AN ABSTRACT OF THE THESIS OF

DONALD FRANCIS ELGER for the degree of Master of Science in Mechanical Engineering presented October 18, 1982

Title: Design and Optimization of a Heating-only Heat Pump for Western Pacific Northwest Applications

Abstract approved: Redacted for Privacy

Gordon M. Reistad

The use of the conventional air-source heat pump for residential space conditioning has increased dramatically over the past decade. Historically these units have been designed as air conditioners which could also be used to provide heating. However, for the climatic region of the Western Pacific Northwest (the geographic region west of the Cascade Moutains), there is very little need for air conditioning. Furthermore, for energy conservation and reliability reasons, it would be advantageous to design a heat pump which could only be used for heating. The design presented in this thesis is for such a heating-only heat pump. The intended application is residential space heating in a climatic region exemplified by the Western Pacific Northwest.

The fundamental design philosophy was to maximize the energetic efficiency of an air-to-air heat pump while constraining the first cost to a value which is comparable with present commercial heat

pumps. This was begun by configuring standard off-the-shelf components in a fashion which, based on the available literature, appeared to be the best for heating-only use. Then this configuration was modeled and extensively evaluated in an effort to achieve an optimum design. This optimization was performed by modeling the heat pump on a detailed heat pump simulation code and coupling this model with a nonlinear optimization algorithm.

The predicted performance of the final design is significantly higher than that of presently available heat pump units. On a steady state basis, the COPH is 3.79 at 8.3 °C (47 °F); this is 35% higher than that of a typical commercial heat pump. On a seasonal basis the proposed design is predicted to use 67% less energy than an electric resistance heating system and 31% to 47% less electrical energy than a typical commercial unit. Because the expected first cost of the heating-only unit is comparable with present commercial units, the economic desirability of this unit is expected to be good.

Design and Optimization of a Heating-Only Heat Pump for Western Pacific Northwest Applications

by

Donald Francis Elger

A THESIS

submitted to

Oregon State University

in partial fulfillment of
the requirements for the
degree of

MASTER OF SCIENCE

Completed October 1982

Commencement June 1983

APPROVED:

Redacted for Privacy

Professor of Mechanical Engineering in charge of major

Redacted for Privacy

Head of Mechanical Engineering Department

Redacted for Privacy

Dean of Graduate School

Date thesis is presented October 18, 1982

Typed by Mary Ann (Sadie) Airth for Donald Francis Elger

ACKNOWLEDGEMENTS

It is quite exciting and satisfying to be on the final stages of completion of this thesis and I wish to acknowledge the individuals and organizations which helped make the work described here possible.

Dr. Gordon Reistad, my major professor, provided a great deal of valuable input while still allowing me the freedom to proceed as I thought best. I also wish to thank him for a very good editing job on this thesis.

Bonneville Power Administration provided the funding for my work. Purdue University allowed me to use OPT at a small fraction of the usual licensing fee and the OSU Computer Center provided additional funding for the adapting of OPT to the OSU computer.

During the course of the project I talked with over one hundred individuals from various companies and universities; I would like to acknowledge their help. In particular, Keith Rice of Oak Ridge National Laboratory and John Hackstedde of Tecumseh Products Company provided a great deal of assistance.

I would also like to thank my committee members, Dr. Lorin Davis, Dr. Alan Robinson and Dr. James Knudsen for taking their time to serve on my committee.

Finally, I would like to thank my wife, Linda, for her moral support and understanding during the course of the project.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

DESIGN AND OPTIMIZATION OF A HEATING-ONLY HEAT PUMP FOR WESTERN PACIFIC NORTHWEST APPLICATIONS

CHAPTER 1

INTRODUCTION

The 1973 energy crisis caught many of us by surprise. Years of prosperity and growth in the post World War II years had decreased our awareness of the limits on our energy resources. Today, the needs for energy conservation are increasing in importance with each passing day. The objective of the work described in this thesis is to design a system for conserving the amount of electrical energy required for residential space heating. This system is the heating-only heat pump.

A heat pump can be defined as any system for moving heat energy from a low temperature reservoir to a high temperature reservoir. The heating-only heat pump, as the name implies, is a heat pump used to provide heating only. This is in contrast to the conventional residential heat pump which is used to provide both heating and air conditioning.

The work described in this thesis was conducted as part of a larger project funded by the Bonneville Power Administration. The overall project was directed towards assessing:

- The status, potential and research needs for different electric heat pump systems and system modifications. The systems reviewed included air, water and ground source systems with possible modifications of capacity control, solar assist, crawl space assist, energy storage and/or combinations of these modifications. For the results of this study see (Elger and Reistad, 1981).

- The design of a heating-only heat pump and an estimation of the potential energy savings as compared to the standard commercial reversible heat pump and electric resistance heating system. This aspect is the subject of this thesis.

- Methods of providing heating at low ambients without using electric resistance heating. The results of this are presented in (Reistad and Elger, 1982).

1.1 Project Objective

There are numerous types of heat pump systems which have been proposed and/or built. The heat pump system considered for this work operates on the most commonly used cycle, the vapor compression or Rankine cycle. Figure 1.1 is a sketch of the simplist possible configuration. As shown, it consists of four components; the compressor, evaporator, condenser and expansion or flow control valve. This project does not propose to evaluate or improve on the individual component designs. Rather the overriding objective is:

> To seek the combination of components and the operating strategy which minimizes the amount of electrical energy required by the heat pump while satisfying acceptable cost constraints. Furthermore, the heat pump system must be designed to achieve an acceptable reliability level.

It is important to note that the design is done entirely by computer simulation and optimization. Due to economic and time limitations it was not possible to perform any experimental work.

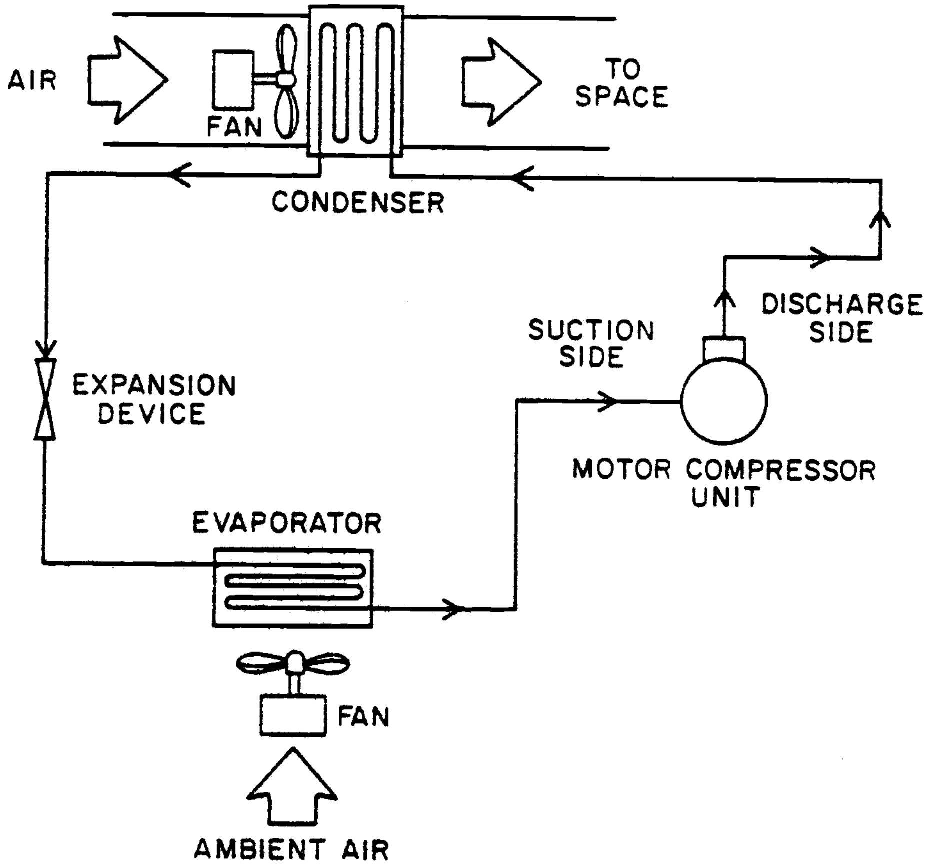


Figure 1.1 Schematic of an air-to-air heat pump

1.2 Project Restrictions

Because there are many possible types of heat pump systems and applications, one must make certain restrictions when performing a design. For this project the restrictions are:

- The heat pump shall be designed for heating only.
- The application is for residential space heating in the Western Pacific Northwest. This area is defined as the regions of Oregon and Washington west of the Cascade Mountains.
- Only an air-to-air vapor compression heat pump configuration will be considered for the design.
- The components of the heat pump system must be commercially available.

The specific reasons why it is advantageous to design a heating-only heat pump are discussed in Section 1.4. In order to put these reasons into perspective, the history of the commercial residential heat pump will be discussed next.

1.3 Historical Background--The Heat Pump

The basic operating principle of the heat pump was defined in Carnot's thesis on the Carnot cycle in 1824. Shortly thereafter, William Thompson of England (later Lord Kelvin) proposed the first practical design of a heat pump for space heating. Thompson conceived of an open cycle heat pump system using air as the working fluid. He claimed that his "heat multiplier" would use only 3% of the energy required by a system which burns a fuel to provide heating!

However, it was not until the early 1920s that the first heat pumps were built and tested. From this time until the early 1950s numerous demonstration systems were built for a wide variety of applications. However, these were basically customized for one specific application. In the early 1950s the commercial unitary heat pump industry began. These units, which were mass produced, were generally marketed in the southern U.S. as air conditioners which could also provide heating.

Figure 1.2 illustrates the growth of residential heat pump sales from the early 1950s until the present. From 1950 to 1963 the new industry did quite well; as shown the growth rate was quite good. However, this prosperity ended in 1964 due to the availability of cheap competing energy sources and more importantly to the very dismal reliability record of the early heat pumps. From 1964 until 1970 the industry stagnated and nearly died. Many manufacturers dropped heat pumps from their lines and other manufacturers restricted sales to the southern states. Then in 1971 the declining energy supply situation, rising fuel costs and better product reliability led to vigorous increases in heat pump sales.

With regards to this work, there are three very important aspects of this history. They are:

(1) As shown by Figure 1.2, the number of heat pumps being used in the United States is growing significantly. For example, in 1978 25% of new housing starts had heat pumps (Martin and O'Neal, 1980). Thus, designing heat pumps for high efficiency can have significant impacts on electrical energy consumption.

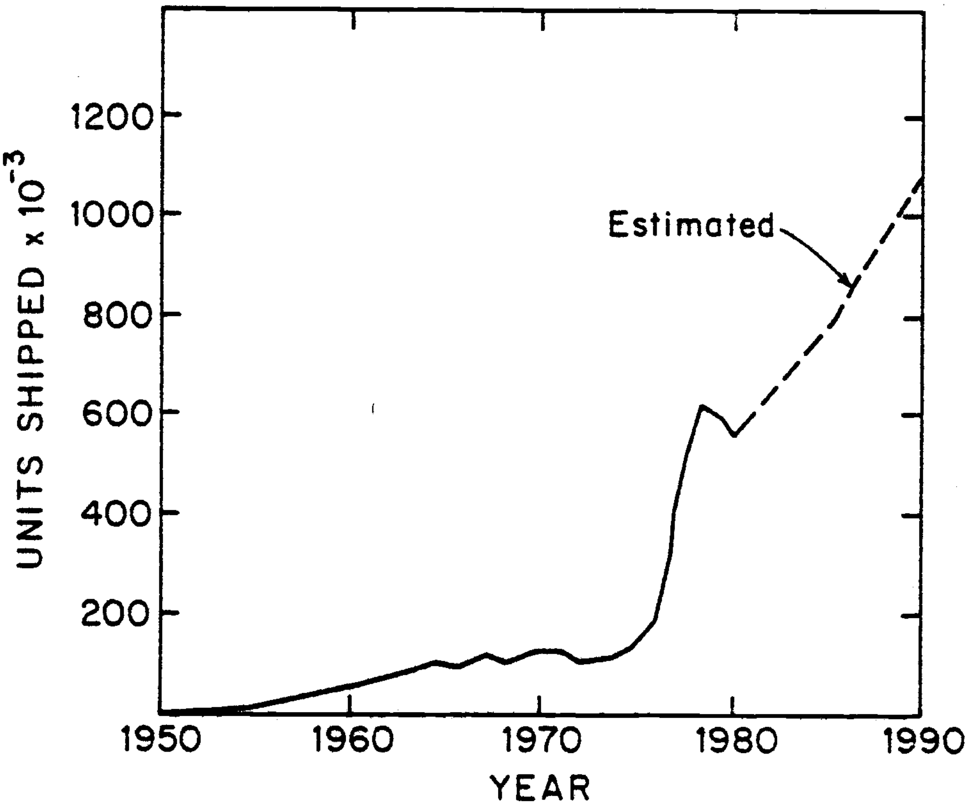


Figure 1.2 Shipment of heat pumps from United States manufacturers (Source: Groff, 1980).

(2) The early heat pumps were plagued by poor reliability. But even today, the reliability problems have not been solved by all manufacturers. Design for system reliability remains a very important design goal.

(3) The early heat pumps were designed and sized for air conditioning use. They were essentially air conditioners which had a reversing valve so they could also provide space heating. This trend is still very evident in many commercial residential heat pumps.

1.4 Advantages of the Heating-only Heat Pump

The differences in the components required for a reversible heat pump as opposed to a heating-only heat pump are very slight. So why is it advantageous to design for heating-only when it is no real problem to design for both heating and air conditioning? The specific advantages of a heating only design are:

- There is very little need for air conditioning in the Western Pacific Northwest. For example, in Seattle 95% of the electrical energy required for residential space conditioning is required for heating and 5% is required for cooling. It is felt that the air conditioning capability of a reverse cycle heat pump will be used when it is not really needed; this will decrease the energetic advantages of the heat pump relative to other heating systems.

- The design of a reversible heat pump is always a compromise between the heating and cooling requirements. For a heating only design, there is no need for compromises and the net result is a higher heating efficiency.

- There are some savings possible in components. The heating only design does not require a reversing valve or an extra expansion valve. (The extra expansion valve is for the cooling cycle.)

- All of the heat pump components except the evaporator can be positioned within the heated air stream. Thus any heat losses from these components will be recovered and used for space heating.

- The sizing of the heat pump for a specific residence can be based entirely on the space heating load; the air conditioning load does not need to be considered.

1.5 Definition of Terms

There are two figures of merit which will be used in this thesis to reflect the energetic performance of the heat pump. They are the Seasonal Performance Factor for Heating, SPFH, and the Coefficient of Performance for Heating, COPH.

COPH is used to rate the steady state efficiency of the heat pump. It is evaluated at a fixed ambient and space temperature. It is defined in words as:

$$\text{COPH} = \frac{\text{Rate at which Heat Energy is Supplied to the Space}}{\text{Rate at which Electrical Energy is used by the Heat Pump System.}}$$

SPFH is used to rate the efficiency of the heat pump for operation over an entire winter. It is a much better performance index than the COPH because the effects of on/off cycling, electric resistance backup heating and defrosting are included. All three of these factors degrade heat pump efficiency and so the SPFH is a more accurate evaluation of performance than the COPH. It is defined as:

$$\text{SPFH} = \frac{\text{Total Amount of Heat Energy Supplied to the Space for the Heating Season}}{\text{Total Electrical Energy Required by the Heating System for the Heating Season}}$$

In equation form these two performance indices are defined as:

$$COPH = \frac{\dot{Q}_{HP}}{\dot{W}_{HP}} \tag{1.1}$$

where: $\dot{Q}_{HP}$ is the heating energy output rate of the heat pump (kW).

$\dot{W}_{HP}$ is the power input to the heat pump unit (kW).

$$SPFH = \frac{Q_{HP} + Q_{AUX}}{W_{HP} + W_{AUX}} \tag{1.2}$$

where:

Q_{HP} is the total heating energy supplied by the heat pump during the heating season.

W_{HP} is the total electrical energy supplied to the heat pump during the heating season.

Q_{AUX} is the total auxiliary heating energy supplied during the heating season.

W_{AUX} is the total electrical energy supplied to the auxiliary heating system during the heating season.

CHAPTER 2

DESIGN FOR HEATING APPLICATIONS--REVIEW OF THE LITERATURE

As discussed in Section 1.3, the commercial residential air source heat pump has historically been designed and sized for air conditioning applications. However, research and design work for systems whose primary application is space heating has been conducted in a number of countries. In England a heating-only heat pump has been designed and a prototype built. In Canada and New Zealand, designs have been proposed which optimize heating performance and the Canadian design has been built and extensively tested. In the United States, Westinghouse, under the sponsorship of EPRI, and Oak Ridge National Laboratory have conducted detailed computer studies of the potential of heat pumps designed primarily for heating. The objective of this chapter is to review and summarize each of these studies.

2.1 THE HEATING-ONLY HEAT PUMP IN ENGLAND

The work on the heating-only heat pump in England has been done at the Electricity Council Research Center at Capenhurst. What appears to be the earliest reference is a study of the optimization of the heat exchangers of a heat pump (Blundell, 1977). Blundell assumed that the heat pump would be used for heating only and described a method of selecting the evaporator and condenser which would minimize the heat pump life cycle cost. The procedure, which is done separately for the evaporator and the condenser, is: (i) Determine the most efficient type of finned tube heat exchanger and for this type (ii) use graphical techniques to select the combination of heat exchanger frontal area, number of tube rows and air flow rate which minimizes the annual cost of operation of the heat pump.

Specifically, Blundell considered five different finned tube heat exchanger geometries and selected the geometry which, for a given heat transfer rate and fan power, had the lowest air-to-refrigerant temperature difference. The geometry selected had staggered circular refrigerant tubes and wavy plate fins with a 0.002 m (0.079 in) pitch.

Before performing the optimization it was first necessary to characterize the performance of the heat pump based on the performance of the evaporator and condenser heat exchangers. Blundell devotes most of his report to a discussion of the modeling of these heat exchangers. The resulting model provides a method to predict heat pump COPH based on heat exchanger area, number of tube rows and air flow rates over the heat exchangers. Then using the COPH values to predict electricity requirements, together with costing information, it is possible to predict the annual cost of operation of the heat pump as a function of the independent heat exchanger variables; where the independent variables are the evaporator and condenser frontal areas, number of tube rows and air flow rates.

The optimization procedure is performed graphically. First the COPH is plotted as a function of air flow rate and frontal area; this is illustrated in Figure 2.1. Next annual cost is plotted as a function of frontal area and air flow rate; this is shown in Figure 2.2. From this figure the most economical heat exchanger may be selected. It is important to note that this procedure is done independently for the evaporator and condenser. In other words, the evaporator geometry is fixed when the condenser optimization is performed and vice-versa.

The major conclusions of Blundell's study are:

- There are several different types of continuous-fin heat exchangers. The most commonly used geometry, wavy plate fins on a 0.002 m (0.079 in) pitch with staggered circular refrigerant tubes, is the most efficient.
- U.S. commercial heat pumps have an outdoor coil of about optimum size but the indoor coil should be doubled in size for use in a British climate. (This would improve seasonal COPH from about 2.4 to about 2.8 to 2.9.)
- COPH would be increased by reducing the size of the outdoor fan to about half that presently used. Very little advantage would be gained from having a variable speed fan.

The next step for the British engineers was to examine the suitability of the then current commercial heat pumps for applications in England. Their general finding was that available units were not very suitable. So specified design requirements were identified and Blundell's optimization procedure was used to design a heating-only unit. A prototype has been built and is currently being tested

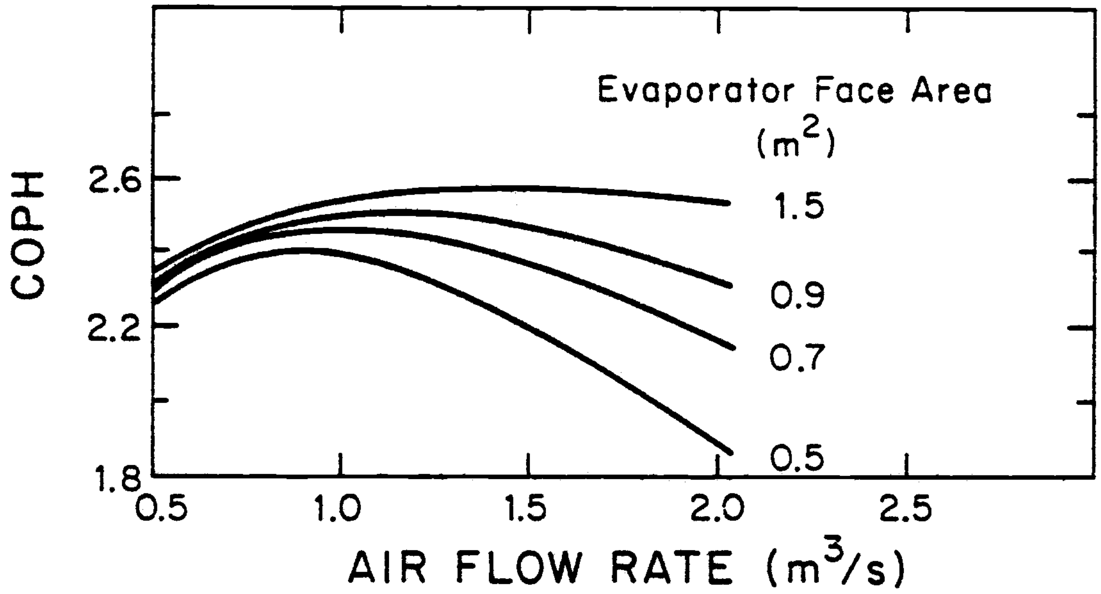


Figure 2.1 COPH as a function of evaporator face area and volumetric air flow rate (Blundell, 1977).

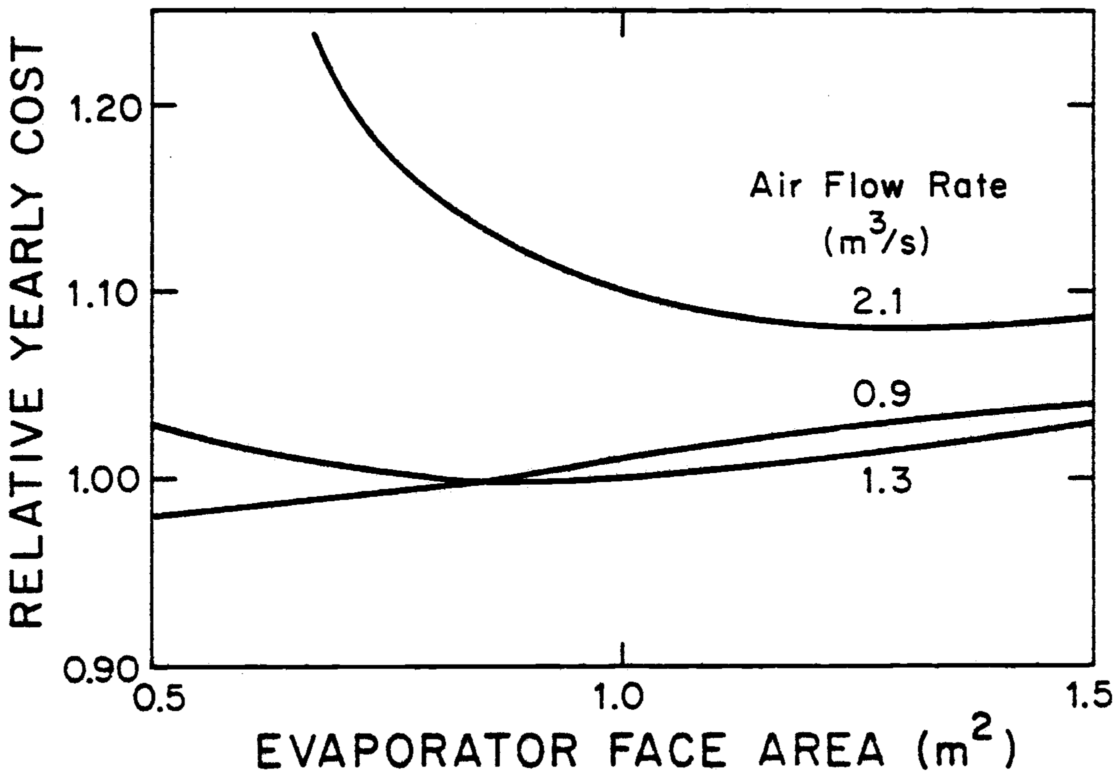


Figure 2.2 Heat pump yearly operating costs as a function of evaporator face area and volumetric air flow rate (Blundell, 1977).

(Blundell, 1977; Blundell, Heap and Goodall, 1977; Heap and Blundell, 1979; Heap, 1979).

The design requirements of the British heat pump are:

- There is no requirement for air conditioning in most British houses. Thus, the heat pump should be designed for heating only.
- The heat pump should have a low first cost in order to be competitive on the domestic heating market.
- The heat pump should not adversely affect the utility's load factor. This means that the heat pump must supply at least 65% of the design heat load at the design temperature. Then the use of supplementary heating will depend more on occupancy patterns than on the weather.
- Insulation should be considered a first priority for reducing electrical energy consumption. Therefore, the design heating load should be characteristic of a well insulated house. For a typical 80 ft^2 (861 ft^2) British house, the design heating load will be 4 to 5 kW (13,600 to 17,000 Btu/hr).

The prototype heating-only heat pump has a 3.21 kW (10,952 Btu/hr) heating capacity and a COPH of 2.84 under British winter conditions (5°C (41°F)). Defrosting is by 4.5 kW (15,354 Btu/hr) electric resistance heaters embedded in the evaporator or, if the outdoor air is warm enough by running the outdoor fan with the compressor stopped. The heat pump uses a demand defrost system which senses an increase in the refrigerant-to-air temperature difference across the evaporator to determine the need for defrosting.

2.2 HEATING OPTIMIZED DESIGN IN NEW ZEALAND

Carrington, in New Zealand, has used the heat exchanger optimization techniques developed by Blundell for the design of a heating optimized heat pump (Carrington, 1978). However his design procedure differed in three ways from Blundell's.

(i) The heat exchangers and compressor were modeled using engineering design data from equipment manufacturers instead of using heat exchanger equations.

(ii) The heating demand is specified using a load line technique and typical winter temperatures. Thus, the effects of supplementary electric resistance heating are included on the design.

(iii) A short term economic performance criteria, simple payback, was used with the life cycle costing criteria to select heat exchangers.

Table 2.1 gives the geometry of the condenser and evaporator heat exchangers for the reference case and for the optimized case. The result of the optimization was to increase the condenser surface area by 131% and the evaporator area by 105%. For use over a typical New Zealand winter, Carrington's design would decrease energy consumption by 20%, reduce supplementary electric resistance heating by 50% and reduce the number of defrost cycles. The payback period of this design is estimated to be 2 months as compared to the baseline heat pump.

