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An Analysis of Efficiency Improvements in Residential Sized Heat Pumps

Final Report

May 1986

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Prepared under Contract No. 4531710
For
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GLOSSARY OF TERMS

APF	ANNUAL PERFORMANCE FACTOR
ARI	AIR CONDITIONING AND REFRIGERATION INSTITUTE
BTU	BRITISH THERMAL UNIT
CD	DEGRADATION COEFFICIENT
CFM	CUBIC FEET PER MINUTE
CL	COOLING
COP	COEFFICIENT OF PERFORMANCE
DOE	DEPARTMENT OF ENERGY
EER	ENERGY EFFICIENCY RATIO
FPI	FINS PER INCH
HSPF	HEATING SEASONAL PERFORMANCE FACTOR
HR	HOUR
HT	HEATING
LBL	LAWRENCE BERKELEY LABORATORY
NBS	NATIONAL BUREAU OF STANDARDS
NECPA	NATIONAL ENERGY CONSERVATION POLICY ACT
OEM	ORIGINAL EQUIPMENT MANUFACTURERS
ORNL	OAK RIDGE NATIONAL LABORATORY
PLF	PART LOAD FACTOR
SAI	SCIENCE APPLICATION INCORPORATED
SEER	SEASONAL ENERGY EFFICIENCY RATIO
SF	SQUARE FEET
SHF	SENSIBLE HEATING FACTOR
TDB	DRY BULB TEMPERATURE
TON	12000 BTU/HR
TXV	THERMAL EXPANSION VALVE
TWB	WET BULB TEMPERATURE

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CHAPTER 1

INTRODUCTION

The National Energy Conservation Policy Act (NECPA) P.L. 95-619, requires the imposition of minimum efficiency standards on major appliances used in the residential sector. The law requires proposed standards that are both technologically feasible and economically justifiable. One of the appliances for standards consideration is the residential sized (65000 Btu/hr and under) air source heat pump. This report summarizes the results of an engineering design analysis used to evaluate the technical feasibility of improving the efficiency of heat pumps. The costs of the heat pump designs resulting from this analysis need to be evaluated to determine the cost effectiveness of the designs and whether minimum efficiency standards need to be imposed.

The objectives of this study included: (1) development of classes of heat pumps, (2) evaluation and selection of a suitable heat pump design model, (3) characterization of suitable baseline heat pump designs, (4) selection of design options that can be used to improve heat pump efficiency, and (5) development of heat pump designs to cover the whole spectrum of efficiencies available today and those that may be technologically feasible in the next few years.

Chapter 2 provides background material on the shipments and installation of heat pumps. This material sheds some light on the current range of efficiencies in heat pumps and possible issues surrounding the evaluation of minimum efficiency standards of heat pumps.

In Chapter 3, a discussion of product classes for heat pumps is presented. A minimum of nine classes of heat pumps are recommended. Because many of these classes do not have finalized test procedures, only four are evaluated in this report: (1) air source split systems, 39000 Btu/hr and under, (2) air source split systems, greater than 39000 Btu/hr, (3) air source single package sytems, 39000 Btu/hr and under, and (4) air source single package systems, greater than 39000 Btu/hr.

Engineering design models can be useful tools for evaluating the technical feasibility of improving the efficiency of heat pumps. For the heat pump, two models were used: a steady state and seasonal performance model. Chapter 4 provides a discussion of both of the models. Validation runs are also provided for several different heat pumps.

Eleven major design options available for improving the efficiency of heat pumps are discussed in Chapter 5. These design options included conventional improvements, such as increased heat exchanger surface area, and advanced options, such as variable speed compressors.

The design methodology and final designs for four classes of heat pumps are discussed in Chapter 6. The performance of the final designs in each class was calculated using the models described in Chapter 4. The final designs started with a baseline model near the minimum efficiency of units on the market in the 1985. The efficiency of each design was incrementally improved to arrive at the units that are efficiencies that would be considered the "maximum technologically feasible".

Major conclusions from this study and recommendations for further study are provided in Chapter 7. Some of the recommendations center on improving the analysis of the four classes in this study. Other recommendations center on preparations for the analysis of the other classes of heat pumps.

CHAPTER 2

THE HEAT PUMP MARKET

This chapter provides a discussion of the residential heat pump market: history, shipments, installations, efficiencies, type of units, etc. Factors that could influence the heat pump standards analysis are discussed where appropriate.

Historical Background and Shipments

The first documented experimental heat pump ever constructed was built by T.G. N. Haldane in Scotland in the mid-1920s[1]. Conditions in Europe, such as lack of fossil fuels and pollution from coal furnaces, promoted development of the heat pump in the 1920s. Sweden, for example had developed about 1100 kwh per person per year of hydroelectric power by 1930, so the heat pump was an obvious device for effective natural resource use.[2]

There were a few heat pump installations in the U.S. in the 1930s[3]. These systems were custom made using existing air-conditioning and refrigeration equipment.

The first complete heat pump for retail distribution in the U.S. was manufactured in 1947 by the Muncie Gear Works, Inc. of Muncie, Indiana[4]. Cost of the system was about $1700 (in 1947 dollars) including installation.

Heat pumps were described as the liberators "from the coal shovel, from dust, smoke and summer heat...."[4] as well as "effortless heating and cooling systems". Numerous advertisements furthered the idea that the heat pump was the final answer for residential space conditioning.

Early promoters of heat pumps claimed heat pump efficiencies 6 to 7 times that of ordinary electric resistance heaters. Because of these claims, early heat pump sales grew steadily. From 1955 to 1963, annual heat pump sales grew from 4746 to approximately 76,000 units (See Figure 2.1).[5]

Heat pumps experienced problems early in their commercial history. The military services installed about 10,000 heat pumps in base housing between mid-1958 and March 1964[6]. Initial failure rates ranged from 22 to 25% per year. Determined effort by the military services reduced failure rates to 10%. As a consequence of this lack of reliability, the Defense Department ceased all purchases of heat pumps. In-warranty compressor failures were about 17.8% per year in 1964. Shipments of heat pumps grew slowly from 1963 to 1970 (76,000 to 98,000 units, respectively). During this same period, sales of central air conditoning systems (split and packaged) grew from approximately 580,000 to 1,616,000 units[5].

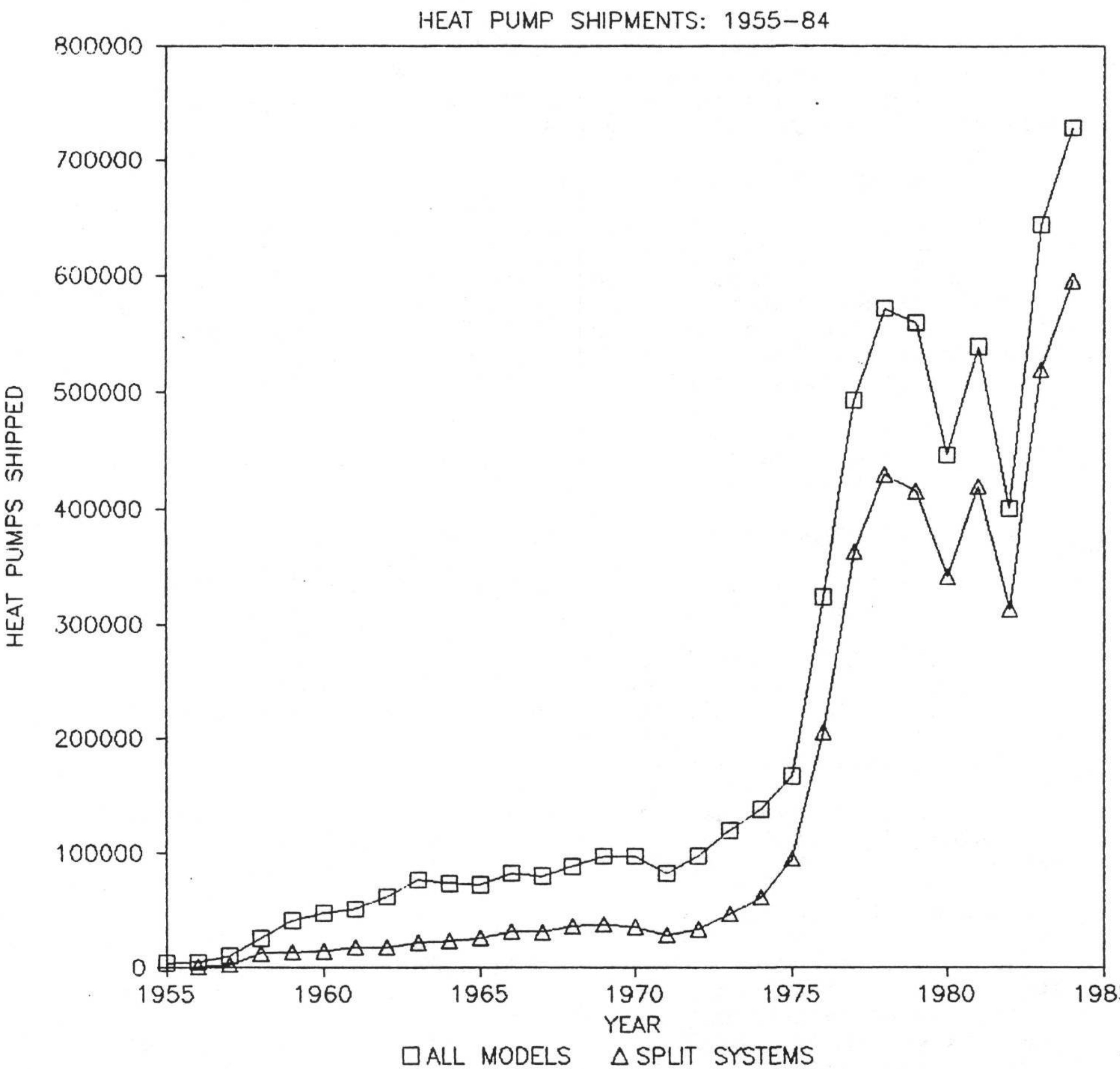


Figure 2.1 - Shipments of heat pumps from 1955 to 1984
(Source: ARI [13])

During the early to mid 70s, the number of heat pump shipments rose dramatically (Figure 2.1). Sales of heat pumps jumped from 97,600 in 1972 to over 500,000 in 1977. Sales fluctuated with the housing market in the late 1970s and early 1980s (Figure 2.1). The peak year for shipments of heat pumps was 1984, when over 729,000 were shipped.

Until 1974, the preferred heat pump was the single package unit (Figures 2.1 and 2.2). During the mid 60s and early 70s, approximately 35% of all heat pumps shipped were split systems. From 1973 to 1977, the market share for split systems increased from 39% to 74%. Since 1977, the market share for split systems has grown to 80%. If recent trends continue, split systems may account for 90% of all heat pumps shipped by 1990. Single package systems are likely to have a much smaller impact on the heat pump market in future years.

Shipments of heat pumps by size have changed over the past eight years (Figures 2.3 and 2.4). From 1976-1984, heat pumps with capacity less than 21900 Btu/hr grew from 6.3% to 15.3% of shipments. The majority of heat pumps are still in the two to three ton size (75% in 1976 and 72% in 1984). The increase in the smaller units could be due to a shift to smaller houses (or apartments), better thermal performance of houses (requiring smaller sized units), or better matching of heat pumps to the thermal loads of houses (i.e., not oversizing the units).

Heat Pump Installations

The number of existing homes (single-family, multi-family, and mobile homes) with heat pumps grew from one million in 1977 to almost 2.2 million in 1983 (Figure 2.5) [8]. Two-thirds of all heat pumps in existing houses are in the South Census Region. The large number of heat pumps in this region might be explained by a combination of factors: (1) gas was not available in some parts of the South, (2) electric prices were traditionally low in those parts of the region served by the Tennessee Valley Authority, (3) the mild winters allow for relative good performance in the heating mode, (4) electric utilities aggressively marketed heat pumps in that part of the country, and (5) that area has a high cooling demand and the heat pump was the only appliance which provided both cooling and heating.

The strong regional bias of heat pump installations can also be seen in construction of new homes. Figure 2.6 shows the penetration of heat pumps in new single-family residences. In the South, heat pumps were installed in approximately four out of every ten new single-family homes in that region in 1984. The heat pump installations in the South accounted for two out of every three heat pumps installed in new single-family residences in the U.S. from 1978 to 1984 (Figure 2.7). Nationally, heat pump installations in new single-family residences have grown from 25% in 1978 to 30% in 1984.

