--December 12, 1989)
c
c   TYPE22NEWERMAD.FOR  100% anti-sweat power to cases, 20 W/ft variable a-s
c   power to low temp cases, 5.0 W/ft variable a-s power to med temp cases.
c   (case model #3, taking store's varrying RH into account)
c
c***********************************************************************
**********

    implicit none        !don't allow implicit definitions
```

```
c     define the usual TRNSYS variables:
      real   time,T,Dtdt
      real   xin(2),out(27),par(4)
      integer info(10)

c     define parameter variables:
      real  lenlow      !length of low-temp, reach in cases
      real  lenmed      !length of medium-temp, single-shelf cases
      real  lenms       !length of medium-temp, multi-shelf cases
      real  ontime      !fraction of time that cases are running

c     define input variables:

      real  To          !outdoor dry-bulb temp.
      real  RH          !relative humidity of the store


c     define ouput and internal variables              Suffix used

c     first variable:  low temperature, reach-in cases       (low, or nothing)
c     second variable: medium temperature, single-shelf cases  (med)
c     third variable:  medium temperature, multi-shelf cases   (ms)

      real  powas        !power consumption of anti-sweat heaters
      real  powasmed
      real  powasms
      real  powas55        !power consumption of anti-sweat heaters at 55% RH
      real  powasmed55
      real  powasms55
      real  miscload      !power consumption of misc. (fans & lights)
      real  miscloadmed
      real  miscloadms
      real  defload       !power consumption of defrost
      real  defload55      !power consumption of defrost at 55% RH
      real  caseload      !refrigeration load of cases
      real  caseloadmed
      real  caseloadms
      real  Ecaseslow      !power consumption of cases (refrigeration)
      real  Ecasesmed
      real  Ecasesms
      real  Etotlow       !total power consumption
      real  Etotmed
      real  Etotms
      real  CAPlow55       !total cooling capacity at 55% RH
      real  CAPmed55
      real  CAPms55
      real  CAPlatlow      !latent cooling capacity
      real  CAPlatmed
      real  CAPlatms
```

```
      real  CClat          !latent case credits
      real  CClatmed
      real  CClatms
      real  CClat55        !latent case credits at 55% RH
      real  CClatmed55
      real  CClatms55
      real  CCsens         !sensible case credits
      real  CCsensmed
      real  CCsensms
      real  COPlow         !actual COP (varies with condensing temp)
      real  COPmed
      real  COPms
      real  perlow         !fraction of anti-sweat power that goes to cases
      real  permed
      real  perms

c     common variables to all cases

      real  Te             !evaporator temp
      real  latcorrlow     !corrections factor for latent loads as fcn of RH
      real  latcorrmed     !corrections factore for latent loads for med T
      real  Etot           !total electrical energy used for cases
      real  Ecases         !total electrical energy used to cool cases
      real  CCsenstot      !total sensible case credit
      real  CClattot       !total latent case credit
      real  miscloadtot    !total miscellaneous load
      real  powastot       !total anti-sweat heater load
      real  caseloadtot    !total case load (refrigeration energy)


c     get parameters

       lenlow = par(1)
       lenmed = par(2)
       lenms = par(3)
       ontime = par(4)


c     if this is the first call of the simulation, check number of inputs and
c          parameters, then set up TRNSYS
      if (info(7) .eq. -1) then
       call typeck(1,info,2,4,0)
       info(6) = 27              !twenty seven outputs
       info(9) = 0               !outputs depend only on inputs
      endif

c     get inputs

      To = xin(1)
```

```
      RH = xin(2)

c     latent credit correction factor
c
c     Foster Miller Correlation
      latcorrlow = -0.1 + 0.02 * RH
      if (latcorrlow .lt. 0.0) latcorrlow = 0.0
      latcorrmed = latcorrlow


c     assign where anti-sweat power goes
      perlow = 1.0
      permed = 1.0
      perms = 1.0

c     **** low temp, reach in cases  **********************************

      if (lenlow .gt. 0.0) then

c     anti-sweat heaters
      powas = 40.0 * lenlow * 3.413
      if ((RH .gt. 40.0) .and. (RH .lt. 55.0)) then
        powas = (40.0 + 1.333 * (RH - 40.0)) * lenlow * 3.413
      endif
      if (RH .ge. 55.0)  powas = 60.0 * lenlow * 3.413
      powas55 = 60.0 * lenlow * 3.413

c     miscellaneous case loads (20 W/ft fans, 15 W/ft lights)
      miscload = 119.455 * lenlow

c     total capacity, for 55% RH
      caplow55 = 600.0 * lenlow

c     find latent capacity, credits (12% for low temp cases)
      caplatlow = 0.12 * caplow55
      CClat55 = caplatlow * ontime
c     correct latent capacity, credit for the relative humidity setpoint
      caplatlow = caplatlow * latcorrlow
      CClat = CClat55 * latcorrlow

c     defrost load (0.24 kWhr/ft-day, Hussman, varies with latent credits)
      defload55 = 34.0 * lenlow * ontime
c     correct according to latent credits (same as correcting for RH)
      defload = defload55 * latcorrlow

c     find sensible credits at the 55% RH level, same as for other RH levels
      CCsens = caplow55 * ontime - miscload - powas55 * perlow
     +        - defload55 - CClat55
```

```
c     correct total case load according to relative humidity setpoint
      caseload = miscload + powas * perlow + defload + CClat + CCsens

c     This correlation between ambient temperature, evaporator temp,
c     and COP was developed from an EES computer program. See master's
c     thesis by Thornton for more info.

c     COP calculation:Low Temperature

      Te = -20.0

      If (Te.eq.-40) then
        COPlow=5.8765-0.15759*To+2.7323*To*To/1000-2.853*To*To*To
     + /(10**5)+1.5746*(To**4)/(10**7)-3.5897*(To**5)/(10**5)/(10**5)
       COPlow=Min(2.074,COPlow)
      endif

      If (Te.eq.-30) then
        COPlow=6.6489-0.16329*To+2.5244*To*To/1000-2.3846*To*To*To
     + /(10**5)+1.2086*(To**4)/(10**7)-2.5641*(To**5)/(10**5)/(10**5)
       COPlow=Min(2.377,COPlow)
      endif

      If (Te.eq.-20) then
        COPlow=5.4231-6.5048*To/100+2.7706*To*To/10000+1.2121*To*To*To
     + /(10**7)-3.7296*(To**4)/(10**9)-9.3402*(To**5)/(10**5)/(10**5)
     + /(10**5)/(10**5)
       COPlow=Min(2.743,COPlow)
      endif