2.3 HEATING OPTIMIZED DESIGN IN CANADA

As part of a project jointly sponsored by the Ontario Hydro and the Canadian Electrical Association, Young and Lange have designed and built a prototype heating optimized heat pump (Young and Lange, 1980; Young, 1980). The motivation for this project was much the same as that of the work in Britian in New Zealand; increased

Table 2.1 Condenser and Evaporator Geometry -- Base Case and Optimized Case: Carrington's Design (Carrington, 1977).

	Base Case		Optimized Case	
EVAPORATOR				
Fin Pitch, m (in)	0.002	(0.0079)	0.002	(0.0079)
Rows	4		3	
Face Area, m^2 (ft^2)	0.33	(3.55)	0.90	(9.69)
Air Flow Rate, m^3/s (cfm)	0.88	(1865)	1.50	(3178)
Fan Power, W (hP)	390	(0.52)	225	(0.30)
CONDENSER				
Fin Pitch, m (in)	0.0018	(0.071)	0.0018	(0.071)
Rows	4		4	
Face Area, m^2 (ft^2)	0.26	(2.80)	0.60	(6.46)
Air Flow Rate, m^3/s (cfm)	0.59	(1250)	0.70	(1483)
Fan Power, W (hP)	660	(0.81)	640	(0.86)

interest in conservation led to a study of the effectiveness of the then current commercial heat pumps. This in turn motivated a project to redesign the heat pump for better heating performance and reliability.

This heat pump, which uses off-the-shelf components in the standard vapor compression system configuration, has COPH values of 3.4 at 8.3°C (47°F) and 2.6 at -8.3°C (17°F). These values are very good, 6 to 30% higher than those of present commercial units.

The high efficiency of this particular design is at least in part due to the use of high efficiency fan motors and compressor as well as large heat exchangers. The basic specifications of the prototype are presented in Table 2.2. In addition to designing for high efficiency the other design criteria were high reliability, an efficient defrost system and ease of maintenance and repair during the cold Canadian winters. There are several aspects of the design which are valuable to discuss.

In place of a reversing valve (which switches between cooling and heating mode operation) a system of four solenoid valves in a bridge arrangement was used. During testing, refrigerant leaks and thermal conduction losses at the conventional sliding-port reversing valve were found to decrease the COPH by 5 to 10%. The solenoid valves have a loss of approximately 0.5%.

Young and Lange performed a fairly extensive study of defrost methods and selected reverse cycle defrosting. A defrost bypass valve, which bypasses the cooling mode expansion valve during defrost

Table 2.2. Basic Specifications of the Heat Pump Designed by Young and Lange (Young and Lange, 1980).

ARRANGEMENT:	Split System			
COMPRESSOR:	Hermetic -- 9.67 kW (33000 Btu/hr) Nominal Capacity			
HEAT EXCHANGERS:				
	Outdoor Coil		Indoor Coil	
Face Area, m^2 (ft^2)	0.66	(7.10)	0.67	(7.21)
Tube Diameter, m (in)	0.0095	(0.372)	0.0095	(0.372)
Number of Rows	3		3	
Number of Circuits	5		5	
Fin Density, fins/cm (fins/in)	3.15	(8)	4.72	(12)
FANS:				
Type	Propeller		Blower	
Rated Power, W (hP)	124	(0.166)	186	(0.249)

was used. The effect of this valve is to decrease defrosting time and more importantly to reduce the high stresses placed on the compressor during reverse cycle defrost.

For a flow control device, Young and Lange examined capillary tubes, a subcooling expansion valve (a Westinghouse proprietary design), an EEV (electronic expansion valve) sensing evaporator superheat and an EEV sensing the refrigerant liquid level in the receiver. Based on tests Young selected the EEV with liquid level sensing. This is a unique method of flow control, no commercial residential heat pumps use such a system. Essentially, it controls the operation of the system to provide a desired subcooling out of the condenser. The use of an accumulator with a heat exchanger and thermal contact between the compressor suction line and the condenser liquid line insure that no liquid is allowed to enter the compressor.

2.4 UNITED STATES RESEARCH WORK

Perhaps the most comprehensive and detailed study of the heating optimized heat pump was conducted by the Westinghouse Corporation under EPRI (Electric Power Research Institute) sponsorship as part of a larger heat pump study (Chapter 2: Kirschbaum and Veyo, 1976). This task of the Westinghouse project had general objectives of optimizing the heating performance of the heat pump under typical U.S. Northern climate conditions and identifying those improvements which have the most significant potential for improving reliability.

The optimization of the heating performance was performed using a classical nonlinear optimization routine, the method of steepest descents. Basically, this method begins with some configuration and moves along the steepest negative gradient of the objective function (variable being optimized; in this case the objective was to minimize life cycle cost) to find an optimum configuration. In simple terms, the technique can select the best combination of any given number of variables.

The objective of the work was to identify heat pump designs which minimize the annual cost of ownership of a heat pump. The annual cost of ownership is a life cycle costing figure of merit which is calculated using first cost, maintenance costs, capital costs, tax credits, energy costs, estimates of the general inflation rate and estimates of the energy inflation rate.

The general strategy begins with a computer model of the heat pump. This is used with an optimization algorithm to design a heat pump which is optimized at a fixed ambient temperature and heating capacity. Next, this heat pump system is applied to a specific residence and the heating capacity is scaled to find the best capacity. Because of this scaling, the heat pump must be reoptimized at a new fixed ambient temperature. This procedure iterates until the heating capacity of the heat pump must no longer be scaled.

Optimizations were carried out for different assumed costs of electricity, different efficiency compressors and for configurations using capacity control. Capacity control is modeled using either a two speed compressor or two parallel equal displacement compressors.

Ten variables were identified as having the dominant role in the efficiency of the heat pump. These are the independent variables of the optimization. They are:

- For each heat exchanger coil:

  effective size
  air flow rate over the coil
  fin spacing
  number of tube rows

- Fraction of the indoor coil devoted to liquid subcooling
- Indoor duct effective size
- The compressor operating map

It was assumed that heat exchanger materials, tube spacing and tube diameter have been refined to such an extent that they could be considered as fixed parameters.

Table 2.3 presents configurations of several of the optimized heat pumps of the Westinghouse study.

The second general objective of the Westinghouse research was to identify those heat pump modifications which have the greatest potential for increasing the reliability under northern climate operating conditions. As a starting point for this, a detailed field study of the frequency and costs of heat pump failure was conducted. Using this information and engineering judgment the recommended modifications were arrived at. The very common indictment of reverse cycle defrosting as a major source of poor reliability comes out of

Table 2.3

Specifications of the Westinghouse and Oak Ridge Computer Optimized Heat Pump Systems

Description*	Outdoor Coil Parameters					Indoor Coil Parameters						Performance at the Given Temperature		
	Frontal Area m^2 (ft^2)	Number of Tube Rows	Fin Density fins/cm (fins/in)	Air Volume m^3/s (cfm)	Fan Power W (hP)	Frontal Area m^2 (ft^2)	Number of Tube Rows	Fin Density fins/cm (fins/in)	Air Volume m^3/s (cfm)	Fan Power W (hP)	Subcooling °C (°F)	Ambient Temperature °C (°F)	COPH	Capacity kW (BTU/hr)
WESTINGHOUSE														
System #1	0.864 (9.30)	3	3.94 (10)	0.911 (1930)	--	0.314 (3.38)	4	5.12 (13)	0.328 (695)	--	20.8 (37.4)	-4.44 (24)	2.96	6.89 (23515)
System #2	0.764 (8.22)	3	3.94 (10)	0.776 (1644)	--	0.295 (3.18)	4	5.12 (13)	0.351 (744)	--	15.5 (27.9)	-2.22 (28)	3.13	6.19 (21126)
System #3	1.384 (14.51)	3	3.94 (10)	1.384 (2933)	--	0.425 (4.57)	4	5.12 (13)	0.469 (994)	--	16.0 (28.8)	-7.22 (19)	2.90	7.82 (26689)
System #4	5.435 (58.50)	3	3.94 (10)	2.204 (4670)	--	1.189 (12.8)	4	5.12 (13)	0.457 (968)	--	15.3 (27.5)	-8.89 (16)	3.09	8.34 (28.464)
OAK RIDGE														
System #5	0.31 (3.34)	3	5.5 (14)	0.944 (2000)	129.0 (0.173)	0.51 (5.49)	3	5.5 (14)	0.755 (1600)	509.5 (0.683)	5.0 (9.0)	8.3 (47)	2.86	11.7 (39932)
System #6	1.25 (13.5)	1	5.5 (14)	1.581 (3350)	62.1 (0.083)	0.41 (4.41)	3	5.5 (14)	0.814 (1725)	436.9 (0.586)	10.0 (18.0)	8.3 (47)	3.11	11.7 (39932)
System #7	1.25 (13.5)	1	5.5 (14)	1.770 (3750)	71.8 (0.096)	0.41 (4.41)	3	5.5 (14)	0.779 (1650)	383.8 (0.515)	8.0 (14.4)	8.3 (47)	3.48	11.7 (39932)
System #8	2.25 (24.2)	1	5.5 (14)	2.266 (4800)	51.9 (0.070)	0.64 (6.89)	4	5.5 (14)	0.708 (1500)	264.6 (0.355)	7.0 (12.6)	8.3 (47)	3.96	11.7 (39932)

*See the next page for a more detailed description of the systems.

Table 2.3 (Continued)

DESCRIPTION OF SYSTEMS

WESTINGHOUSE OPTIMIZED SYSTEMS

System #1:

Single level-E compressor; the heat pump system was optimized at an ambient temperature of -4.4°C (24°F) and an assumed running time of 2600 hours. Electricity cost was assumed to be 3.6¢/kWh.

Note: A level E compressor is a postulated compressor which is intended to reflect the performance increases due to:

- Clearance volume reduction
- An adiabatic suction line
- Wall ports
- Reduced pressure drops in the suction and discharge lines
- An increased Stroke/Bore ratio.

System #2:

Single state-of-the-art compressor; the heat pump system was optimized at -2.22 °C (28 °F) with an assumed running time of 3000 hr. Electricity was assumed to cost 3.6¢/kWh.

System #3:

Same as System #2 except: Optimization was performed at -7.2 °C (19 °F) with a running time of 2200 hr and an electricity cost of 10¢/kWh.

System #4:

Same as System #2 except: Optimization was performed at -8.9 °C (16 °F) with a running time of 2000 hr and an electricity cost of 100¢/kWh.

ORNL OPTIMIZED SYSTEMS

<u>System #5</u>:

A limited optimization with base case heat exchangers. Compressor, evaporator fan and condenser fan efficiencies are 48%, 34% and 28%, respectively. Total heat exchanger area is constrained to 0.2 m^2/kW (6.3 x 10^{-4} ft^2/Btu/hr).

<u>System #6</u>:

Same as #5 except the heat exchanger geometry was also optimized.

<u>System #7</u>:

Same as #6 except that the compressor efficiency is 56%.

<u>System #8</u>:

Same as #7 except that the total heat exchanger area is constrained to 0.4 m^2/kW (12.6 x 10^{-4} ft^2/Btu/hr).

this study. The most significant modifications on a cost/benefit scale are:

- Indoor siting of the compressor and controls.
- Use of a suction line accumulator with an oil return heat exchanger.
- An improved defrosting system.

At Oak Ridge National Laboratory, ORNL, a design optimization study has been performed which, like the Westinghouse study, uses nonlinear optimization techniques (Rice, Fischer, Jackson and Ellison, 1981). This study had three primary objectives.

(i) To calculate the COPH for an optimized vapor compressor heat pump using both existing components and components which will be available in the future.

(ii) To perform an analysis to find the sensitivity of the optimum designs to tradeoff among the design variables.

(iii) To assess the extent to which future designs could approach the performance of the Carnot heat pump.

The optimization had a general objective of maximizing COPH given component constraints. A series of optimizations were performed as the constraints were varied to reflect the component efficiencies and sizes relative to current, short term and long term technologies and economics.

Especially significant is that, contrary to previous optimization work, the objective of the optimization was to design the most energy efficient heat pump rather than the most economical heat pump. Furthermore, none of the constraints were economic. The philosophy of this procedure is that premature introduction of cost constraints could obscure designs which are nearly as cost effective as the economically optimized heat pump but more energy efficient. Economics should be used together with sensitivity analysis to determine the best tradeoffs among the design variables around the optimum configuration.

The optimization was performed by coupling the Oak Ridge heat pump model to a proprietary optimization routine (Fischer and Rice, 1981). Ten design parameters were allowed to vary simultaneously during optimization. They are:

- Compressor displacement
- Subcooling at the condenser exit
- For each heat exchanger

  Frontal area
  Volumetric air flow rate
  Number of tube rows
  Number of parallel refrigerant channels

The constraints are:

- The total heat transfer area is constrained to one of three values (total heat transfer area is equal to the evaporator frontal area times the number of tube rows plus the condenser frontal area times the number of tube rows).

  0.21 m^2/kW of nominal heating capacity--base case (6.6 x 10 ft^2/Btu/hr)

  0.42 m^2/kW of nominal heating capacity --- short term projection (13.2 x 10 ft^2/Btu/hr)

  0.84 m^2/kW of nominal heating capacity -- long term projection (26.4 x 10 ft^2/Btu/hr)

- The nominal heating capacity is fixed at 11.7 kW (39920 Btu/hr).

- The compressor overall efficiency is fixed at one of three values:

  48% -- base case
  56% -- short term projection
  65% -- long term projection

- The overall fan efficiencies (fan plus fan motors) were fixed at one of two sets of values:

	Indoor fan	Outdoor fan	
Level 1	17%	14%	-- base case
Level 2	34%	28%	-- average of a short term and long term projection

The optimizations were carried out for an ambient air temperature of 8.3°C (47°F) and an indoor temperature of 21.1°C (70°F). Test runs have shown that the optimum configuration at 8.3°C (47°F) is near optimum at -8.3°C (17°F). Thus, optimizing at one fixed temperature is valid.

Table 2.3, referred to previously, presents configurations of several of the optimized heat pumps of the Oak Ridge study.

A nonlinear optimization routine will find a single best design. However, there are typically other designs with COPH values close to that of the optimum design. Sensitivity analysis is used to identify these designs and the corresponding tradeoffs possible among the design variables.

To illustrate sensitivity analysis, the ORNL researchers performed this analysis for one of their optimized systems. The results are presented graphically in their report as contour lines of constant COPH as two independent design variables are perturbed around their optimum values. The detailed results of the sensitivity analysis will not be discussed here. The general findings were that there are many feasible tradeoffs possible among the design variables.

The major conclusions of the Oak Ridge study are:

- Compared to the current state of the art commercial heat pumps, short term improvements in designs and component performance could result in a 28% increase in efficiency. Using projected long term improvements, a 56% efficiency improvement is possible.

- The validity of the nonlinear optimization technique was established by optimizing a system representative of current commercial heat pumps. The effect of the optimization was to increase COPH by 20%, without changing the overall heat exchanger area or compressor size.

- There are many tradeoffs possible among the design variables which results in near optimum performance.

- Three promising modifications to the vapor compression system are capacity modulation, multi-stage vapor compression and the use of nonazeotropic refrigerant mixtures.

It should also be noted that developmental work on the heating-only heat pump has not been limited to research. Janitrol, a U.S. company which is no longer in business, marketed a heating-only heat pump in the 1970s. The system was designed as an add-on to an existing fossil fuel or electric furnace. It had a heating capacity of 10.8 kW (36850 Btu/hr) and a COPH of 3.2 at 8.3°C (47°F); and a COPH of 2.3 at -8.3°C (17°F) (Gordian, 1978). Defrosting was performed by a hot gas bypass system which functions by routing the hot compressor discharge gas directly through the evaporator during defrost. It has been indicated that getting the defrost system to function properly

CHAPTER 3

A DESCRIPTION OF MATHEMATICAL PROGRAMMING

3.1 THE CONCEPT OF A WORKABLE-VS-OPTIMUM DESIGN

In many engineering projects the objective is to design a system which satisfies the requirements of the intended application while meeting certain constraints. These constraints may involve cost, raw material availability, marketing decisions, size, manufacturing limitations, etc. Because in many cases it is a very difficult problem to arrive at a design which will perform the intended function, the major emphasis is often on a system which is workable as opposed to one which is optimum. Optimization, when it is performed, typically consists of parametric studies and/or "engineering judgment". Because most systems of engineering interest are so complicated, it is not feasible to examine all possible combinations of the design variables using this approach. In short, there usually is room for design improvement by using more advanced optimization techniques.

The design considered here is somewhat different from many other projects in that it involves a system which has been commercially available for approximately 30 years (this is true if one considers that the design of a heating-only heat pump is very similar to that of a reversible heat pump). It would be relatively simple to design a functional heating-only heat pump which has a performance similar to those of commercially available units. Then, by performing a parametric study for various values of the design variables this performance could be improved.

However, consider that the designer has 10 to 20 or more independent variables to specify. Furthermore, the performance of all six of the basic heat pump components is coupled. In other words, changing the performance on any given component in the heat pump system will influence all other components to some degree. Trying to select the optimum combination of the design variables is not an easy problem. To simplify these complications, a mathematical programming approach was selected for the work described here.

Mathematical programming is the general field of optimization using mathematical and numerical techniques. The general problem is to optimize some parameter (for example COPH) which can be a function of any number of design variables, subject to user specified constraints (for example equipment costs). The mathematical programming technique will select the best combination of the design variables. In general, mathematical programming algorithms are implemented on computers. A general description of mathematical programming is presented in the next section and a discussion of the specifics of how the design of a heat pump is formulated as a mathematical programming problem is presented in Chapter 4.

3.2 THE GENERAL OPTIMIZATION PROBLEM

The general optimization problem is to minimize (or maximize) some function, called the objective function, which has one or more independent variables. In addition, constraint functions involving the independent variables may be specified. These functions may be expressed as equality or inequality constraints. If no constraint functions are required, the problem is called an unconstrained problem; otherwise it is called a constrained problem.

There are two general categories of mathematical programming problems. If the objective and the constraint functions are all linear functions of the independent variables then the problem is a linear programming problem. If some of these functions are nonlinear then the problem is a nonlinear programming problem.

Linear problems are relatively easy to solve. There is a single algorithm, the Simplex method, which will solve all types of linear problems, if a solution exists. Unfortunately the optimization of a heat pump using mathematical programming techniques is a nonlinear problem, as is the case with most engineering optimization problems.

Two rather serious difficulties with nonlinear problems should be mentioned. First, there is no general solution scheme for the nonlinear problem. Instead, there are numerous algorithms from which to choose and there is no guarantee that a given method will find a solution, even if a solution does exist. In other words, for a given problem some algorithms will work while other algorithms will fail. Also, the solution times required can differ considerably between different algorithms and which particular method is fastest can depend on the particular problem. In short, choosing an algorithm is not trivial. The second difficulty is that if a given algorithm has found an optimum point, there is no general way to determine if this point is a local or global optimum. Thus, it is entirely possible to converge to a local optimum and miss the actual solution to the problem.

Because of the enormous amount of information which would be required, it would be futile here to attempt to present a worthwhile discussion of all the different algorithms for solving nonlinear problems. Instead the approach taken is to present a detailed discussion of: (i) the optimization procedure used for this project and (ii) the method of implementing this procedure for the design of a heating-only heat pump.

3.3 THE GENERALIZED REDUCED GRADIENT METHOD

The algorithm selected for this design is the Generalized Reduced Gradient Method, GRGM. The remainder of this section is devoted to the mathematical and numerical basis of this method.

The reduced gradient method, for solving a problem with a nonlinear objective function and linear constraint functions, was first proposed by Wolfe (Wolfe, 1963). This method was later generalized to handle nonlinear constraints by Abadie and Carpentier and thus the name, the Generalized Reduced Gradient Method (Abadie and Carpentier, 1969). In the following discussion of the method, mathematical rigor has been sacrificed for hopefully, increased clarity. For other references on the GRGM, see (Gabriele and Ragsdell, 1977), (Bazarra and Shetty, 1979) and (Himmelblau, 1972). Note that the majority of the following discussion follows from information presented in the first of these three references.

The GRGM is designed to solve the general constrained nonlinear optimization problem. Any problem of this type can be written in the following form:

Minimize:

$$f(\bar{x}): \quad \bar{x} = (x_1, x_2, x_3, \ldots, x_N)^T$$

Subject to:

$$g_1 (\bar{x}) = 0$$

$$g_2 (\bar{x}) = 0$$

$$\vdots$$

$$g_M (\bar{x}) = 0.$$

and

$$\bar{A} < \bar{x} < \bar{B}$$

Note that f is the objective function, x is a column vector containing the design variables, g_1 through g_m are the constraint functions and A and B are column vectors which contain the upper and lower bounds of the values of the design variables.

As stated the problem has N independent design variables and M constraint functions which must be satisfied by these independent variables. For the problem to be of interest as an optimization problem N must be greater than M. In other words, these must be more unknowns than there are equations to be satisfied. Thus, if the problem has a solution at all, it has an infinite number of solutions and the optimization consists of selecting the solution which satisfies the constraints and minimizes the objective function.

The basic idea of the GRGM is to reduce the stated problem to an unconstrained problem with N-M independent variables. M design variables are required to satisfy the M constraint equations and hence N-M variables are completely independent. Then, the unconstrained problem in N-M variables can be solved using a technique for this class of problem. This is advantageous because in general, an unconstrained problem is much easier to solve than a constrained problem.

In more detail this procedure begins with an arbitrary division of the design variables into a basic group containing M variables and a nonbasic group containing the remaining N-M variables. Before proceeding further, the following matrices will be defined to simplify the notation:

$\bar{y}^T = (y_1,\ldots,y_M) = (x_1,\ldots,x_M)$:	The Basic Variables: These are used to satisfy the M constraint functions.
$\bar{x}^T = (z_1,\ldots,z_Q) = (x_{M+1},\ldots x_N)$:	The Nonbasic Variables: These are the independent variables of the optimization.

(Note: $Q = N - M$; Q is the number of nonbasic variables)

$$\frac{\partial \bar{g}}{\partial \bar{y}} = \begin{bmatrix} \frac{\partial g_1}{\partial y_1} & \frac{\partial g_1}{\partial y_2} & \cdot & \cdot & \cdot & \frac{\partial g_1}{\partial y_M} \\ \cdot & \cdot & & & & \cdot \\ \cdot & \cdot & & & & \cdot \\ \cdot & \cdot & & & & \cdot \\ \frac{\partial g_M}{\partial y_1} & \frac{\partial g_M}{\partial y_2} & \cdot & \cdot & \cdot & \frac{\partial g_M}{\partial y_M} \end{bmatrix}$$

$$\frac{\partial \bar{g}}{\partial \bar{z}} = \begin{bmatrix} \frac{\partial g_1}{\partial z_1} & \frac{\partial g_1}{\partial z_2} & \cdot & \cdot & \cdot & \frac{\partial g_1}{\partial z_Q} \\ \cdot & \cdot & & & & \cdot \\ \cdot & \cdot & & & & \cdot \\ \frac{\partial g_M}{\partial z_1} & \frac{\partial g_M}{\partial z_2} & \cdot & \cdot & \cdot & \frac{\partial g_M}{\partial z_Q} \end{bmatrix}.$$

$$\nabla_{\bar{y}} f = [\frac{\partial f}{\partial y_1}, \ldots, \frac{\partial f}{\partial y_M}]^T$$

$$\nabla_{\bar{z}} f = [\frac{\partial f}{\partial z_1}, \ldots, \frac{\partial f}{\partial z_Q}]^T$$

$$(\frac{df}{d\bar{z}}) = [\frac{df}{dz_1}, \frac{df}{dz_2}, \ldots, \frac{df}{dz_Q}]^T = (\bar{g}_r(\bar{x}))$$

(Note: $\bar{g}_{\bar{r}}(\bar{x})$ is the reduced gradient matrix)

$$d\bar{y} = [dy_1, dy_2, \ldots, dy_M]^T$$

$$d\bar{z} = [dz_1, dz_2, \ldots, dz_Q]^T$$

$$d\bar{g} = [dg_1, dg_2, \ldots, dg_M]^T$$

For differential changes in the design variables the corresponding change in the objective function is given by Eq. (3.1) and the changes in the constraint functions are given by Eq. (3.2).

$$df = (\nabla_{\bar{y}} f)\, d\bar{y} + (\nabla_{\bar{z}} f) d\bar{z} \tag{3.1}$$

$$d\bar{g} = \left(\frac{\partial \bar{g}}{\partial \bar{y}}\right) d\bar{y} + \left(\frac{\partial \bar{g}}{\partial \bar{z}}\right) d\bar{z} \tag{3.2}$$

Since the constraint functions must always be equal to zero, it follows that $d\bar{g}$ must equal the zero vector. Thus, solving Eq. (3.2) for $d\bar{y}$ yields:

$$d\bar{y} = \left[\frac{\partial \bar{g}}{\partial \bar{y}}\right]^{-1} \left[\frac{\partial \bar{g}}{\partial \bar{z}}\right] d\bar{z} \tag{3.3}$$

Eq. (3.3) gives the changes required in the values of the basic variables to satisfy the constraint equations as the nonbasic variables (independent variables) are perturbed. Note that for finite changes in the design variables Eq. (3.3) is only a linear approximation. In the algorithm it is used as an estimate for the required changes in the basic variables and subsequent refinement is performed using Newton's method.

Substituting Eq. (3.3) into (3.1) and rearranging yields the reduced gradient:

$$\bar{g}_r(\bar{x}) = (\nabla_{\bar{z}} f) - (\nabla_{\bar{y}} f) \left[\frac{\partial \bar{g}}{\partial \bar{y}}\right]^{-1} \left[\frac{\partial \bar{g}}{\partial \bar{z}}\right] \tag{3.4}$$

The reduced gradient is a vector which gives the change in the value of the objective function for changes in the nonbasic variables with the basic variables adjusted to satisfy the constraints. For the unconstrained problem the gradient vector points in the direction of maximum increase of the objective function. Similarly, for the constrained problem, the reduced gradient given by (3.4) points in the direction, which results in the maximum rate of increase of the objective function while still satisfying the constraint functions.

The problem has now been reduced from a constrained optimization problem in N independent variables to an unconstrained problem in N-M variables and the reduced gradient vector can be used by an unconstrained optimization algorithm to determine a search direction. A line search is performed in this direction until a local minimum is found, at which point a new reduced gradient is calculated and the procedure iterates until convergence is obtained.

The minimum of an unconstrained optimization problem occurs when the elements of the gradient vector vanish. Similarly, for the constrained problem, the necessary condition for a minimum is that the Kuhn-Tucker conditions be satisfied. It can be shown that when the elements of the projected reduced gradient vanish, the Kuhn-Tucker conditions are satisfied and hence for the GRGM this is the convergence criteria (Gabriele and Ragsdell, 1977).