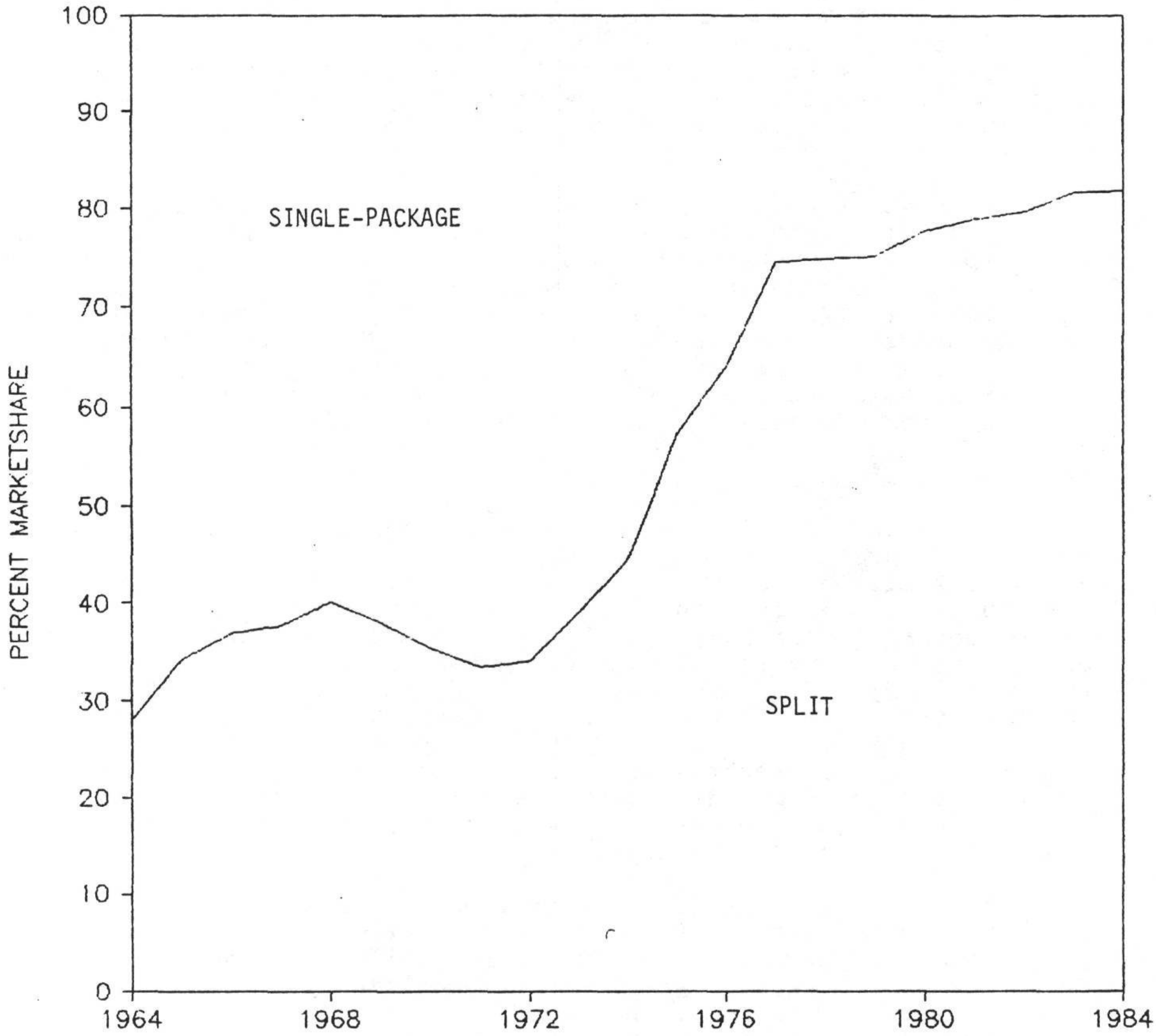


Figure 2.2 - Historical market share of split and single package heat pumps (Source: ARI [13])

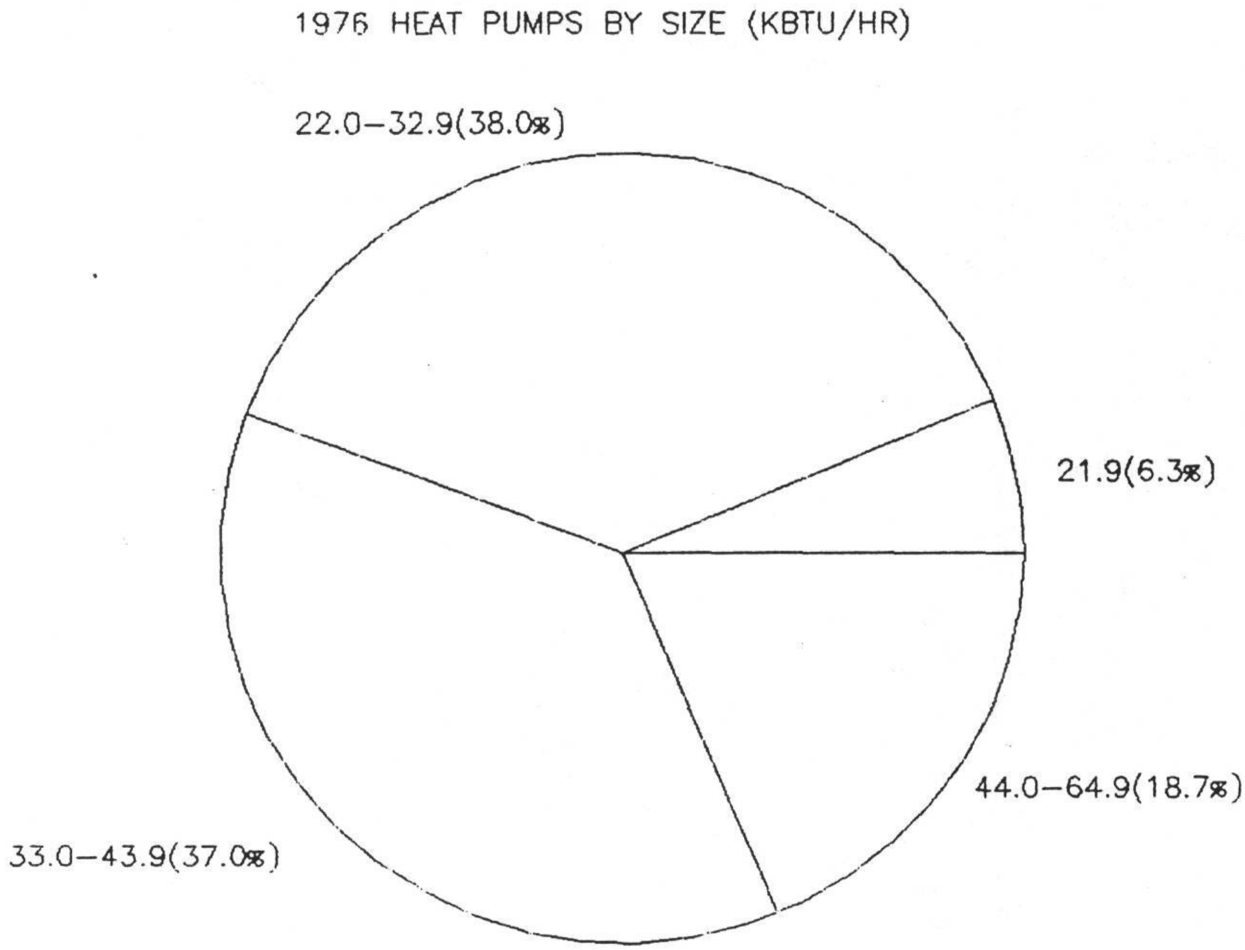


Figure 2.3 - Shipments of heat pumps by capacity in 1976
(Source: ARI [13])

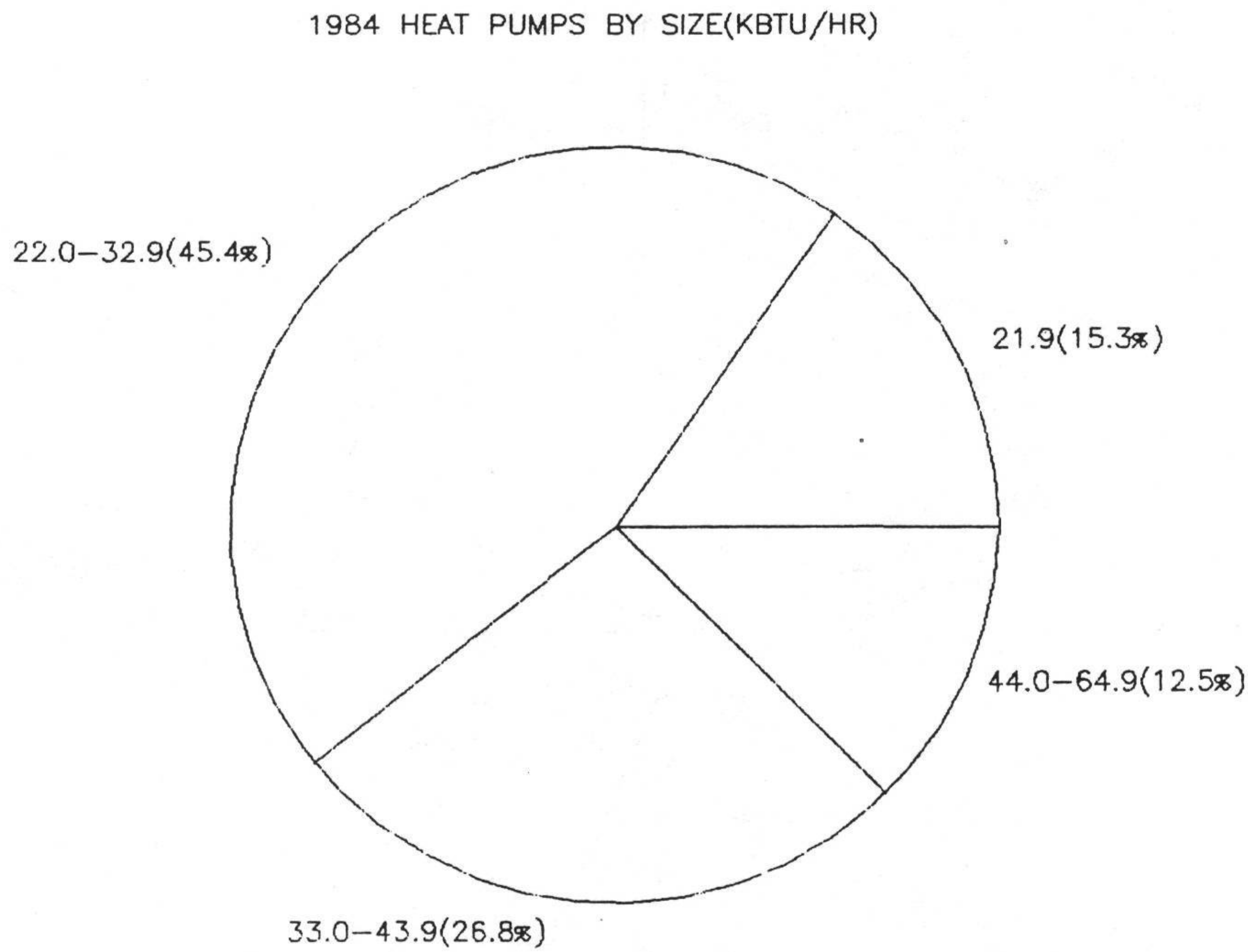


Figure 2.4 - Shipments of heat pumps by capacity in 1984
(Source: ARI [13])

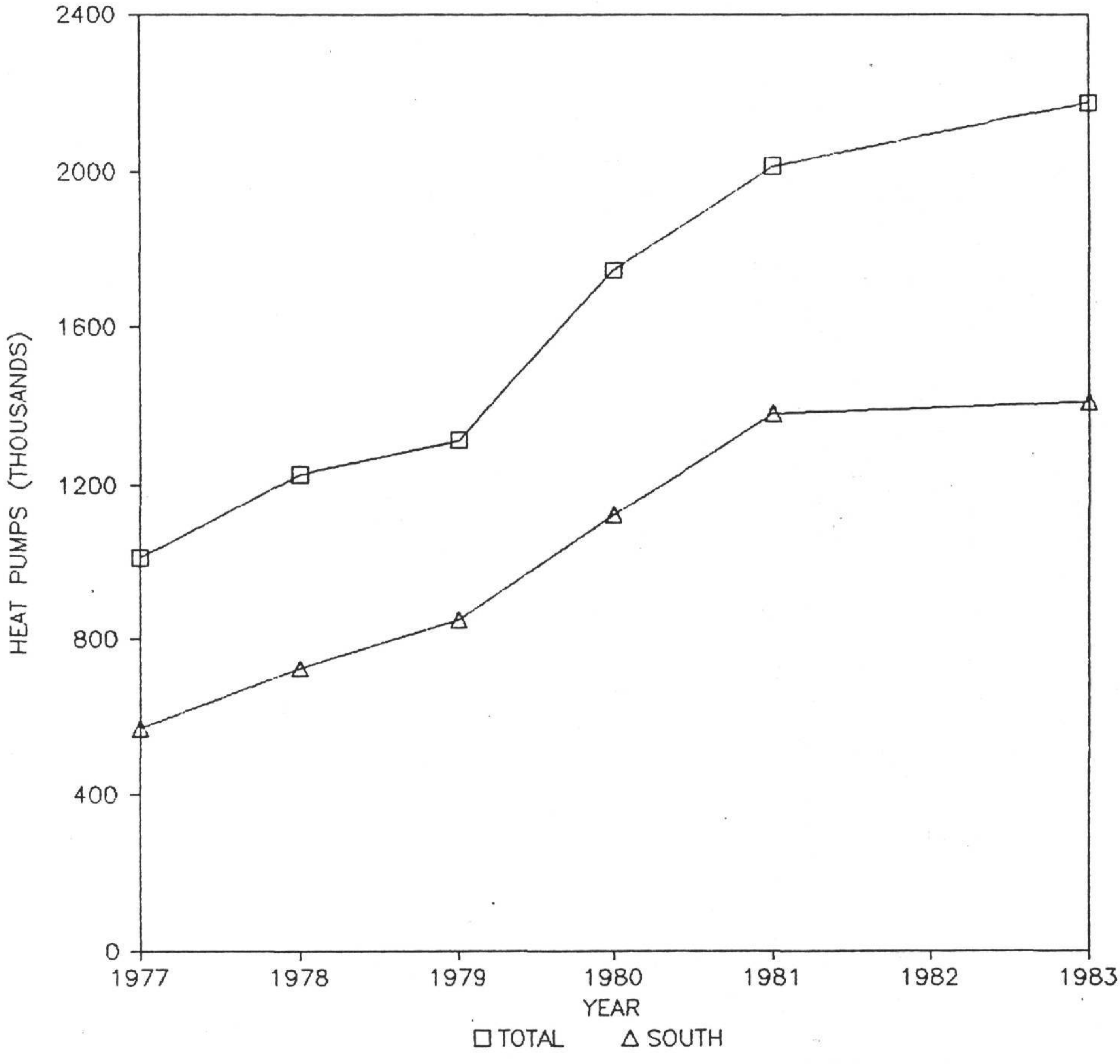


Figure 2.5 - Number of heat pumps in the U.S. Housing Stock
(Source: Bureau of Census [8])