      If (Te.eq.-10) then
        COPlow=10.428-0.26955*To+3.9607*To*To/1000-3.3874*To*To*To
     + /(10**5)+1.5329*(To**4)/(10**7)-2.8718*(To**5)/(10**5)/(10**5)
       COPlow=Min(3.191,COPlow)
      endif

      If (Te.eq.0) then
        COPlow=12.317-0.30713*To+4.283*To*To/1000-3.515*To*To*To
     + /(10**5)+1.5515*(To**4)/(10**7)-2.8718*(To**5)/(10**5)/(10**5)
       COPlow=Min(3.753,COPlow)
      endif

      If (Te.eq.10) then
        COPlow=17.564-0.49303*To+7.369*To*To/1000-6.2779*To*To*To
     + /(10**5)+2.8401*(To**4)/(10**7)-5.3333*(To**5)/(10**5)/(10**5)
       COPlow=Min(4.476,COPlow)
      endif
```

```
      If (Te.eq.20) then
        COPlow=34.239-1.2204*To+20.927*To*To/1000-19.233*To*To*To
     + /(10**5)+9.0909*(To**4)/(10**7)-17.436*(To**5)/(10**5)/(10**5)
       COPlow=Min(5.441,COPlow)
      endif

c     energy consumption
      Ecaseslow = caseload/COPlow
      Etotlow = Ecaseslow + miscload + powas + defload

      endif


c     ****  medium temperature, single shelf cases  ************************

      if (lenmed .gt. 0.0) then

c     anti-sweat heaters (5 W/ft at 40% RH, 10 W/ft at 55% RH, linear between)
      powasmed = 5.0 * lenmed * 3.413
       if ((RH .gt. 40.0) .and. (RH .lt. 55.0)) then
         powasmed = (5.0 + 0.33333 * (RH - 40.0)) * lenmed * 3.413
      endif
      if (RH .ge. 55.0) powasmed = 10.0 * lenmed * 3.413
      powasmed55 = 10.0 * lenmed * 3.413

c     miscellaneous case loads (20 W/ft fans, 15 W/ft lights)
      miscloadmed = 119.455 * lenmed

c     total capacity, for 55% RH
      capmed55 = 600.0 * lenmed

c     find latent capacity, credits (19% for medium temp cases)
      caplatmed = 0.19 * capmed55
      CClatmed55 = caplatmed * ontime
c     correct latent capacity,credit for the relative humidity setpoint
      caplatmed = caplatmed * latcorrmed
      CClatmed = CClatmed55 * latcorrmed

c     no  defrost load

c     find sensible credits at the 55% RH level, same as for other RH levels
      CCsensmed = capmed55 * ontime - miscloadmed -
     +          powasmed55 * permed - CClatmed55

c     correct total case load according to the relative humidity setpoint
      caseloadmed = miscloadmed + powasmed*permed + CClatmed + CCsensmed

c     COP calculation:
      Te = 20.0
```

```
      If (Te.eq.-40) then
        COPmed=5.8765-0.15759*To+2.7323*To*To/1000-2.853*To*To*To
     + /(10**5)+1.5746*(To**4)/(10**7)-3.5897*(To**5)/(10**5)/(10**5)
       COPmed=Min(2.074,COPmed)
      endif

      If (Te.eq.-30) then
        COPmed=6.6489-0.16329*To+2.5244*To*To/1000-2.3846*To*To*To
     + /(10**5)+1.2086*(To**4)/(10**7)-2.5641*(To**5)/(10**5)/(10**5)
       COPmed=Min(2.377,COPmed)
      endif

      If (Te.eq.-20) then
        COPmed=5.4231-6.5048*To/100+2.7706*To*To/10000+1.2121*To*To*To
     + /(10**7)-3.7296*(To**4)/(10**9)-9.3402*(To**5)/(10**5)/(10**5)
     + /(10**5)/(10**5)
       COPmed=Min(2.743,COPmed)
      endif


      If (Te.eq.-10) then
        COPmed=10.428-0.26955*To+3.9607*To*To/1000-3.3874*To*To*To
     + /(10**5)+1.5329*(To**4)/(10**7)-2.8718*(To**5)/(10**5)/(10**5)
       COPmed=Min(3.191,COPmed)
      endif

      If (Te.eq.0) then
        COPmed=12.317-0.30713*To+4.283*To*To/1000-3.515*To*To*To
     + /(10**5)+1.5515*(To**4)/(10**7)-2.8718*(To**5)/(10**5)/(10**5)
       COPmed=Min(3.753,COPmed)
      endif

      If (Te.eq.10) then
        COPmed=17.564-0.49303*To+7.369*To*To/1000-6.2779*To*To*To
     + /(10**5)+2.8401*(To**4)/(10**7)-5.3333*(To**5)/(10**5)/(10**5)
       COPmed=Min(4.476,COPmed)
      endif

      If (Te.eq.20) then
        COPmed=34.239-1.2204*To+20.927*To*To/1000-19.233*To*To*To
     + /(10**5)+9.0909*(To**4)/(10**7)-17.436*(To**5)/(10**5)/(10**5)
       COPmed=Min(5.441,COPmed)
      endif

c     energy consumption
      Ecasesmed = caseloadmed/COPmed
      Etotmed = Ecasesmed + miscloadmed + powasmed
```

```
      endif

c     **** medium temperature, multi-shelf cases ************************

      if (lenms .gt. 0.0) then

c     anti-sweat heaters (5 W/ft at 40% RH, 10 W/ft at 55% RH, linear between)
      powasms = 5.0 * lenms * 3.413
      if ((RH .gt. 40.0) .and. (RH .lt. 55.0)) then
        powasms = (5.0 + 0.33333 * (RH - 40.0)) * lenms * 3.413
      endif
      if (RH .ge. 55.0) powasms = 10.0 * lenms * 3.413
      powasms55 = 10.0 * lenms * 3.413

c     miscellaneous case loads (20 W/ft fans, 15 W/ft lights)
      miscloadms = 119.455 * lenms

c     total capacity, for 55% RH
      capms55 = 1500 * lenms

c     find latent capacity, credits (19% for medium cases)
      caplatms = 0.19 * capms55
      CClatms55 = caplatms * ontime
c     correct latent capacity, credit for the relative humidity setpoint
      caplatms = caplatms * latcorrmed
      CClatms = CClatms55 * latcorrmed

c     no defrost load

c     find sensible credits at the 55% RH level, same as for other RH levels
      CCsensms = capms55 * ontime - miscloadms - powasms55 * perms
     +          - CClatms55

c     correct total case load according to relative humidity setpoint
      caseloadms = miscloadms + powasms * perms + CClatms + CCsensms

c     COP calculation:

      Te = 20.0

      If (Te.eq.-40) then
        COPms=5.8765-0.15759*To+2.7323*To*To/1000-2.853*To*To*To
     + /(10**5)+1.5746*(To**4)/(10**7)-3.5897*(To**5)/(10**5)/(10**5)
        COPms=Min(2.074,COPms)
      endif