The projected reduced gradient, $\overline{gr}_p(\overline{x})$ is a vector which is determined from the reduced gradient using the following prescription.

$$\bar{g}r_p(\bar{x})_i = 0 \quad \begin{cases} \text{if } z_i = b_i \text{ and } \bar{g}_r(\bar{x})_i < 0 \\ \text{or } z_i = a_i \text{ and } \bar{g}_r(\bar{X})_i > 0 \end{cases} \tag{3.5}$$

$$\bar{g}r_p(\bar{x})_i = \bar{g}_r(\bar{x})_i \quad \text{otherwise}$$

The description of the mathematical basis of the GRGM is now completed. The algorithm for the method can be divided into the following steps:

(i) Specify the M basic and N-M nonbasic variables and their numerical values. This division is arbitrary except for the requirements that the $(\frac{\partial \bar{g}}{\partial y})^{-1}$ matrix be nonsingular and no basic variables can be set at their upper or lower limits.

(ii) Calculate the reduced gradient using either Eq. (3.4) or the following formula:

$$(\bar{g}r(\bar{x}))_i = \frac{f(z_1, z_2, \ldots, \tilde{z}_i, \ldots z_Q, \tilde{y}_1, \tilde{y}_2, \ldots \tilde{y}_M) - F_1}{\delta} \tag{3.6}$$

where: $F_1 = f(\bar{z}, \bar{y}) = f(z_1, .., z_Q, \; y_1, \ldots, y_m)$

δ = step size

$\tilde{z}_i = z_i + \delta$

$\tilde{y}_i = y_1 + \delta Dmi$

$Dmi = \left[[\frac{\partial \bar{g}}{\partial \bar{y}}]^{-1} \; [\frac{\partial \bar{g}}{\partial \bar{z}}] \right]_{mi}$

$m = (1, 2, 3, \ldots M)$

Eq. (3.6) is typically used because it requires N - M + 1 evaluations of the objective function while Eq. (3.4) requires N + 1 evaluations

(iii) Form the projected reduced gradient using Eq. (3.5) and test for convergence. For this work the convergence criteria is that the L_2 norm of the projected reduced gradient vector be less than some small value.

(iv) Use the reduced gradient of step (ii) with an unconstrained minimization technique to determine a line search direction. For this work the unconstrained algorithm is the Fletcher Reeves conjugate gradient method.

(v) Perform a line search for a local minimum in the direction determined in step (iv). This is performed by incrementing the values of the nonbasic variables by a small amount and adjusting the basic variables to satisfy the constraints. The basic variables are adjusted to satisfy the constraint functions by using Newton's iteration method with the linear approximation of Eq. (3.3) used as the initial guess. Specifying the step size is not entirely straight forward. It must be small enough that the problem remains feasible and Newton's method converges, and yet large enough to yield a reasonable solution time.

(vi) At the conclusion of step (v) a local minimum has been found along the search direction determined in step (iv). If any of the basic variables are at their upper or lower bounds they must be switched with a nonbasic variable. In either case, this is the end of one reduced gradient stage and the procedure returns to step (ii) for the next iteration.

CHAPTER 4

OPTIMUM DESIGN OF A HEATING-ONLY HEAT PUMP

In this chapter the decisions and procedures involved in the design of the heating-only heat pump are discussed. This begins with a statement of the design objectives. Next a discussion of heat pump sizing, component selection, air delivery temperature and economics is presented. Note that in order to properly model the problem these four areas needed to be considered prior to performing the design work. A description of the computer models used for the design is discussed in the next session. Finally the description of how the design was formulated as a nonlinear optimization problem is presented. Chapter 5 presents the configuration and performance of the final design.

4.1 DESIGN OBJECTIVES

The design objectives for this project are:

- To optimize the energy performance of a vapor compressor heat pump for residential space heating.
- To configure the heat pump system to achieve maximum reliability.
- To constrain the first cost of the heat pump system to a value which is comparable to commercial residential heat pump units.

4.2 SIZING

Sizing refers to the specification of the capacity (heating or cooling) of a heat pump in relation to the requirements of a given dwelling. In optimizing the energy performance of a heat pump for residential heating, achieving the proper match between the heat pump capacity and the residence heating load is very important. In fact, the sizing has a greater influence of the SPFH than essentially any other single design variable. However conventional heat pump sizing is based primarily on the requirements of the air conditioning cycle; there is no sizing criteria for the heating-only heat pump. Thus, before an optimization could be performed it was necesary to develop a sizing philosophy. The objective of this section is to discuss the conventional sizing procedure and to propose a rational sizing philosophy for the heating-only unit.

Historically, heat pumps have been marketed as air conditioners which could also be used to provide space heating. Hence, the conventional sizing philosophy is to size the heat pump to meet the design air conditioning load. To see if this sizing is adequate for the heating season, the balance point temperature is compared with the peak temperature of a temperature histogram for the heating season. If the balance point temperature lies above this peak, the capacity of the unit is increased so that the balance point temperature is at or below the peak histogram temperature (Kirshbaum and Veyo, 1977).

Using the conventional sizing philosophy, the heating capacity of the heat pump will not be sufficient to satisfy heating loads of the dwelling for all ambient air temperatures. Consequently, electric resistance strip heaters are added to the system to supply the extra heating capacity for low temperature operation. In addition, most heat pumps have a low temperature cutout (for example, -6 °C (21°F)) below which the heat pump is turned off and heating is provided by electric strip heaters. This is done because at these low temperatures most of the heating is already being provided by the electric resistance heaters and shutting down the heat pump protects the compressor from the higher stresses inherent in low temperature operation. Of course, any use of electric resistance heating decreases the energetic efficiency of the heat pump.

For the Western Pacific Northwest, heat pumps are usually sized to have a balance point temperature of 0 to 4.4 °C (32 to 40 °F). This is well above a typical ASHRAE 97.5% design temperature, -5 °C (23 °F), for the climatic region considered for this work. For heating operation it would be advantageous to have a balance point temperature at or below 0 °C (32 °F) but then the unit would be sized too large for the cooling loads of summer. Thus, even in a climate where the space conditioning load is almost entirely a heating load, the conventional sizing practice is a compromise between the cooling and heating loads.

For a heating-only heat pump, sizing is dictated only by the heating requirements of the dwelling under consideration. One sizing procedure would be to size the heat pump to meet the design heating load, thus avoiding almost all use of electric resistance backup heating. However, it turns out that it is more efficient to size smaller than this.

To determine a sizing criteria for heating-only units it is necessary to evaluate the SPFH for a given dwelling as a function of heat pump size. The size which yields the highest SPFH will be the one which uses the least amount of electrical energy and hence is the optimum size.

SPFH was calculated by first determining the hourly heating loads for a typical residence and then calculating the hourly heat pump electrical energy requirements to meet these loads. Electric resistance heating was added to the electrical requirements whenever the heat pump could not meet the heating load. Next, the hourly heating loads and electrical requirements were summed for the heating season and SPFH was calculated using Eq. (1.2) in Section 1.5. Note that the losses due to on/off cycling and defrosting were included in this calculation.

The dwelling heating loads were intended to be characteristic of an average single family residence; the residence floor area is 139 m^2 (1500 ft^2), an average level of insulation is used and it is assumed that the dwelling is occupied by a family of four. The residence heat load was calculated as a function of ambient air temperature by:

$$\dot{Q}_h = UA(Ti - To) - Q''' \qquad (4.1)$$

where:

$\dot{Q}_h$ = The dwelling heat load

UA = The overall heat transfer coefficient (0.348 kW/°C, 625 Btu/(hr-°F))

T_i = The temperature in the interior of the dwelling (20 °C, 68 °F)

T_o = The ambient air temperature

$\dot{Q}'''$ = The internal heat generation due to occupants, lighting, appliances, etc. 1.35 kW (4608 Btu/hr) was the value used. This is a typical value given in (Kirshbaum and Veyo, 1977)

The COPH and capacity curves for the representative heat pump are:

$$\text{COPH} = 0.04819 * T_o\ (°\text{C}) + 3.0 \qquad (4.2)$$

$$\text{Capacity} = 0.3675 * T_i\ (°\text{C}) + \text{Base} \qquad (4.3)$$

T_o = ambient air temperature (^{o}C)

Note that the values given by Eqs. (4.2) and (4.3) are representative of a "near optimum" heating-only heat pump and the value of "Base" in Eq. (4.3) was the parameter adjusted to model different sizes of heat pumps.

When the heat pump did not have sufficient heating capacity to meet the heating load, it was assumed that the difference between the capacity and the load would be met using electric resistance heating. In other words, when the heat pump operated below the balance point temperature, the heat pump operated all of the time and electric resistance heating would cycle on and off to provide the extra heating required. In practice, contractors will control the electric resistance heating in several ways: They may set the heat pump and resistance heat to cycle together below a certain ambient temperature or they may set the heat pump to operate on the first stage of a thermostat, with the resistance heating operating on the second stage. For the second method the resistance heat only operates when the heat pump cannot meet the load and consequently it is the most efficient and so it was the one selected for this model.

After calculating the resistance heating requirements the hourly electrical energy consumption, $\dot{W}_{hp}$, of the heat pump was calculated using:

$$\dot{W}_{hp} = \frac{(\dot{Q}_h(T) - \dot{Q}_{aux})}{COPH(T_o)*PLF*KD(T_o)} \qquad (4.4)$$

$\dot{Q}_h$ = The dwelling heat loss rate calculated from Eq. (4.1)

$\dot{Q}_{aux}$ = The electrical resistance backup heating required for the hour.

PLF = The part load factor: This correlation models the cycling losses.

$KD(T_o)$ = The defrost degradation factor: This correlation models the additional energy required by reverse cycle defrosting.

The equation for the part load factor, PLF, was derived empirically from the results of extensive testing at the National Bureau of Standards; it is (Parken, Kelly and Didion, 1980):

$$PLF = 1.0 - CD*(1.0-X) \qquad (4.5)$$

CD = The degradation factor (the recommended value of 0.25 was used).

X = The fraction of time that the heat pump was on for the given hour.

The defrost degradation factor, KD, was derived from a curve fit of a figure presented in (Martin and O'Neal, 1980). It is given by:

$$
\begin{aligned}
KD &= 1.0 && \text{for } T_o > 7\ ^\circ C \\
&= 0.03333T_o\ (^\circ C) + 0.76667 && \text{for } 4 < T_o < 7\ ^\circ C \\
&= 0.908333T_o\ (^\circ C) - 0.002083 && \text{for } T_o < 4\ ^\circ C
\end{aligned}
\qquad (4.6)
$$

T_o = ambient air temperature (°C)

The results of the SPFH evaluation are illustrated in Figure 4.1 which shows SPFH as a function of balance point temperature. As shown, the most efficient heat pump size is one which yields a balance point temperature in the range -2 to 0 °C (28.2 to 32 °F). It is apparent that the usual sizing philosophy can result in a decrease of up to 19% in the seasonal performance as compared to the optimum sizing philosophy.

Although the optimum size of heat pump will require more electrical resistance backup heating than a larger unit, it is still more efficient because the losses due to on/off cycling are significantly less than those of a larger unit. Cycling the heat pump on and off decreases the efficiency for a number of reasons; thermal and mechanical inertia is lost each time the heat pump is stopped, heat is lost to the ambient while the heat pump is not running and the high and low pressure sides must be equalized before the heat pump is started to avoid high compressor starting torques.

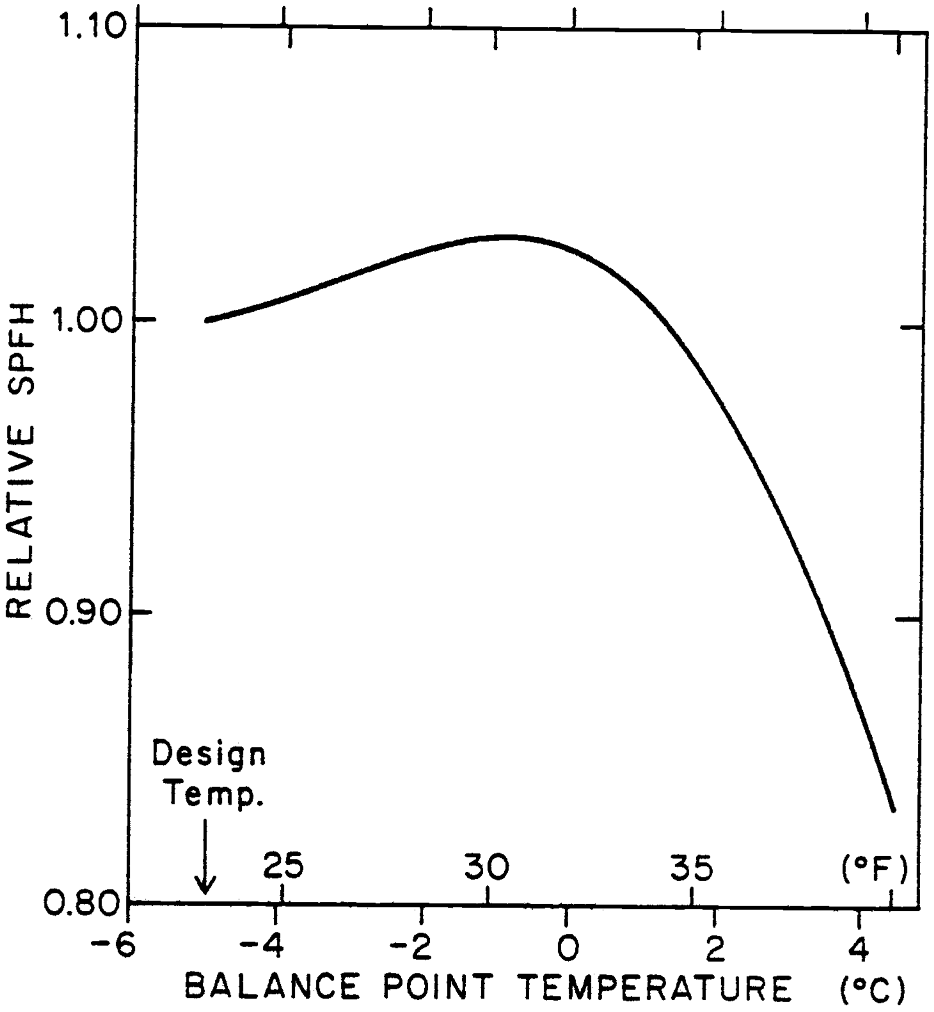


Figure 4.1 The effect of balance point temperature on SPFH.

The magnitude of cycling losses is from 0 to 33% and depends primarily on the fraction of time per hour that the heat pump is in operation (Bullock and Reedy, 1978; Goldschmidt, et al., 1980; Kelly and Bean, 1977; Parken et al., 1977).

4.3 COMPONENT CONSIDERATIONS

Prior to the heat pump system optimization it was necessary to select the general types of components which would make up the system and decide on some of the details about these components. This was done to insure that it would be possible to build the resulting design with off the shelf components. Obviously, redesigning the individual components as well as the system configuration might have resulted in higher efficiency but this was not within the scope of the present project.

4.3.1 The Compressor

The compressor is the heart of a vapor compression heat pump system. Because of its almost universal application in heat pump design, the hermetic reciprocating compressor was selected. The major criteria in selecting the particular compressor to be modeled were the efficiency and the capacity of the unit.

Figure 4.2 illustrates the isentropic efficiency of five compressor models for heating mode operation. These curves are for a single condensing temperature (34 °C, 100 °F) and are derived from curve fits to the performance maps supplied by the manufacturers.

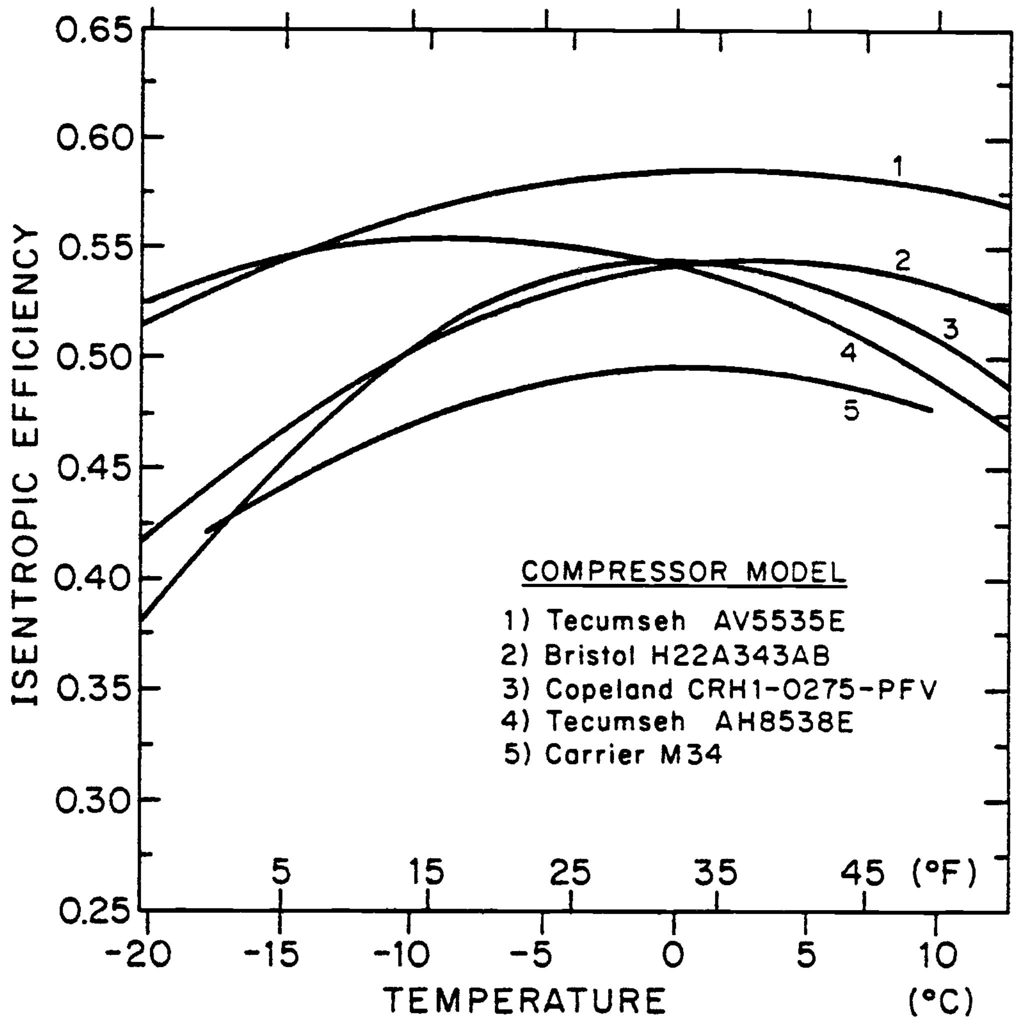


Figure 4.2 Compressor efficiency as a function of evaporating temperature for fixed condensing temperature.

Clearly, the Tecumseh AV5535E compressor has higher efficiencies than the other models and hence the AV line was selected.

The compressor displacement is a major variable in the determination of the heat pump heating capacity. Based on a desired balance point temperature of -1 to -2 °C (28 to 30 °F) and the representative house as discussed in Section 4.2, a compressor displacement in the range of 55.7 to 57.4 cm^3/rev (3.4 to 3.5 in^3/rev) is desirable. Hence, the Tecumseh AV5532E, which has a displacement of 57.6 cm^3/rev (3.516 in^3/rev) and a nominal refrigeration capacity of 9.38 kW (32000 Btu/hr), was the particular compressor selected for the design.

4.3.2 The Condenser and Evaporator Heat Exchangers

The choice of heat exchangers has a significant effect on the efficiency and cost of a heat pump system. In deciding on the type of heat exchanger, the selection was limited to the conventional finned tube geometry which consists of copper refrigerant tubes and continuous aluminum fins. Blundell has evaluated the performance of finned tube heat exchangers for heat pump applications and based on the conclusions of this work, the particular geometry selected for modeling has staggered refrigerant tubes and wavy plate fins (Blundell, 1977).

It would have perhaps been interesting to optimize the geometry of this type of heat exchanger (the variables are the tube diameter, row spacing, vertical tube space and fin spacing). The previous optimization work discussed in Chapter 2 did not optimize these variables and it would be valuable to know if the results would change significantly. The primary reason this was not done in this project was because it was felt that it would be beneficial to model one of the standard geometries of a coil manufacturer to insure equipment availability. In addition, computer costs for this more comprehensive optimization would have increased the project cost substantially.

In specifying the standard geometry to model it was noted that in order to optimize the performance of a finned tube heat exchanger the tube diameter and fin spacing should be set as small as possible (McQuiston, 1981). However as the fin spacing is decreased, the frost holding capacity of a heat exchanger is decreased. Thus, for the evaporator the fin spacing was set at a value intermediate between a typical small value (which has a small frost holding capacity) and a value used in the cold room chiller industry (Young and Lange, 1981). For the condenser the small fin spacing was selected.

The final geometry selected was the standard geometry of a major coil OEM (Original Equipment Manufacturer) which has the smallest refrigerant tube diameter (McQuay, 1978). Specifically, it is:

	Condenser		Evaporator	
Row spacing cm (in)	1.588	(0.625)	1.588	(0.625)
Vertical tube spacing cm (in)	2.54	(1.0)	2.54	(1.0)
Tube outside diameter cm (in)	0.794	(0.3125)	0.794	(0.3125)
Fin density fins/cm (fins/in)	6.30	(16)	3.15	(8)

4.3.3 The Fans and Fan Motors

Fans are added to both the condenser and evaporator heat exchangers to increase the air side heat transfer rates and to supply heated air to the conditioned space. Because significant electrical power is drawn by the fans, proper fan selection is quite important to system efficiency.

The condenser (indoor) fan has the requirements of quiet operation and moving a volumetric air flow rate, in the range of 0.47 to 1.0 m3/s (1000 to 2000 cfm), against a constant but relatively high static pressure drop. This pressure drop, which typically is 75 to 150 Pa (0.3 to 0.6 in of water), is due to the individual pressure drops across the condenser coil, condenser case, air filters, resistance heater racks and the air ducts leading to and from the conditioned space.

A direct drive centrifugal blower driven by a high efficiency PSC (positive split capacitor) motor is well suited to these requirements and hence it was the fan modeled for the optimization.

The choice of an evaporator (outdoor) fan is not nearly as straightforward as that of the condenser fan. The fan has the requirement of moving a volumetric air flow rate, in the range of 0.7 to 1.4 m3/s (1480 to 3000 cfm), across a variable pressure drop. The pressure drop is variable due to frost formation on the evaporator coil; this will occur whenever the ambient air temperature drops below about 7°C (45°F). With no frost present, the pressure drop ranges from 7.5 to 50 Pa (0.03 to 0.2 in of water) and as frost begins to form this may increase by a factor of six or more before the coil is defrosted. Thus, it is important that the volumetric air flow rate delivered by this fan remain high as the pressure drop increases.

The type of fan unit selected for the optimization is a four-bladed axial flow fan driven by a high efficiency PSC motor. The particular fan type modeled was the Torin series N four-bladed fan (Torin). A typical fan performance curve is illustrated in Figure 4.3 and as shown, it has the desired characteristic of maintaining a high air flow rate as the pressure drop increases. Furthermore, it has the characteristic that the efficiency will also increase as the pressure drop increases (within the operating range for a heat pump). It is important to mention that frosting effects were not modeled as part of the optimization.* However, selection of this type of fan is an important step in insuring that the optimized outdoor unit will perform well under frosting conditions.

4.4 AIR DELIVERY TEMPERATURE

Air delivery temperature refers to the temperature at which the air heated by the heat pump is supplied to the conditioned space.

---

*Although they were included in all SPFH calculations.

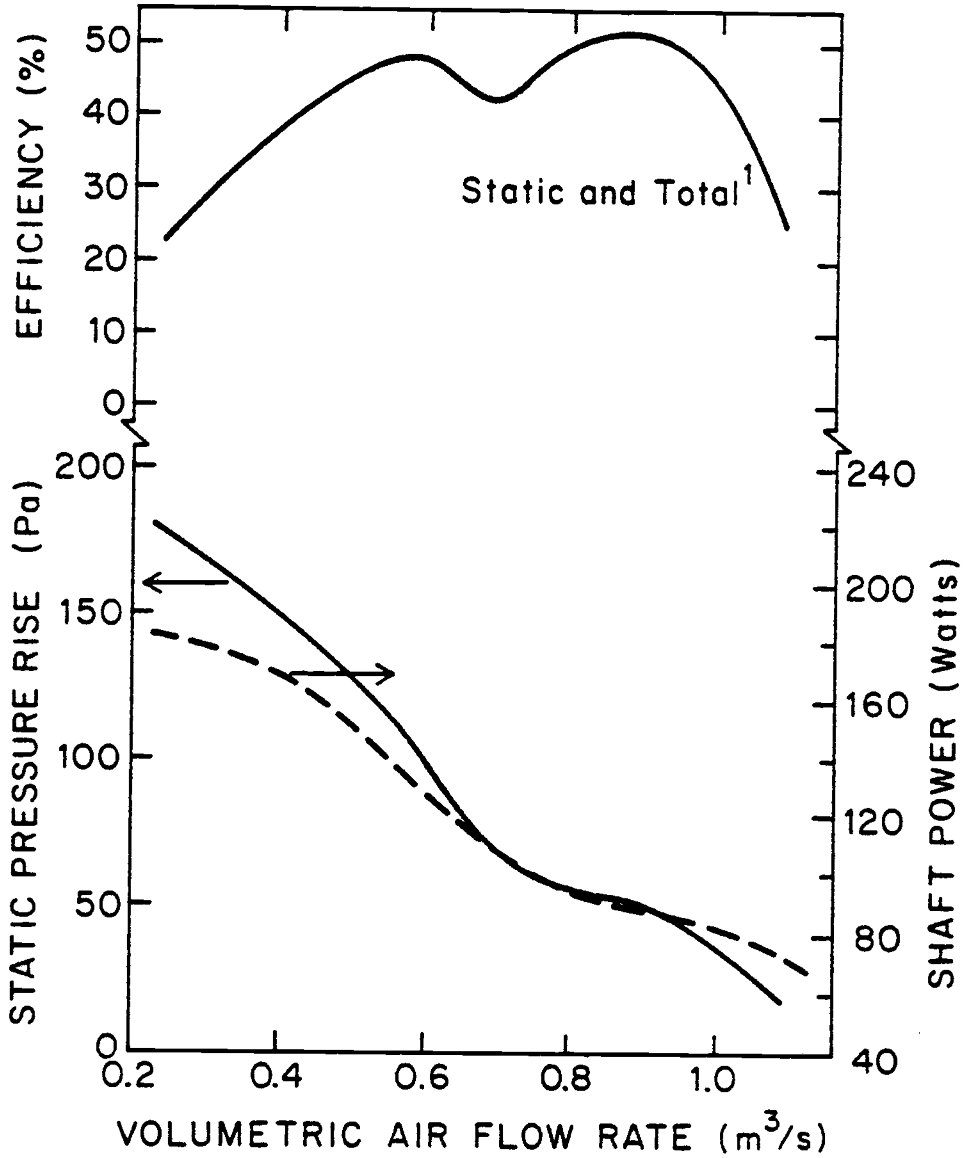


Figure 4.3 Characteristic performance curves of the evaporator fan [1](static and total efficiency curves are too close together to draw separate curves).