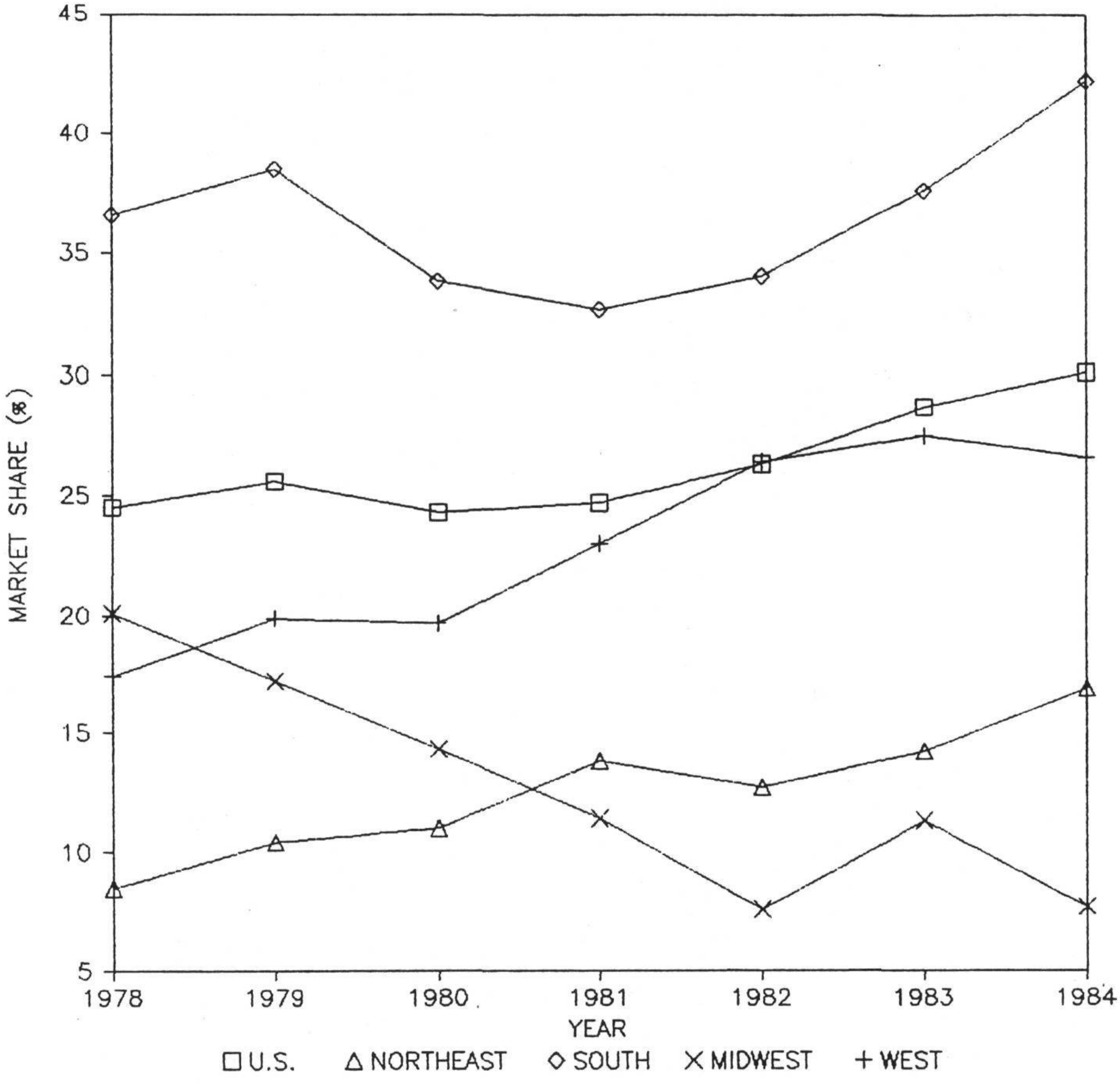


Figure 2.6 - Market share of heat pumps for the U.S. and four Census regions (Source: Bureau of Census [8])

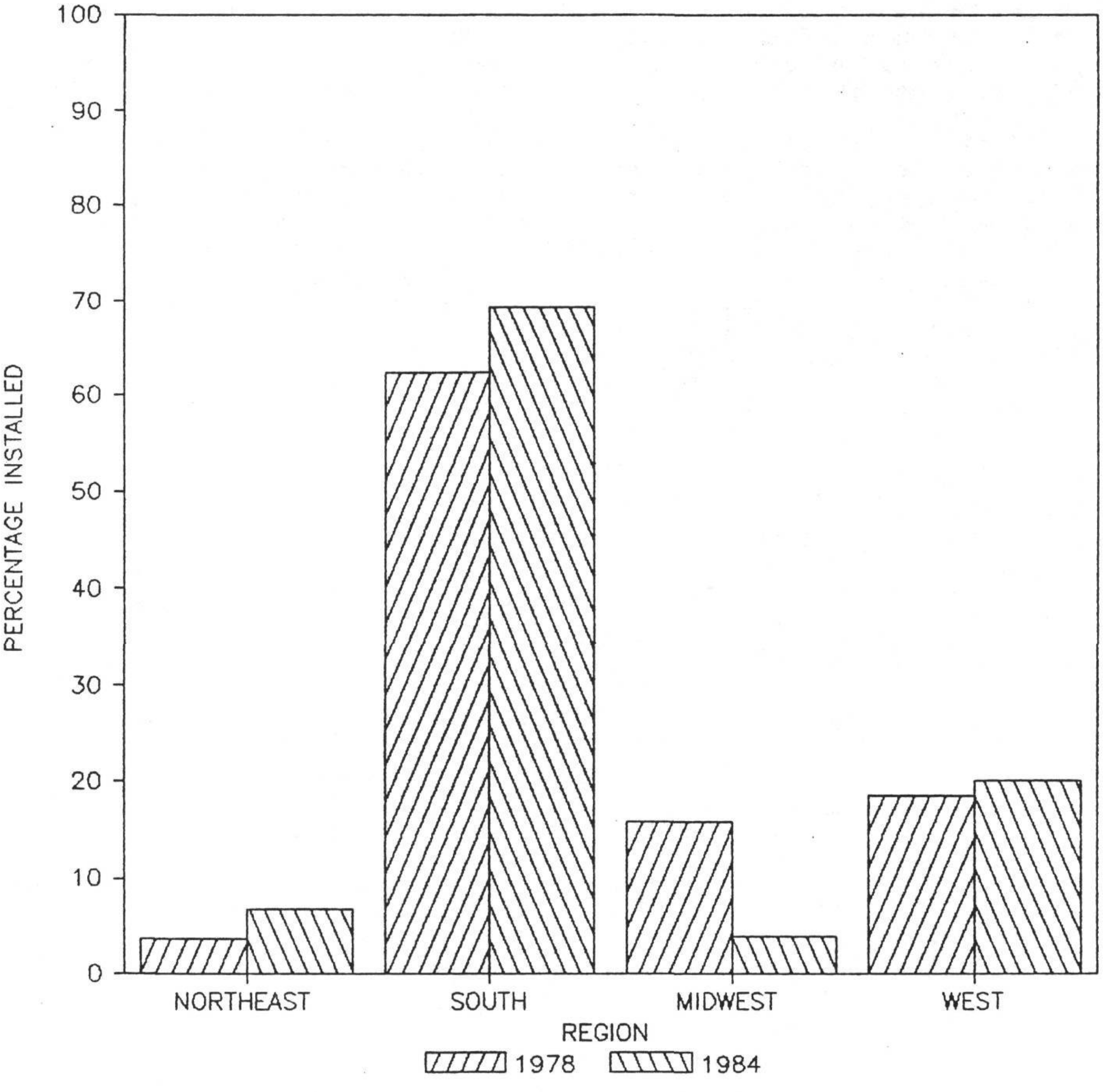


Figure 2.7 - Geographic distribution of heat pumps installed in new single-family residences built in 1978 and 1984 (Source: Bureau of Census [9])

Figure 2.8 shows the distribution of heat pump installations in new multi-family homes in 1980 and 1984. The South Census Region accounted for five out of every ten installations of heat pumps in 1980 and seven out of every ten in 1984. Over 90% of all installations in 1984 were in the combined West and South Regions.

There are at least two possible implications of the large number of heat pumps installed in the South. First, the Department of Energy may want to consider evaluating "average" heat pump performance based on weather data weighted more toward the South rather than the national average. Because the heating performance of heat pumps is strongly dependent on winter climate, estimating heat pump performance should be based on a weather region which is typical of where most heat pumps are installed. DOE Region III which is weighted more towards the South might be a better choice for representing heat pump performance rather than Region IV which DOE has proposed (Figure 2.9).

Another implication of the strong regional bias of the installation of heat pumps is in the forecast of installations of heat pumps. The analysis the Department of Energy performed for central air conditioners used a national forecast of sales and energy use[11]. If the South is the dominant region for new heat pumps installations, these new heat pumps may use more energy for air conditioning and less energy for space heating than existing installations. Rather than doing one national forecast, regional forecasts should be done.

Heat Pump Performance

The major sources of heat pump performance data are the Unitary Directory produced by the Air Conditioning and Refrigeration Institute (ARI), the annual summaries produced by ARI, and manufacturers catalogs and data sheets. Currently, ARI does not report a combined shipment weighted heating and cooling performance efficiency. The only alternative efficiency measure is the cooling Seasonal Energy Efficiency Ratio (SEER). ARI cooling data since 1981 has been reported SEER. Before 1981, ARI reported the Energy Efficiency Ratio (EER). Recently, ARI has started reporting the Heating Seasonal Performance Factor (HSPF).

Figure 2.10 shows the average efficiency in split and single-package heat pumps from 1976 to 1984[12]. From 1976 to 1980, the data are in EER, while for 1981 to 1984, the data are in SEER. The EER and SEER are slightly different measures of the efficiency of heat pumps. The average efficiencies of split and packaged systems have both improved. For instance, the SEER has increased by 0.7 for both split and packaged systems from 1981 to 1984. There does not appear to be any significant differences in the sales weighted average efficiencies of either the packaged or split systems.

When comparing the distribution of efficiencies found in heat pumps shipped in 1984, there do not appear to be any significant differences between split and package systems. The most common

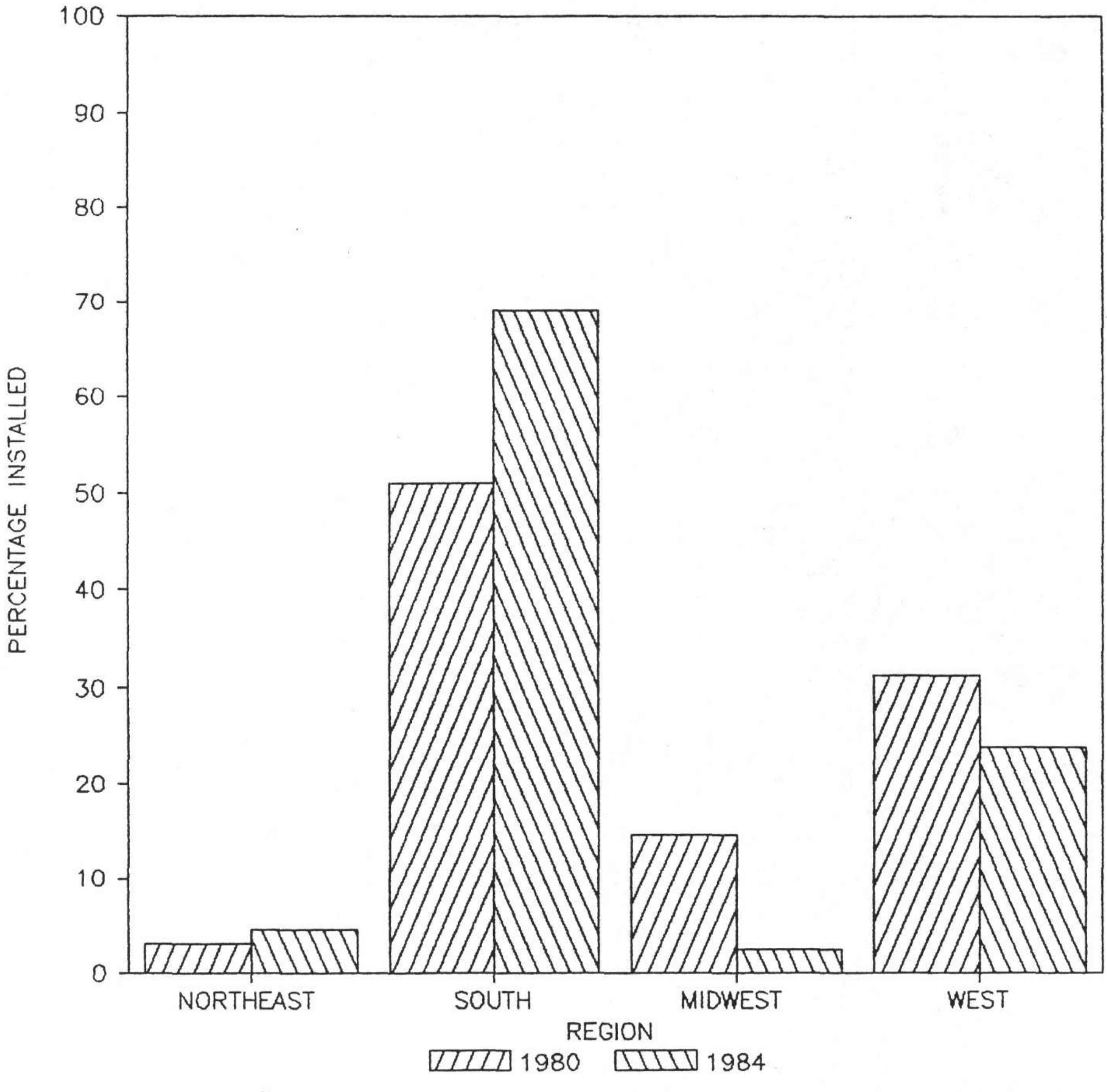
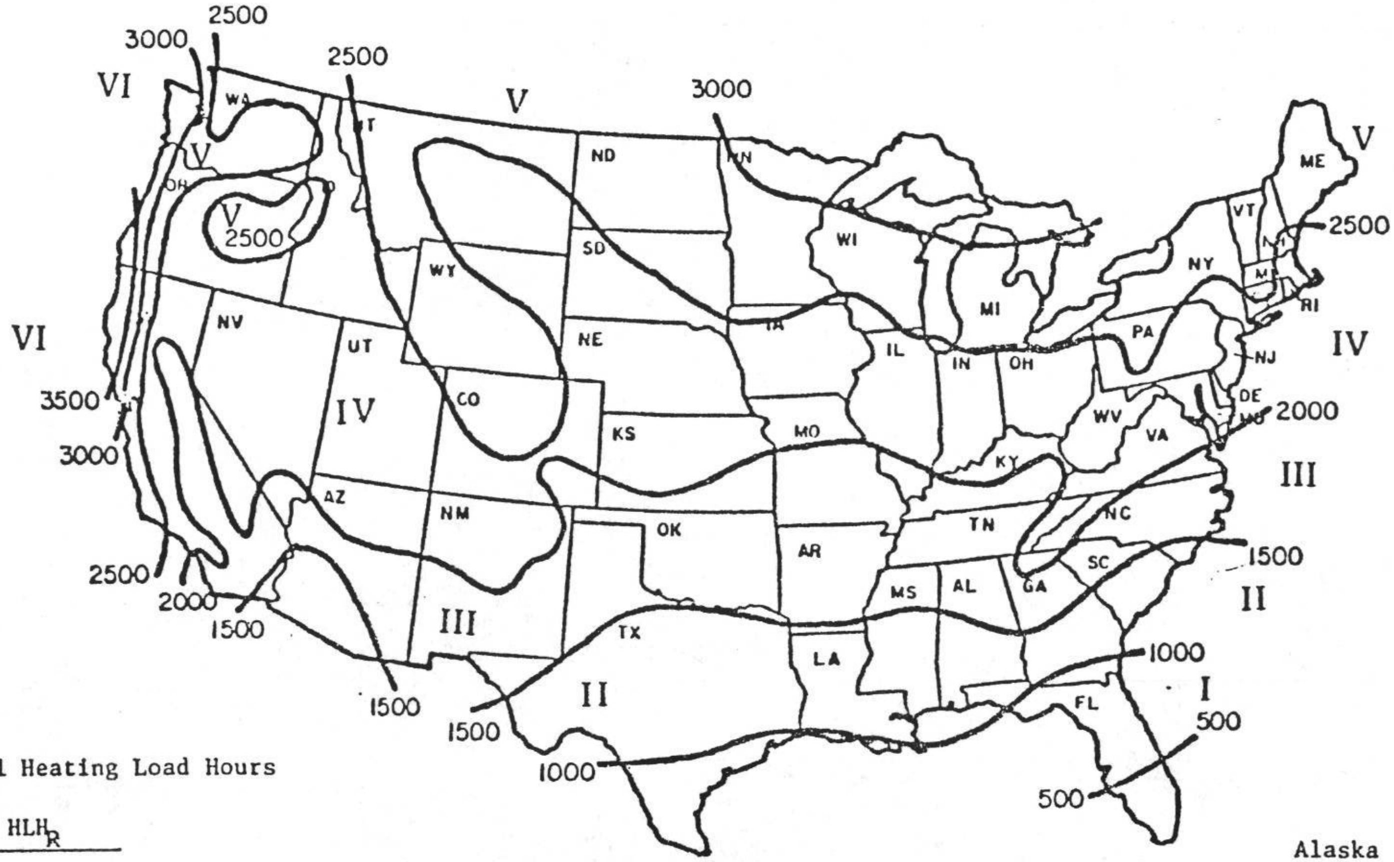