      If (Te.eq.-30) then
        COPms=6.6489-0.16329*To+2.5244*To*To/1000-2.3846*To*To*To
```

```
     + /(10**5)+1.2086*(To**4)/(10**7)-2.5641*(To**5)/(10**5)/(10**5)
        COPms=Min(2.377,COPms)
       endif

       If (Te.eq.-20) then
          COPms=5.4231-6.5048*To/100+2.7706*To*To/10000+1.2121*To*To*To
     + /(10**7)-3.7296*(To**4)/(10**9)-9.3402*(To**5)/(10**5)/(10**5)
     + /(10**5)/(10**5)
        COPms=Min(2.743,COPms)
       endif


       If (Te.eq.-10) then
          COPms=10.428-0.26955*To+3.9607*To*To/1000-3.3874*To*To*To
     + /(10**5)+1.5329*(To**4)/(10**7)-2.8718*(To**5)/(10**5)/(10**5)
        COPms=Min(3.191,COPms)
       endif

       If (Te.eq.0) then
          COPms=12.317-0.30713*To+4.283*To*To/1000-3.515*To*To*To
     + /(10**5)+1.5515*(To**4)/(10**7)-2.8718*(To**5)/(10**5)/(10**5)
        COPms=Min(3.753,COPms)
       endif

       If (Te.eq.10) then
          COPms=17.564-0.49303*To+7.369*To*To/1000-6.2779*To*To*To
     + /(10**5)+2.8401*(To**4)/(10**7)-5.3333*(To**5)/(10**5)/(10**5)
        COPms=Min(4.476,COPms)
       endif

       If (Te.eq.20) then
          COPms=34.239-1.2204*To+20.927*To*To/1000-19.233*To*To*To
     + /(10**5)+9.0909*(To**4)/(10**7)-17.436*(To**5)/(10**5)/(10**5)
        COPms=Min(5.441,COPms)
       endif

c      energy consumption (miscloadms, powasms already in power,Btu/hr, terms)
       Ecasesms = caseloadms/COPms
       Etotms = Ecasesms + miscloadms + powasms

       endif

c      **** find exit state ************************************************

       CCsenstot = CCsens + CCsensmed + CCsensms
       CClattot = CClat + CClatmed + CClatms


c      **** sum case terms, output table  ************************************
```

```
c    add up total electrical energy consumption
     Etot = Etotlow + Etotmed + Etotms
     Ecases = Ecaseslow + Ecasesmed + Ecasesms
     miscloadtot = miscload + miscloadmed + miscloadms
     powastot = powas + powasmed + powasms
     caseloadtot = caseload + caseloadmed + caseloadms

c    print table of values to error file first time unit is called

     if (info(7) .eq. -1) then
       write(7,*)
       write(7,*)'         Summary of Case Loads and Credits'
       write(7,*)'         for a fixed head pressure system'
       write(7,*)' '
       write(7,*)'                low          Med          med,ms
    +           total'
       write(7,*)
       write(7,*)'case load   : ',caseload, caseloadmed, caseloadms,
    +          caseloadtot
       write(7,*)'misc load   : ',miscload, miscloadmed, miscloadms,
    +          miscloadtot
       write(7,*)'anti-sweat  : ',powas, powasmed, powasms,powastot
       write(7,*)'defrost     : ',defload
       write(7,*)'CClatent    : ',CClat, CClatmed, CClatms,CClattot
       write(7,*)'CCsensible  : ',CCsens, CCsensmed, CCsensms,CCsenstot
       write(7,*)
       write(7,*)'COP         : ',COPlow, COPmed, COPms
       write(7,*)'Total Energy: ',Etotlow, Etotmed, Etotms, Etot
     endif

c    **** output ****************************************************


c    assign ouput array
c      note that CCsenstot output is not really the case credits--2 * the anti-
c       sweat power that goes to the store is subtracted off so that the right
c      amount of credit is given to the store when it is divided by 2

     out(1) = Etot
     out(2) = CCsenstot - 2* ((1.0-perlow) * powas - (1.0-permed)
    +       * powasmed - (1.0 - perms) * powasms)
     out(3) = CClattot
     out(4) = powastot
     out(5) = miscloadtot
     out(6) = Ecases
     out(7) = defload
     out(8) = caseloadtot
```

```
c    for low temperature cases (all defrost load is here also)
     out(9) = Etotlow
     out(10) = Ecaseslow
     out(11) = miscload
     out(12) = powas
     out(13) = caseload

c    for medium temperature cases, single-shelf
     out(14) = Etotmed
     out(15) = Ecasesmed
     out(16) = miscloadmed
     out(17) = powasmed
     out(18) = caseloadmed

c    for medium temperature cases, multi-shelf
     out(19) = Etotms
     out(20) = Ecasesms
     out(21) = miscloadms
     out(22) = powasms
     out(23) = caseloadms

c    real sensible case credits
     out(24) = CCsenstot

     out(25)=COPlow
     out(26)=COPmed
     out(27)=COPms

     return
     end
```

APPENDIX B

FIXED HEAD PRESSURE

STEADY-STATE MODEL

```
{This is the latest model describing the relations between a
fixed head pressure refrigeration system, refrigerated case and the entering store
conditions. The date is 1-29-91}


{parameters}

caseloadlow=180000
Tamb=80
dsh=7.0

{evaporator}

Tevap=-20

T5=Tevap+dsh
P5=Pressure(R12,T=Tevap,x=0)
h5=Enthalpy(R12,T=T5,P=P5)
s5=Entropy(R12,T=T5,P=P5)

P4=P5

mref=caseloadlow/(h5-h4)

{compressor}

s1id=s5
h1id=Enthalpy(R12,P=P1,s=s1id)

Wcompid=mref*(h1id-h5)
Iseff=0.8
Wcomp=Wcompid/Iseff

h1=h5+Wcomp/mref
T1=Temperature(R12,P=P1,h=h1)

{condenser}

UAcond=15000
UAcond=(maircond*CPaircond)*NTUcond

maircond=3800*caseloadlow/12000
CPaircond=SpecHeat(Air,T=Tamb)

Effcond=1-exp(-NTUcond)
```

```
DeltaTLM=(Taircondout-Tamb)/ln((T2a-Tamb)/(T2a-Taircondout))

Qcond=mref*(h1-h2)
Qcond=maircond*CPAircond*(Taircondout-Tamb)
Qcond=UAcond*DeltaTLM

x2=0.0
P2a=Pressure(R12,T=T2a,x=x2)
h2=Enthalpy(R12,T=T2,x=x2)
P2=P1

Pset=100
Tset=Temperature(R12,P=Pset,x=0)
T2=if(T2a,Tset,Tset,Tset,T2a)
P2=if(P2a,Pset,Pset,Pset,P2a)