For a heat pump the air delivery temperature is not constant but rather decreases as the ambient air temperature decreases. Commercial heat pumps have air delivery temperatures in the range 32 to 43.3 °C (90 to 110° F) and somewhat higher, while other heating systems have constant air delivery temperatures typically in the range 43 to 60°C (110 to 140° F).

None of the other heating optimized heat pump work described in Chapter 2 discussed or constrained air delivery temperature. This seems like an oversight because the objective of a heat pump is to provide conditions which are comfortable to people and air delivery temperature certainly affects comfort.

For a heat pump system there are two conflicting requirements for the air delivery temperature; it should be kept high to insure comfortable conditions but it should be kept low to maximize the COPH and heating capacity of the unit. Analyzing for maximum efficiency is relatively easy but deciding on what conditions are comfortable appears to be an unresolved question.

The problem can be stated as follows: Suppose a person is in a house with an inside air temperature of 20 ° C (68 ° F) and the supply air from a duct is blowing on this person; for what range of air velocities and air supply temperatures will this person be comfortable?

ASHRAE has sponsored a great deal of work to evaluate the ranges of indoor conditions for which people are comfortable (ASHRAE, 1981).

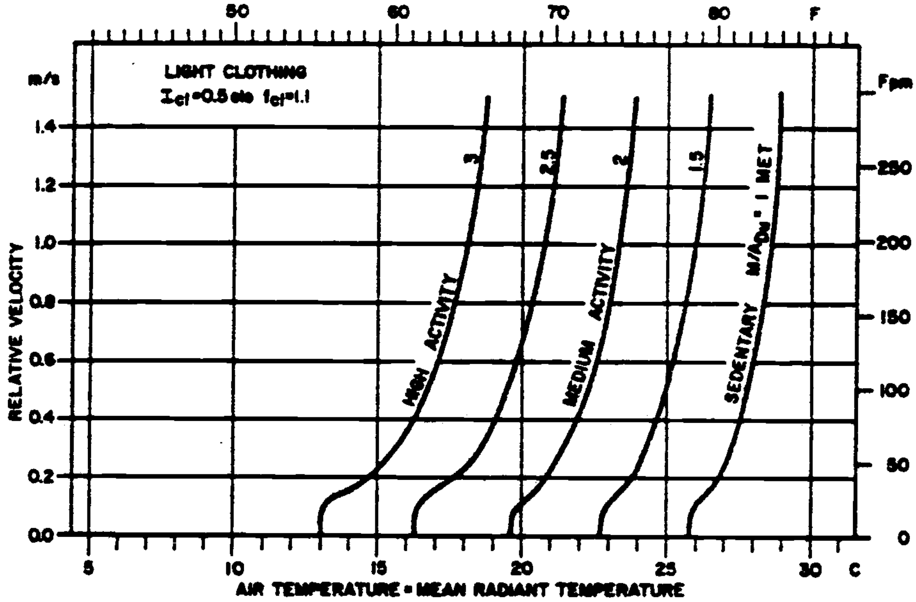
However, it does not appear that they have addressed the effects of air delivery temperature on comfort for heating. There is, however, some information which is valuable. Figure 4.4 is an ASHRAE diagram which illustrates the combined influence of air temperature and air movement on comfort. From this figure it would appear for the range of air velocities considered, the most that the air temperature would need to increase to insure comfort is about 5 °C (9° F). Furthermore, since most room air diffusers do not create high air velocities over the occupants of the room, it could be postulated that the air delivery temperature does not need to be substantially higher than the room air temperature.

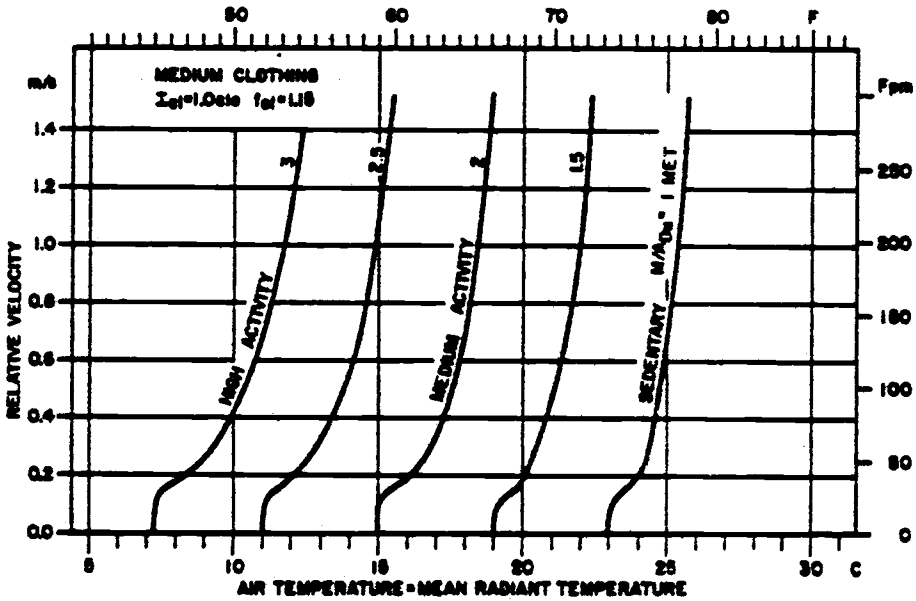
Several heating and air conditioning contractors were contacted to obtain their views on air delivery temperature. The general consensus was that the air delivery temperature range of present commercial heat pumps is satisfactory. However, one contractor did indicate that he had received complaints and he recommended a minimum air delivery temperature of 46 °C (115 °F).

In light of the above discussion it would seem reasonable to constrain the air delivery temperature to be above 32.2° C (90° F) for operation above the balance point temperature.

4.5 ECONOMIC CONSIDERATIONS

Economics are without a doubt the single most important factor determining the level of market penetration, and hence the effects on





[a]The comfort lines corresponding to five different activity levels are curves through different combinations of relative air velocity and ambient temperature which provide optimal thermal comfort. The two charts apply for persons wearing 0.5 and 1.0 clo at 50% rh.

Figure 4.4 Influence of air velocity and ambient temperature on comfort (from ASHRAE, 1981).

energy conservation, of heat pumps. For any residential heating system it is essential that economics be considered. In the initial phases of this project the possibility of optimizing some economic parameter was considered but then rejected for a variety of reasons. The usual economic performance index in the recent literature is Life Cycle Cost, LCC. However, a LCC minimization was not performed for the following reasons.

(i) LCC is a very attractive parameter to the engineer because it is analytical, unambiguous and is the most logical criteria in making a buying decision. However, it is not felt that LCC is considered by the consumer in his decision of whether to buy or not. Rather the marginal first cost of a heat pump, as compared to the alternatives, is weighed against the amount of money saved per year for electricity. In other words, simple payback is used. As an illustration of this, consider the number of people who use a LCC analysis to evaluate the alternatives when purchasing an automobile.

(ii) The uncertainty in LCC is quite high due to the number of long-term economic parameters which must be estimated. These include the general inflation rate, energy inflation rate, maintenance costs, discount rate and the system life. One approach to bypass this particular problem would be to do the analysis for a range of different values and let the person interested in the results choose whatever values they thought were best. However, one LCC optimization would be quite expensive in terms of computer time and doing the analysis for a range of economic parameters would be prohibitively expensive.

(iii) Equipment manufacturers were typically very reluctant to give out their OEM prices to a university. To do a LCC analysis, one would need OEM prices as a function of component size and configuration; attempts to gather this information were not very successful.

(iv) An optimization will lead to a single design. However, the results of sensitivity analysis indicate that there is not a single optimum but rather there is a cluster of designs around the optimum. Thus if a cost parameter is the primary variable being optimized, it would be possible to not consider configurations which are as cost effective as the optimum but more energy efficient (Rice, et al., 1981).

The costing philosophy for the design reported here is to constrain the heat pump system first cost to a value which is comparable with commercial units. Rather than estimating the first cost of the heating-only heat pump from OEM costs, the approach taken was to directly compare the types and sizes of components between the heating-only heat pump and present commercial units.

The heat pump system costs can be subdivided into the following individual costs:

Compressor unit
Evaporator heat exchanger
Evaporator fan and fan motor
Condenser heat exchanger
Condenser fan and fan motor
Controls
Refrigerant lines, valves and other plumbing
Flow control valve
Cabinets
Accumulator and receiver
Resistance heaters
Duct work
Assembly, testing, and packaging
Shipping and installation

For a fixed capacity of heat pump the majority of these costs are more or less fixed. The costs which are variable are the quality of the fan motors and compressor (a high efficiency unit costs more than a standard performance unit); and the heat exchanger sizes.

As previously discussed, high efficiency fan motors and compressors were selected. Thus, the only variable left is heat exchanger size. To assess the heat exchanger sizes of commercially available units, sizes for 15 representative units were found from manufacturers' literature. The variable which is used to characterize heat exchanger cost is the total heat exchanger area; this is equal to the product of the tube rows and frontal area for the evaporator plus the product of the tube rows and frontal area for the condenser. The statistics of the commercial units are:

Sample Mean

$\overline{Y}$ = 0.291 kW/m^2 of heating capacity at 8.3 °C
(0.918 ft^2/(1000 BTU of heating capacity at 47 °F)

Sample standard deviation:

s = 0.0939 kW/m^2 of heating capacity at 8.3 °C
(0.2962 ft^2/(1000 BTU of heating capacity at 47 °F))

Based on these values, the total heat exchanger area for the heating-only heat pump was constrained to 0.35 m^2/kW (1.104 $ft^2/1000$ Btu) which is slightly less than one standard deviation above the mean value.

The net results of this approach is that it is expected that the first cost of the heating-only heat pump, if it were to be manufactured, would be between the typical cost of the middle of the road heat pump and the high efficiency heat pump. Since the expected performance is significantly higher than that of any commercially available units (see Section 5.2), the payback period would be reduced significantly and consequently it should be quite attractive to the potential buyer.

4.6 THE OPTIMIZATION MODEL

The optimization model, which is written in FORTRAN 5 code and conceptually illustrated in Figure 4.5, consists of a heat pump model, a generalized reduced gradient optimization algorithm, a function subroutine for evaluation of the objective function, a subroutine for evaluation of the constraint functions and an interface subroutine. The heat pump model and optimization algorithm were obtained from Oak Ridge National Laboratory and Purdue University (Mechanical Engineering Department), respectively, and are discussed in the following two subsections. Note that both of these codes were supplied in FORTRAN 4 code but were modified for compatibility with

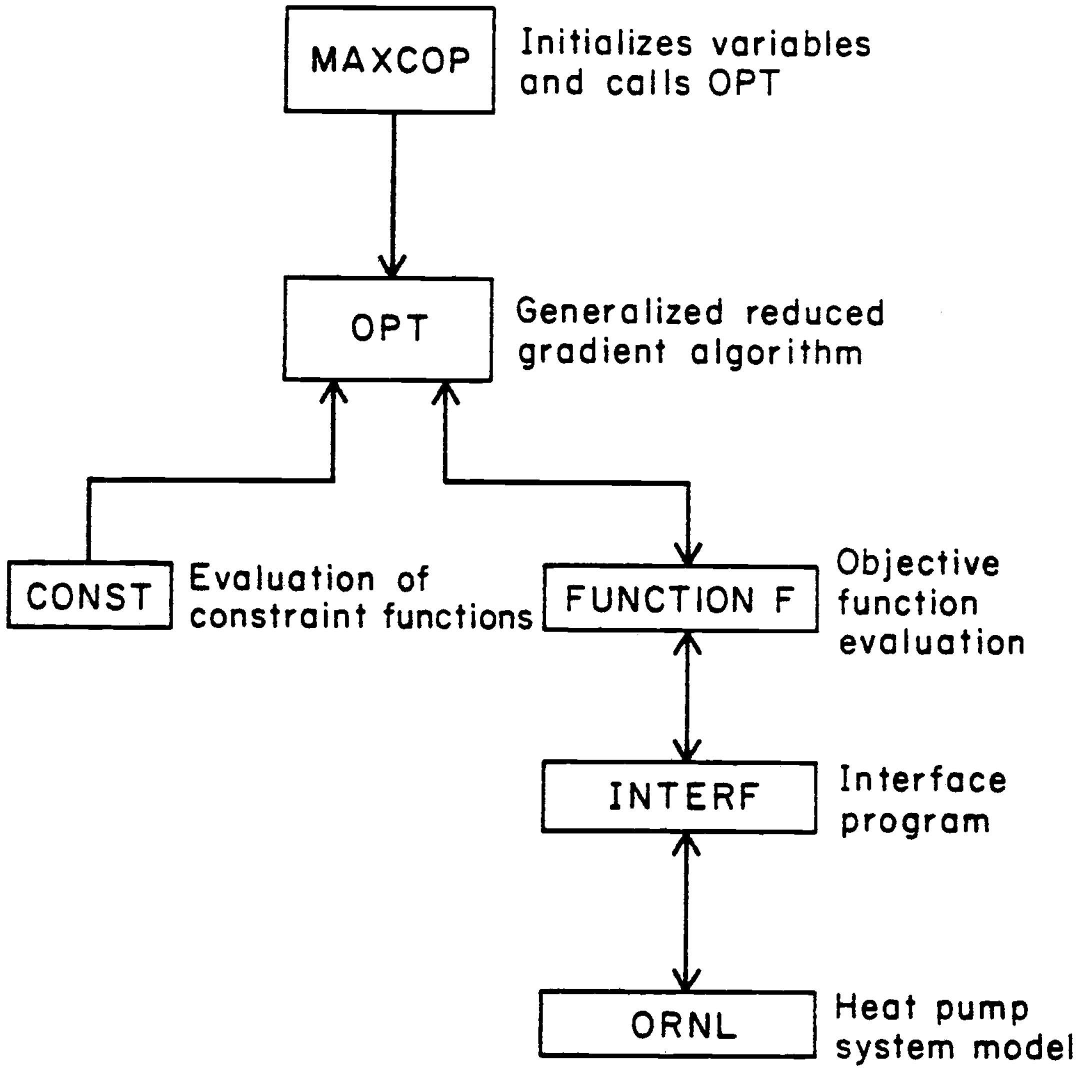


Figure 4.5 Block diagram of the optimization model

FORTRAN 5. A listing of the objective function, constraint functions and interface subroutines is presented in Appendix A.

4.6.1 The Heat Pump System Model

The heat pump system model was developed at Oak Ridge National Laboratory (Fischer and Rice, 1981). It models the steady-state performance of a conventional, electric, vapor compression, air-to-air heat pump operating in either the heating or cooling mode. The basic modeling philosophy is to use underlying physical principles, as opposed to empirical correlations based on measured data, whenever it is possible. The code has evolved over a period of approximately five years and the accuracy has been experimentally verified against laboratory data. It is reported to be accurate to within 3% (Ellison, 1979; Dabiri, 1982).

The heat pump system is simulated by modeling the individual components and then performing an iteration to establish a balance point among these components. This procedure is similar in idea to the usual graphical methods of analyzing a refrigeration or heat pump system. Of course, the level of detail used in the computer model greatly exceeds what one could use with hand calculations.

The components modeled by the code are the condenser, evaporator, compressor, refrigerant lines and a flow control device. The components which are not directly modeled are the accumulator, receiver, fans and fan motors. The accumulator and receiver are not modeled because the steady-state operation point of the heat pump is not influenced by these components because they are simply reservoirs for extra refrigerant.

Compressor: The compressor can be modeled in one of two ways; either curve fits to a compressor map (supplied by a compressor manufacturer) can be input or heat loss and efficiency parameters can be used to characterize a compressor. For this work the map-based model was used. The heat losses from the compressor shell are specified as input data by the user. For this work the code was modified to add these heat losses to the indoor air stream. This is consistent with the proposed configuration which locates the compressor in the indoor unit.

Heat Exchangers: The heat exchanger models are quite detailed. They model the conventional finned tube heat exchanger by separating it into equivalent parallel circuits with unmixed flow on both the refrigerant and the air side. Calculations on the refrigerant side are performed separately for regions of subcooled, two phase and superheated conditions. Calculations on the air side are performed separately for the portion of the heat exchanger which is dry and the portion which is wet due to dehumidification.

Expansion Device: The code will model either a capillary tube, a short tube orifice or a thermal expansion valve; or the subcooling at the exit of the condenser can be specified and the code will size the flow control device for the given operating conditions. The latter choice was used for all calculations performed for this work.

Evaporator Fan: In place of a fan unit model in the code the user is required to input volumetric air flow rates for both the evaporator and condenser coils and a fan unit efficiency for the condenser. For the evaporator fan unit the user has the option of specifying an efficiency or allowing this parameter to be calculated by the code. Previous nonlinear optimization work using a constant evaporator efficiency resulted in a fan specification for the optimized design which could not be met with existing designs (Rice et al., 1981). Consequently, in order to insure that this did not happen, it was ncessary to develop an evaporator fan model. The specific reasons why a model was needed are (i) to insure that an existing fan could be sized to supply the volumetric air flow rate for the evaporator geometry as specified by the optimization results and (ii) to accurately model the evaporator fan unit efficiency.

The objective of the evaporator fan model is to calculate the fan efficiency and the diameter of fan for operation at a given pressure drop and volumetric air flow rate. This is done by using dimensionless groups to correlate the performance of the selected fan type, the Torin series N four-bladed axial flow fan. It was assumed for this model that the total head developed by the fan was equal to the pressure head (the velocity head was neglected) and the fan total efficiency was equal to the static efficiency. Note that this assumption was made because the velocity head is typically less than 5% of the pressure head and the total efficiency is at most 0.2% above the static efficiency.

With the above assumptions the dimensionless groups used are:

Specific speed, N_S:

$$N_s = \frac{N \rho \dot{Q}}{\Delta P} \tag{4.7}$$

Head coefficient, C_h:

$$C_h = \frac{\Delta P}{\rho N^2 D^2} \tag{4.8}$$

Fan efficiency, η:

$$\eta = \frac{\dot{Q} \Delta P}{\dot{W}} \tag{4.9}$$

where:

N = fan motor speed

$\dot{Q}$ = volumetric air flow rate

ΔP = static pressure drop across the evaporator

ρ = air density

$\dot{W}$ = shaft power delivered to the fan

D = fan diameter

By examining the fan performance curves for a number of fan sizes, in the fan series of interest, it was found that the fan efficiency and head coefficient could be correlated as a function of specific speed. For the head coefficient it was found that for a given specific speed there is a range of possible head coefficients. This is because a given flow rate and pressure drop can be produced with fans that have diameters within a certain range. The range of diameters is a consequence of changes in the fan blade pitch. The range of head coefficients, at a specific speed, is defined here as the range between $C_{h,max}$ and $C_{h,min}$; the maximum and minimum head coefficients for a given specific speed, respectively. The correlations for $C_{h,max}$, $C_{h,min}$ and the fan efficiency derived from the fan performance data are:

$$C_{h,max} = 0.0411 \exp(-0.1352 N_S) \quad (4.10)$$

$$C_{h,min} = 0.0881 N_S \quad (4.11)$$

$$\eta = 0.789 \exp(-0.0805 N_S) \quad (4.12)$$

Now, for a given volumetric air flow rate, evaporator geometry and fan motor speed the specific speed is calculated using Eq. (4.7). Then the fan efficiency, and the maximum and minimum head coefficients are calculated using Eqs. (4.10), (4.11), and (4.12). Finally, the minimum and maximum fan diameters which can be sized to operate at the given point are calculated by solving Eq. (4.8) for fan diameter.

The required input to the heat pump model is necessarily quite detailed. Rather than discussing all of the assumptions and details used to create the input data file, sample inputs and outputs for the final design are included as Appendices B and C. The constants used in the "BLOCK DATA" input section of the code are the recommended values from the user's manual (Fischer and Rice, 1981).

4.6.2 OPT: The GRGM Algorithm

The program OPT, which utilizes the generalized reduced gradient method (GRGM), was obtained from Purdue University for use as the optimization algorithm (Gabriele and Ragsdell, 1976). It is a proprietary code which was purchased for this project and has been available for license since 1976. The method used by the code is presented in detail in Chapter 3 and will not be discussed further here.

The following values of user supplied parameters required by OPT were found to work well:

CRIT	(convergence criteria)	0.01
EPSLS	(line search stopping criteria)	0.0005
EPSBD	(tolerance on active constraint region)	0.02
EPS	(differing parameter used in the numerical evaluation of first derivatives)	0.03 to 0.04

The value of EPS, the differencing parameter for calculating derivatives, is quite a bit higher than the value of 1×10^{-6} recommended in the users manual. This was done to eliminate the effects of "white noise" (due to convergence tolerances) inherent in a code such as the Oak Ridge heat pump model which performes an iterative solution requiring many separate convergences. However, with this large of a step size, the convergence of the projected reduced gradient was never achieved. So, convergence was assumed when the line search could not find a better value of the objective function in the direction of search. In any case, it was not much of a problem because the coding was set up to allow the user to monitor the progress of the optimization and halt the run at any point, if desired. In other words, the progress of the optimization was followed and as long as it was proceeding as desired the run was continued; when it appeared that a run was bouncing around an optimum point, it could be halted.

It should be noted that the user supplied parameter values for OPT listed previously were not the results of critical examination and analysis. Computing costs and time constraints precluded such an analysis. Thus, while the values given worked well, it is quite possible that there are better values to use.

4.7 OPTIMIZATION FORMULATION

The objective of this section is to describe how the design of a heating-only heat pump was formulated as a nonlinear optimization problem.

4.7.1 Objective Statement

The objective of the optimization was to maximize the COPH of a heating-only heat pump for an ambient air temperature of 4.83 °C (40.7 °F). This particular temperature is the degree day weighted average temperature for Portland, Oregon. In other words, one half of the heating degree days occur at temperatures less than this value and one half occur at temperatures greater than it.

At first glance, it seems that the optimization should have been based on optimizing the SPFH since this parameter is the best index of energetic efficiency. However, this would have increased computing costs by a factor of two to four times. Furthermore, previous optimization work has indicated that an optimization at a fixed ambient air temperature results in a configuration which is near optimum for other temperatures (Rice et al., 1981). Thus, it is felt that the optimization at a fixed ambient temperature together with the evaluation of the optimum energetic sizing will achieve comparable results to an SPFH optimization, but at a substantially lower cost.

4.7.2 Choice of Independent Variable

The independent variables for a heat pump system design are:

For each heat exchanger:

- tube diameter
- row spacing
- fin spacing
- vertical tube spacing
- number of tube rows
- number of parallel circuits
- frontal area
- volumetric air flow rate (actually fan size)

compressor displacement
subcooling at the condenser exit
indoor duct size
refrigerant line diameters

Indoor duct size was not considered. Furthermore, based on the component considerations discussed in Section 4.3, the following variables were fixed at constant values:

For each heat exchanger:

tube diameter
row spacing
vertical tube spacing
fin spacing

Compressor displacement

Refrigerant line diameters were fixed based on reliability considerations; they must be sized to provide continuous oil return to the compressor under all operating conditions.

Based on the results of previous optimization work, the number of parallel refrigerant channels was fixed at three for the condenser and seven for the evaporator. These particular values provide pressure drops below the recommended maximum values of 103 kPa (15 psi) for the condenser and 48 kPa (7 psi) for the evaporator (Rice et al., 1981). In addition, the number of tube rows for the condenser was fixed at four to constrain the size of the indoor unit to a reasonable value.

The remaining six variables are the independent variables of the optimization. They are:

evaporator frontal area
number of tube rows for the evaporator
condenser frontal area
subcooling at the condenser exit
volumetric air flow rate over the evaporator
volumetric air flow rate over the condenser

The program OPT is not set up to handle variables which are constrained to integer values, hence the number of tube rows for the evaporator was modeled as being continuously variable and the final value was determined by rounding off to the nearest integer.

4.7.3 Constraint Functions

Only two constraint functions were required, one nonlinear and one linear. The first constrains the total heat exchanger area to a value of 3.63 m^2 (39.1 ft^2). This value will yield the desired capacity normalized heat exchanger area of 0.35 m^2/kW of nominal heating capacity (1.104 $ft^2/1000$ Btu of nominal heating capacity), as discussed in Section 4.5. The second constraint was used to constrain the allowable fan specific speed to a value between 4 and 35.

CHAPTER 5

THE HEATING-ONLY HEAT PUMP: FINAL DESIGN

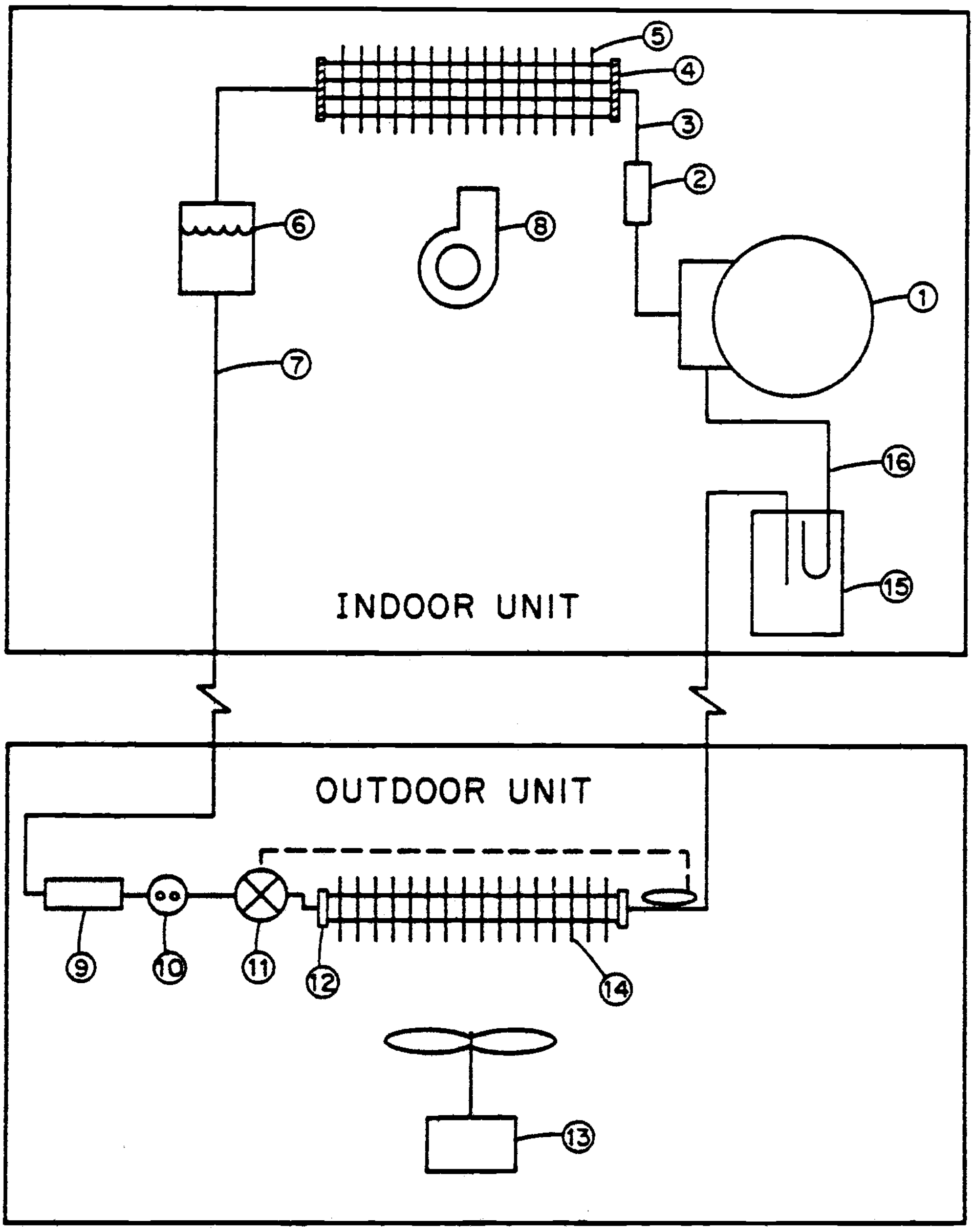
The design optimization described in the previous chapter was accomplished for a heat pump configured for heating only. This chapter presents the configuration, the resulting final design and a discussion of various aspects of this final design.