Figure 2.8 - Geographic distribution of heat pump installations in new multi-family residences in 1980 and 1984
(Source: Bureau of Census [9])



This map is reasonably accurate for most parts of the United States but is necessarily highly generalized, and consequently not too accurate in mountainous regions, particularly in the Rockies.

Regional Heating Load Hours

Region	HLH_R
I	750
II	1250
III	1750
IV	2250
V	2750
VI	2750

Alaska	-	3500 HLH
Hawaii and Territories	-	0 HLH

Figure 2.9 - Map of the United States showing the Department of Energy's climate regions.

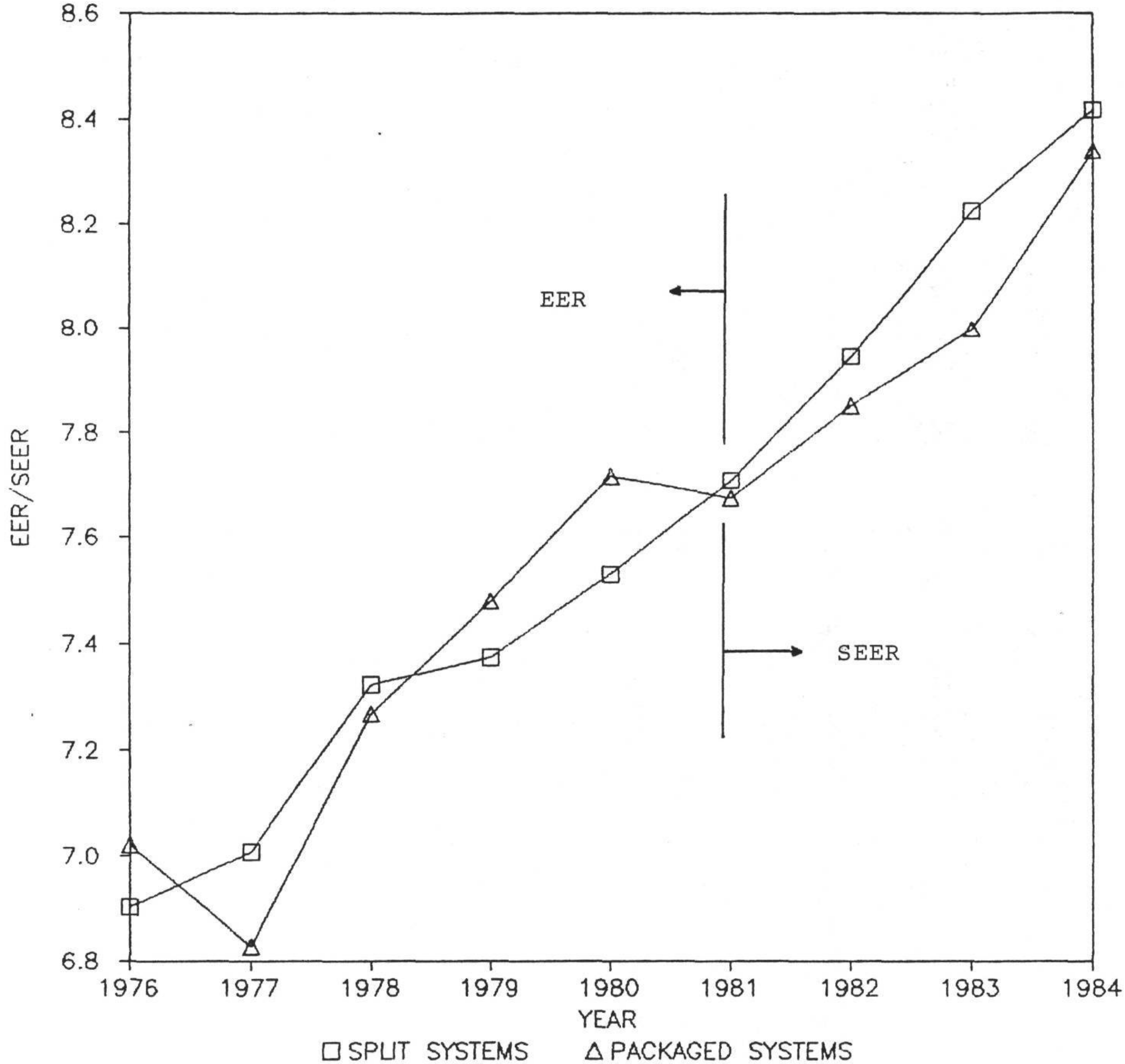


Figure 2.10 - Average cooling efficiency of residential sized split and single-package heat pumps (Source: ARI [13])

unit shipped for both split and package units was one with an SEER of between 8.0 and 8.4. 50.6% of package and 57.6% of split systems were in this SEER range. There was a slightly higher percentage of split systems with SEERs greater than 9.0 than package units (21.0% versus 15.7%). The maximum efficiency package system was between a SEER of 10.0 and 10.4 while for a split system it was between 11.0 and 11.4.

The distribution of efficiencies for both split and single-packaged systems have changed dramatically over the past eight years. In 1976, it was not possible to buy a residential sized split system with an EER above 8.4 or a packaged system with an EER above 9.4[12]*. In 1984, heat pumps were shipped for split and packaged systems with SEERs as high as 11.3 and 10.2, respectively. In 1985, it was possible to purchase a split system heat pump with a SEER of 13.20. In 1976, 50.4% of packaged and 91.8% of split systems had EERs less than 7.4. In 1984, only 0.8% of packaged and 5.7% of split systems had SEERs less than 7.4[12]. Thus, manufacturers have reduced the number of lower efficiency systems.

The lowest efficiency systems on the market in 1985 are listed in Table 2.1. This "bottom" end of the market is important for the Engineering Analysis because it serves as a minimum threshold for consideration of efficiency improvements. The starting point of the analysis should be close to the levels in Table 2.1 and incrementally move up in efficiency until the peak efficiency is reached.

Table 2.1 - Minimum Efficiency Heat Pumps Available in 1985.

Size(tons)	Type	SEER	HSPF
3	split	5.80	5.95
3	package	6.55	5.85
5	split	6.80	6.10
5	package	7.60	5.60

Figure 2.11 shows a plot of SEER versus the HSPF for three ton splite systems heat pumps offered in 1985[12]. The SEER and HSPF values are from the ARI directory which publishes results for Region IV.

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*While EERs and SEERs do not exactly correspond to each other for an individual unit, one study has indicated that the values may be within 6% of each other when making comparisons on a large number of units[13].

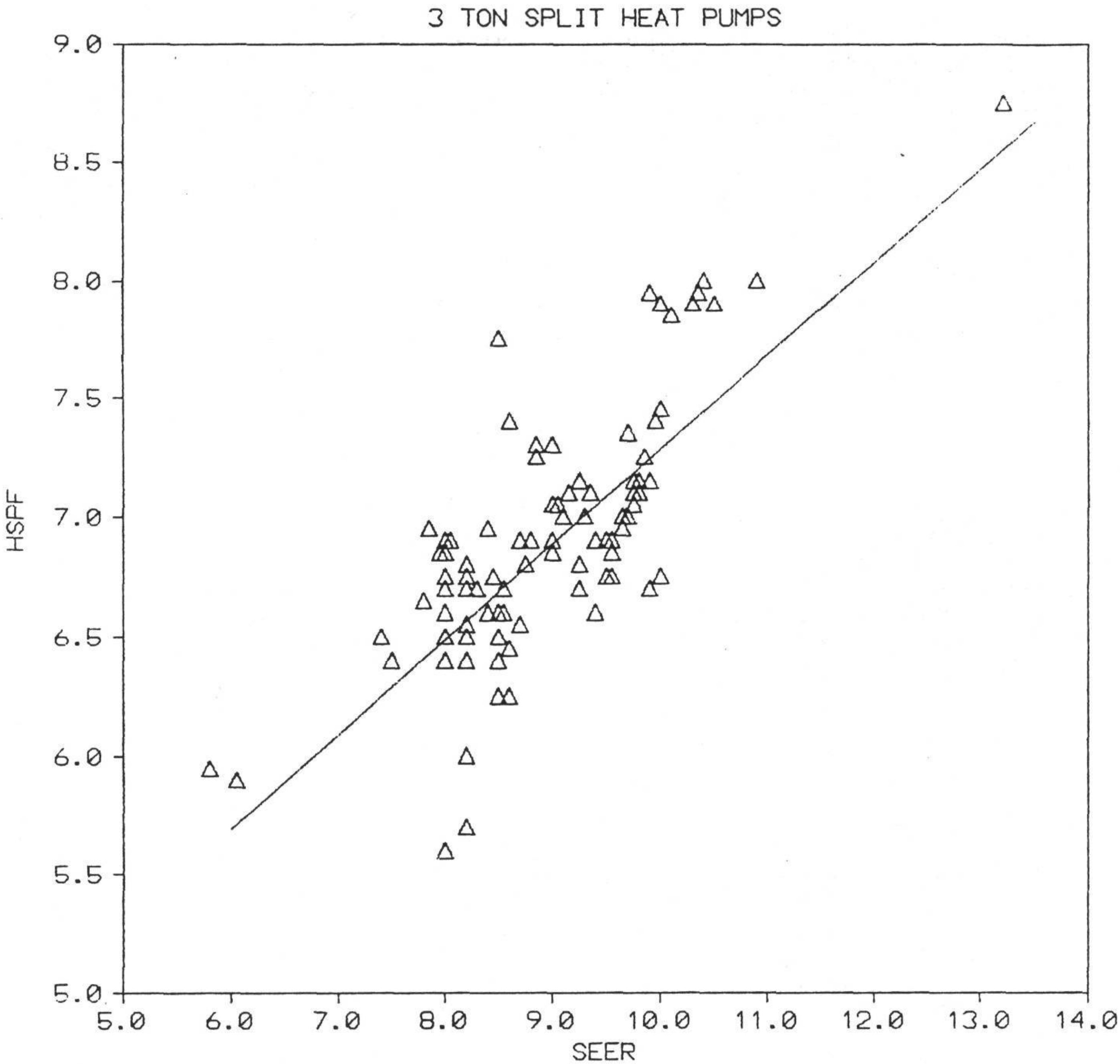

Figure 2.11 - Relationship between SEER and HSPF for 3 ton split system heating pumps sold in 1985. (Source: ARI [12])

There is a definite trend in the data - as the SEER increases, the HSPF increases. However, there is also considerable scatter in the data. For instance, in the 3 ton split system, it is possible to have two heat pumps with equal SEERs of 8.2, but have HSPFs ranging from 5.7 to 7.8. A linear equation was fit to the data with the form:

$$HSPF = A + B*SEER \tag{2.1}$$

where,

A, B = coefficients from the least squares fit

Table 2.2 provides the coefficients for 1.5 and 3.0 ton systems. The coefficient B is the slope of the line which gives the change in HSPF for a given change in SEER. For the systems considered, this quantity varied from 0.27 to 0.84. This wide variation would indicate that it would be difficult to justify use of only one value to relate average changes in SEER and HSPF for all heat pumps. Even for a specific size and type (split and single-package) of system, the r-squared of the regressions are all less than 0.61. Such low correlations indicate the relationship between the SEER and HSPF is dependent on other variables. These variables may include type of expansion device, circuiting in heat exchangers, etc.