P3=P2
h3=h2
T3=T2

{Expansion Valve}

h4=h3

{extras}

COP=caseloadlow/Wcomp
```

SOLUTION AT TAMB = 80°F, TEVAP = -20°F

caseloadlow = 180000.000
COP = 1.902
CPAircond = 0.240

DeltaTLM = 18.311
dsh = 7.000

Effcond = 0.666

h1 = 98.431
h1id = 93.960
h2 = 33.565
h3 = 33.565

h4 = 33.565
h5 = 76.075

Iseff = 0.800

maircond = 57000.000
mref = 4234.261

NTUcond = 1.097

P1 = 151.390
P2 = 151.390
P2a = 151.390
P3 = 151.390
P4 = 15.263
P5 = 15.263
Pset = 100.000

Qcond = 274658.934

s1id = 0.173
s5 = 0.173

T1 = 167.665
T2 = 110.149
T2a = 110.149
T3 = 110.149
T5 = -13.000
Taircondout = 100.078
Tamb = 80.000
Tevap = -20.000
Tset = 80.771

UAcond = 15000.000

Wcomp = 94658.934
Wcompid = 75727.147

x2 = 0.000

SOLUTION AT TAMB = 40°F, TEVAP = 20°F

caseloadlow = 180000.000

COP = 5.441
CPAircond = 0.239

DeltaTLM = 14.206
dsh = 7.000

Effcond = 0.667

h1 = 90.344
h1id = 88.363
h2 = 26.542
h3 = 26.542
h4 = 26.542
h5 = 80.438

Iseff = 0.800

maircond = 57000.000
mref = 3339.794

NTUcond = 1.101

P1 = 100.000
P2 = 100.000
P2a = 76.526
P3 = 100.000
P4 = 35.729
P5 = 35.729
Pset = 100.000

Qcond = 213084.153

s1id = 0.169
s5 = 0.169

T1 = 109.626
T2 = 80.771
T2a = 63.429
T3 = 80.771
T5 = 27.000
Taircondout = 55.635
Tamb = 40.000
Tevap = 20.000
Tset = 80.771

UAcond = 15000.000

Wcomp = 33084.153
Wcompid = 26467.322

x2 = 0.000

PERFORMANCE CURVES

NOTE: 'x' represents the ambient temperature

Tevap = -20:

COP = 5.4231 - 6.5048e-2x + 2.7706e-4x^2 + 1.2121e-7x^3 - 3.7296e-9x^4 - 9.3402e-20x^5 R^2 = 1.000

Tevap = 20

COP = 34.239 - 1.2204x + 2.0927e-2x^2 - 1.9233e-4x^3 + 9.0909e-7x^4 - 1.7436e-9x^5 R^2 = 1.000

The COP is represented by the equations above, or the set point COP, whichever is the minimum.

For Tevap = -20, COPset = 2.743

For Tevap = 20, COPset = 5.441

APPENDIX C

FLOATING HEAD PRESSURE

STEADY-STATE MODEL

```
{This is the latest model describing the relations between a
floating head pressure rerigeration system, refrigerated cases
and the entering store conditions. The date is 1-29-91}


{parameters}

caseloadlow=180000
Tamb=80
dsh=7.0

{evaporator}

Tevap=-20

T5=Tevap+dsh
P5=Pressure(R12,T=Tevap,x=0)
h5=Enthalpy(R12,T=T5,P=P5)
s5=Entropy(R12,T=T5,P=P5)

P4=P5

mref=caseloadlow/(h5-h4)

{compressor}

s1id=s5
h1id=Enthalpy(R12,P=P1,s=s1id)

Wcompid=mref*(h1id-h5)
Iseff=0.8
Wcomp=Wcompid/Iseff

h1=h5+Wcomp/mref
T1=Temperature(R12,P=P1,h=h1)

{condenser}

UAcond=15000
UAcond=(maircond*CPaircond)*NTUcond

maircond=3800*caseloadlow/12000
CPaircond=SpecHeat(Air,T=Tamb)

Effcond=1-exp(-NTUcond)
```

```
DeltaTLM=(Taircondout-Tamb)/ln((T2-Tamb)/(T2-Taircondout))

Qcond=mref*(h1-h2)
Qcond=maircond*CPAircond*(Taircondout-Tamb)
Qcond=UAcond*DeltaTLM

x2=0.0
P2=Pressure(R12,T=T2,x=x2)
h2=Enthalpy(R12,T=T2,x=x2)
P2=P1

P3=P2
h3=h2
T3=T2

{Expansion Valve}

h4=h3

{extras}

COP=caseloadlow/Wcomp
```

SOLUTION AT TAMB = 80°F, TEVAP = -20°F

caseloadlow = 180000.000
COP = 1.902
CPAircond = 0.240

DeltaTLM = 18.311
dsh = 7.000

Effcond = 0.666

h1 = 98.431
h1id = 93.960
h2 = 33.565
h3 = 33.565
h4 = 33.565
h5 = 76.075

Iseff = 0.800

maircond = 57000.000
mref = 4234.261

NTUcond = 1.097

P1 = 151.390
P2 = 151.390
P3 = 151.390
P4 = 15.263
P5 = 15.263

Qcond = 274658.934

s1id = 0.173
s5 = 0.173

T1 = 167.665
T2 = 110.149
T3 = 110.149
T5 = -13.000
Taircondout = 100.078
Tamb = 80.000
Tevap = -20.000

UAcond = 15000.000

Wcomp = 94658.934
Wcompid = 75727.147

x2 = 0.000

SOLUTION AT TAMB = 40°F, TEVAP = 20°F

caseloadlow = 180000.000
COP = 8.223
CPAircond = 0.239

DeltaTLM = 13.459
dsh = 7.000

Effcond = 0.667

h1 = 87.513
h1id = 86.098

h2 = 22.263
h3 = 22.263
h4 = 22.263
h5 = 80.438

Iseff = 0.800

maircond = 57000.000
mref = 3094.136

NTUcond = 1.101

P1 = 75.033
P2 = 75.033
P3 = 75.033
P4 = 35.729
P5 = 35.729

Qcond = 201890.644

s1id = 0.169

s5 = 0.169
T1 = 85.699
T2 = 62.198
T3 = 62.198
T5 = 27.000
Taircondout = 54.813
Tamb = 40.000
Tevap = 20.000

UAcond = 15000.000

Wcomp = 21890.644
Wcompid = 17512.515

x2 = 0.000

PERFORMANCE CURVES

NOTE: 'x' represents the ambient temperature.

For Tevap = -20

$$COP = 7.5561 - 0.23099x + 6.2141e\text{-}3x^2 - 1.0526e\text{-}4x^3 + 8.7242e\text{-}7x^4 - 2.7089e\text{-}9x^5$$

$$R^2 = 0.999$$

For Tevap = 20

$$COP = 45.373 - 2.2331x + 5.2910e\text{-}2x^2 - 6.5551e\text{-}4x^3 + 4.0622e\text{-}6x^4 - 9.9298e\text{-}9x^5$$

$$R^2 = 1.000$$

The floating head pressure performance curves are listed above. For the floating head pressure system, there is no minimum COP corresponding to a set point.

APPENDIX D

AMBIENT SUBCOOLING SYSTEMS

STEADY-STATE MODEL