5.1 CONFIGURATION AND COMPONENT SPECIFICATIONS

The configuration of components for the final design of the heating-only heat pump is illustrated in Figure 5.1. The heat pump is organized into a split system geometry with the indoor unit being comprised of the compressor, condenser fan, condenser coil, receiver and accumulator; and the outdoor unit consisting of the evaporator fan, evaporator coil and thermal expansion valve. The configuration of the heating-only heat pump is different than the usual commercial geometry, for which only the condenser coil and the condenser fan unit are located in the indoor unit with the remainder of the components in the outdoor unit. The configuration proposed here has two primary advantages over the conventional one:

(i) The indoor location of the compressor unit will increase the reliability of this unit (Kirschbaum and Veyo, 1976). Furthermore, if servicing is required in the winter, it will be performed more easily if the unit is located indoors.

(ii) The heat losses from the compressor shell and the compressor discharge line are recovered and used for heating



1) COMPRESSOR
2) MUFFLER
3) DISCHARGE LINE
4) DISTRIBUTOR
5) CONDENSER
6) RECEIVER
7) LIQUID LINE
8) CENTRIFUGAL FAN
9) FILTER DRIER
10) SIGHT GLASS
11) THERMAL EXP. VALVE
12) DISTRIBUTOR
13) AXIAL FLOW FAN
14) EVAPORATOR
15) ACCUMULATOR

Figure 5.1 The heating-only heat pump system.

The specifications for the components of the finalized design are presented in Table 5.1. When this unit is compared to an "average" commercial unit a number of differences are apparent. The optimized design has:

- smaller fan units for both the indoor and outdoor fans
- greater fin spacing on the evaporator coil
- a significantly greater fraction of the total air side heat transfer area located on the condenser side. (For the design presented here 65% of the total air side heat transfer area is on the condenser side; for an average commercial unit this value is 48%).
- fewer evaporator coil tube rows
- no reversing valve or cooling mode expansion valve

It should be noted that there are differences between the values of the design variables used for the final design and those found by the optimization algorithm due to (i) requirements for an integer number of tube rows for the evaporator (the number of tubes rows was allowed to take on fractional values for the optimization) and (ii) fan sizing considerations. As discussed earlier, the number of tube rows was rounded to the nearest integer value. In addition, the evaporator fan was sized approximately 5.1 cm (2 in) larger than specified by the optimization results. While the increase in fan size decreases the COPH slightly, it results in a significantly increased volumetric air flow rate; this will increase the time required between defrost cycles and increase the air delivery temperature to the conditioned space.

Table 5.1 Specifications of the Heating-Only Heat Pump Final Design

COMPRESSOR UNIT:

Type - Hermetic, reciprocating, high efficiency heat pump compressor

Refrigerant - R22

Displacement - 57.62 cm^3/rev (3.516 in^3/rev)

Motor Speed - 3450 rpm

Protection - High and low pressure cutout switch
Protection against short cycling

Recommended Model - Tecumesh AV5532E

CONDENSER HEAT EXCHANGER:

Type - Finned tube: continuous, wavy, aluminum, plate fins; staggered copper refrigerant tubing.

Frontal area - 0.444 m^2 (4.774 ft^2)

Number of tube rows - 4

Number of parallel refrigerant circuits - 3

Fin pitch - 6.3 fins/cm (16 fpi)

Fin thickness - 0.01524 cm (0.006 in)

Tube nominal outside diameter - 0.794 cm (5/16 in)

Row spacing - 1.588 cm (0.625 in)

Vertical tube spacing - 2.54 cm (1.0 in)

Tube wall thickness - 0.0406 cm (0.016 in)

Subcooling - 9.5 °C at 4.83 °C ambient air temperature
(17.1 °F at 40.7 °F ambient air temperature)

Table 5.1 (Continued)

EVAPORATOR HEAT EXCHANGER:

Type - Finned tube: continuous, wavy, aluminum plate fins; staggered copper refrigerant tubing

Frontal area - 0.927 m^2 (9.99 ft^2)

Number of tube rows - 2

Number of parallel refrigerant circuits - 7

Fin pitch - 3.15 fins/cm (8 fpi)

Fin thickness - 0.01524 cm (0.006 in)

Tube nominal outside diameter - 0.794 cm (5/16 in)

Row spacing - 1.588 cm (0.625 in)

Vertical tube spacing - 2.54 cm (1 in)

Tube wall thickness - 0.0406 cm (0.016 in)

CONDENSER FAN UNIT:

Type - Direct drive centrifugal blower driven by an eight-pole high efficiency PSC motor.

Volumetric air flow rate - 0.528 m^3/s (1180 cfm)

Static pressure drop - 104.4 Pa (0.419 in H_2O)

Motor speed - 825 rpm

Motor rated power - 149 W (1/5 hp)

Fan - Torin PD 0909; 0.241 x 0.241 m (9.5 x 9.5 in) blower

EVAPORATOR FAN UNIT

Type - Direct drive propeller fan driven by a six-pole high efficiency PSC motor. The motor should be totally enclosed by a case.

Volumetric air flow rate - 1.09 m^3/s (2310 cfm)

Static pressure drop - 15.94 Pa (0.064 in H_2O)

Motor speed - 1000 rpm

Motor rated power - 149 W (1/5 hp)

Fan - Torin, Series N, 4 blade 0.508 m (1.67 ft) propeller fan

THERMAL EXPANSION VALVE:

Type - Sporlan valve incorporating the RPB feature and an external equalizer. The permanent bleed factor should be 1.15.

Nominal capacity - 7.03 kW (2 ton)

Rated operating superheat - 6.11 °C (11 °F)

Static superheat - 3.33 °C (6 °F)

As previously discussed, system reliability is quite an important consideration for a heat pump. Aspects of this design configuration and components which contribute to improving reliability are:

- The use of a suction line accumulator. (Virtually all reliability information available indicates that an accumulator is essential to protect the compressor from liquid refrigerant entrained in the suction vapor)
- The indoor siting of the compressor
- The use of direct drive fan units. (Broken fan belts are a major source of service calls)
- The use of a compressor designed for heat pump applications. (An air conditioning compressor is not engineered to withstand the inherently higher operating stresses required for heat pump operation)
- The refrigerant lines are sized to insure continuous compressor oil return under any operating conditions which might be encountered in the Western Pacific Northwest.
- The system design is such that the compressor discharge temperature, which is a maximum under cold ambient conditions, is only 96.1 °C for operation when the ambient temperature is -12.2 °C (205 °F for an ambient temperature of 10 °F). (This is well below the temperature, approximately 139 °C (283 °F), at which refrigerant breakdown begins)
- A high and low pressure cutout to protect the compressor.
- A lockout switch to prevent short cycling
- A totally enclosed outdoor fan motor.

5.2 PERFORMANCE

The performance of the heating optimized heat pump is quite good. COPH and heating capacity curves as predicted by the Oak Ridge computer model are presented in Figure 5.2. The COPH values show a marked improvement over those of commercially available heat pumps:

	COPH	
	at 8.3 °C (47 °F)	at -8.3 °C (17 °F)
Heating-only unit	3.8	3.0
Average commercial unit	2.7	1.8
Best COPH performance commercial unit	3.1	2.4

It is important to note the values for the heating-only heat pump were predicted by the Oak Ridge computer model and the values for the commercial units were obtained using the American Refrigeration Institute (ARI) standard rating procedure. The ARI ratings, which involve a laboratory test, were not possible for the work described here because no prototype heating-only unit was built. However, it is not expected that the COPH values determined by this test would be much different than those calculated by the code. The expected difference would be at the low temperature rating point, -8.3 ° C (17 ° F), where defrosting losses would reduce the COPH value predicted by the Oak Ridge model.

The nominal heating capacity of the heating optimized design is 10.5 kW (35800 Btu/hr) at an 8.3 ° C (47 ° F) ambient air temperature. Using the linear heat loss rate for the representative residence as given by equation 4.1, the corresponding balance point temperature is -4.6 °C (23.7 °F). Note that this is several degrees below the

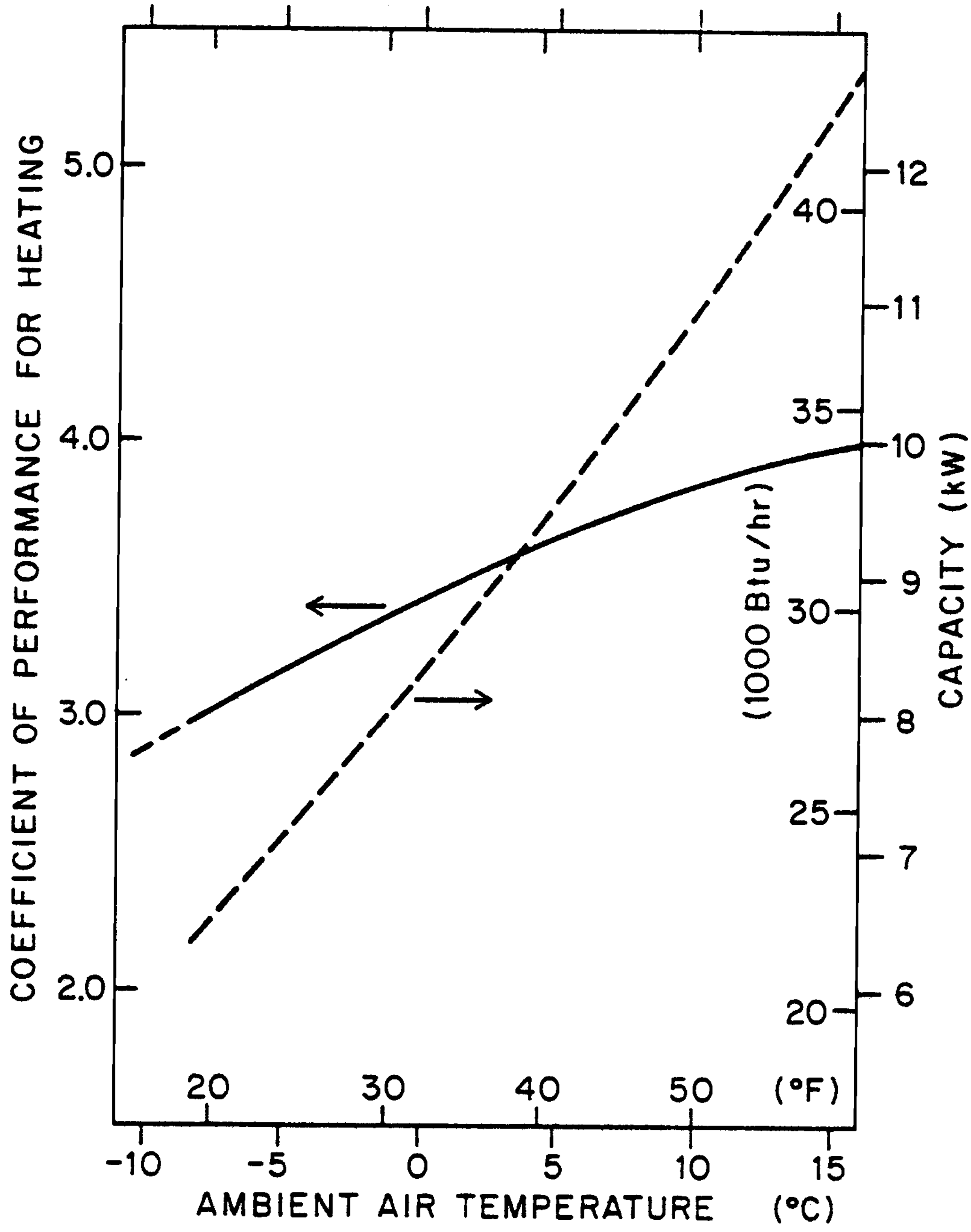


Figure 5.2 Heating capacity and COPH of optimized heating-only heat pump as a function of ambient temperature.

optimum range of balance point (-2 to 0 °C (28.2 to 32 °F)) as discussed in Section 4.2, This results in a decrease in SPFH of less than 3%. (Adjustment of sizing is addressed in Section 5.4).

The air delivery temperatures for the heating optimized unit are illustrated in Figure 5.3. As shown, the air delivery temperature at the balance point is 31.1 °C (88 °F) which is slightly less than the design specification of 32.2 °C (90 °F). This was due to the characteristics of the centrifugal fans and PSC motor considered for this design. The lowest speed PSC motor available is an eight pole motor which operates at 825 RPM and to increase the air delivery temperature would require a motor which would operate at a lower speed. A belt driven fan or some type of gearing system was not considered due to their inherent lower efficiencies. In any case, it is felt that a minimum air delivery temperature of 31.1 °C (88 °F) should be acceptable.

The SPFH values, as defined by Eq. (1.2), have been calculated for the heating-only heat pump, an average commercial heat pump and for the best performance commercial heat pump. The procedure for this calculation is described in Section 4.2. The results are:

	SPFH
Heating-only heat pump.................. (optimum sizing--balance point temperature of -1 °C (30.2 °F))	2.97
Heating-only heat pump.................. (sizing of finalized design as in Table 5.1--balance point temperature of -4.6 °C (23.7 °F))	2.89

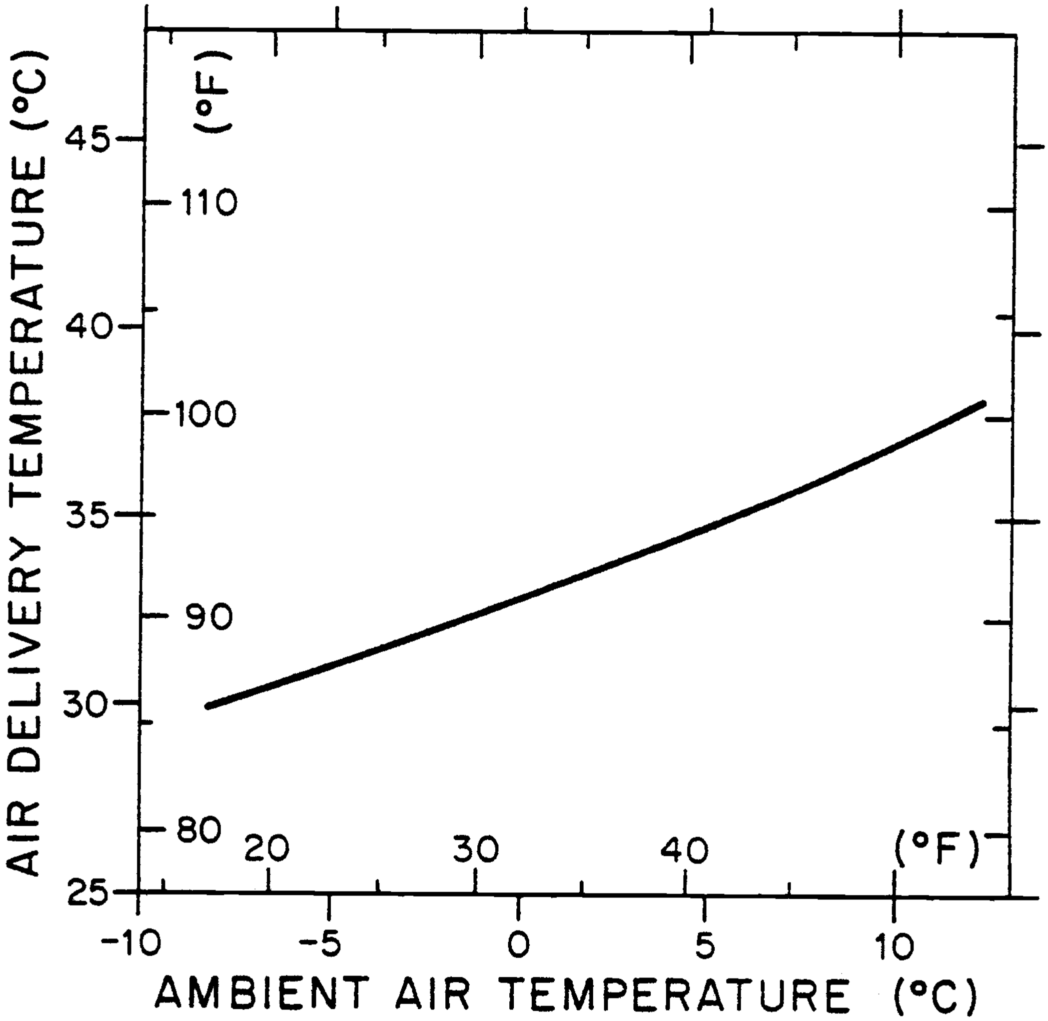


Figure 5.3 Air delivery temperature of the heating-only heat pump.

Average commercial heat pump............2.04
(average sizing--balance point
temperature of 2.22 °C (36 °F))

Best Performance commercial heat pump...2.33
(average sizing--balance point
temperature of 2.22 °C (36 °F))

In simple terms, these numbers indicate that over the heating season the heating-only heat pump, when it is sized for a balance point temperature of -1 °C (30.2 °F), will consume 67% less electrical energy than an electric resistance heating system, 31% less energy than an average commercial heat pump and 22% less energy than the "best performance" commercial heat pump for the heating season. When the electrical energy used by the commercial heat pumps for air conditioning is considered, the conservation potential of the heating-only heat pump is even more striking.

A detailed economic comparative type analysis was not carried out in this work. However, Table 5.2 presents some annual energy cost values intended to provide an estimate of how the proposed heating-only heat pump system compares with the alternatives.

In addition to the information already presented, Appendix C contains detailed results from the Oak Ridge heat pump model for the finalized design at ambient air temperatures of -8.33, -1.67, 4.83, 8.33 and 16 °C (17, 29, 40.7, 47, and 60.8 °F). These are included to provide guidance in the construction of a prototype unit.

Table 5.2 Annual Energy Costs of Residential Space Heating Systems*

System	Residence Annual Space Heating Energy Consumption kWh (10^6 Btu)	Electricity Consumed (kWh)	Fuel Consumed	Annual Energy Cost ($)
Heating-Only Heat Pump	14000 (47.78)	4714	0	$189
Average Commercial Heat Pump	14000 (47.78)	6863 to 8922**	0	$275 to 357**
Best Performance Commercial Heat Pump	14000 (47.78)	6008 to 7812**	0	$240 to 313**
Central Electric Furnace	14000 (47.78)	14286	0	$571
Natural Gas Furnace	14000 (47.78)	412	2.2×10^3 m^3 (7.8×10^4 ft^3)	$458
Oil Furnace	14000 (47.78)	355	2260 ℓ (597 gallons)	$651

* ASSUMPTIONS

Performance:

SPF Values for heat pump systems are: Heating-Only - 2.97; Average Commercial - 2.04; Best Performance Commercial - 2.33

Efficiency values for the furnaces are: Central Electric Furnace - 98%; Natural Gas Furnace - 65%; Oil Furnace - 60%

Energy Costs:

Oil - $0.282/ℓ ($1.066/gallon)
Natural Gas - $0.21/kWh ($0.60/10^5 BTU)
Electricity - $0.04/kWh

Heating Values of Fuels:

Oil - 10.3 kWh/ℓ (133,300 BTU/gal)
Natural Gas - 10.6 kWh/m^3 (1021 BTU/ft^3)

** In the larger of these values, the calculated electric consumption for the two commercial heat pump systems has been increased by 30% to provide an upper bound for the energy consumed by air conditioning.

5.3 DEFROSTING CONSIDERATIONS

The importance of a good defrosting system should not be underestimated; the defrosting system could be called the Achilles tendon of the heat pump system. Due to the transient nature of the defrosting cycle, proper design is quite important to protect the compressor from harmful operating conditions. Analysis, modeling and design of a defrosting system is made difficult by the lack of published literature on the topic. The engineers with the greatest wealth of knowledge, those in the commercial heat pump industry, typically do not publish. Furthermore, from a theoretical viewpoint, an analysis of frost formation and/or melting is a very complicated problem; it involves heat transfer, mass transfer, phase change phenomena, moving boundaries, a complex three-dimensional geometry which is continually changing and changing boundary conditions.

For the Western Pacific Northwest, where the winter weather is greatly influenced by the nearby Pacific Ocean, the warm winter air temperature and high relative humidities combine to create conditions which require a large number of defrosting cycles over the course of a winter. The trends in defrosting frequency as a function of air temperature and relative humidity are shown in Figure 5.4. As shown, when the ambient air temperature is -4 to 4 °C (25 to 39 °F) and the air is moist, the greatest number of defrost cycles are required.

Commercial heat pumps are defrosted almost exclusively using the reverse cycle defrost system. This is not surprising because this defrost system only requires the addition of the proper controls to

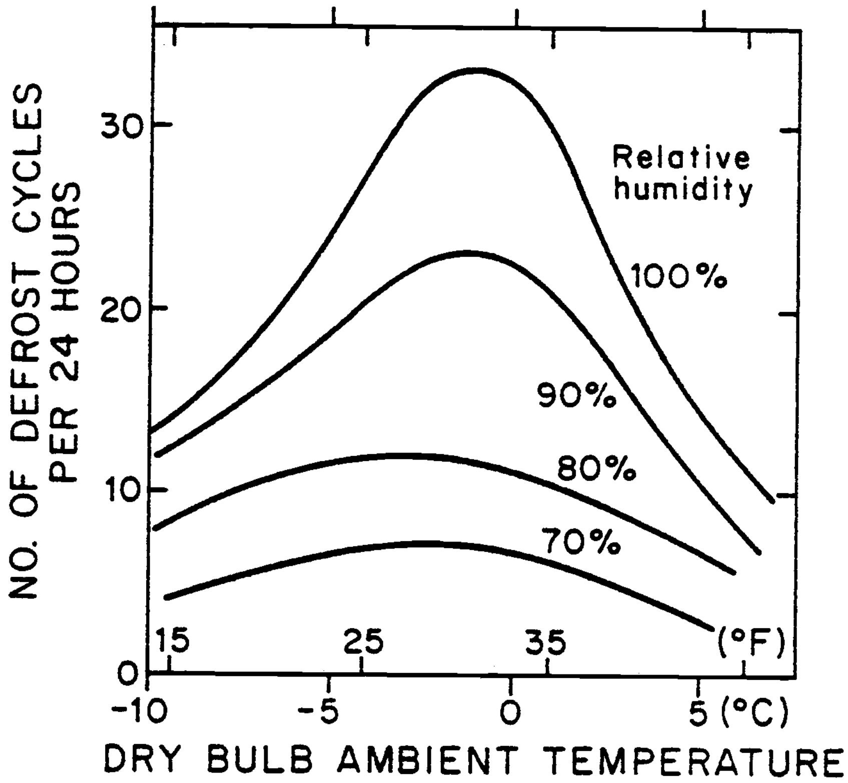


Figure 5.4 Defrost frequency as a function of air conditions (from Reay and MacMichael, 1979).

the heat pump system in order to function. However, reverse cycle defrosting has been cited as a major cause of compressor failure (Kirschbaum and Veyo, 1976). But, with proper system design, this defrosting system will function in an acceptable manner. Basically, this proper design consists of providing a bypass line to route refrigerant flow past the cooling mode expansion valve during the defrost cycle (Kirshbaum and Veyo, 1976; Young and Lange, 1980).

The only other defrosting system used on commercial residential heat pumps is the hot gas bypass system. This system, illustrated in Figure 5.5, functions by routing the compressor discharge gas directly into the frosted evaporator coil. This hot gas condenses, thus providing the heat energy required for defrosting, and is then drawn into the accumulator and finally back into the compressor. When the defrost cycle is initiated the compressor discharge line is suddenly connected to the low pressure evaporator coil. The three-way valve, shown in Figure 5.5, is used to prevent the high pressure refrigerant in the condenser coil from flowing into the compressor discharge line (Young and Lange, 1980).

There are two problems with hot gas bypass defrosting which should be mentioned: (i) care in system design must be taken to insure that liquid refrigerant levels in the compressor suction vapor remain low or damage to the compressor may result, and (ii) the time required to complete a defrost cycle can be substantially longer than the time required by the reverse cycle defrost system. This occurs because once defrosting is initiated the compressor suction vapor

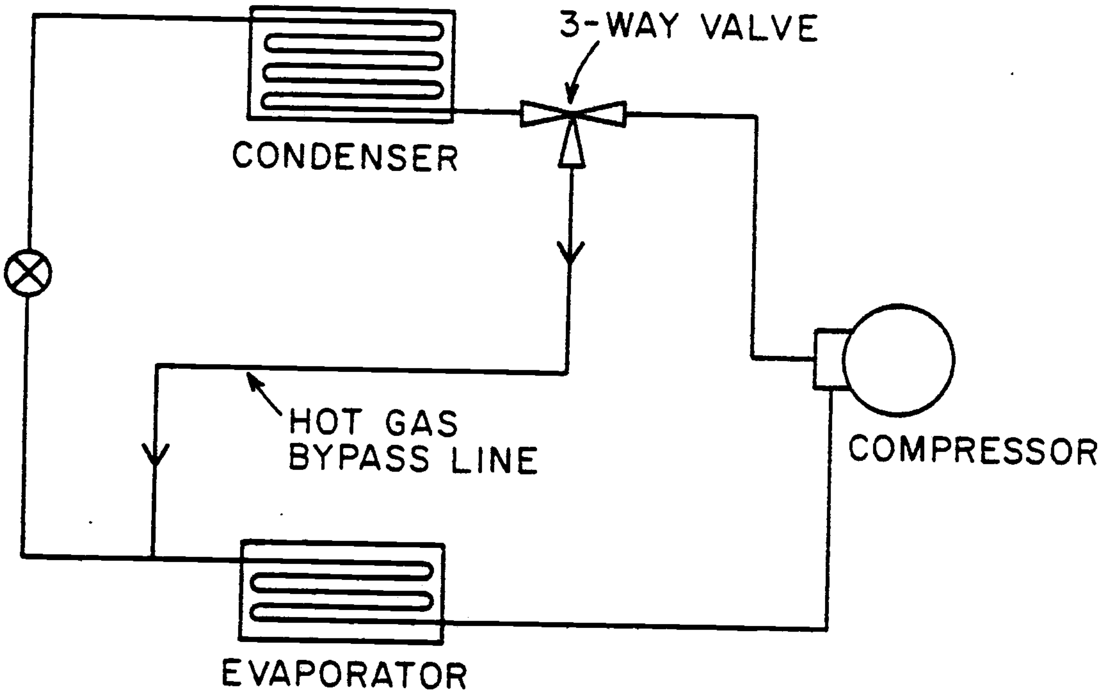


Figure 5.5 Hot gas bypass system.

quickly reaches the vapor pressure of the refrigerant at 0°C (32°F). Consequently, the refrigerant mass flow rate is low and the rate of heat transfer to the frost on the evaporator coil is low.