Table 2.2 - Regression coefficients for split and package systems.

System	A	B	r-squared
1.5 Ton Split	-1.05	0.84	0.47
3.0 Ton Split	3.31	0.40	0.61
1.5 Ton Package	0.71	0.64	0.55
3.0 Ton Package	4.21	0.27	0.16

The lack of a simple relationship between SEER and HSPF may pose some difficulties in forecasting the energy impact of heat pumps. Both the heating and cooling energy of heat pumps must be known when forecasting the installation and energy savings of heat pumps. Without a simple relationship between the heating and cooling efficiency, a more complex model must be developed. Such a model could include probability distributions of HSPFs around a given SEER. However, data for this type of model is not readily available in the public domain. This issue should be given more consideration by the contractors doing the forecast of the energy impacts of heat pumps.

REFERENCES

1. R. G. Heap, Heat Pumps, E. & F. N. Spon, Ltd, London, 1979.

2. M. Cole and E. Smith, Eds., Democratic Sweden, The Greystone Press, 1939.

3. J. F. Sandfort, The Heat Pumps, Scientific American, May 1951.

4. H. Manchester, "Fireless Heat for the Home", Scientific American, May 1947.

5. Air Conditioning and Refrigeration Institute, shipments data, May 15, 1984

6. A. Trask, "Ten Design Principles for Air-to-Air Heat Pumps", Proceedings of the 2nd Annual Heat Pump Technology Conference, October 18-19, 1976, Oklahoma State University, Stillwater, Oklahoma.

7. H. S. Kirshbaum and S. E. Veyo, "An Investigation of Methods to Improve Heat Pump Performance and Reliability in a Northern Climate, Vols. I, II, and III (EPRI Report No. EM-319, January 1977).

8. U.S. Bureau of the Census, Annual Housing Survey, 1977 and later editions, U.S. Department of Commerce, Washington, D.C.

9. U.S. Bureau of the Census, Construction Report, Series C25, "Characteristic of New Housing", 1975 and later editions, U.S. Department of Commerce, Washington, D.C.

10. "Test Procedure for Central Air Conditioners, Including Heat Pumps", Federal Register, Dec. 27, 1979, pp. 76700-76723.

11. "Economic Analysis Document", DOE/CE-0029, U.S. Department of Energy, March 1982.

12. Directory of Certified Unitary Air Conditioners, Air Source Heat Pumps, and Sound-Rated Outdoor Unitary Equipment, Air Conditioning and Refrigeration Institute, Jan. 1 - June 30, 1985, Arlington, VA.

13. "Comparative Study of Energy Efficiency Ratios", Air Conditioning and Refrigeration Institute, Apr. 1984, Arlington, VA.

14. H.V. Nguyen, et al, "Trends of Residential Air-Conditioning Cycle Tests", ASHRAE Transactions, Vol. 88, Pt. 2, pp. 954-972.

15. "Energy Conservation Program for Consumer Products", Federal Register, pp. 43976-44087, June 30, 1980.

16. "Engineering Analysis", DOE/CS-0166, U.S. Department of Energy, June 1980.

17. "Energy Conservation Program for Consumer Products", <u>Federal Register</u>, pp. 14424-14460, April 2, 1982.

18. "Engineering Analysis", DOE/CE-0030, U.S. Department of Energy, March 1982.

CHAPTER 3

HEAT PUMP CLASSES

The Department of Energy currently has defined three classes of heat pumps[1]:

(1) Air source, split system,
(2) Air source, single package system, and
(3) Air source, split system, heating only.

To create any new classes, a particular heat pump model must satisfy the following criteria[2]:

(1) have a different primary energy source, i.e., oil or gas,
(2) have a different capacity or other performance related feature which affects efficiency and utility, or
(3) have features providing utility that also affect the efficiency of the model.

A potential new class of heat pumps could be the gas driven heat pumps. These would include those heat pumps that use gas in an absorption cycle, or in an engine to drive a refrigeration cycle. However, because neither of these technologies are not fully developed and because these heat pumps do not have a finalized test procedure to rate their performance, it would be premature to include them as a new heat pump class.

The second criteria can be used to justify the creation of most classes of heat pumps. Heat pumps available today use either the air, ground, or water for a heat sink(or source). Water(or ground) source heat pumps use the water(or ground) for a heat source in the winter and a heat sink in the summer. Air source heat pumps use the air as both a heat source or sink. Because the ground has smaller seasonal and daily temperature fluctuations than the outdoor air, ground source heat pumps tend to have higher seasonal efficiencies for both heating and cooling than air source units. Similarly, water source heat pumps also tend to have higher seasonal efficiencies than air units because their heat source(or sink) is a well or lake water, which has smaller seasonal and daily temperature fluctuations than the ambient air. The construction of water(or ground) source heat pumps is also different from air source heat pumps. Both water and ground source heat pumps use a water-to-refrigerant heat exchanger in place of the outdoor coil usually found on air source heat pumps. The water circulated through this heat exchanger is then used to transfer heat to(or from) the ground or water via tubes in the ground or water. Because of the differences in heat source and differences in construction, there should be seperate classes for water(or ground) source heat pumps and air source units.

The second criteria can also been used to justify the divisions between single package and split heat pumps. Single-package designs have inherent heat transfer and infiltration losses between the indoor and outdoor sections due to the close proximity of the air flow paths. Because both the evaporator and condenser coils must be placed within the single package "box", there will be more constraints on the increased size of heat exchangers in single package than in split systems.

In the earlier rulemaking for central air conditioners, split and single package central air conditioners units were divided into classes by capacity[3,4]. A seperate class was created for units with capacities greater and less than 39000 Btu/hr capacity. While not explicitly stated in the Engineering Analysis Support Documentation, the classes based on capacity were created because of engineering considerations relating to the size of the indoor evaporator coil[4].* This coil must fit within the air handling section which is matched to a furnace. This matching puts physical constraints on the face area of the evaporator. For the 39,000 Btu/hr and under, the face area was limited to 5 square feet. For units larger than 39,000 Btu/hr, the face area was limited to 7 square feet.

While the indoor coil on a heat pump is not matched to a furnace, it must fit within the box or the ductwork containing the air handling unit. Limitations on the size of the indoor coil will limit the maximum efficiency that is feasible for a heat pump. Figure 3.1 shows the maximum SEER for optimized air source, single-speed, split heat pump systems built with "off-the-shelf" components. These values were calculated with the performance models described in Chapter 4. The maximum size of the indoor coil ranged from a face area of 3.8 sf for the 1.5 ton heat pump to 7.0 sf for the 5.0 ton heat pump. These maximum sizes are within the practical limitations for the size of the air handlers for those capacity systems. Given these constraints, it is possible to build a 1.5 ton heat pump with an SEER as high as 15.8 while for a 5.0 ton heat pump, the maximum SEER is only 10.7. The restraint on the indoor coil size has a much smaller effect on the HSPF than the SEER. The outdoor coil, which controls the amount of heat absorbed from the ambient air, is oversized with respect to the indoor coil in the heating mode. Figure 3.2 shows the HSPF for the same optimized systems shown in Figure 3.1.

We recommend a division of at least two heat pump classes by capacity similar to that done with air conditioners. Because of the precedent set with the central air conditioners, the 39,000 Btu/hr break in capacity should be kept. The Department of Energy might also want to consider another division below 39,000 Btu/hr. A large fraction of the units in the smaller capacity class have capacities less than 27,000 Btu/hr. Forty-four percent of split and 25% of single package systems had capacities below 27,000

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*The Engineering Analysis Document cited "manufacturers of larger air conditioner units would bear a disproportionate cost in meeting the proposed standards" as justification[4].