```
{This is the latest model describing the relations between  a
ambient subcooling refrigeration system, refrigerated case and the entering store
conditions. The date is 3-29-91}


{parameters}

caseloadlow=180000
Tamb=80
dsh=7.0

{evaporator}

Tevap=-20

T5=Tevap+dsh
P5=Pressure(R12,T=Tevap,x=0)
h5=Enthalpy(R12,T=T5,P=P5)
s5=Entropy(R12,T=T5,P=P5)

P4=P5

mref=caseloadlow/(h5-h4)

{compressor}

s1id=s5
h1id=Enthalpy(R12,P=P1,s=s1id)

Wcompid=mref*(h1id-h5)
Iseff=0.8
Wcomp=Wcompid/Iseff

h1=h5+Wcomp/mref
T1=Temperature(R12,P=P1,h=h1)

{condenser}

UAcond=15000
UAcond=(maircond*CPaircond)*NTUcond

maircond=3800*caseloadlow/12000
CPaircond=SpecHeat(Air,T=Tamb)

Effcond=1-exp(-NTUcond)
```

```
DeltaTLM=(Taircondout-Tamb)/ln((T2-Tamb)/(T2-Taircondout))

Qcond=mref*(h1-h2)
Qcond=maircond*CPAircond*(Taircondout-Tamb)
Qcond=UAcond*DeltaTLM

x2=0.0
P2=Pressure(R12,T=T2,x=x2)
h2=Enthalpy(R12,T=T2,x=x2)
P2=P1

UAsub=750
UAsub=(mairsub*CPairsub)*NTUsub

mairsub=3800*caseloadlow/12000
CPairsub=SpecHeat(Air,T=Tamb)

Effsub=1-exp(-NTUsub)
DeltaTLMsub=((T2-Tairsubout)-(T3-Tamb))/ln((T2-Tairsubout)/(T3-Tamb))

Qsub=mref*(h2-h3)
Qsub=mairsub*CPairsub*(Tairsubout-Tamb)
Qsub=UAsub*DeltaTLMsub

P2=P3
h3=Enthalpy(R12,T=T3,P=P3)

{Expansion Valve}

h4=h3

{extras}

COP=caseloadlow/Wcomp
DSC=T2-T3
```

SOLUTION AT TAMB = 80°F, TEVAP = -20°F

caseloadlow = 180000.000
COP = 2.128
CPAircond = 0.240
CPairsub = 0.240

DeltaTLM = 16.710
DeltaTLMsub = 18.601

DSC = 15.055
dsh = 7.000

Effcond = 0.666
Effsub = 0.053

h1 = 98.062
h1id = 93.665
h2 = 32.919
h3 = 29.294
h4 = 29.294
h5 = 76.075

Iseff = 0.800

maircond = 57000.000
mairsub = 57000.000
mref = 3847.651

NTUcond = 1.097
NTUsub = 0.055

P1 = 146.121
P2 = 146.121
P3 = 146.121
P4 = 15.263
P5 = 15.263

Qcond = 250645.337
Qsub = 13950.821

s1id = 0.173
s5 = 0.173

T1 = 164.530
T2 = 107.513
T3 = 92.458
T5 = -13.000
Taircondout = 98.323
Tairsubout = 81.020
Tamb = 80.000
Tevap = -20.000

UAcond = 15000.000
UAsub = 750.000

Wcomp = 84596.157
Wcompid = 67676.926

x2 = 0.000

SOLUTION AT TAMB = 40°F, TEVAP = 20°F

caseloadlow = 180000.000
COP = 8.954
CPAircond = 0.239
CPairsub = 0.239

DeltaTLM = 12.720
DeltaTLMsub = 12.394
DSC = 14.110
dsh = 7.000

Effcond = 0.667
Effsub = 0.054

h1 = 87.321
h1id = 85.945
h2 = 21.986
h3 = 18.803
h4 = 18.803
h5 = 80.438

Iseff = 0.800

maircond = 57000.000
mairsub = 57000.000
mref = 2920.429

NTUcond = 1.101
NTUsub = 0.055

P1 = 73.576
P2 = 73.576
P3 = 73.576
P4 = 35.729
P5 = 35.729

Qcond = 190806.322
Qsub = 9295.547

s1id = 0.169
s5 = 0.169

T1 = 84.093
T2 = 60.979
T3 = 46.869
T5 = 27.000
Taircondout = 54.000
Tairsubout = 40.682
Tamb = 40.000
Tevap = 20.000

UAcond = 15000.000
UAsub = 750.000

Wcomp = 20101.870
Wcompid = 16081.496

x2 = 0.000

PERFORMANCE CURVES

NOTE: 'x' represents the ambient temperature.

For Tevap=-20

$$y = 8.1961 - 0.25421x + 7.0240e\text{-}3x^2 - 1.2048e\text{-}4x^3 + 1.0044e\text{-}6x^4 - 3.1267e\text{-}9x^5$$
$$R^2 = 0.999$$

For Tevap=20

$$y = 51.306 - 2.5733x + 6.1623e\text{-}2x^2 - 7.6803e\text{-}4x^3 + 4.7778e\text{-}6x^4 - 1.1709e\text{-}8x^5$$
$$R^2 = 1.000$$

The ambient subcooling performance curves are listed above. For the ambient subcooling system with floating head pressure, there is no minimum COP corresponding to a set point.

APPENDIX E

DEDICATED MECHANICAL SUBCOOLING

STEADY-STATE MODEL

```
{This is the latest model describing the relations between  a
dedicated mechanical subcooling rerigeration system. The date is 3-6-91}


{parameters}

caseloadlow=180000           {The design refrigeration load - 15 tons}
Tamb=80                      {The ambient temperature}
dsh=7.0                      {The degrees of superheat leaving the evaporator}

{evaporator}

Tevap=-20                        {The evaporator set point}
T4=Tevap

T5=Tevap+dsh
P5=Pressure(R12,T=Tevap,x=0)
h5=Enthalpy(R12,T=T5,P=P5)
s5=Entropy(R12,T=T5,P=P5)