The Janitrol heating-only heat pump (see Chapter 2) and an experimental residential heating-only heat pump in Northern Ireland (Morgan and McMullan, 1980) both use a hot gas bypass system. Engineers involved in the design of both of these heat pumps were contacted and asked about this defrost system. It was reported that the Janitrol unit sometimes had problems with a long defrosting cycle and that the Northern Ireland units defrosting system functioned well. In Germany, hot gas bypass is the common defrost method, but most of their units are hybrid units which do not operate below about 0 °C (32 °F).

At this point, with the limited literature, it is not possible to determine if the hot gas bypass defrost system will function satisfactorily and hence the recommendation is to test a hot gas defrosting system on a prototype unit. If this system does not achieve acceptable defrosting times, then reverse cycle defrosting could be used. If this is the case, then it is recommended that four solenoid valves in a bridge arrangement be used in place of the reversing valve in order to avoid the performance penalties imposed by the reversing valve (Young and Lange, 1980). The unit should be configured such that it is still a heating-only heat pump.

5.4 SIZING FOR OTHER HEATING LOADS

For the work reported here, a specific residence was modeled and based on this "average house" an optimum sizing philosophy was developed and a heat pump with a specific heating capacity was designed. The question arises, what about other dwellings with different heating loads? Since the optimum sizing philosophy is characterized in terms of a balance point temperature and not tied to the specific heating requirements of a residence, the results of this work are valid for any residence under consideration. Thus, the remaining problem is how to use the results of the final design to configure other heat pump systems with different heating capacities in order to satisfy this sizing philosophy. A procedure for scaling a heat pump system such that the COPH values are unchanged but the heating capacity values are changed has been developed at Oak Ridge National Laboratory (Rice et al., 1981). Rather than repeating the procedure here, the interested reader is referred to this reference.

5.5 THE ADVANTAGES OF DESIGNING FOR HEATING ONLY

The objective of this section is to quantify the advantages of (i) designing for heating only and (ii) using improved performance components. All results reported in this section are for heat pump operation with an ambient air temperature of 8.3 °C (47 °F). The specific aspects which contribute to improved energetic performance are:

Sizing advantages of a heating-only unit

Recovery of discharge line heat losses

Recovery of compressor shell heat losses

No reversing valve

Optimum matching or components for heating

High efficiency fan units

High efficiency compressor

Sizing: As previously discussed, the current sizing practice, which for the Western Pacific Northwest is a compromise between the requirements of the heating and air conditioning season, results in a decreased seasonal performance. Since the heating-only heat pump's sizing can be based only on the heating load, it seems reasonable that the resulting advantages in performance can be credited to this unit. Thus, based purely on sizing considerations the heating-only unit can be expected to increase seasonal performance, SPFH, by an average of 3 to 7%.

Discharge Line Heat Losses: The refrigerant discharge line, which connects the outlet port of the compressor to the condenser coil, carries the highest temperature refrigerant in the system, typically above 93 ^{o}C (200 ^{o}F). For the commercial split system heat pump this line runs between the outdoor unit and the indoor unit and consequently heat energy is lost from the line to the ambient. For the proposed heating-only unit, this line is located entirely within the indoor unit and so any heat losses are recovered and used for heating. The Oak Ridge heat pump model was used to assess the effects of recovering this heat loss on COPH. An average discharge line heat loss, 0.437 kW (1494 Btu/hr), based on experimental data from four commercial heat pumps was used (Dabiri, 1982). The result is that by recovering this loss, the COPH for the heating-only heat pump was increased by 4.5%.

Compressor Shell Heat Losses: Recovering the heat energy lost from the compressor shell also increases the performance of the heating-only heat pump relative to commercial units. The shell heat loss rate will range from 10 to 40% of the compressor input power and for the commercial units may be lost to the ambient or added to the outdoor air before the air passes over the outdoor heat exchanger. Based on this range of heat losses, computer analysis was used to predict that by recovering the shell heat losses the COPH for the heating-only unit was increased by 2 to 9%.

Reversing Valve Losses: The reversing valve, which is used to switch the heat pump from the air conditioning to the heating cycle, is of course, not required by a heating-only heat pump. This valve decreases performance in several ways, it increases the pressure drop in both the suction and discharge lines and it allows heat transfer from the discharge line to the suction line. It has also been experimentally found in one study that a significant amount of refrigerant leaks from the high pressure side of this valve to the low pressure side which creates a relatively high performance degradation (Young and Lange, 1980). It is not known if this is the general case or the result of one bad valve. However, if it were the general case it would not be too surprising considering the stresses imposed on the reversing valve every time it is switched to initiate a defrost cycle.

The reversing valve typically creates a pressure drop of 6.9 to 20.7 kPa (1 to 3 psi) in both the suction and discharge lines and increases the discharge line heat loss by 17 to 25% (Dabiri, 1982). For this work, a pressure drop of 10.3 kPa (1.5 psi) and a heat loss rate of 21% were assumed. Then, based primarily on the magnitude of the refrigerant leak rate, the COPH of the heating-only heat pump is increased 1.4 to 10% by the elimination of this valve.

Optimum Matching of Components for Heating: There are other performance advantages of the heating-only heat pump, as compared to the conventional reverse cycle units, which are implicit within the design proposed here. These appear because all design decisions were made with respect to heating with no restrictions for cooling. These have resulted in increased air side heat transfer area of the condenser, smaller fans, and fewer evaporator-coil tube rows. The improvement from these factors can be as much as 20%.

Improved Efficiency Components: The benefits of improve efficiency components has also been estimated. The improved performance compressor results in up to a 15% increase in COPH and the improved performance fan/motor units result in a 5 to 10% increase in COPH.

A summary of the improvements discussed in the section is presented in Table 5.3.

Table 5.3 Summary of Performance Improvements for the Heating-only Heat Pump.

DESIGN OF UNIT:

	Increase in COPH (at 8.3 °C (47 °F))*
Recovery of discharge line heat losses	4.5%
Recovery of compressor shell heat losses	2-9%
No reversing valve	1.4-10%
Improved efficiency compressor	5-15%
Improved efficiency fan units	5-10%
Optimum component matching for heating • condenser sizing • fan sizing • fewer evaporator-coil tube rows	up to 20%

SIZING RELATIVE TO SPACE LOAD:

Lower balance point temperature appropriate to heating-only unit	SPFH is increased by 3-7%

* The percentage increase in COPH values are not intended to be summed and multiplied times the COPH value for a commercial heat pump. Rather, they are estimates of the merits of each of the design aspects considered. Obviously, the better performance commercial heat pumps will have incorporated some of these advantages to a certain extent.

5.6 CLOSURE

The optimized heating-only air-to-air heat pump design appears quite promising in terms of improved efficiency and potential for high reliability. Introduction of this type of unit would be expected to save substantial amounts of electricity in space heating applications in the Western Pacific Northwest. Thus, in the opinion of the author of this thesis, the development of this unit should be pursued.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

The objective of the work described in this thesis was to design a heat pump to be used for residential space heating in a climatic region exemplified by the Western Pacific Northwest. This has been done using a heat pump simulation model together with a nonlinear optimization algorithm. This chapter summarizes the conclusions of the design work and gives recommendations for future work on the heating-only heat pump.

6.1 CONCLUSIONS

(i) The basic philosophy of the design, to optimize the energy performance of the heat pump while constraining the system first cost, is the most logical design approach to take.

(ii) On a steady state efficiency basis, the performance of the proposed heating-only heat pump is substantially above that of present reverse cycle heat pump designs. The design described here is estimated to have COPH values 18.5% above those of the best commercially available heat pump units and 35.1% above that of an average commercial unit.

(iii) On a seasonal efficiency basis, the heating-only heat pump will use 67% less electrical energy than an electric resistance heating system, 31% to 42% less than an average commercial heat pump and 21% to 32% less than a state of the art commercial heat pump.

(iv) The first cost of the heating only heat pump will be about midway between that of an average commercial heat pump and that of a "high efficiency" heat pump.

(v) The high efficiency of the heating-only heat pump as compared to commercial reverse cycle units results from:

Optimization of the system component matching for heating.

No reversing valve in the system.

Recovery of the discharge line heat losses.

Recovery of the compressor shell heat losses.

Smaller fan motors for both the condenser and the evaporator.

Use of high efficiency fans and fan motors.

Use of a high efficiency compressor.

(vi) For the Western Pacific Northwest's heating season the present sizing practice results in a heat pump which is not the most efficient from an energy conservation viewpoint. In general, the small air conditioning load requires a unit which is smaller than a sizing based only on the heating load would indicate. However, the heating-only unit is not constrained by such a requirement.

(vii) The optimum sizing for the heating-only heat pump is one which yields a balance point temperature of -2 to 0 °C (28.4 to 32°F)

(viii) The minimum air delivery temperature to insure comfort to the occupants of a space is somewhat uncertain. For the design presented here the minimum air delivery temperature (above the balance point temperature) is 31.1 °C (88 °F).

(ix) The final selection of a defrosting system will have to result from the testing of a prototype unit. If the recommended system, hot gas bypass defrost, does not achieve acceptable defrosting times; a reverse cycle system using four solenoid valves in place of a reversing valve should be used.

6.2 RECOMMENDATIONS

The design and related work described in this thesis was the result of approximately five man months of effort. This was not a great deal of time and consequently it was not possible to analyze a number of identified tasks. The tasks recommended for future study are:

(i) A study of alternative refrigerants; note that R-12 and R-502 are better low temperature refrigerants than the refrigerant, R-22, proposed for this design.

(ii) An assessment of the energetic advantages and/or disadvantages of using a capillary tube or a short tube orifice as the flow control device. Note that a capillary tube will allow some liquid refrigerant into the suction vapor when the evaporating temperature is low. This results in better compressor cooling and hence increased reliability.

(iii) A study of the benefits of using spine finned heat exchangers. This type of heat exchanger has been indicated to be more efficient than the more conventional finned tube heat exchanger.

(iv) An acoustic analysis and design of the indoor unit.

(v) The design of a control system. A very important part of this is the design of a reliable demand defrost system.

REFERENCES CITED

Abadie, J. and J. Carpentier, "Generalization of the Wolfe Reduced Gradient Method to the Case of Nonlinear Constraints," in Optimization, edited by R. Fletcher, London: Academic Press, 1969.

ASHRAE Handbook of Fundamentals, ASHRAE, Atlanta, 1981.

Baxter, V.D., L.A. Abbatiello, and R.E. Minturn, "Comparison of Field Performance for Two Dealer-Installed Air-to-Air Heat Pumps," ASHRAE Transactions, 1982, V. 87, Pt. 2.

Bazarra, Mokhtar S., and C.M. Shetty, Nonlinear Programming: Theory and Algorithms, New York: Wiley, 1979.

Blundell, C.J., "Optimizing Heat Exchangers for Air-to-Air Space-Heating Heat Pumps in the United Kingdom," International J. of Energy Research, Vol. 1, 69-94, John Wiley & Sons, Ltd., 1977.

Blundell, C.J., R.D. Heap, and E.G.A. Goodall, "Heat Pumps for Space Heating in the UK Research and Application," Conference, Electric Space Heating and Air Conditioning, UNIPEDE/UIE, Bordeaux, October 1977, paper 2.5.2.

Bullock, C. and W. Reedy, "Heat Pump Cyclic Performance and its Influence on Seasonal Operation," Proceedings of 3rd Annual Heat Pump Technology Conference, Oklahoma State University, Stillwater, Oklahoma, April 10-11, 1978.

Carrington, C.G., "Optimizing a Heat Pump for Heating Purposes," International Journal of Energy Research, Vol. 2, No. 2, April-June 1978.

Dabiri, A.E., "A Steady State Computer simulation Model for Air-to-Air Heat Pumps," ASHRAE Transactions, V. 88, Pt. 2, 1982.

Elger, Donald F. and Gordon M. Reistad, "Electric Heat Pump Systems for Western Pacific Northwest Applications: Review and Recommendations," Prepared for the USDOE-Bonneville Power Administration by Oregon State University. November, 1981 (Unpublished).

Fischer, S.K. and C.K. Rice, A Steady-State Computer Design Model for Air to Air Heat Pumps, ORNL/CON-80, Dec., 1981.

Gabriele, G.A. and K.M. Ragsdell, "The Generalized Reduced Gradient Method: A Reliable Tool for Optimal Design," Journal of Engineering for Industry, 99 (May 1977): 394-400.

Goldschmidt, V.W., G.H. Hart, and R.C. Reiner, "A Note on the Transient Performance and Degradation Coefficient on a Field Tested Heat Pump-Cooling and Heating Mode," ASHRAE Transactions, 1980, V. 86, Pt. 2.

Gordian Associates, Heat Pump Technology, A Survey of Technical Developments, Market Prospects, and Research Needs, Report HCP/M2121-01 for the U.S. Department of Energy, June 1978.

Groff, C.G., "Heat Pumps in the USA, 1950-1990," The Electric Heat Pump, German/American Conference on Technology and Applications, Dusseldorf, June 18-20, 1980.

Heap , R.D., and C.J. Blundell, "The Use of Heat Pumps for Domestic Heating," IFF/IIR Commission E1, Belgrade, November 1977.

Heap , R.D., "Development of a Domestic Heating-Only Heat Pump," Proceedings of 15th International Congress of Refrigeration, Venice, September 1979.

Himmelblau, D.M. Applied Nonlinear Programming, New York: McGraw Hill Book Company, 1972.

Kelly, G.E., and J. Bean, Dynamic Performance of a Residential Air-to-Air Heat Pump, NBS Building Science Series, 93, March 1977.

Kirschbaum, H.S., and S.E. Veyo, An Investigation of Methods to Improve Heat Pump Performance and Reliability in a Northern Climate, EPRI Report No. EM-319, 1977.

Martin, J.K., and D.L. O'Neal, Energy and Cost Analysis of Residential Heat Pumps in Northern Climates, Oak Ridge National Laboratory, Contract W-7405-ENG-26, 1980.

McQuiston, Faye C., "Finned Tube Heat Exchangers: State of the Art for the Air Side," ASHRAE Transactions, Vol. 87, Part 1, 1981.

McQuay Group, "OEM Coil Engineering Manual," McQuay, Perfex, Inc., Minneapolis, MN, 1978.

Morgan, R. and J.T. McMullan, "Operating Characteristics of Air-to-Water Heat Pumps in Belfast during the Winters of 1978/79 and 1979-80," (Paper presented at International Conference: Copenhagen, April 1980), New University of Ulster: Coleraine, N. Ireland, 1980.

Parken, W.H., G.E. Kelly, and D.A. Didion, "Method of Testing, Rating and Estimating the Heating Seasonal Performance of Heat Pumps," NBS Report NS.BSIR 80-2002, 1980.

Parken, W.H., R.W. Beausoliel, and G.E. Kelly, "Factors Affecting the Performance of a Residential Air-to-Air Heat Pump," ASHRAE Trans., Vol. 83, No. 1, pp. 839-850, 1977.

Reay, D.J. and D.B.A. MacMichael, Heat Pumps: Design and Application, Pergamon Press, Oxford, 1979.

Reistad, Gordon M. and Donald F. Elger "Final Report: Heating Only Heat Pumps for the Western Pacific Northwest," Prepared for the U.S.D.O.E. - Bonneville Power Administration by Oregon State University (Unpublished).

Rice, C.K., S.K. Fischer, R.D. Ellison, and W.L. Jackson, Design Optimization of Conventional Heat Pumps: Application to Steady-State Heating Efficiency, Oak Ridge National Lab., Tennessee, Contract W-7405-ENG-26, 1981.

Torin Corporation, "Torin A/N/B Series Fans," Torrington, CT.

Young, D.J., "Development of a Northern Climate Residential Air Source Heat Pump," ASHRAE Transactions, Vol. 86, Part 1, 1980.

Young, D.J., and H.F. Lange, Optimization and Evaluation of a Northern Climate Residential Air-Source Heat Pump, Final Report for Canadian Electrical Association, October 1980.

APPENDICIES

APPENDIX A. THE OPTIMIZATION COMPUTER MODEL

```
      PROGRAM MAXCOP(INPUT,OUTPUT,TAPE5,TAPE6,TAPE7,TAPE8)
C
C   DRIVER PROGRAM FOR HEAT PUMP OPTIMIZATION USING OPT AND ORNL
C
      COMMON /A1/ NF,NC
      COMMON /DON/ D(214)
      COMMON / PARI / CRIT, EPS, IPR, MAXM, IDATA, NE,
     1                NI, LBD, NCON, EPSLS, EPSBD
      DIMENSION X(6), XMAX(6), XMIN(6)
      DATA N, NE, NI, IPR, MAXM, IDATA, LBD / 6,1,1,-1,50,1,0/
      DATA CRIT, EPSLS, EPSBD, EPS / 0.01, 0.0005, 0.02, 0.035/
      NF=0
      NC=0
C
C   DEFINE THE UPPER, NOMINAL AND LOWER BOUNDS OF THE DESIGN VARIABLES
C
      DATA (XMAX(I),I=1,6) /1.667, 1.85, 1.667, 1.75, 2.78,
     1 5.1 /, (XMIN(I),I=1,6) /0.3, 0.436, 0.332, 0.50, 0.67,
     2  0.326/, (X(I),I=1,6) / 6*1.00 /
C
C   CALL SUBROUTINE INTERF TO READ IN THE ORNL INPUT FILE
C
      CALL INTERF(X,0)
      CALL OPT (X, XMAX, XMIN, N)
      STOP
      END
C
C   CONSTRAINT FUNCTIONS EVALUATION
C
      SUBROUTINE CONST(X,CON)
      COMMON /A1/ NF,NC
      COMMON / INITIAL / BASE(6), FANETA
      DIMENSION X(6), CON(2)
      NAMELIST / FAN / SPECIF, HCTOP, HCBTM, FANETA,
     1                 CON,DELTAP,DIAMMI,DIAMMX,FLOW
      DATA RPM /1000.0/
      NC=NC+1
C
C   HEAT EXCHANGER AREA CONSTRAINT
C
      CON(1) = 37.18 - X(6)*BASE(6)*X(3)*BASE(3) - X(2)*BASE(2)*4.0
C
C   CALCULATE THE EVAPORATOR FAN SPECIFIC SPEED
C   NOTE: DELTAP IS AN APPROXIMATION TO THE AIR SIDE PRESSURE DROP FOR
C   THE EVAPORATOR (IN H2O)
C
      VELOC=X(4)*BASE(4)/(X(6)*BASE(6))
      XNTE2=(X(3)*BASE(3))/2.0
      DELTAP = 6.2190E-6*VELOC**1.7*XNTE2**0.7
```

```
      SPECIF = 4.3252E-5 * RPM * SQRT(X(4)*BASE(4)) / DELTAP**0.75
      IF (SPECIF.LE.25.0 .AND. SPECIF.GE.4.0) GO TO 200
      IF (SPECIF.LE.4.0) CON(2)=SPECIF - 4.0
      IF (SPECIF.GE.25.0) CON(2) = 25.0 - SPECIF
      GO TO 300
200   CONTINUE
C
C   CALCULATE EVAPORATOR FAN EFFICIENCY
C
      FANETA = 0.789191*EXP(-0.08047*SPECIF)
C
C    CALCULATE THE TOP AND BOTTOM OF THE FAN PERFORMANCE BAND
C    FOR AN PROPELLER FAN, (HCTOP AND HCBTM), FOR THE GIVEN
C    SPECIFIC SPEED.  NOTE THAT DIAMMI AND DIAMMX ARE THE MINIMUM
C    AND THE MAXIMUM FAN DIAMETERS WHICH CAN BE USED.
C
      HCTOP = 0.041118*EXP(-0.135223*SPECIF)
      HCBTM = 0.088050*SPECIF**(-1.107977)
      DIAMMI = SQRT(2.7855E7*DELTAP/HCTOP)/RPM
      DIAMMX = SQRT(2.7855E7*DELTAP/HCBTM)/RPM
      DELTA1 = SPECIF - 4.0
      DELTA2 = 16.0 - SPECIF
      CON(2) = AMIN1 (DELTA1, DELTA2)
      FLOW = X(4)*BASE(4)
300   WRITE (8,FAN)
      RETURN
      END
C
C    OBJECTIVE FUNCTION EVALUATION
C
      FUNCTION F(X)
      DIMENSION X(6)
      COMMON /A1/ NF,NC
      COMMON / PRNT8 / EINDF, EOUTF, POW2, RESIST, COP, DP,SS,
     &                  COPHP, QAIR,  FANOUT
      COMMON / EVAPTR / TAIIE,  TIE, TSATEI, HIE, PIE, XIE,
     &                   TAOE,  TROE, TSATEO, HOE, POE, XOE
      NAMELIST /RESULTS/ COPHP,QAIR,COPADJ,TAIIE,OBJECT
      NF = NF + 1
      CALL INTERF(X,1)
      CALL ORNL
C
C    ADJUST COPH VALUE TO ONE CORRESPONDING TO 40.7 F AIR TEMP.
C
      COPADJ = COPHP + 0.02667*(40.7 - TAIIE)
      OBJECT = COPADJ/3.44
      F = -OBJECT
      PRINT RESULTS
      WRITE(8,RESULTS)
      CALL INTERF(X,2)
      RETURN
      END
```

```
C
C   SUBROUTINE INERF INTERFACES BETWEEN OPT AND THE OAK RIDGE PROGRAM.
C   THE VALUE OF NTASK DETERMINES WHICH TASK THE SUBROUTINE DOES
C
      SUBROUTINE INTERF(XDES,NTASK)
C
C   ORNL COMMON BLOCKS AND SPECIFICATION STATEMENTS
C
      LOGICAL PRINT
      REAL L,NTE,NSECTE,NTC,NSECTC
      DIMENSION ERRMSG(3)
      COMMON / AA1 / PRINT
      COMMON / COMPR / VR,     SYNC,    FLMOT,   EFFMMX, ETAISN, ETAMEC,
     &                 ETAVLA, ETAVLB, POW,     CANFAC, HILOFC, QCAN,
     &                 QHILO,  DISPL, MTRCLC,  NSPEED
      COMMON / CONDEN /  DEAC,   DERC, DELTAC,    FPC,  XKFC,   AAFC,
     &                   NTC, NSECTC, HCONTC,     STC,    WTC,  SIGAC,
     &                    PC,  ARFTC,  ARHTC, ALFARC, ALFAAC,   FARC,
     &                  CARC,    QAC, RTBCND,     DZC, FANEFC,   RHIC,
     &                FINTYC, MUNITC
      COMMON / CONDS / HAC, SEFFXC, XMAC,    QC, PDAIRC, PDC,
     &                HSPC,   QSPC, FSPC, CPSPC, CPSP,
     &                HTPC,   QTPC, FTPC,
     &                HSCC,   QSCC, FSCC, CPSCC,
     &               TRVDS
      COMMON / CONDSR / TAIIC,  TIC, TSATCI, HIC, PIC, XIC,
     &                  TAOC,  TROC, TSATCO, HOC, POC, XOC
      COMMON /EVAPOR /  DEAE,  DERE,DELTAE,    FPE,  XKFE,  AAFE,   NTE,
     &                NSECTE,HCONTE,   STE,    WTE, SIGAE,     PE, ARFTE,
     &                 ARHTE,ALFARE,ALFAAE,  FARE,  CARE,   QAE,RTBEVP,
     &                   DZE,FANEFE,  RHIE, FINTYE, MUNITE
      COMMON / EVAPS / HAE, XMAE,  QE, PDAIRE, PDE,
     &                HSPE, QSPE, FSPE, CPSPE,
     &                HTPE, QTPE, FTPE
      COMMON / EVAPTR / TAIIE,  TIE, TSATEI, HIE, PIE, XIE,
     &                  TAOE,  TROE, TSATEO, HOE, POE, XOE
      COMMON / FANMOT / COFAN, C1FAN, C2FAN,.EFFMOT, ETAS, RPMFAN
      COMMON / FLOWBA / DTROC, SUPER, CAPFLO, ORIFD, XMR, NCAP,
     &                  IREFC, ICOMP
      COMMON / MPASS / CNDCON, AMBCON, EVPCON, CONMST, CMPCON, FLOCON,
     &                 TOLS,   TOLH,   LPRINT, NCORH,  MCMPOP, MFANIN,
     &                 MFANOU, MFANFT
      COMMON / PRNT8 / EINDF, EOUTF, POW2, RESIST, COP, DP,SS,
     &                 COPHP, QAIR,  FANOUT
      COMMON / REFRIG /  NR
      COMMON  DDUCT, FIXCAP, ITITLE(20)
      EXTERNAL CNDNSR, EVPTR
C
C   END ORNL SPECIFICATION STATEMENTS
C
      COMMON / PARI / CRIT, EPS, IPR, MAXM, IDATA, NE,
     1                NI, LBD, NCON, EPSLS, EPSBD
```

```
      COMMON /INITIAL/ BASE(6), FANETA
      DIMENSION XDES(6),CON(3)
      CHARACTER*10 CONTRL
      NAMELIST /ZAP/ NSPEED,CRIT,EPSLS,EPS
      NAMELIST /DESIGN/ AAFE,AAFC,NTE,QAE,QAC,DTROC
C
C   IF NTASK EQUALS ZERO, READ THE ORNL INPUT DATA, INITIALIZE
C   ORNL VARIABLES AND INITIALIZE OPT'S DESIGN VARIABLE ARRAY.
C
      IF (NTASK.EQ.0) THEN
      CALL DATAIN
      CALL TABLES(NR)
      PDAIRE = 0.0
      NSPEED = 1.0
C
C   DEFINE THE VARIABLES TO BE OPTIMIZED AND CALL SUBROUTINE
C   CONST TO CALCULATE THE INITIAL FAN EFFICIENCY.
C
      BASE(1) = DTROC
      BASE(2) = AAFC
      BASE(3) = NTE
      BASE(4) = QAE
      BASE(5) = QAC
      BASE(6) = AAFE
      CALL CONST(XDES,CON)
C
C    IF NTASK.EQ.1 SET THE ORNL VARIABLES TO VALUES DETERMINED BY OPT
C
      ELSE IF (NTASK.EQ.1)  THEN
      DTROC = XDES(1)*BASE(1)
      AAFC = XDES(2)*BASE(2)
      NTE  = XDES(3)*BASE(3)
      QAE  = XDES(4)*BASE(4)
      QAC  = XDES(5)*BASE(5)
      AAFE= XDES(6)*BASE(6)
C
C   RESET DESIRED AMBIENT AIR TEMPERATURE
C
      TAIIE=40.7
      FANEFE = FANETA*EFFMOT
      WRITE(6,DESIGN)
      PRINT*,'ORNL CALLED-VALUES OF DESIGN VARS='
      PRINT DESIGN
C
C   IF NTASK.EQ.2, ALLOW THE USER TO CONTINUE THE RUN,
C   STOP THE RUN OR CHANGE THE CONTROL VARIABLES
C
      ELSE IF (NTASK.EQ.2) THEN
      PRINT*,'INPUT STOP, GO OR CHANGE'
      READ*,CONTRL
      IF(CONTRL.EQ.'STOP') STOP
      IF (CONTRL.EQ.'CHANGE') THEN
```

```
PRINT ZAP
PRINT*,'INPUT CHANGES IN NAMELIST FORMAT'
READ (*,ZAP)
WRITE(7,ZAP)
ENDIF
ENDIF
RETURN
END
```