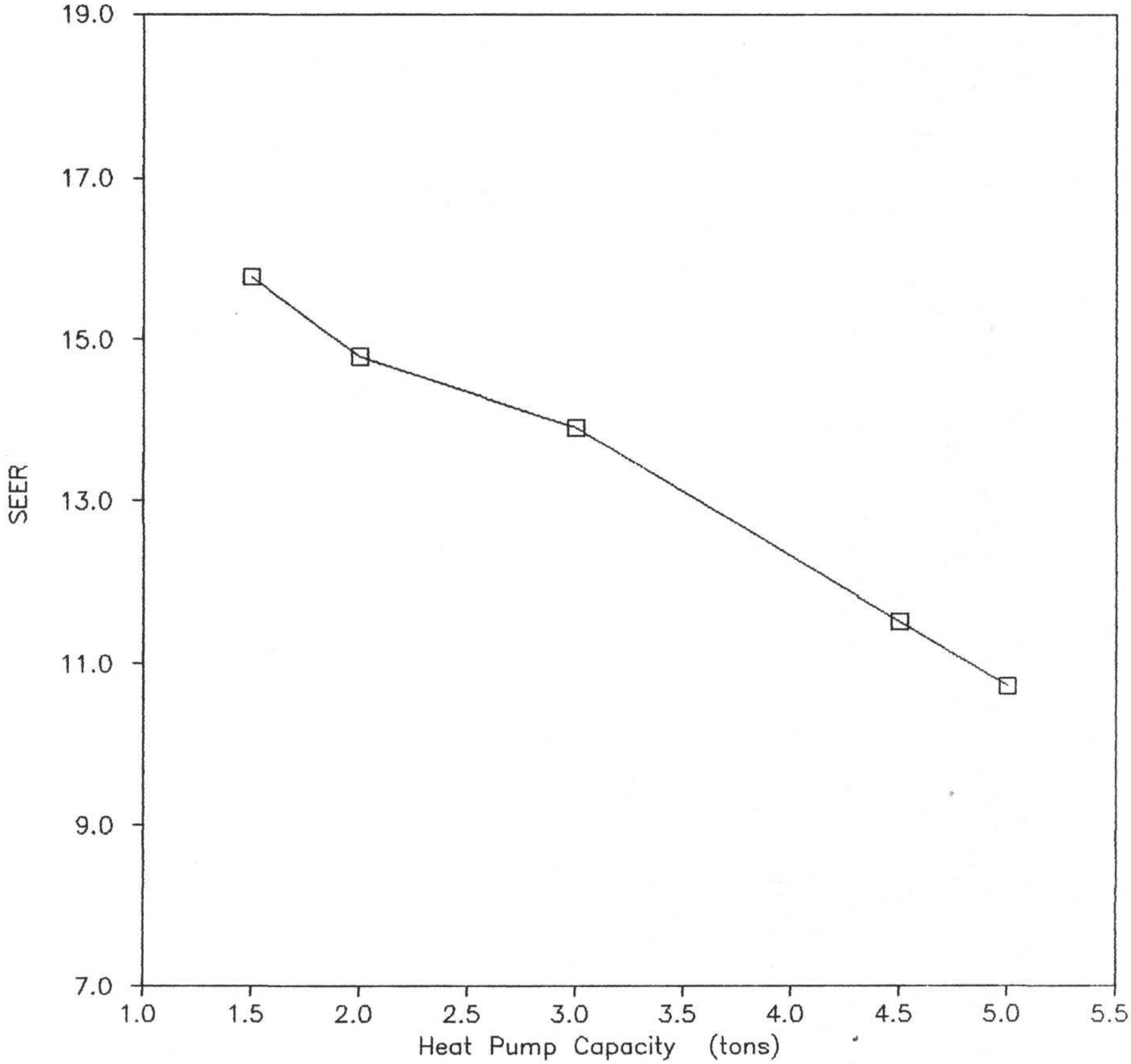

Figure 3.1 - Maximum calculated SEERs for single-speed heat pumps

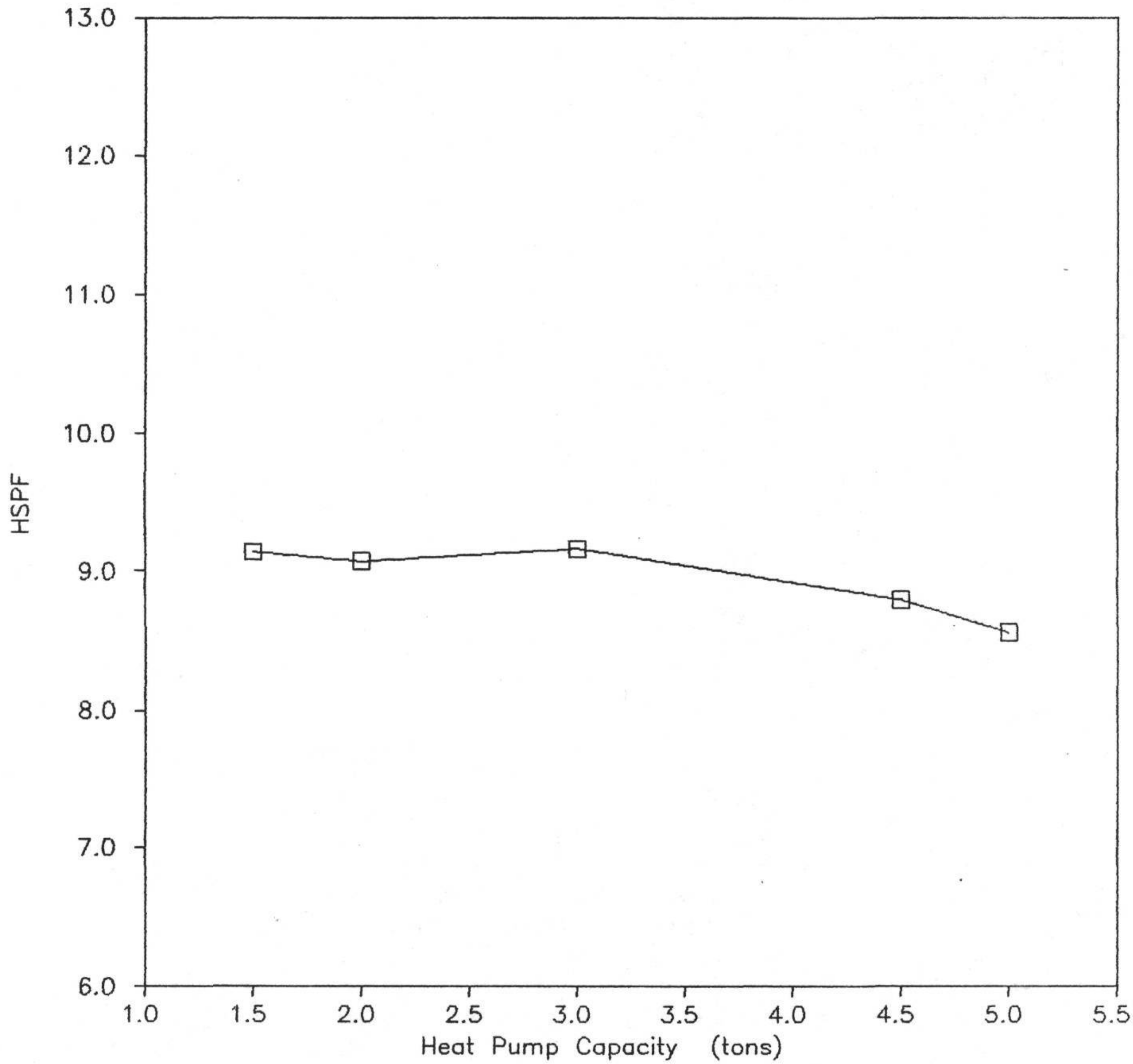

Figure 3.2 - HSPFs for the single-speed heat pumps in Figure 3.1

Btu/hr in 1984[5]. The units under 27,000 Btu/hr have a better potential for reaching higher efficiencies than do the units between 27,000 and 39,000 Btu/hr.

The existing air source heating only heat pump class should be retained. This heat pump does not have the same consumer utility as other heat pumps because of its lack of air conditioning. However, because there may not be any heating only heat pumps in production at this time, the Department of Energy should consider the analysis of this class a low priority. It also does not appear that this class of heat pumps should be divided by capacity as some of the other heat pumps. Because they are heating only, the constraints on the size of the indoor coil will not significantly affect their performance as they do those that also have cooling.

A class should be created for multi-zone heat pumps. These units both provide utility to the consumer and affect the energy use of the units. With a multi-zone heat pump, a consumer has the option of cooling or heating only the particular rooms(or zones) in a house that he chooses. Each room(or zone) has a seperate thermostat and indoor coil unit. Energy use for these units can be smaller because only the occupied rooms in the house need to be conditioned. At the same time, it can improve the occupants comfort, because the environment in each room can be controlled by the occupant of the room.

The current recommendations for classes are listed in Table 3.1. The analysis of efficiency improvements in this report explicitly covers the first four classes. It should be possible to use the heating side of the analysis of the air source package and split systems to evaluate the efficiency improvements in heating only heat pumps.

Table 3.1 - Heat pump class recommendations

Heat Source/Sink	Type	Capacity (Btu/hr)
air	split	less than 39,000
air	split	39,000 to 65,000
air	package	less than 39,000
air	package	39,000 to 65,000
air	split, heating only	less than 65,000
air	package, heating only	less than 65,000
air	multi-zone	less than 65,000
Water		less than 39,000
Water		39,000 to 65,000

REFERENCES

1. "Energy Conservation Program for Consumer Products", Federal Register, pp. 43976-44087, June 30, 1980.

2. "Engineering Analysis", DOE/CS-0166, U.S. Department of Energy, June 1980.

3. "Energy Conservation Program for Consumer Products", Federal Register, pp. 14424-14460, April 2, 1982.

4. "Engineering Analysis", DOE/CE-0030, U.S. Department of Energy, March 1982.

5. "Comparative Study of Energy Efficiency Ratios", Air Conditioning and Refrigeration Institute, April 1985, Arlington, VA.

CHAPTER 4

SELECTION AND VALIDATION OF PERFORMANCE MODELS

Improvements in heat pump performance can be most easily evaluated with a good system simulation model. For those design changes that are beyond the capabilities of the simulation model, engineering judgement can be used. Both approaches are used in this analysis. For conventional design options, computer models are used to estimate energy use(and efficiency) of the heat pump. For some advanced technology design options, engineering judgement is used because data are not available.

Two models of heat pumps are required for this analysis. The first is a steady state model that provides estimates of efficiency and capacity at specified temperatures. The second model is a seasonal performance model (SPM) that uses the steady state information plus the DOE test procedure to produce estimates of the Seasonal Energy Efficiency Ratio(SEER) and Heating Seasonal Performance Factor(HSPF). Both models are described below. A detailed description of the SPM is in Appendix B. An example of the steady state model output is provided in Appendix A.

Steady State Model Selection

Two steady state public domain heat pump models have received acceptance in the engineering community. One was developed by Oak Ridge National Laboratory (ORNL) and was completed in 1981[1]. The other was developed at the National Bureau of Standards (NBS) and was finished in 1983[2]. Nearly every manufacturer has their own proprietary model or a modified version of one of the above models.

After a thorough review of both the ORNL and NBS heat pump models, we chose the ORNL model. The primary reasons for using the ORNL model over NBS centered on the ORNL's: (1) methodology, (2) compressor model, (3) expansion device models, (4) fan model, (5) detailed output. These items are discussed below.

Model Methodology

The two heat pump models were developed with different objectives. The NBS model was developed as a Ph.D. dissertation with a primary objective being to provide a system model that incorporates a good capillary tube model[2]. The ORNL model was intended for use in evaluating the performance of systems using actual hardware, with component performance provided by empirical or measured relationships. The NBS model is more academic in that it is fundamental in how it handles certain components, particularly the compressor. Much data needed for the NBS model cannot be obtained from available sources. The ORNL model is more practical to implement for this study.

Compressor Models

The Oak Ridge model provides two choices of compressor models. One model uses manufacturer's compressor maps to estimate refrigerant flow rates and power consumption. Software is also provided to curvefit a polynomial to the compressor performance data.

The other ORNL compressor model does an energy balance on the compressor, and requires the motor efficiency, mechanical efficiency, and isentropic compression efficiency, etc., as input. These values are usually not measured during compressor efficiency tests. For existing hardware, the map based model would be more accurate in generating output conditions.

The NBS compressor model is similar to the latter ORNL compressor model discussed above, but is more fundamental. It computes compression efficiencies from polytropic relationships. It also requires detailed motor performance characteristics that are not readily available. The NBS model cannot readily handle variable speed, screw, rotary, or scroll compressors.

Condenser and Evaporator Models

The ORNL model uses a simpler approach to determining heat exchanger performance than does the NBS model. Effectiveness as a function of the number of transfer units (NTU) is computed for the dry parts of the heat exchangers. A modified NTU form is used for the portions producing condensation. Heat transfer coefficients are calculated with existing correlations[3,4,5].

Because of its simplicity, the ORNL heat exchanger model is faster than the NBS. The ORNL model allows for the use of wavy, louvered, and other fin designs if performance data are available. The NBS model is limited to smooth fins. It also does not compute the air-side pressure drop in the coils. Either model should be able to produce acceptable trends for changing variables, such as the fin density or number of tube rows, but they cannot be expected to handle the hundreds of fin shapes and configurations currently in production.

The two models treat refrigerant circuiting differently. The number of parallel circuits is estimated by the user and input to the ORNL model. In the NBS model, performance of the coil is estimated on a tube-by-tube basis. This accounts for performance differences associated with complex tube circuiting where tubes are combined or split to optimize pressure drop characteristics.

In comparing the NBS and ORNL heat exchanger models, the NBS model emphasizes refrigerant side performance while ORNL emphasizes the total system effect. ORNL used much simpler algorithms to determine the coil heat transfer rate, but included calculations of air side pressure drop and computed fan power. NBS used very detailed refrigerant side calculations on a tube-by-tube basis, but ignored the fan power calculation, opting to have it as an input variable. Overall, the NBS model should give results that are better than those from the ORNL model for conventional smooth plate coils, provided the fan power is known. From a design perspective, the tedious specifications needed for the NBS model would be hard to justify, unless the same tube-by-tube configurations were simply specified for every coil. The ORNL model would probably give better results for the more exotic fin configurations and fin spacings where coil air-side pressure drop and fan power are unknown.

Expansion Devices

Three expansion devices are widely used in heat pumps: the capillary tube, fixed orifice, and thermal expansion valve(TXV). Capillary tubes consist of one or more small bore (typically about 0.05" inside diameter) copper tubes. The fixed orifice is normally a small plug that fits into a modified flare connector just upstream of the evaporator. A TXV adjusts its flow area to compensate for changes in operating conditions. It can operate more efficiently over a wider range of conditions than orifice or capillary tubes. TXVs are found on many of the higher efficiency heat pumps.

The ORNL model allows for modeling of all three devices. The NBS model only handles capillary tubes. The TXV is designed to maintain a constant temperature difference between the suction gas and the evaporator phase change temperature. The TXV algorithm in the ORNL model adds the superheat value to the two phase temperature as one of the specifications in the iteration process.

The ORNL orifice model uses a simple orifice equation with empirical data from Mei [6]. The accuracy of the orifice model is probably limited to the orifice designs studied in Mei[8].

The ORNL model uses a simple capillary tube model based on correlations found in the ASHRAE Equipment Handbook [7]. These correlations should be most accurate near the design cooling condition(82 F). The accuracy of these equations in the heating mode is probably poorer than in the cooling mode. The NBS model uses a more sophisticated capillary tube model based on a study by Erth[8]. The NBS procedure is costly in computational time and programming length.

In summary, the NBS model does a thorough job of computing capillary tube performance. However, it has no provisions for either fixed orifices or TXV's. The ORNL model should do an adequate job of predicting capillary tube performance in the cooling mode. The greater the deviation from this mode of operation, the greater the expected error. The ORNL model should do a good job of predicting TXV performance if certain rated flow conditions are known for the valve.

Refrigerant Charge Inventory

The performance of a heat pump system is sensitive to the refrigerant charge in the system [9]. The ORNL model has no charge inventory accounting procedures. It is equivalent to modeling systems with liquid receivers where refrigerant charge is unimportant and state conditions are determined solely by external operating conditions. Not accounting for refrigerant charge might make cooling predictions for systems somewhat inaccurate. If manufacturers optimize refrigerant charge at the 82 F rating point, errors should be small since it is in the middle of its normal cooling range.

The NBS model does calculate charge inventory. The model uses input data on internal volumes, computed refrigerant densities where single phase conditions exist, and vapor void fraction in the two phase portions of the heat exchangers.

Fan Power Calculations

The two fans in a heat pump typically demand about 15% of the total system power for cooling operation, and as much as 25% for heating operation. The ORNL model computes variable fan power using correlations from several sources [10,11,12]. These correlations account for fin type, fin pitch, air velocity and tube rows. Pressure drops for ducts, filters, heater elements, and cabinet pressure drops are also computed. Outdoor air density is included in the calculation of outdoor fan power. Fan power is computed using a combined fan/motor efficiency (the product of fan static efficiency and motor efficiency).

The NBS model does not calculate fan power. Values are calculated by hand and input for indoor and outdoor coils.

Model Output

The ORNL model output in its shortened form includes detailed state point data at each component inlet and outlet(Appendix A). Compressor power, flow rate and efficiency data are shown. Coil heat transfer data are shown for each part of the coil (superheated, saturated, subcooled) along with pressure drop data. Air side pressure drop and fan power are also given. Total, sensible and latent heat transfers are listed separately. A final summary of heating or cooling capacities, electrical power consumption, and COP is tabulated.

The NBS output is substantially less detailed and harder to follow than the ORNL model. Complete specifications (temperature, pressure, enthalpy, entropy and quality) are given for 13 locations around the heat pump cycle. In addition, saturation temperatures at the compressor inlet and outlet are given, along with suction gas superheat. Refrigerant charge and mass flow rate are given along with capacity, total power, and COP. Intermediate values can be printed, however, they are generally hard to follow for one not intimately familiar with the variable names in the model. Few refrigerant heat transfer coefficients, air side heat transfer coefficients, line heat losses, etc. are available for inspection.

Conclusions of Model Selection

Based on the previous comparison of the two models and what is required to evaluate the performance of heat pumps, the ORNL model provides the best results. The NBS model has several excellent features: 1) it performs state-of-the-art computation of capillary tube performance, 2) it computes tube-by-tube heat transfer and pressure drop calculations on the refrigerant side of the coils, 3) it keeps a refrigerant charge inventory and balances the system performance according to a prescribed charge. For this analysis, it also has some very major shortcomings: 1) it only handles reciprocating compressors, 2) it only handles capillary tube expansion devices, 3) it requires as input the indoor and outdoor fan power values, 4) many of the compressor parameters are easily obtainable. Items 1 and 2 alone are sufficient to eliminate it from consideration, since many heat pumps use TXV expansion devices or rotary compressors.

While the ORNL model wins by default, it does have its own shortcomings: (1) lack of refrigerant charge inventory and (2) simplistic capillary tube model. Off-design points will be somewhat suspect since the effect of refrigerant charge is not modeled. For systems with an accumulator, charge should have little impact on heating operation when excess refrigerant usually exists in the accumulator. Because SEER ratings are based on 82 F tests, refrigerant charge is probably optimized for this condition, so that all results expressed by SEER are consistent with proper charge.