P4=P5

mref=caseloadlow/(h5-h4)

{compressor}


s1id=s5
h1id=Enthalpy(R12,P=P1,s=s1id)

Wcompid=mref*(h1id-h5)
Iseff=0.8
Wcomp=Wcompid/Iseff

h1=h5+Wcomp/mref
T1=Temperature(R12,P=P1,h=h1)

{condenser}

UAcond=15000
UAcond=(maircond*CPaircond)*NTUcond

{condenser flow rates = 3800 lbm/hr per ton of refrigeration}
maircond=3800*caseloadlow/12000
CPaircond=SpecHeat(Air,T=Tamb)
```

```
{Effectiveness of condenser}
Effcond=1-exp(-NTUcond)
DeltaTLM=(Tamb-Taircondout)/ln((T2-Taircondout)/(T2-Tamb))

Qcond=mref*(h1-h2)
Qcond=maircond*CPaircond*(Taircondout-Tamb)
Qcond=UAcond*DeltaTLM

x2=0.0
P2=Pressure(R12,T=T2,x=x2)
h2=Enthalpy(R12,T=T2,x=x2)
P2=P1

{subcooler}

P3=P2
Uasub=2000
Ps=Pressure(R12,t=80,x=0)
Cpr=SpecHeat(R12,T=80,P=Ps+100)
Cmin=Cpr*mref
Ntusub=Uasub/Cmin
Effsub=(1-exp(Ntusub))/(-exp(Ntusub))
T6=30
P6=Pressure(R12,T=T6,x=0)

DeltaTLMsub=((T2-T7)-(T3-T6))/ln((T2-T7)/(T3-T6))
Qsub=mref*(h2-h3)
Qsub=mref2*(h7-h6)
Qsub=Uasub*DeltaTLMsub
T3=Temperature(R12,P=P3,h=h3)
P7=P6
x7=1.0
T7a=Temperature(R12,x=x7,P=P7)
T7=T7a+dsh
s7=Entropy(R12,t=T7,P=P7)
h7=Enthalpy(R12,T=T7,P=P7)

{Expansion Valve}

h4=h3

{expansion valve 2}

h9=h6

{compressor 2}

s8id=s7
```

```
h8id=Enthalpy(R12,P=P8,s=s8id)

Wcomp2id=mref2*(h8id-h7)
Wcomp2=Wcomp2id/Iseff

h8=h7+Wcomp2/mref2
T8=Temperature(R12,P=P8,h=h8)

{condenser 2}

UAcond2=5000
UAcond2=(maircond2*CPaircond)*NTUcond2

maircond2=3800*Qsub/12000

Effcond2=1-exp(-NTUcond2)
DeltaTLM2=(Tamb-Taircond2out)/ln((T9-Taircond2out)/(T9-Tamb))

Qcond2=mref2*(h8-h9)
Qcond2=maircond2*CPaircond*(Taircond2out-Tamb)
Qcond2=UAcond2*DeltaTLM2

x9=0.0
P9=Pressure(R12,T=T9,x=x9)
h9=Enthalpy(R12,T=T9,x=x9)
P9=P8

{extras}

COP1=caseloadlow/Wcomp
COP2=Qsub/Wcomp2
COPtot=caseloadlow/(Wcomp+Wcomp2)

UAratio=UAcond/(UAcond+UAcond2)
mratio=mref/(mref+mref2)