APPENDIX B. SAMPLE INPUT DATA

```
RUN3F: OPTIMUM CONFIG., 40.7 F AMBIENT: TECH AV5532E COMP
  20000.0      8.00     9.413
        2         1         3         1         1         0
        0      17.1
   0.3725     30.00      0.00    350.00    350.00
    0.785      29.9    0.4300       2.9
    0.785       0.1    0.4300       0.1
   22.427    101.42    312.07
        2     3.516   3450.00       0.0     0.150
-1.575E-4 3.5758E-2 2.2140E-6 -3.328-02 4.9446E-4 -1.292E-1     3.516      20.0
-3.739E-2 6.9513E00 9.0037E-3 1.2808E01 -4.351E-2 -1.987E02
      2.0    0.2970    0.2650     0.006      16.0     137.0     4.774
    4.000     3.000
  30000.0      1.00     .6250
   1120.0      68.0     0.500    124.00     0.225
      2.0    0.2970    0.2650     0.006       8.0     132.0    9.9860
    2.000     7.000
 30000.00       1.0     0.625
   2310.0     40.70     0.750     76.00    0.1340
```

```
***** INPUT DATA *****
     RUN3F: OPTIMUM CONFIG., 40.7 F AMBIENT: TECH AV5532E COMP
     HEATING MODE OF OPERATION (NCORH=2)

     THE HOUSE LOAD IS  20000. BTU/H
     DIAMETER OF 6 EQUIVALENT DUCTS- 8.000 IN
     POWER TO THE INDOOR FAN ADDED TO AIR BEFORE CROSSING THE COIL.
     POWER TO THE OUTDOOR FAN ADDED TO AIR BEFORE CROSSING THE COIL.
     CONDENSER SUBCOOLING IS HELD FIXED AT   17.10 F
     EVAPORATOR SUPERHEAT IS HELD FIXED AT    9.41 F

     DESCRIPTION OF CONNECTING TUBING:
       LIQUID LINE FROM INDOOR TO OUTDOOR HEAT EXCHANGER
           ID                          .37250 IN
           EQUIVALENT LENGTH            30.00 FT
       FROM INDOOR COIL TO REVERSING VALVE            FROM OUTDOOR COIL TO REVERSING VALVE
           ID                          .43000 IN              ID                        .78500 IN
           EQUIVALENT LENGTH             2.90 FT              EQUIVALENT LENGTH          29.90 FT
       FROM REVERSING VALVE TO COMPRESSOR INLET       FROM REVERSING VALVE TO COMPRESSOR OUTLET
           ID                          .78500 IN              ID                        .43000 IN
           EQUIVALENT LENGTH              .10 FT              EQUIVALENT LENGTH            .10 FT

       HEAT LOSS IN DISCHARGE LINE        0.0 BTU/H
       HEAT GAIN IN SUCTION LINE        350.0 BTU/H
       HEAT LOSS IN LIQUID LINE         350.0 BTU/H

     ESTIMATE OF:
       REFRIGERANT MASS FLOW RATE                  312.070 LBM/H
       SATURATION TEMPERATURE INTO CONDENSER       101.420 F
       SATURATION TEMPERATURE OUT OF EVAPORATOR     22.427 F

     COMPRESSOR CHARACTERISTICS:
       TOTAL DISPLACEMENT            3.516 CUBIC INCHES
       SYNCHRONOUS MOTOR SPEED 3450.000 RPMS

     MAP-BASED COMPRESSOR INPUT:
       POWER CONSUMPTION=    -1.575E-04*CONDENSING  TEMPERATURE**2 +  3.576E-02*CONDENSING  TEMPERATURE
                           +  2.214E-06*EVAPORATING TEMPERATURE**2 + -3.328E-02*EVAPORATING TEMPERATURE
                           +  4.945E-04*CONDENSING  TEMPERATURE*EVAPORATING TEMPERATURE + -1.292E-01

       MASS FLOW RATE=       -3.739E-02*CONDENSING  TEMPERATURE**2 +  6.951E+00*CONDENSING  TEMPERATURE
                           +  9.004E-03*EVAPORATING TEMPERATURE**2 +  1.281E+01*EVAPORATING TEMPERATURE
                           + -4.351E-02*CONDENSING  TEMPERATURE*EVAPORATING TEMPERATURE + -1.987E+02

       CORRECTION FACTOR FOR SUCTION GAS                  .330
       CORRECTION FACTOR FOR VOLUMETRIC EFFICIENCY        .750
       BASE SUPERHEAT FOR COMPRESSOR MAP                20.000 F
       BASE DISPLACEMENT FOR COMPRESSOR MAP              3.516 CU IN

       HEAT REJECTED FROM COMPRESSOR SHELL IS  .15 TIMES THE COMPRESSOR POWER
```

INDOOR UNIT: CONDENSER

OD OF TUBES IN HX	.29700	IN
ID OF TUBES IN HX	.26500	IN
FRONTAL AREA OF HX	4.774	SQ FT
NUMBER OF PARALLEL CIRCUITS	3.00	
NUMBER OF TUBES IN DIRECTION OF AIR FLOW	4.00	
NUMBER OF RETURN BENDS	124.00	
AIR FLOW RATE	1120.00	CFM
INLET AIR TEMPERATURE	68.000	F
WAVY FINS		
FIN THICKNESS	.00600	IN
FIN PITCH	16.00	FINS/IN
THERMAL CONDUCTIVITY OF FINS	137.000	BTU/H-FT-F
CONTACT CONDUCTANCE	30000.0	BTU/H-SQ FT-F
HORIZONTAL TUBE SPACING	.625	IN
VERTICAL TUBE SPACING	1.000	IN
FAN EFFICIENCY	.22500	
RELATIVE HUMIDITY	.50000	

OUTDOOR UNIT: EVAPORATOR

OD OF TUBES IN HX	.29700	IN
ID OF TUBES IN HX	.26500	IN
FRONTAL AREA OF HX	9.986	SQ FT
NUMBER OF PARALLEL CIRCUITS	7.00	
NUMBER OF TUBES IN DIRECTION OF AIR FLOW	2.00	
NUMBER OF RETURN BENDS	76.00	
AIR FLOW RATE	2310.00	CFM
INLET AIR TEMPERATURE	40.700	F
WAVY FINS		
FIN THICKNESS	.00600	IN
FIN PITCH	8.00	FINS/IN
THERMAL CONDUCTIVITY OF FINS	132.000	BTU/H-FT-F
CONTACT CONDUCTANCE	30000.0	BTU/H-SQ FT-F
HORIZONTAL TUBE SPACING	.625	IN
VERTICAL TUBE SPACING	1.000	IN
FAN EFFICIENCY	.13400	
RELATIVE HUMIDITY	.75000	

CALCULATED HEATING-ONLY HEAT PUMP PERFORMANCE AT -8.3°C (17°F)

APPENDIX C. RESULTS OF THE OAK RIDGE HEAT PUMP SIMULATION MODEL FOR AMBIENT AIR TEMPERATURES OF -8.3, -1.7, 4.83, 8.3 and 16 oC

***** CALCULATED HEAT PUMP PERFORMANCE *****

COMPRESSOR OPERATING CONDITIONS:

		EFFICIENCY	
COMPRESSOR POWER	1.730 KW	VOLUMETRIC	.6262
MOTOR SPEED	3450.000 RPM	OVERALL	.5572

REFRIGERANT MASS FLOW RATE	196.083 LBM/H	POWER PER UNIT MASS FLOW	30.24001 BTU/LBM
COMPRESSOR SHELL HEAT LOSS	885.825 BTU/H	POWER CORRECTION FACTOR	.9966
		MASS FLOW RATE CORRECTION FACTOR	1.0007

SYSTEM SUMMARY	REFRIGERANT TEMPERATURE	SATURATION TEMPERATURE	REFRIGERANT ENTHALPY	REFRIGERANT QUALITY	REFRIGERANT PRESSURE	AIR TEMPERATURE
COMPRESSOR SUCTION LINE INLET	12.143 F	4.018 F	106.161 BTU/LBM	1.0000	42.030 PSIA	
SHELL INLET	23.224	3.673	107.946	1.0000	41.732	
SHELL OUTLET	199.857	89.652	133.546	1.0000	182.188	
CONDENSER INLET	199.834 F	89.580 F	133.546 BTU/LBM	1.0000	182.000 PSIA	68.000 F
OUTLET	77.690	87.725	32.413	0.0000	177.226	85.501
EXPANSION DEVICE	71.685 F	87.647 F	30.628 BTU/LBM	0.0000	177.027 PSIA	
EVAPORATOR INLET	6.561 F	6.561 F	30.628 BTU/LBM	.1986	44.278 PSIA	17.000 F
OUTLET	12.143	4.018	106.161	1.0000	42.030	12.199

PERFORMANCE OF EACH CIRCUIT IN THE CONDENSER

INLET AIR TEMPERATURE	68.000 F
HEAT LOSS FROM FAN	839.6 BTU/H
AIR TEMPERATURE CROSSING COIL	69.401 F
OUTLET AIR TEMPERATURE	85.501 F

TOTAL HEAT EXCHANGER EFFECTIVENESS .7309

	SUPERHEATED REGION	TWO-PHASE REGION	SUBCOOLED REGION
NTU	1.1076	1.5684	1.1060
HEAT EXCHANGER EFFECTIVENESS	.5183	.7916	.5479
CR/CA	.6978		.7001
FRACTION OF HEAT EXCHANGER	.0441	.8869	.0690
HEAT TRANSFER RATE	872.8 BTU/H	5538.0 BTU/H	199.3 BTU/H
OUTLET AIR TEMPERATURE	117.088 F	84.610 F	76.434 F

AIR SIDE:		REFRIGERANT SIDE:	
MASS FLOW RATE	1684.6 LBM/H	MASS FLOW RATE	65.4 LBM/H
PRESSURE DROP	.4206 IN H2O	PRESSURE DROP	4.774 PSI
HEAT TRANSFER COEFFICIENT	8.866 BTU/H-SQ FT-F	HEAT TRANSFER COEFFICIENT	
		VAPOR REGION	79.764 BTU/H-SQ FT-F
		TWO PHASE REGION	447.676 BTU/H-SQ FT-F
		SUBCOOLED REGION	93.677 BTU/H-SQ FT-F

UA VALUES:

VAPOR REGION (BTU/H-F)		TWO PHASE REGION (BTU/H-F)		SUBCOOLED REGION (BTU/H-F)	
REFRIGERANT SIDE	55.897	REFRIGERANT SIDE	6312.099	REFRIGERANT SIDE	102.798
AIR SIDE	116.875	AIR SIDE	2351.520	AIR SIDE	183.016
COMBINED	37.813	COMBINED	1713.260	COMBINED	65.825

FLOW CONTROL DEVICE - CONDENSER SUBCOOLING IS SPECIFIED AS 10.030 F

CORRESPONDING TXV RATING PARAMETERS:		CORRESPONDING CAPILLARY TUBE PARAMETERS:		CORRESPONDING ORIFICE PARAMETER:	
RATED OPERATING SUPERHEAT	11.000 F	NUMBER OF CAPILLARY TUBES	1	ORIFICE DIAMETER	.0450 IN
STATIC SUPERHEAT RATING	6.000 F	CAPILLARY TUBE FLOW FACTOR	2.074		
PERMANENT BLEED FACTOR	1.150				
TXV CAPACITY RATING INCLUDING NOZZLE AND TUBES	2.001 TONS				

PERFORMANCE OF EACH CIRCUIT IN THE EVAPORATOR

INLET AIR TEMPERATURE	17.000 F
HEAT LOSS FROM FAN	469.3 BTU/H
AIR TEMPERATURE CROSSING COIL	17.169 F
OUTLET AIR TEMPERATURE	12.199 F

MOISTURE REMOVAL OCCURS

SUMMARY OF DEHUMIDIFICATION PERFORMANCE (TWO-PHASE REGION)

	LEADING EDGE OF COIL	POINT WHERE MOISTURE REMOVAL BEGINS		LEAVING EDGE OF COIL	
	AIR	AIR	WALL	AIR	WALL
DRY BULB TEMPERATURE	17.169 F	17.169 F	10.521 F	11.894 F	8.356 F
HUMIDITY RATIO	.00139	.00139	.00134	.00130	.00121
ENTHALPY	5.610 BTU/LBM	5.610 BTU/LBM	3.961 BTU/LBM	4.248 BTU/LBM	3.292 BTU/LBM

RATE OF MOISTURE REMOVAL	.1303 LBM/H
FRACTION OF EVAPORATOR THAT IS WET	1.0000
LATENT HEAT TRANSFER RATE IN TWO-PHASE REGION	142. BTU/H
SENSIBLE HEAT TRANSFER RATE IN TWO-PHASE REGION	1938. BTU/H
SENSIBLE TO TOTAL HEAT TRANSFER RATIO FOR TWO-PHASE REGION	.9318

OVERALL SENSIBLE TO TOTAL HEAT TRANSFER RATIO .9329

OVERALL CONDITIONS ACROSS COIL

	ENTERING AIR	EXITING AIR
DRY BULB TEMPERATURE	17.169 F	12.199 F
WET BULB TEMPERATURE	15.649 F	11.690 F
RELATIVE HUMIDITY	.744	.896
HUMIDITY RATIO	.00139	.00131

TOTAL HEAT EXCHANGER EFFECTIVENESS (SENSIBLE) .4444

	SUPERHEATED REGION	TWO-PHASE REGION
NTU	1.0157	.5837
HEAT EXCHANGER EFFECTIVENESS	.6127	.4422
CR/CA	.1514	
FRACTION OF HEAT EXCHANGER	.0749	.9251
HEAT TRANSFER RATE	35.8 BTU/H	2079.9 BTU/H
AIR MASS FLOW RATE	123.47 LBM/H	1524.87 LBM/H
OUTLET AIR TEMPERATURE	15.965 F	11.894 F

AIR SIDE		REFRIGERANT SIDE	
MASS FLOW RATE	1648.3 LBM/H	MASS FLOW RATE	28.0 LBM/H
PRESSURE DROP	.068 IN H2O	PRESSURE DROP	2.247 PSI
HEAT TRANSFER COEFFICIENT		HEAT TRANSFER COEFFICIENT	
DRY COIL	14.079 BTU/H-SQ FT-F	VAPOR REGION	32.144 BTU/H-SQ FT-F
WET COIL	15.273 BTU/H-SQ FT-F	TWO PHASE REGION	403.617 BTU/H-SQ FT-F

DRY FIN EFFICIENCY	.778
WET FIN EFFICIENCY	.752

UA VALUES:	VAPOR REGION	TWO PHASE REGION
REFRIGERANT SIDE	40.036	6208.280 BTU/H-F
AIR SIDE		
DRY COIL	160.344	0.000 BTU/H-F
WET COIL		2076.576 BTU/H-F
COMBINED		
DRY COIL	32.037	0.000 BTU/H-F
WET COIL		1556.088 BTU/H-F

SUMMARY OF ENERGY INPUT AND OUTPUT:

RUN3H: OPTIMUM CONFIG., 17.0 F AMBIENT: TECH AV5532E COMP

AIR TEMPERATURE INTO EVAPORATOR	17.00 F
HEAT FROM CONDENSER TO AIR	19830. BTU/H
HEAT TO EVAPORATOR FROM AIR	14810. BTU/H
POWER TO INDOOR FAN	840. BTU/H
POWER TO OUTDOOR FAN	469. BTU/H
POWER TO COMPRESSOR MOTOR	5906. BTU/H
COMPRESSOR SHELL HEAT LOSS	886. BTU/H
TOTAL HEAT TO/FROM INDOOR AIR	21556. BTU/H

SYSTEM EFFICIENCY:		HEAT OUTPUT:	
COP (HEATING)	2.988	HEAT FROM HEAT PUMP	21555.80 BTU/H
COP (WITH RESISTANCE HEAT)	2.988	RESISTANCE HEAT	0.00 BTU/H
		HOUSE LOAD	20000.00 BTU/H

CALCULATED HEATING-ONLY HEAT PUMP PERFORMANCE AT -1.67°C (28°F)

***** CALCULATED HEAT PUMP PERFORMANCE *****

COMPRESSOR OPERATING CONDITIONS:

		EFFICIENCY	
COMPRESSOR POWER	1.996 KW	VOLUMETRIC	.6725
MOTOR SPEED	3450.000 RPM	OVERALL	.5760

REFRIGERANT MASS FLOW RATE	253.775 LBM/H	POWER PER UNIT MASS FLOW	26.90226 BTU/LBM
COMPRESSOR SHELL HEAT LOSS	1021.854 BTU/H	POWER CORRECTION FACTOR	1.0020
		MASS FLOW RATE CORRECTION FACTOR	1.0041

SYSTEM SUMMARY

	REFRIGERANT TEMPERATURE	SATURATION TEMPERATURE	REFRIGERANT ENTHALPY	REFRIGERANT QUALITY	REFRIGERANT PRESSURE	AIR TEMPERATURE
COMPRESSOR SUCTION LINE INLET	22.334 F	13.525 F	107.210 BTU/LBM	1.0000	50.909 PSIA	
SHELL INLET	30.568	13.127	108.589	1.0000	50.510	
SHELL OUTLET	191.436	95.645	131.407	1.0000	198.275	
CONDENSER INLET	191.385 F	95.548 F	131.407 BTU/LBM	1.0000	198.007 PSIA	68.000 F
OUTLET	79.016	92.981	32.812	0.0000	191.000	89.823
EXPANSION DEVICE	74.392 F	92.865 F	31.433 BTU/LBM	0.0000	190.687 PSIA	
EVAPORATOR INLET	16.556 F	16.556 F	31.433 BTU/LBM	.1814	54.020 PSIA	28.956 F
OUTLET	22.334	13.525	107.210	1.0000	50.909	23.016

PERFORMANCE OF EACH CIRCUIT IN THE CONDENSER

INLET AIR TEMPERATURE	68.000 F
HEAT LOSS FROM FAN	838.4 BTU/H
AIR TEMPERATURE CROSSING COIL	69.510 F
OUTLET AIR TEMPERATURE	89.823 F

TOTAL HEAT EXCHANGER EFFECTIVENESS .7281

	SUPERHEATED REGION	TWO-PHASE REGION	SUBCOOLED REGION
NTU	1.3039	1.6453	1.3017
HEAT EXCHANGER EFFECTIVENESS	.4304	.8070	.5954
CR/CA	1.0072		.6757
FRACTION OF HEAT EXCHANGER	.0407	.8662	.0931
HEAT TRANSFER RATE	886.3 BTU/H	7092.5 BTU/H	361.5 BTU/H
OUTLET AIR TEMPERATURE	122.612 F	89.452 F	78.962 F

AIR SIDE:		REFRIGERANT SIDE:	
MASS FLOW RATE	1684.6 LBM/H	MASS FLOW RATE	84.6 LBM/H
PRESSURE DROP	.4200 IN H2O	PRESSURE DROP	7.007 PSI
HEAT TRANSFER COEFFICIENT	8.880 BTU/H-SQ FT-F	HEAT TRANSFER COEFFICIENT	
		VAPOR REGION	100.164 BTU/H-SQ FT-F
		TWO PHASE REGION	538.518 BTU/H-SQ FT-F
		SUBCOOLED REGION	115.130 BTU/H-SQ FT-F

UA VALUES:

VAPOR REGION (BTU/H-F)		TWO PHASE REGION (BTU/H-F)		SUBCOOLED REGION (BTU/H-F)	
REFRIGERANT SIDE	64.735	REFRIGERANT SIDE	7415.885	REFRIGERANT SIDE	170.465
AIR SIDE	107.935	AIR SIDE	2299.853	AIR SIDE	247.279
COMBINED	40.466	COMBINED	1755.445	COMBINED	100.905

FLOW CONTROL DEVICE - CONDENSER SUBCOOLING IS SPECIFIED AS 13.960 F

CORRESPONDING TXV RATING PARAMETERS:		CORRESPONDING CAPILLARY TUBE PARAMETERS:		CORRESPONDING ORIFICE PARAMETER:	
RATED OPERATING SUPERHEAT	11.000 F	NUMBER OF CAPILLARY TUBES	1	ORIFICE DIAMETER	.0503 IN
STATIC SUPERHEAT RATING	6.000 F	CAPILLARY TUBE FLOW FACTOR	2.462		
PERMANENT BLEED FACTOR	1.150				
TXV CAPACITY RATING INCLUDING NOZZLE AND TUBES	2.003 TONS				

PERFORMANCE OF EACH CIRCUIT IN THE EVAPORATOR

INLET AIR TEMPERATURE	28.956 F
HEAT LOSS FROM FAN	459.1 BTU/H
AIR TEMPERATURE CROSSING COIL	29.125 F
OUTLET AIR TEMPERATURE	23.016 F

MOISTURE REMOVAL OCCURS

SUMMARY OF DEHUMIDIFICATION PERFORMANCE (TWO-PHASE REGION)

	LEADING EDGE OF COIL	POINT WHERE MOISTURE REMOVAL BEGINS		LEAVING EDGE OF COIL	
	AIR	AIR	WALL	AIR	WALL
DRY BULB TEMPERATURE	29.125 F	29.125 F	21.263 F	22.681 F	18.652 F
HUMIDITY RATIO	.00245	.00245	.00228	.00222	.00201
ENTHALPY	9.636 BTU/LBM	9.636 BTU/LBM	7.549 BTU/LBM	7.831 BTU/LBM	6.630 BTU/LBM

RATE OF MOISTURE REMOVAL	.3469 LBM/H
FRACTION OF EVAPORATOR THAT IS WET	1.0000
LATENT HEAT TRANSFER RATE IN TWO-PHASE REGION	376. BTU/H
SENSIBLE HEAT TRANSFER RATE IN TWO-PHASE REGION	2320. BTU/H
SENSIBLE TO TOTAL HEAT TRANSFER RATIO FOR TWO-PHASE REGION	.8606

OVERALL SENSIBLE TO TOTAL HEAT TRANSFER RATIO .8632

OVERALL CONDITIONS ACROSS COIL

	ENTERING AIR	EXITING AIR
DRY BULB TEMPERATURE	29.125 F	23.016 F
WET BULB TEMPERATURE	26.792 F	22.289 F
RELATIVE HUMIDITY	.744	.904
HUMIDITY RATIO	.00245	.00224

TOTAL HEAT EXCHANGER EFFECTIVENESS (SENSIBLE) .4682

	SUPERHEATED REGION	TWO-PHASE REGION
NTU	.8919	.6283
HEAT EXCHANGER EFFECTIVENESS	.5589	.4665
CR/CA	.2124	
FRACTION OF HEAT EXCHANGER	.0726	.9274
HEAT TRANSFER RATE	51.5 BTU/H	2696.0 BTU/H
AIR MASS FLOW RATE	116.77 LBM/H	1491.24 LBM/H
OUTLET AIR TEMPERATURE	27.297 F	22.681 F

AIR SIDE		REFRIGERANT SIDE	
MASS FLOW RATE	1608.0 LBM/H	MASS FLOW RATE	36.3 LBM/H
PRESSURE DROP	.066 IN H2O	PRESSURE DROP	3.110 PSI
HEAT TRANSFER COEFFICIENT		HEAT TRANSFER COEFFICIENT	
DRY COIL	14.478 BTU/H-SQ FT-F	VAPOR REGION	40.494 BTU/H-SQ FT-F
WET COIL	15.706 BTU/H-SQ FT-F	TWO PHASE REGION	467.597 BTU/H-SQ FT-F

DRY FIN EFFICIENCY	.774
WET FIN EFFICIENCY	.733

UA VALUES:	VAPOR REGION	TWO PHASE REGION
REFRIGERANT SIDE	48.894	7210.205 BTU/H-F
AIR SIDE		
DRY COIL	158.905	0.000 BTU/H-F
WET COIL		2085.265 BTU/H-F
COMBINED		
DRY COIL	37.390	0.000 BTU/H-F
WET COIL		1617.475 BTU/H-F

SUMMARY OF ENERGY INPUT AND OUTPUT:

RUN3J: OPTIMUM CONFIG., 29.0 F AMBIENT: TECH AV5532E COMP

AIR TEMPERATURE INTO EVAPORATOR	28.96	F
HEAT FROM CONDENSER TO AIR	25021.	BTU/H
HEAT TO EVAPORATOR FROM AIR	19232.	BTU/H
POWER TO INDOOR FAN	838.	BTU/H
POWER TO OUTDOOR FAN	459.	BTU/H
POWER TO COMPRESSOR MOTOR	6812.	BTU/H
COMPRESSOR SHELL HEAT LOSS	1022.	BTU/H
TOTAL HEAT TO/FROM INDOOR AIR	26881.	BTU/H

SYSTEM EFFICIENCY:		HEAT OUTPUT:	
COP (HEATING)	3.315	HEAT FROM HEAT PUMP	26881.03 BTU/H
COP (WITH RESISTANCE HEAT)	3.315	RESISTANCE HEAT	0.00 BTU/H
		HOUSE LOAD	20000.00 BTU/H

CALCULATED HEATING-ONLY HEAT PUMP PERFORMANCE AT 4.83°C (40.7°F)

***** CALCULATED HEAT PUMP PERFORMANCE *****

COMPRESSOR OPERATING CONDITIONS:

		EFFICIENCY	
COMPRESSOR POWER	2.256 KW	VOLUMETRIC	.7030
MOTOR SPEED	3450.000 RPM	OVERALL	.5872

REFRIGERANT MASS FLOW RATE	315.819 LBM/H	POWER PER UNIT MASS FLOW	24.60628 BTU/LBM
COMPRESSOR SHELL HEAT LOSS	1154.851 BTU/H	POWER CORRECTION FACTOR	.9969
		MASS FLOW RATE CORRECTION FACTOR	1.0062

SYSTEM SUMMARY	REFRIGERANT TEMPERATURE	SATURATION TEMPERATURE	REFRIGERANT ENTHALPY	REFRIGERANT QUALITY	REFRIGERANT PRESSURE	AIR TEMPERATURE
COMPRESSOR SUCTION LINE INLET	32.311 F	22.867 F	108.225 BTU/LBM	1.0000	60.959 PSIA	
SHELL INLET	38.709	22.429	109.333	1.0000	60.457	
SHELL OUTLET	187.514	101.830	130.055	1.0000	215.950	
CONDENSER INLET	187.450 F	101.707 F	130.055 BTU/LBM	1.0000	215.590 PSIA	68.000 F
OUTLET	81.266	98.382	33.492	0.0000	205.960	94.374
EXPANSION DEVICE	77.567 F	98.219 F	32.384 BTU/LBM	0.0000	205.497 PSIA	
EVAPORATOR INLET	26.159 F	26.159 F	32.384 BTU/LBM	.1660	64.835 PSIA	40.749 F
OUTLET	32.311	22.867	108.225	1.0000	60.959	33.567

PERFORMANCE OF EACH CIRCUIT IN THE CONDENSER

INLET AIR TEMPERATURE	68.000 F
HEAT LOSS FROM FAN	837.2 BTU/H
AIR TEMPERATURE CROSSING COIL	69.617 F
OUTLET AIR TEMPERATURE	94.374 F

TOTAL HEAT EXCHANGER EFFECTIVENESS .7589

	SUPERHEATED REGION	TWO-PHASE REGION	SUBCOOLED REGION
NTU	1.3511	1.7049	1.3479
HEAT EXCHANGER EFFECTIVENESS	.4943	.8182	.5956
CR/CA	1.4064		.7215
FRACTION OF HEAT EXCHANGER	.0369	.8537	.1094
HEAT TRANSFER RATE	894.6 BTU/H	8715.0 BTU/H	555.9 BTU/H
OUTLET AIR TEMPERATURE	128.665 F	94.477 F	81.996 F

AIR SIDE:

MASS FLOW RATE	1684.6 LBM/H
PRESSURE DROP	.4194 IN H20
HEAT TRANSFER COEFFICIENT	8.899 BTU/H-SQ FT-F

REFRIGERANT SIDE:

MASS FLOW RATE	105.3 LBM/H
PRESSURE DROP	9.629 PSI
HEAT TRANSFER COEFFICIENT	
VAPOR REGION	121.859 BTU/H-SQ FT-F
TWO PHASE REGION	627.801 BTU/H-SQ FT-F
SUBCOOLED REGION	137.118 BTU/H-SQ FT-F

UA VALUES:

	VAPOR REGION (BTU/H-F)	TWO PHASE REGION (BTU/H-F)	SUBCOOLED REGION (BTU/H-F)
REFRIGERANT SIDE	71.479	8520.959	238.388
AIR SIDE	98.140	2270.857	290.880
COMBINED	41.357	1793.014	131.016

FLOW CONTROL DEVICE - CONDENSER SUBCOOLING IS SPECIFIED AS 17.100 F

CORRESPONDING TXV RATING PARAMETERS:

RATED OPERATING SUPERHEAT	11.000 F
STATIC SUPERHEAT RATING	6.000 F
PERMANENT BLEED FACTOR	1.150
TXV CAPACITY RATING INCLUDING NOZZLE AND TUBES	2.002 TONS

CORRESPONDING CAPILLARY TUBE PARAMETERS:

NUMBER OF CAPILLARY TUBES	1
CAPILLARY TUBE FLOW FACTOR	2.839

CORRESPONDING ORIFICE PARAMETER:

ORIFICE DIAMETER	.0552 IN

PERFORMANCE OF EACH CIRCUIT IN THE EVAPORATOR

INLET AIR TEMPERATURE	40.749 F
HEAT LOSS FROM FAN	449.3 BTU/H
AIR TEMPERATURE CROSSING COIL	40.918 F
OUTLET AIR TEMPERATURE	33.567 F

MOISTURE REMOVAL OCCURS

SUMMARY OF DEHUMIDIFICATION PERFORMANCE (TWO-PHASE REGION)

	LEADING EDGE OF COIL	POINT WHERE MOISTURE REMOVAL BEGINS		LEAVING EDGE OF COIL	
	AIR	AIR	WALL	AIR	WALL
DRY BULB TEMPERATURE	40.918 F	40.918 F	31.742 F	33.196 F	28.738 F
HUMIDITY RATIO	.00400	.00400	.00373	.00360	.00324
ENTHALPY	14.154 BTU/LBM	14.154 BTU/LBM	11.637 BTU/LBM	11.855 BTU/LBM	10.390 BTU/LBM

RATE OF MOISTURE REMOVAL	.5818 LBM/H
FRACTION OF EVAPORATOR THAT IS WET	1.0000
LATENT HEAT TRANSFER RATE IN TWO-PHASE REGION	627. BTU/H
SENSIBLE HEAT TRANSFER RATE IN TWO-PHASE REGION	2724. BTU/H
SENSIBLE TO TOTAL HEAT TRANSFER RATIO FOR TWO-PHASE REGION	.8129

OVERALL SENSIBLE TO TOTAL HEAT TRANSFER RATIO .8168

OVERALL CONDITIONS ACROSS COIL

	ENTERING AIR	EXITING AIR
DRY BULB TEMPERATURE	40.918 F	33.567 F
WET BULB TEMPERATURE	37.597 F	32.378 F
RELATIVE HUMIDITY	.745	.904
HUMIDITY RATIO	.00400	.00363

TOTAL HEAT EXCHANGER EFFECTIVENESS (SENSIBLE) .4892

	SUPERHEATED REGION	TWO-PHASE REGION
NTU	.8079	.6704
HEAT EXCHANGER EFFECTIVENESS	.5173	.4885
CR/CA	.2796	
FRACTION OF HEAT EXCHANGER	.0720	.9280
HEAT TRANSFER RATE	70.6 BTU/H	3351.4 BTU/H
AIR MASS FLOW RATE	113.11 LBM/H	1457.01 LBM/H
OUTLET AIR TEMPERATURE	38.340 F	33.196 F

AIR SIDE		REFRIGERANT SIDE	
MASS FLOW RATE	1570.1 LBM/H	MASS FLOW RATE	45.1 LBM/H
PRESSURE DROP	.065 IN H20	PRESSURE DROP	3.874 PSI
HEAT TRANSFER COEFFICIENT		HEAT TRANSFER COEFFICIENT	
DRY COIL	14.938 BTU/H-SQ FT-F	VAPOR REGION	49.394 BTU/H-SQ FT-F
WET COIL	16.206 BTU/H-SQ FT-F	TWO PHASE REGION	524.904 BTU/H-SQ FT-F

DRY FIN EFFICIENCY .769
WET FIN EFFICIENCY .715

UA VALUES:	VAPOR REGION	TWO PHASE REGION
REFRIGERANT SIDE	59.164	8098.920 BTU/H-F
AIR SIDE		
DRY COIL	161.566	0.000 BTU/H-F
WET COIL		2101.250 BTU/H-F
COMBINED		
DRY COIL	43.306	0.000 BTU/H-F
WET COIL		1668.390 BTU/H-F

SUMMARY OF ENERGY INPUT AND OUTPUT:
RUN3F: OPTIMUM CONFIG., 40.7 F AMBIENT: TECH AV5532E COMP

AIR TEMPERATURE INTO EVAPORATOR	40.75	F
HEAT FROM CONDENSER TO AIR	30496.	BTU/H
HEAT TO EVAPORATOR FROM AIR	23954.	BTU/H
POWER TO INDOOR FAN	837.	BTU/H
POWER TO OUTDOOR FAN	449.	BTU/H
POWER TO COMPRESSOR MOTOR	7699.	BTU/H
COMPRESSOR SHELL HEAT LOSS	1155.	BTU/H
TOTAL HEAT TO/FROM INDOOR AIR	32488.	BTU/H

SYSTEM EFFICIENCY:		HEAT OUTPUT:	
COP (HEATING)	3.616	HEAT FROM HEAT PUMP	32488.45 BTU/H
COP (WITH RESISTANCE HEAT)	3.616	RESISTANCE HEAT	0.00 BTU/H
		HOUSE LOAD	20000.00 BTU/H

CALCULATED HEATING-ONLY HEAT PUMP PERFORMANCE AT 8.3°C (47°F)

***** CALCULATED HEAT PUMP PERFORMANCE *****

COMPRESSOR OPERATING CONDITIONS:

		EFFICIENCY	
COMPRESSOR POWER	2.389 KW	VOLUMETRIC	.7179
MOTOR SPEED	3450.000 RPM	OVERALL	.6010

REFRIGERANT MASS FLOW RATE	353.063 LBM/H	POWER PER UNIT MASS FLOW	23.45896 BTU/LBM
COMPRESSOR SHELL HEAT LOSS	1223.174 BTU/H	POWER CORRECTION FACTOR	.9914
		MASS FLOW RATE CORRECTION FACTOR	1.0070

SYSTEM SUMMARY	REFRIGERANT TEMPERATURE	SATURATION TEMPERATURE	REFRIGERANT ENTHALPY	REFRIGERANT QUALITY	REFRIGERANT PRESSURE	AIR TEMPERATURE
COMPRESSOR SUCTION LINE INLET	37.628 F	27.882 F	108.765 BTU/LBM	1.0000	66.937 PSIA	
SHELL INLET	43.322	27.424	109.757	1.0000	66.373	
SHELL OUTLET	186.044	105.507	129.389	1.0000	226.992	
CONDENSER INLET	185.973 F	105.370 F	129.389 BTU/LBM	1.0000	226.574 PSIA	68.000 F
OUTLET	82.586	101.618	33.892	0.0000	215.327	97.041
EXPANSION DEVICE	79.287 F	101.426 F	32.901 BTU/LBM	0.0000	214.761 PSIA	
EVAPORATOR INLET	31.304 F	31.304 F	32.901 BTU/LBM	.1575	71.265 PSIA	47.000 F
OUTLET	37.628	27.882	108.765	1.0000	66.937	39.196

PERFORMANCE OF EACH CIRCUIT IN THE CONDENSER

INLET AIR TEMPERATURE	68.000 F
HEAT LOSS FROM FAN	836.5 BTU/H
AIR TEMPERATURE CROSSING COIL	69.672 F
OUTLET AIR TEMPERATURE	97.041 F

TOTAL HEAT EXCHANGER EFFECTIVENESS .7715

	SUPERHEATED REGION	TWO-PHASE REGION	SUBCOOLED REGION
NTU	1.3770	1.7321	1.3745
HEAT EXCHANGER EFFECTIVENESS	.5301	.8231	.5964
CR/CA	1.7104		.7427
FRACTION OF HEAT EXCHANGER	.0343	.8464	.1193
HEAT TRANSFER RATE	880.6 BTU/H	9663.7 BTU/H	694.4 BTU/H
OUTLET AIR TEMPERATURE	132.182 F	97.477 F	83.844 F

AIR SIDE:		REFRIGERANT SIDE:	
MASS FLOW RATE	1684.6 LBM/H	MASS FLOW RATE	117.7 LBM/H
PRESSURE DROP	.4190 IN H20	PRESSURE DROP	11.247 PSI
HEAT TRANSFER COEFFICIENT	8.911 BTU/H-SQ FT-F	HEAT TRANSFER COEFFICIENT	
		VAPOR REGION	134.816 BTU/H-SQ FT-F
		TWO PHASE REGION	675.571 BTU/H-SQ FT-F
		SUBCOOLED REGION	149.886 BTU/H-SQ FT-F

UA VALUES:

VAPOR REGION (BTU/H-F)		TWO PHASE REGION (BTU/H-F)		SUBCOOLED REGION (BTU/H-F)	
REFRIGERANT SIDE	73.532	REFRIGERANT SIDE	9090.112	REFRIGERANT SIDE	284.329
AIR SIDE	91.354	AIR SIDE	2253.680	AIR SIDE	317.728
COMBINED	40.740	COMBINED	1805.940	COMBINED	150.051

FLOW CONTROL DEVICE - CONDENSER SUBCOOLING IS SPECIFIED AS 18.970 F

CORRESPONDING TXV RATING PARAMETERS:		CORRESPONDING CAPILLARY TUBE PARAMETERS:		CORRESPONDING ORIFICE PARAMETER:	
RATED OPERATING SUPERHEAT	11.000 F	NUMBER OF CAPILLARY TUBES	1	ORIFICE DIAMETER	.0578 IN
STATIC SUPERHEAT RATING	6.000 F	CAPILLARY TUBE FLOW FACTOR	3.033		
PERMANENT BLEED FACTOR	1.150				
TXV CAPACITY RATING INCLUDING NOZZLE AND TUBES	2.000 TONS				

PERFORMANCE OF EACH CIRCUIT IN THE EVAPORATOR

INLET AIR TEMPERATURE	47.000 F
HEAT LOSS FROM FAN	444.2 BTU/H
AIR TEMPERATURE CROSSING COIL	47.169 F
OUTLET AIR TEMPERATURE	39.196 F

MOISTURE REMOVAL OCCURS

SUMMARY OF DEHUMIDIFICATION PERFORMANCE (TWO-PHASE REGION)

	LEADING EDGE OF COIL	POINT WHERE MOISTURE REMOVAL BEGINS		LEAVING EDGE OF COIL	
	AIR	AIR	WALL	AIR	WALL
DRY BULB TEMPERATURE	47.169 F	47.169 F	37.457 F	38.808 F	34.148 F
HUMIDITY RATIO	.00509	.00509	.00470	.00456	.00411
ENTHALPY	16.847 BTU/LBM	16.847 BTU/LBM	14.065 BTU/LBM	14.242 BTU/LBM	12.637 BTU/LBM

RATE OF MOISTURE REMOVAL	.7686 LBM/H
FRACTION OF EVAPORATOR THAT IS WET	1.0000
LATENT HEAT TRANSFER RATE IN TWO-PHASE REGION	826. BTU/H
SENSIBLE HEAT TRANSFER RATE IN TWO-PHASE REGION	2917. BTU/H
SENSIBLE TO TOTAL HEAT TRANSFER RATIO FOR TWO-PHASE REGION	.7793

OVERALL SENSIBLE TO TOTAL HEAT TRANSFER RATIO .7841

OVERALL CONDITIONS ACROSS COIL

	ENTERING AIR	EXITING AIR
DRY BULB TEMPERATURE	47.169 F	39.196 F
WET BULB TEMPERATURE	43.428 F	38.107 F
RELATIVE HUMIDITY	.745	.914
HUMIDITY RATIO	.00509	.00460

TOTAL HEAT EXCHANGER EFFECTIVENESS (SENSIBLE) .4999

	SUPERHEATED REGION	TWO-PHASE REGION
NTU	.7753	.6930
HEAT EXCHANGER EFFECTIVENESS	.4994	.4999
CR/CA	.3181	
FRACTION OF HEAT EXCHANGER	.0726	.9274
HEAT TRANSFER RATE	82.7 BTU/H	3743.3 BTU/H
AIR MASS FLOW RATE	112.60 LBM/H	1438.15 LBM/H
OUTLET AIR TEMPERATURE	44.143 F	38.808 F

AIR SIDE		REFRIGERANT SIDE	
MASS FLOW RATE	1550.7 LBM/H	MASS FLOW RATE	50.4 LBM/H
PRESSURE DROP	.064 IN H2O	PRESSURE DROP	4.325 PSI
HEAT TRANSFER COEFFICIENT		HEAT TRANSFER COEFFICIENT	
DRY COIL	15.214 BTU/H-SQ FT-F	VAPOR REGION	54.680 BTU/H-SQ FT-F
WET COIL	16.505 BTU/H-SQ FT-F	TWO PHASE REGION	555.413 BTU/H-SQ FT-F

DRY FIN EFFICIENCY	.766
WET FIN EFFICIENCY	.704

UA VALUES:	VAPOR REGION	TWO PHASE REGION
REFRIGERANT SIDE	66.015	8564.374 BTU/H-F
AIR SIDE		
DRY COIL	165.194	0.000 BTU/H-F
WET COIL		2104.019 BTU/H-F
COMBINED		
DRY COIL	47.166	0.000 BTU/H-F
WET COIL		1689.064 BTU/H-F

SUMMARY OF ENERGY INPUT AND OUTPUT:

RUN3G: OPTIMUM CONFIG., 47.0 F AMBIENT: TECH AV5532E COMP

AIR TEMPERATURE INTO EVAPORATOR	47.00	F
HEAT FROM CONDENSER TO AIR	33716.	BTU/H
HEAT TO EVAPORATOR FROM AIR	26782.	BTU/H
POWER TO INDOOR FAN	836.	BTU/H
POWER TO OUTDOOR FAN	444.	BTU/H
POWER TO COMPRESSOR MOTOR	8154.	BTU/H
COMPRESSOR SHELL HEAT LOSS	1223.	BTU/H
TOTAL HEAT TO/FROM INDOOR AIR	35776.	BTU/H

SYSTEM EFFICIENCY:		HEAT OUTPUT:	
COP (HEATING)	3.792	HEAT FROM HEAT PUMP	35775.99 BTU/H
COP (WITH RESISTANCE HEAT)	3.792	RESISTANCE HEAT	0.00 BTU/H
		HOUSE LOAD	20000.00 BTU/H

CALCULATED HEATING-ONLY HEAT PUMP PERFORMANCE AT 16°C (60.8°F)

***** CALCULATED HEAT PUMP PERFORMANCE *****

COMPRESSOR OPERATING CONDITIONS:

		EFFICIENCY	
COMPRESSOR POWER	2.800 KW	VOLUMETRIC	.7204
MOTOR SPEED	3450.000 RPM	OVERALL	.5752

REFRIGERANT MASS FLOW RATE	435.249 LBM/H	POWER PER UNIT MASS FLOW	22.10424 BTU/LBM
COMPRESSOR SHELL HEAT LOSS	1433.638 BTU/H	POWER CORRECTION FACTOR	1.0013
		MASS FLOW RATE CORRECTION FACTOR	1.0079

SYSTEM SUMMARY

	REFRIGERANT TEMPERATURE	SATURATION TEMPERATURE	REFRIGERANT ENTHALPY	REFRIGERANT QUALITY	REFRIGERANT PRESSURE	AIR TEMPERATURE
COMPRESSOR SUCTION LINE INLET	50.346 F	39.711 F	110.005 BTU/LBM	1.0000	82.787 PSIA	
SHELL INLET	54.767	39.240	110.809	1.0000	82.107	
SHELL OUTLET	190.633	113.328	129.474	1.0000	251.856	
CONDENSER INLET	190.540 F	113.160 F	129.474 BTU/LBM	1.0000	251.301 PSIA	68.000 F
OUTLET	86.862	108.485	35.195	0.0000	236.237	103.148
EXPANSION DEVICE	84.210 F	108.223 F	34.391 BTU/LBM	0.0000	235.412 PSIA	
EVAPORATOR INLET	43.169 F	43.169 F	34.391 BTU/LBM	.1402	87.911 PSIA	60.800 F
OUTLET	50.346	39.711	110.005	1.0000	82.787	52.085

PERFORMANCE OF EACH CIRCUIT IN THE CONDENSER

INLET AIR TEMPERATURE	68.000 F
HEAT LOSS FROM FAN	835.1 BTU/H
AIR TEMPERATURE CROSSING COIL	69.841 F
OUTLET AIR TEMPERATURE	103.148 F

TOTAL HEAT EXCHANGER EFFECTIVENESS .7782

	SUPERHEATED REGION	TWO-PHASE REGION	SUBCOOLED REGION
NTU	1.3002	1.7797	1.2973
HEAT EXCHANGER EFFECTIVENESS	.5721	.8313	.5604
CR/CA	2.1073		.8572
FRACTION OF HEAT EXCHANGER	.0351	.8359	.1289
HEAT TRANSFER RATE	1010.6 BTU/H	11682.8 BTU/H	984.9 BTU/H
OUTLET AIR TEMPERATURE	139.907 F	103.872 F	88.441 F

AIR SIDE:		REFRIGERANT SIDE:	
MASS FLOW RATE	1684.6 LBM/H	MASS FLOW RATE	145.1 LBM/H
PRESSURE DROP	.4183 IN H20	PRESSURE DROP	15.064 PSI
HEAT TRANSFER COEFFICIENT	8.934 BTU/H-SQ FT-F	HEAT TRANSFER COEFFICIENT	
		VAPOR REGION	163.532 BTU/H-SQ FT-F
		TWO PHASE REGION	773.282 BTU/H-SQ FT-F
		SUBCOOLED REGION	177.126 BTU/H-SQ FT-F

UA VALUES:

VAPOR REGION (BTU/H-F)		TWO PHASE REGION (BTU/H-F)		SUBCOOLED REGION (BTU/H-F)	
REFRIGERANT SIDE	91.309	REFRIGERANT SIDE	10276.582	REFRIGERANT SIDE	363.094
AIR SIDE	93.725	AIR SIDE	2230.763	AIR SIDE	344.095
COMBINED	46.251	COMBINED	1832.893	COMBINED	176.670

FLOW CONTROL DEVICE - CONDENSER SUBCOOLING IS SPECIFIED AS 21.700 F

CORRESPONDING TXV RATING PARAMETERS:		CORRESPONDING CAPILLARY TUBE PARAMETERS:		CORRESPONDING ORIFICE PARAMETER:	
RATED OPERATING SUPERHEAT	11.000 F	NUMBER OF CAPILLARY TUBES	1	ORIFICE DIAMETER	.0633 IN
STATIC SUPERHEAT RATING	6.000 F	CAPILLARY TUBE FLOW FACTOR	3.468		
PERMANENT BLEED FACTOR	1.150				
TXV CAPACITY RATING INCLUDING NOZZLE AND TUBES	1.993 TONS				

PERFORMANCE OF EACH CIRCUIT IN THE EVAPORATOR

INLET AIR TEMPERATURE	60.800 F
HEAT LOSS FROM FAN	431.8 BTU/H
AIR TEMPERATURE CROSSING COIL	60.967 F
OUTLET AIR TEMPERATURE	52.085 F

MOISTURE REMOVAL OCCURS

SUMMARY OF DEHUMIDIFICATION PERFORMANCE (TWO-PHASE REGION)

	LEADING EDGE OF COIL	POINT WHERE MOISTURE REMOVAL BEGINS		LEAVING EDGE OF COIL	
	AIR	AIR	WALL	AIR	WALL
DRY BULB TEMPERATURE	60.967 F	60.967 F	50.585 F	51.647 F	46.891 F
HUMIDITY RATIO	.00848	.00848	.00780	.00752	.00678
ENTHALPY	23.887 BTU/LBM	23.887 BTU/LBM	20.609 BTU/LBM	20.567 BTU/LBM	18.607 BTU/LBM

RATE OF MOISTURE REMOVAL	1.3367 LBM/H
FRACTION OF EVAPORATOR THAT IS WET	1.0000
LATENT HEAT TRANSFER RATE IN TWO-PHASE REGION	1427. BTU/H
SENSIBLE HEAT TRANSFER RATE IN TWO-PHASE REGION	3159. BTU/H
SENSIBLE TO TOTAL HEAT TRANSFER RATIO FOR TWO-PHASE REGION	.6889

OVERALL SENSIBLE TO TOTAL HEAT TRANSFER RATIO .6965

OVERALL CONDITIONS ACROSS COIL

	ENTERING AIR	EXITING AIR
DRY BULB TEMPERATURE	60.967 F	52.085 F
WET BULB TEMPERATURE	56.098 F	50.865 F
RELATIVE HUMIDITY	.746	.922
HUMIDITY RATIO	.00848	.00760

TOTAL HEAT EXCHANGER EFFECTIVENESS (SENSIBLE) .5222

	SUPERHEATED REGION	TWO-PHASE REGION
NTU	.7794	.7409
HEAT EXCHANGER EFFECTIVENESS	.4941	.5233
CR/CA	.3745	
FRACTION OF HEAT EXCHANGER	.0805	.9195
HEAT TRANSFER RATE	115.3 BTU/H	4585.6 BTU/H
AIR MASS FLOW RATE	121.56 LBM/H	1388.07 LBM/H
OUTLET AIR TEMPERATURE	57.084 F	51.647 F

AIR SIDE		REFRIGERANT SIDE	
MASS FLOW RATE	1509.6 LBM/H	MASS FLOW RATE	62.2 LBM/H
PRESSURE DROP	.062 IN H2O	PRESSURE DROP	5.121 PSI
HEAT TRANSFER COEFFICIENT		HEAT TRANSFER COEFFICIENT	
DRY COIL	15.921 BTU/H-SQ FT-F	VAPOR REGION	66.554 BTU/H-SQ FT-F
WET COIL	17.272 BTU/H-SQ FT-F	TWO PHASE REGION	608.883 BTU/H-SQ FT-F

DRY FIN EFFICIENCY	.758
WET FIN EFFICIENCY	.669

UA VALUES:	VAPOR REGION	TWO PHASE REGION
REFRIGERANT SIDE	89.110	9308.740 BTU/H-F
AIR SIDE		
DRY COIL	189.783	0.000 BTU/H-F
WET COIL		2075.245 BTU/H-F
COMBINED		
DRY COIL	60.638	0.000 BTU/H-F
WET COIL		1696.938 BTU/H-F

SUMMARY OF ENERGY INPUT AND OUTPUT:

RUN3E: OPTIMUM CONFIG., 60.8 F AMBIENT: TECH AV5532E COMP

AIR TEMPERATURE INTO EVAPORATOR	60.80	F
HEAT FROM CONDENSER TO AIR	41035.	BTU/H
HEAT TO EVAPORATOR FROM AIR	32906.	BTU/H
POWER TO INDOOR FAN	835.	BTU/H
POWER TO OUTDOOR FAN	432.	BTU/H
POWER TO COMPRESSOR MOTOR	9558.	BTU/H
COMPRESSOR SHELL HEAT LOSS	1434.	BTU/H
TOTAL HEAT TO/FROM INDOOR AIR	43304.	BTU/H

SYSTEM EFFICIENCY:		HEAT OUTPUT:	
COP (HEATING)	4.001	HEAT FROM HEAT PUMP	43303.53 BTU/H
COP (WITH RESISTANCE HEAT)	4.001	RESISTANCE HEAT	0.00 BTU/H
		HOUSE LOAD	20000.00 BTU/H