The strong points of the ORNL model are that it uses available compressor maps, has good error diagnostics and printouts, executes quickly and computes fan power reasonably well. The compressor maps permit a prototype compressor to be estimated by simply taking an existing compressor map and shifting a curve up or down to reflect an improvement in performance. The output is also in a form that is easily read and understood and allows the user to quickly assess the performance of the system.

Steady State Model Validation

To examine the accuracy of the ORNL model, detailed hardware data(coil size, fin density, compressor maps, etc.) were obtained for five heat pumps on the market in 1985. The ORNL model was used to compute system capacity and COP for each heat pump. Some of the experimental data needed for validating the ORNL model are more detailed than what manufacturers usually measure. Consequently, some input data input was not received from the manufacturers or was not in the form that was needed. Detailed operating conditions were not given, so it was not possible to compare most of the model`s intermediate calculations. In each case, the available data was input and appropriate assumptions were made to satisfy all other input requirements. For example, the ORNL model requires a combined fan-motor efficiency as input. In one case, only a fan motor efficiency of 55-60% was given. It was necessary to assume a fan efficiency to get reasonable fanpower consumptions.

There are several input variables to the ORNL model that can only be approximated. One is the equivalent number of parallel circuits in each coil. In most coils, tubes either combine as the refrigerant condenses or splits as refrigerant evaporates. A coil may have six inlet tubes and a single outlet tube, or the reverse could be true. The number of equivalent circuits is a convenient way to estimate an average Reynolds number in the coil, so that pressure drop and heat transfer can be calculated. When tubes either combine or split in a coil, the only way to determine what the effective number of parallel circuits should be is by comparing the computed refrigerant pressure drop through the coil. In general, the pressure drop in an evaporator should be approximately 5 psi or less. A somewhat larger loss of 20 psi or more can be acceptable in the condenser. To minimize compressor power, zero pressure drop is optimum. However, lower pressure drops result from lower flow velocities which also produce lower heat transfer. Consequently, proper coil circuiting becomes a compromise between the negative effects of pressure losses and the positive effects of improved heat transfer. Every manufacturer has different philosophies about coil circuiting, so using the ORNL model to reproduce performance characteristics becomes a trial-and-error procedure of guessing the equivalent circuits for their coil configuration. Circuiting differences can account for possibly a 10% variation in capacity or COP.

Tables 4.1 and 4.2 summarize the comparisons of the ORNL model predictions with data from two manufacturers. Case 1 is a high efficiency unit which has a two speed compressor. The data shown are for high speed only. No SEER's were computed because they are found from steady state data after including cycling effects which cannot be determined analytically. Case 1 shows agreement in capacity and COP which is consistently on the high side, but within 8% of measured COP in the worst case. Capacity is consistently within 4%. It is difficult to say whether any significance should be attributed to the 7 and 8% errors in COP in the cooling mode versus the 3% errors in the heating mode. Much of the errors could be attributable to the compressor curves supplied by manufacturers. The curves typically have errors of plus or minus 5% on both capacity and efficiency.

Case 2 is a package system with a capillary tube expansion device. Capacity is seen to vary from 8% high in the cooling mode to 6% low in the heating mode. COP is as much as 11% high. This particular unit was difficult to model accurately because it used two capillary tubes in series in the cooling mode. This arrangement is not common, and is not directly accounted for in the ORNL model. For the heating mode, the capillary flow factor was adjusted until capacity and COP was reasonable at 47 F. The same flow factor also gave reasonable results at 17 F. The series

Table 4.1 - Comparison between the ORNL model output and test data for three residential sized heat pumps.

Case 1: Split System, TXV
Two Speed Compressor
10.11 sf, 3 Row Outdoor/6.39 sf, 3 Row Indoor Coil
High Efficiency Fan Motors

Temp	Measured Capacity	Measured COP	Calculated Capacity	Calculated COP	Percent Error Capacity	Percent Error COP
95	36,500	2.78	37,700	3.01	3.3	8.3
82	38,800	3.14	39,600	3.36	2.1	7.0
47	33,600	3.17	34,900	3.27	3.9	3.2
17	19,600	2.29	20,100	2.37	2.6	3.5

Case 2: Package System, Capillary Tube
Single Speed Compressor, Medium Efficiency
11.1 sf, 2 Row Outdoor/3.0 sf, 3 Row Indoor Coil
Medium Efficiency Fan Motors

Temp	Measured Capacity	Measured COP	Calculated Capacity	Calculated COP	Percent Error Capacity	Percent Error COP
95	35,900	2.19	38,800	2.44	8.1	11.4
82	40,100	2.64	41,200	2.80	2.7	6.1
47	39,500	2.64	37,000	2.67	-6.3	1.1
17	24,500	2.02	23,900	2.19	-2.4	8.4

Case 3: Split System, TXV
Single Speed Compressor, High Efficiency
9.24 sf, 2 Row Outdoor/4.22 sf Three Row Indoor Coil
High Efficiency Fan Motors

Temp	Measured Capacity	Measured COP	Calculated Capacity	Calculated COP	Percent Error Capacity	Percent Error COP
95	36,500	2.42	36,600	2.45	0.3	1.2
82	38,900	2.77	39,500	2.81	1.5	1.4
47	35,400	2.73	38,200	2.89	7.9	5.9
17	19,300	1.98	23,400	2.12	21.2	7.1

Table 4.2 - Comparison between the ORNL model output and test data for two residential sized heat pumps.

Case 4: Split System, TXV
Single Speed Compressor, high efficiency
15.0 sf, 2 Row Outdoor/3.8 sf, 3 Row Indoor Coil
High Efficiency Fan Motors

Temp	Measured Capacity	Measured COP	Calculated Capacity	Calculated COP	Percent Error Capacity	Percent Error COP
95	36,800	2.81	34,500	2.83	-6.3	0.7
82	39,500	3.26	37,000	3.27	-6.3	0.3
47	39,000	3.15	36,500	3.14	-6.4	-0.3
17	22,000	2.35	21,500	2.45	-2.3	4.3

Case 5: Split System, Capillary Tube
Single Speed Compressor, Medium Efficiency
15.0 sf, 2 Row Outdoor/3.8 sf, 3 Row Indoor Coil
Medium Efficiency Fan Motors

Temp	Measured Capacity	Measured COP	Calculated Capacity	Calculated COP	Percent Error Capacity	Percent Error COP
95	36,800	2.20	36,900	2.38	0.2	8.2
82	38,650	2.54	39,200	2.63	1.4	3.5
47	38,000	2.59	38,900	2.83	2.4	9.3
17	22,800	1.90	22,400	2.10	-1.8	10.5

arrangement does not follow normal single capillary performance trends, as capacity was low in the heating mode versus high in the cooling mode.

Case 3 is a moderately efficient unit with a single speed compressor. Capacity and COP are both within 8% except at 17 F. Although the 17 F capacity is taken from the manufacturer, it exhibits markedly different trends than those in either case 1 or case 2. For instance, the capacity of case 2 at 17 F is 27% higher than case 3, while it is only 12% higher at 47 F. Case 3 measured COP is from 5 to 10% higher than for case 2 except at 17 F, which is 2% lower. Despite the divergence from this data point, the model produces believable trends, with good accuracy at the other three rating conditions.

Case 4 is a high efficiency unit with an oversized outdoor coil. The model is consistent in both capacity and COP.

Case 5 uses capillary tubes for both heating and cooling. While capacity is very acceptable over the entire range, COP tends to be high, particularly at the extreme temperatures. This tendency is the same as that shown for case 2 which also used capillary tubes. The capillary tube model is taken from the performance curves in chapter 20 of the ASHRAE Equipment Handbook [7]. Several factors can contriburte to errors in modeling capillary tube performance. First, the curves are averaged for both refrigerants 12 and 22. Also, average refrigerant densities within the range of -40 and 140 F are used to calculate pressure drop in the capillary tube. Finally, the curves in the ASHRAE handbook are generated by empirical curve fits. These factors can combine to produce larger errors at the extreme operating conditions than at the middle of the range.

The bottom line of these performance comparisons is that the ORNL model can be expected to give reasaonable results for conventional technology. It appears that the COP results may be high by a few percent. The effect of not having a charge inventory subroutine is that the efficiency at the 95 F rating point is usually higher than measured performance by slightly more than it would have been with correct charge accounting. However, this increase is only 2-4%, and may not stand out in the comparisons when many other variables are also changing.

Seasonal Performance Model (SPM)

The calculation of seasonal performance required making two sets of calculations. First, the ORNL heat pump model had to be used to generate steady state performance at five points (Table 4.1)*. Then, a SPM had to be run which used the steady state input. The equations used in the SPM are described in Appendix B. The five steady state points corresponded to the four major steady state conditions used in the DOE test procedure plus the frosting

Table 4.3 - Major steady state performance points run with the ORNL heat pump model.

Steady State Run	Outdoor Temp.(F) Tdb	Outdoor Temp.(F) Twb	Indoor Temp.(F) Tdb	Indoor Temp.(F) Twb
1	95	75	80	57
2	82	65	80	57
3	47	43	70	60
4	17	15	70	60
5	35	33	70	60

accumulation test conditions[13]. The first two points are tests A and B in the cooling mode in the DOE test procedure for heat pumps. The 95 F rating point is the traditional central air conditioning rating point used by ARI before the DOE test procedure was developed[14]. In the DOE test procedure, the 95 F run is required to obtain information on the nominal capacity of the heat pump in the cooling mode. This point is also needed in the calculation of the annual performance factor (APF). The second point provides the energy efficiency ratio (EER) that is needed in the calculation of the SEER of the heat pump as specified in the test procedure:

$$SEER = (1 - 0.5*CD) * EER(@82\ F) \qquad (4.1)$$

CD is the degradation coefficient which is a quantity unique to a particular unit and its hardware configuration. The degradation coefficient is a measure of the efficiency losses caused by on/off cycling in a heat pump or air conditioner. It is not possible at this time to analytically predict this value with any public domain computer models. Studies attempting to relate the cooling degradation factor for air conditioners and heat pumps to SEER or

* For the two speed heat pump, there is an extra steady state point at 62 F.

EER have shown considerable scatter(See Figure 4.1)[15,16]. Survey data from reference 15 indicated that the average CD for central air conditioners was about 0.15 and had little dependence on the flow control device(TXV, orifice, or capillary tube).

All heat pump units modeled in this study were assigned CD values of 0.25, 0.20, or 0.15 for the cooling mode. A cooling CD value of 0.25 is the default value specified in the test procedure when no cycling tests are performed[13]. The lower efficiency base units (described later) were assigned CDs of 0.25, while the high efficiency units were assigned CDs of 0.15. The CD values for the high efficiency models are slightly higher than the 0.12 value that DOE used for high efficiency air conditioners in evaluating efficiency standards for central air conditioners[17]. Recent data from the California Energy Commission indicated that some heat pumps had CDs as low as 0.05 in the cooling mode[18]. Thus is appears that a CD of 0.15 should be a reasonable assumption for higher efficiency units. Some of the mid-efficiency units were also assigned CDs of 0.20.

Three rating points are used in the DOE test procedure for calculating of the HSPF: 47, 35, and 17 F[13]. The third and fourth steady state values shown in Table 4.3 (47 and 17 F) are the traditional ARI rating points for air source heat pumps[19]. In the DOE test procedure, the 47 F point is used initially to estimate a design heating load of a hypothetical building that would be used with the heat pump. For example, if the 47 F capacity is 40,000 Btu/hr, then the DOE test procedure would estimate the HSPF for the heat pump installed in a house that had a heating load of 40,000 Btu/hr at design conditions. The 47 F and 17 F points are both used in estimating the capacity and COP as a function of outdoor temperature.

The calculated steady state point at 35 F serves as an estimate of the performance of the heat pump without any frosting. Corrections are made to the capacity and COP at 35 F to account for frosting. These corrections depend on whether the heat pump uses a timed or demand defrost control. A timed defrost is initiated after fixed compressor run times. This interval is usually set by the manufacturer and typically varies from 60 to 90 minutes. A demand defrost control senses changes in operating conditions such as drop in evaporator wall temperature that results from frost buildup on the evaporator coil. Because the demand defrost is initiated only when the coil needs defrosting, it provides a slight boost in efficiency over the timed defrost. The effect of the timed defrost on the performance of a system at 35 F was estimated by using the Science Applications Inc.'s heat pump heating seasonal performance model[20]. At 35 F, the frosting degradation was 1% on the power and 10% on the capacity. Demand defrost controls were assumed to provide the 7% boost in capacity over the timed defrost control unit as specified in the DOE test procedure[13].