DSC=T2-T3
mtot=mref+mref2
```

SOLUTION AT TAMB = 80°F, TEVAP = -20°F

caseloadlow = 180000.000 [Btu/hr]
Cmin = 699.242 [Btu/hr °F]
COP1 = 2.834
COP2 = 4.562

COPtot = 2.444
CPaircond = 0.240 [Btu/lbm °F]
Cpr = 0.233 [Btu/lbm °F]

DeltaTLM = 13.158 [°F]
DeltaTLM2 = 11.255 [°F]
DeltaTLMsub = 23.079 [°F]
DSC = 66.823 [°F]
dsh = 7.000 [°F]

Effcond = 0.666
Effcond2 = 0.760
Effsub = 0.943

h1 = 97.232 [Btu/lbm]
h1id = 93.000 [Btu/lbm]
h2 = 31.499 [Btu/lbm]
h3 = 16.126 [Btu/lbm]
h4 = 16.126 [Btu/lbm]
h5 = 76.075 [Btu/lbm]
h6 = 31.368 [Btu/lbm]
h7 = 81.497 [Btu/lbm]
h8 = 92.484 [Btu/lbm]
h8id = 90.287 [Btu/lbm]
h9 = 31.368 [Btu/lbm]

Iseff = 0.800

maircond = 57000.000 [lbm air/hr]
maircond2 = 14616.900 [lbm air/hr]
mratio = 0.765
mref = 3002.507 [lbm refr./hr]
mref2 = 920.798 [lbm refr./hr]
mtot = 3923.305 [lbm refr./hr]

NTUcond = 1.097
NTUcond2 = 1.425
Ntusub = 2.860

P1 = 134.915 [psia]
P2 = 134.915 [psia]
P3 = 134.915 [psia]
P4 = 15.263 [psia]
P5 = 15.263 [psia]
P6 = 43.140 [psia]
P7 = 43.140 [psia]
P8 = 133.907 [psia]
P9 = 133.907 [psia]
Ps = 98.855 [psia]

Qcond = 197362.898 [Btu/hr]
Qcond2 = 56276.058 [Btu/hr]
Qsub = 46158.631 [Btu/hr]

s1id = 0.173 [Btu/lbm °R]
s5 = 0.173 [Btu/lbm °R]
s7 = 0.169 [Btu/lbm °R]
s8id = 0.169 [Btu/lbm °R]

T1 = 157.526 [°F]
T2 = 101.664 [°F]
T3 = 34.841 [°F]
T4 = -20.000 [°F]
T5 = -13.000 [°F]
T6 = 30.000 [°F]
T7 = 37.000 [°F]
T7a = 30.000 [°F]
T8 = 130.526 [°F]
T9 = 101.121 [°F]
Taircond2out = 96.043 [°F]
Taircondout = 94.428 [°F]
Tamb = 80.000 [°F]
Tevap = -20.000 [°F]

UAcond = 15000.000 [Btu/hr °F]
UAcond2 = 5000.000 [Btu/hr °F]
UAratio = 0.750
Uasub = 2000.000 [Btu/hr °F]

Wcomp = 63521.529 [Btu/hr]
Wcomp2 = 10117.426 [Btu/hr]
Wcomp2id = 8093.941 [Btu/hr]
Wcompid = 50817.223 [Btu/hr]

x2 = 0.000
x7 = 1.000
x9 = 0.000

SOLUTION AT TAMB = 40°F, TEVAP = 20°F

caseloadlow = 180000.000 [Btu/hr]
Cmin = 645.616 [Btu/hr °F]
COP1 = 9.664
COP2 = 15.057

COPtot = 9.103
CPaircond = 0.239 [Btu/lbm °F]
Cpr = 0.233 [Btu/lbm °F]

DeltaTLM = 12.088 [°F]
DeltaTLM2 = 3.690 [°F]
DeltaTLMsub = 8.650 [°F]
DSC = 27.886 [°F]
dsh = 7.000 [°F]

Effcond = 0.667
Effcond2 = 0.978
Effsub = 0.955

h1 = 87.157 [Btu/lbm]
h1id = 85.813 [Btu/lbm]
h2 = 21.749 [Btu/lbm]
h3 = 15.509 [Btu/lbm]
h4 = 15.509 [Btu/lbm]
h5 = 80.438 [Btu/lbm]
h6 = 20.496 [Btu/lbm]
h7 = 81.497 [Btu/lbm]
h8 = 85.548 [Btu/lbm]
h8id = 84.738 [Btu/lbm]
h9 = 20.496 [Btu/lbm]

Iseff = 0.800

maircond = 57000.000 [lbm air/hr]
maircond2 = 5478.577 [lbm air/hr]
mratio = 0.907
mref = 2772.240 [lbm refr./hr]
mref2 = 283.617 [lbm refr./hr]
mtot = 3055.857 [lbm refr./hr]

NTUcond = 1.101
NTUcond2 = 3.817
Ntusub = 3.098

P1 = 72.347 [psia]
P2 = 72.347 [psia]
P3 = 72.347 [psia]
P4 = 35.729 [psia]
P5 = 35.729 [psia]
P6 = 43.140 [psia]
P7 = 43.140 [psia]
P8 = 66.073 [psia]
P9 = 66.073 [psia]
Ps = 98.855 [psia]

Qcond = 181324.907 [Btu/hr]
Qcond2 = 18449.766 [Btu/hr]
Qsub = 17300.768 [Btu/hr]

s1id = 0.169 [Btu/lbm °R]
s5 = 0.169 [Btu/lbm °R]
s7 = 0.169 [Btu/lbm °R]
s8id = 0.169 [Btu/lbm °R]

T1 = 82.716 [°F]
T2 = 59.937 [°F]
T3 = 32.051 [°F]
T4 = 20.000 [°F]
T5 = 27.000 [°F]
T6 = 30.000 [°F]
T7 = 37.000 [°F]
T7a = 30.000 [°F]
T8 = 70.872 [°F]
T9 = 54.401 [°F]
Taircond2out = 54.084 [°F]
Taircondout = 53.304 [°F]
Tamb = 40.000 [°F]
Tevap = 20.000 [°F]

UAcond = 15000.000 [Btu/hr °F]
UAcond2 = 5000.000 [Btu/hr °F]
UAratio = 0.750
Uasub = 2000.000 [Btu/hr °F]

Wcomp = 18625.675 [Btu/hr]
Wcomp2 = 1148.998 [Btu/hr]
Wcomp2id = 919.198 [Btu/hr]
Wcompid = 14900.540 [Btu/hr]

x2 = 0.000
x7 = 1.000
x9 = 0.000

PERFORMANCE CURVES

NOTE: 'x' represents the ambient temperature.

For Tevap=-20

$$y = 7.0038 - 0.12130x + 1.4869e\text{-}3x^2 - 1.2661e\text{-}5x^3 + 6.2228e\text{-}8x^4 - 1.3356e\text{-}10x^5$$
$$R^2 = 1.000$$

For Tevap=20

$$y = 41.130 - 1.7743x + 3.8219e\text{-}2x^2 - 4.4096e\text{-}4x^3 + 2.5856e\text{-}6x^4 - 6.0505e\text{-}9x^5$$
$$R^2 = 1.000$$

The dedicated subcooling performance curves are listed above. For the dedicated subcooling system with fixed head pressure, the COP is the minimum of the equation listed above or the COP at the set point.

The COP at the set point is:

Tevap=-20 COP=3.068
Tevap= 20 COP=5.628

UA OPTIMIZATION

Design Temperature	$UA_{cond,main}$	$UA_{cond,sub}$	$UA_{subcooler}$
30°F	18,711	1204	2084
40°F	17,951	1835	2215
50°F	17,053	2564	2383
60°F	16,395	3183	2422
70°F	15,657	3863	2479
80°F	14,967	4528	2505
90°F	14,238	5236	2527
100°F	13,496	5969	2535

Note: UA's are in BTU/hr °F

APPENDIX F

PART LOAD RATIO

STEADY-STATE MODEL

{This is the latest model describing the relations between a
dedicated mechanical subcooling rerigeration system with part load ratio effects.}

{parameters}

caseloadlow=180000
Tamb=80
dsh=7.0

{If Tevap=20, adjust Wcomp}
CAPrated1=80060
CAP1=mref*(h1-h5)
plr1=CAP1/CAPrated1
plr_f1=0.25*plr1+0.75

CAPrated2=35307
CAP2=mref2*(h8-h7)
plr2=CAP2/CAPrated2
plr_f2=0.25*plr2+0.75

{evaporator}

Tevap=-20
T4=Tevap

T5=Tevap+dsh
P5=Pressure(R12,T=Tevap,x=0)
h5=Enthalpy(R12,T=T5,P=P5)
s5=Entropy(R12,T=T5,P=P5)

P4=P5

mref=caseloadlow/(h5-h4)

{compressor}

s1id=s5
h1id=Enthalpy(R12,P=P1,s=s1id)

Wcompid=mref*(h1id-h5)
Iseff=0.8
Wcomp=Wcompid/Iseff/plr_f1

h1=h5+Wcomp/mref

```
T1=Temperature(R12,P=P1,h=h1)

{condenser}

UAcond=15000
UAcond=(maircond*CPaircond)*NTUcond

maircond=3800*caseloadlow/12000
CPaircond=SpecHeat(Air,T=Tamb)

Effcond=1-exp(-NTUcond)
DeltaTLM=(Tamb-Taircondout)/ln((T2-Taircondout)/(T2-Tamb))

Qcond=mref*(h1-h2)
Qcond=maircond*CPaircond*(Taircondout-Tamb)
Qcond=UAcond*DeltaTLM

x2=0.0
P2=Pressure(R12,T=T2,x=x2)
h2=Enthalpy(R12,T=T2,x=x2)
P2=P1

{subcooler}

P3=P2
Uasub=2000
Ps=Pressure(R12,t=80,x=0)
Cpr=SpecHeat(R12,T=80,P=Ps+100)
Cmin=Cpr*mref
Ntusub=Uasub/Cmin
Effsub=(1-exp(Ntusub))/(-exp(Ntusub))
T6=30
P6=Pressure(R12,T=T6,x=0)

DeltaTLMsub=((T2-T7)-(T3-T6))/ln((T2-T7)/(T3-T6))
Qsub=mref*(h2-h3)
Qsub=mref2*(h7-h6)
Qsub=Uasub*DeltaTLMsub
T3=Temperature(R12,P=P3,h=h3)
P7=P6
x7=1.0
T7a=Temperature(R12,x=x7,P=P7)
T7=T7a+dsh
s7=Entropy(R12,t=T7,P=P7)
h7=Enthalpy(R12,T=T7,P=P7)

{Expansion Valve}

h4=h3
```

```
{expansion valve 2}

h9=h6

{compressor 2}

s8id=s7
h8id=Enthalpy(R12,P=P8,s=s8id)

Wcomp2id=mref2*(h8id-h7)
Wcomp2=Wcomp2id/Iseff/plr_f2

h8=h7+Wcomp2/mref2
T8=Temperature(R12,P=P8,h=h8)

{condenser 2}

UAcond2=5000
UAcond2=(maircond2*CPaircond2)*NTUcond2

maircond2=3800*Qsub/12000
CPaircond2=SpecHeat(Air,T=Tamb)

Effcond2=1-exp(-NTUcond2)
DeltaTLM2=(Tamb-Taircond2out)/ln((T9-Taircond2out)/(T9-Tamb))

Qcond2=mref2*(h8-h9)
Qcond2=maircond2*CPaircond2*(Taircond2out-Tamb)
Qcond2=UAcond2*DeltaTLM2

x9=0.0
P9=Pressure(R12,T=T9,x=x9)
h9=Enthalpy(R12,T=T9,x=x9)
P9=P8

{extras}

COP1=caseloadlow/Wcomp
COP2=Qsub/Wcomp2
COPtot=caseloadlow/(Wcomp+Wcomp2)

UAratio=UAcond/(UAcond+UAcond2)
mratio=mref/(mref+mref2)

DSC=T2-T3
mtot=mref+mref2
```

SOLUTION AT TAMB = 80°F, TEVAP = -20°F

CAP1 = 66475.759
CAP2 = 12318.367
CAPrated1 = 80060.000
CAPrated2 = 35307.000
caseloadlow = 180000.000
Cmin = 699.294
COP1 = 2.708
COP2 = 3.764
COPtot = 2.284
CPaircond = 0.240
CPaircond2 = 0.240
Cpr = 0.233

DeltaTLM = 13.341
DeltaTLM2 = 11.737
DeltaTLMsub = 23.184
DSC = 67.104
dsh = 7.000

Effcond = 0.666
Effcond2 = 0.758
Effsub = 0.943

h1 = 98.214
h1id = 93.035
h2 = 31.572
h3 = 16.130
h4 = 16.130
h5 = 76.075
h6 = 31.573
h7 = 81.497
h8 = 94.760
h8id = 90.380
h9 = 31.573

Iseff = 0.800

maircond = 57000.000
maircond2 = 14682.955
mratio = 0.764
mref = 3002.732
mref2 = 928.762
mtot = 3931.494

NTUcond = 1.097
NTUcond2 = 1.419

Ntusub = 2.860

P1 = 135.476
P2 = 135.476
P3 = 135.476
P4 = 15.263
P5 = 15.263
P6 = 43.140
P7 = 43.140
P8 = 135.486
P9 = 135.486
plr1 = 0.830
plr2 = 0.349
plr_f1 = 0.958
plr_f2 = 0.837
Ps = 98.855

Qcond = 200108.532
Qcond2 = 58685.594
Qsub = 46367.227

s1id = 0.173
s5 = 0.173
s7 = 0.169
s8id = 0.169

T1 = 163.231
T2 = 101.965
T3 = 34.861
T4 = -20.000
T5 = -13.000
T6 = 30.000
T7 = 37.000
T7a = 30.000
T8 = 143.662
T9 = 101.970
Taircond2out = 96.654
Taircondout = 94.628
Tamb = 80.000
Tevap = -20.000

UAcond = 15000.000
UAcond2 = 5000.000
UAratio = 0.750
Uasub = 2000.000

Wcomp = 66475.759
Wcomp2 = 12318.367
Wcomp2id = 8250.579

Wcompid = 50924.742

x2 = 0.000
x7 = 1.000
x9 = 0.000

PERFORMANCE CURVES

NOTE: 'x' represents the ambient temperature.

FIXED HEAD PRESSURE

y = 5.7028 - 0.11992x + 1.6794e-3x^2 - 1.4997e-5x^3 + 7.3602e-8x^4 - 1.5273e-10x^5 R^2 = 1.000

y = 18.785 - 0.54110x + 8.1721e-3x^2 - 6.8955e-5x^3 + 3.0627e-7x^4 - 5.6109e-10x^5 R^2 = 1.000

FLOATING HEAD PRESSURE

y = 5.8897 - 0.17027x + 4.5664e-3x^2 - 7.7327e-5x^3 + 6.4061e-7x^4 - 1.9883e-9x^5 R^2 = 0.999

y = 34.069 - 1.6453x + 3.8944e-2x^2 - 4.8238e-4x^3 + 2.9889e-6x^4 - 7.3054e-9x^5 R^2 = 1.000

AMBIENT SUBCOOLING

y = 6.4491 - 0.18786x + 5.1715e-3x^2 - 8.8697e-5x^3 + 7.3937e-7x^4 - 2.3017e-9x^5 R^2 = 0.999

y = 38.609 - 1.9001x + 4.5464e-2x^2 - 5.6667e-4x^3 + 3.5255e-6x^4 - 8.6410e-9x^5 R^2 = 1.000

DEDICATED SUBCOOLING

y = 5.8346 - 8.7016e-2x + 9.4024e-4x^2 - 7.4720e-6x^3 + 3.5888e-8x^4 - 7.6457e-11x^5 R^2 = 1.000

$$y = 29.108 - 1.1157x + 2.2220e\text{-}2x^2 - 2.4169e\text{-}4x^3 + 1.3528e\text{-}6x^4 - 3.0483e\text{-}9x^5 \quad R^2 = 1.000$$

ANNUAL ENERGY CONSUMPTION

<u>MADISON (kWh/year)</u>

Fixed Head Pressure (plr)	505,420
Floating Head Pressure (plr)	405,508
Ambient Subcooling (plr)	365,954
Dedicated Subcooling (plr)	339,291

<u>MIAMI (kWh/year)</u>

Fixed Head Pressure (plr)	626,428
Floating Head Pressure (plr)	624,377
Ambient Subcooling (plr)	566,950
Dedicated Subcooling (plr)	506,592

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