The heat pump seasonal performance model follows the temperature bin calculation procedure in the DOE test

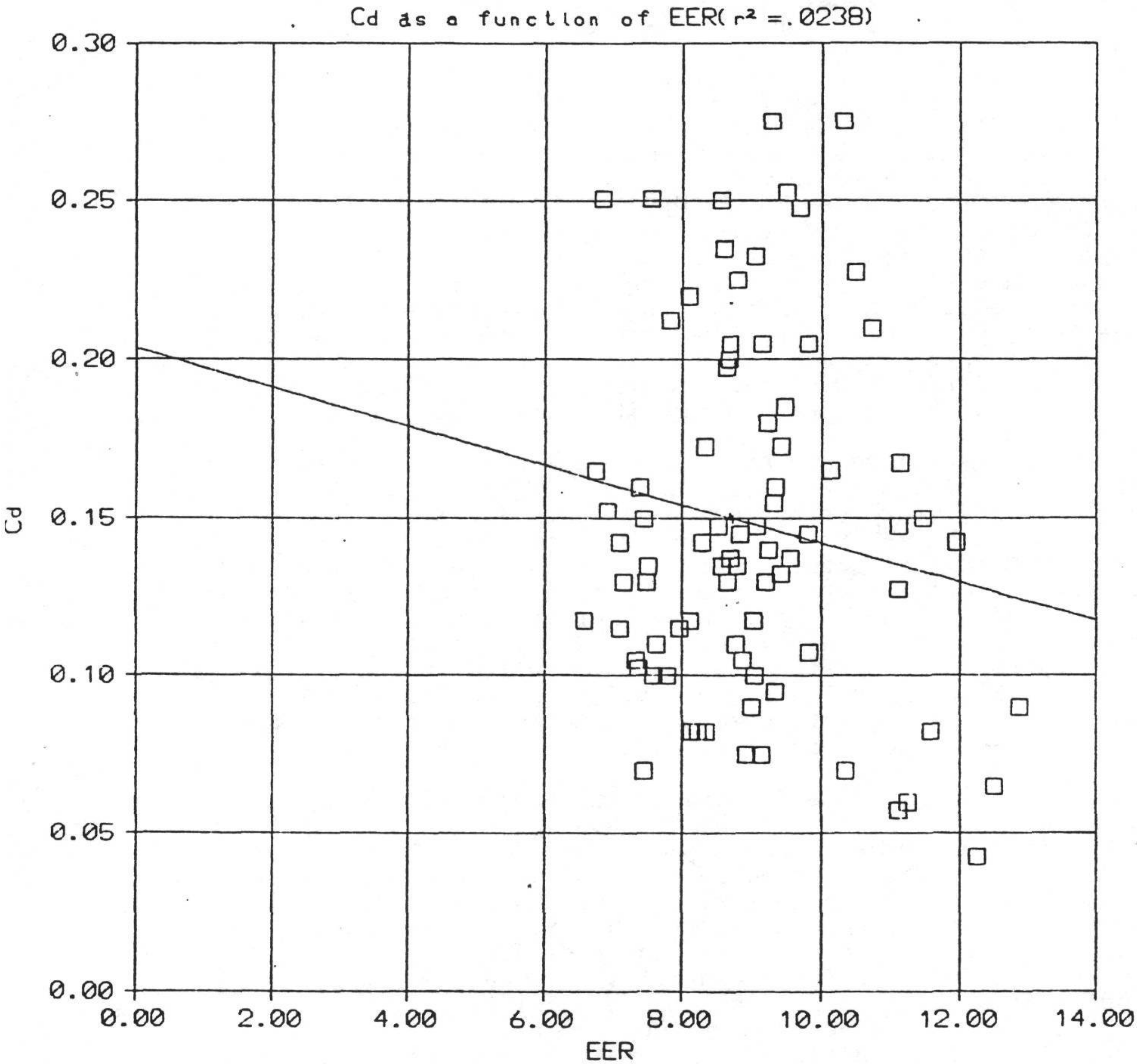

Figure 4.1 - Plot of degradation coefficient as a function of the 82F EER for central air conditioners (Source: Ref. 4)

procedure[13]. Heating degradation coefficients were limited to values of 0.25 and 0.20. From limited industry data we have seen, heating CDs of less than 0.20 should not be assumed, even for the higher efficiency units. While some units drop below 0.20, many use the default 0.25 value, indicating that the actual CD is larger than 0.25. Region IV was selected for the represtative estimates of the cooling and heating seasonal performance. This is the region that is used for reporting HSPF data in the ARI directory[21]. Other regions are also output with the seasonal performance model.

The SPM is microcomputer based, and takes less than 5 seconds to run on a standard IBM-PC. The SPM can handle both single and two speed heat pumps. For single speed units, the user inputs the capacity and efficiency at the five conditions in Table 4.3, the degradation coefficient, and whether the unit uses a timed or demand defrost system. Two speed systems are handled similarly, but the data are required for the two different speeds of the system.

Seasonal Performance Model Validation

The seasonal performance model was first tested with detailed empirical data provided by several manufacturers on the units listed in Tables 4.1 and 4.2. The object of this first test was to see if the seasonal performance model could closely reproduce the SEERs and HSPFs that the manufacturers had calculated for their heat pumps. Manufacturers provided data on the COP and Capacity at the five rating points in Table 4.3, plus the heating and cooling CDs. Heat pump 1 was not included in the comparison because the manufacturer of that unit did not provide enough performance data at both the low and high speeds to make a seasonal peformance calcualation. Table 4.4 summarizes the results.

Table 4.4 - Validation runs of the seasonal performance model.

Heat Pump	Cooling CD	Heating CD	SEER* Model	SEER* Actual	HSPF* Model	HSPF* Actual
2	0.120	0.230	8.46	8.46	6.08	6.10
3	0.090	0.160	9.03	9.03	6.45	6.40
4	0.127	0.115	10.42	10.42	8.16	8.16
5	0.152	0.250	8.08	8.08	6.90	6.92

*The SEER and HSPF data shown above may differ from values in the ARI directory because the above values are the "measured", not the "certified" values found in the directory. It is not unusual for the "certified" values to be slightly less than the "measured" values.

As seen in Table 4.4, the SPM produces accurate HSPF and SEER values when using empirical data provided by the manufacturers. The SEERs are exact because the single-speed SEER calculations only involve one step(see Equation 4.1). The SPM was within 1% of the "measured" HSPF for the four heat pumps. It was high in the second decimal point for heat pumps 3 and low for 2 and 5. The small differences in HSPF could be due to roundoff error in either the SPM or the rounded off values for CDs that we obtained from manufacturers. For example, for heat pumps 2 and 3, we were only provided CDs rounded off to two digits, while for 4 and 5, CDs of four digits were provided.

The second validation test included the combined steady state and seasonal performance models. This could be considered a test of the system of models to accurately predict the heat pumps' HSPFs and SEERs. The calculated capacity and COP values given in Tables 4.1 and 4.2 are used along with the actual CDs to estimate the HSPF and SEER. The results are shown in Table 4.5.

Table 4.5 - Validation results of combined steady state and SPM modeling system.

Heat Pump	SEER Model	SEER Actual	SEER %Error	HSPF Model	HSPF Actual	HSPF %Error
2	8.98	8.46	6.1	6.58	6.10	7.9
3	9.16	9.03	1.4	6.80	6.40	6.3
4	10.45	10.42	0.3	8.09	8.16	-0.9
5	8.29	8.08	3.5	6.78	6.92	-2.1

The combine steady state/SPM system compared favorably with the SEER and HSPF values calculated from measured data. The system was high on SEERs and HSPFs for units 2 and 3, and low for units 4 and 5. The largest error was 6.1% for the SEER(unit 1) and 7.9% for the HSPF(unit 1). These small errors are acceptable considering the complexities in accurately modeling a heat pump system.

REFERENCES

1. Fischer, S.K. and Rice, C.K., "A Steady-State Computer Design Model for Air-to-Air Heat Pumps", Oak Ridge National Laboratory, ORNL/CON-80, December 1981.

2. Domanski, Piotr and Didion, David, "Computer Modeling of the Vapor Compression Cycle With Constant Flow Area Expansion Device", NBS Building Science Series 155, May 1983.

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

4. Travis, D.P., Baron, A.B., and Rohsenow, W.M., "Forced Convection Condensation Inside Tubes: A Heat Transfer Equation for Condenser Design", ASHRAE Transactions, Vol. 79, Pt. 1, 1973.

5. Geary, D.F., "Return Bend Pressure Drop in Refrigeration Systems", ASHRAE Transactions, Vol. 81, Pt. 1, 1975.

6. Mei, V.C., "Short Tube Refrigerant Flow Restrictors", Internal ORNL report, taken from [1].

7. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Equipment Handbook, 1983.

8. Erth, R.A., "Two-Phase Flow in Refrigeration Capillary Tubes: Analysis and Prediction", Doctoral Dissertation, Purdue University, January 1970.

9. Houcek, J. and Thedford, M., "A Research Into a New Method of Refrigerant Charging and the Effects of Improper Charging", Proceedings of the First Annual Symposium on Efficient Utilization of Energy in Residential and Commercial Buildings, Texas A&M University, August 14-15, 1984.

10. McQuiston, F.C., "Correlation of Heat, Mass, and Momentum Transport Coefficients for Plate-Fin-Tube Heat Transfer Surfaces with Staggered Tubes", ASHRAE Transactions, Vol. 84, Pt. 1, 1978.

11. Hosoda, T., Uzuhashi, H., and Kobayashi, N., "Louver Fin Heat Transfer Japanese Research, Vol. 6, No. 2, pp. 67-77, 1977.

12. Kirschbaum, H.S., and Veyo, S.E., "An Investigation of Methods to Improve Heat Pump Performance in a Northern Climate", Vol. I-III, Electric Power Research Institute, EPRI EM-319, January 1977.

13. "Test Procedure for Central Air Conditioners, Including Heat Pumps", Federal Register, Dec. 27, 1979, pp. 76700-76723.

14. "Unitary Air-Conditioning Equipment", Standard 210-79, Air Conditioning and Refrigeration Institute, Arlington, VA.

15. H.V. Nguyen, et al, "Trends of Residential Air-Conditioning Cyclic Tests", ASHRAE Transactions, Vol. 88, Pt. 2, pp. 954-972.

16. B. Bittle and V. Goldsmitdh, "Effect of Cycling and Frost Formation on Heat Pump Performance", ASHRAE Transactions, DC-83-13, No. 1, V. 89, Pt. 2, 1983.

17. "Supplement to: March 1982 Consumer Products Efficiency Standards Engineering Analysis and Economic Analysis Documents", DOE/CE-0045, U.S. Dept. of Energy, July 1983.

18. "Directory of Central Air Conditioners and Heat Pumps", P400-00-021 California Energy Commission, Sacremento, CA, June, 1985.

19. "Air-Source Unitary Heat Pump Equipment", Standard 240-77, Air Conditioning and Refrigeration Institute, Arlington, VA.

20. "User's Manual for Heat Pump Seasonal Performance Model with Selected Parametric Examples", Science Applications, Inc., La Jolla, CA., June 30, 1982.

21. Directory of Certified Unitary Air Conditioners, Air Source Heat Pumps, and Sound-Rated Outdoor Unitary Equipment, ARI, Jan. 1-June 30, 1985, Arlington, VA.

CHAPTER 5

DESIGN OPTIONS

Design options used to improve the performance of the heat pumps are discussed below. Appendix C discusses some of the incremental improvements these design options provide when implemented on a system. The analysis of the incremental effects was used as a basis for developing the final designs discussed in the next chapter. The design options considered included: (1) Increased condenser and evaporator heat exchanger performance, (2) Decreased compressor size, (3) Increased combined fan and motor efficiency, (4) Demand defrost control systems, (5) High efficiency compressors, (6) Two speed compressors, and (7) Variable speed compressors, (8) Scroll compresssors, (9) Two speed and variable speed fan motors, (10) Electronic expansion valves, and (11) New/mixed refrigerants.

Options seven through eleven are considered "advanced technolgy" options. These options are not widely used on U.S. manufactured residential sized heat pumps. For some of the options, the technology should be available in the next few years. Both the performance improvements and additional costs expected with these options are still uncertain. Estimates were made on their potential for improving heat pump performance and to arrive at the "maximum technologically feasible" units.

1) <u>Increased Condenser and Evaporator Heat Exchanger Performance</u>.

One of the easiest methods of increasing a heat pump's efficiency is to increase the heat transfer of the heat exchangers. A simplified form of the equation governing the rate of heat transfer in a coil is :

$$q = U*A*(Tref - Tair) \qquad (5.1)$$

where,

q = capacity of the heat transfer (btu/hr)
U = overall heat transfer coefficient (btu/hr-sf-F)
A = coil surface area (sf)
Tref = average refrigerant temperature (F)
Tair = temperature of ambient air (F)

The actual expressions used to calculate heat transfer in a coil are much more complicated, but the above equation illustrates the major influences on heat transfer. Improving the heat transfer is accomplished either by increasing the heat exchanger surface area or by increasing the overall heat transfer coefficent. To maintain a constant capacity, increasing either U or A must be accompanied with a decrease in the temperature difference, Tcoil - Tair. This temperature reduction means that the condensing temperature in the condenser must be lowered and the evaporating temperature in the evaporator must be raised. Lowering the condenser temperature or increasing the evaporating temperature raises the compressor efficiency, which, in turn, raises the efficiency of the heat pump.

Increasing the surface area in a heat exchanger can be accomplished by adding more frontal area, adding tube rows, or increasing the fin density. Each is discussed below.

1A) Increased heat exchanger frontal area

Adding more frontal area increases the area for air to contact the fins and tubes of the heat exchanger. The added frontal area increases the distance the refrigerant must flow. This increases the pressure drop on the refrigerant side of the heat exchanger unless the refrigerant tubes are recircuited.

For three ton systems, increasing frontal area of the outdoor coil beyond ten to fifteeen square feet improves the cooling effiency, but has little effect on the heating. The primary reason for the smaller impact for heating is that the outdoor coil is usually oversized at 10 square feet when used as an evaporator in the heating mode.

Limits were imposed on the maximum size of both the indoor and outdoor coils(see Chapter 6). The limits either equalled or slightly exceeded the maximum coil sizes for systems on the market. The principal reasons for the limitations were the effect of coil size on latent cooling capacity and physical constraints of the ductwork.

1B) Increased tube rows

Another option for increasing the surface area of the heat exchanger is to increase the number of tube rows. The amount of copper tubing and fin material increases, but the overall dimensions of the heat pump chasis remain small. Because the cabinet for a single package unit must contain both the evaporator and condenser, many manufacturers have chosen this option to improve performance. The incremental improvement of each new tube row is smaller than the improvement provided by the previous tube row. Four or five tube rows are not uncommon for the indoor coil while the outdoor coil is seldom more than three or four rows.

1C) Increased fin density

The last approach to increasing the surface area of a heat exchanger is by increasing fin density. Low efficiency units on the market in 1985 typically had 15 fins per inch (fpi) in the outdoor heat exchanger while the high efficiency units had as many as 21 fpi. For the indoor heat exchanger, most units have 12 to 14 fpi. It is unlikely that outdoor heat exchangers will exceed 21 or 22 fpi. Any closer spacing of fins has two major penalties. It requires more fan power to draw the air through the heat exchanger. Secondly, it increases the frosting losses. With closer spacing, the air passages are narrower, allowing the frost to more quickly block the passages. The heat pump then has to defrost more often. Indoor fin densities are not likely to increase since enough space is needed to allow for condensation to form and drop off the heat exchanger.

1D) Increased heat transfer coefficient

The overall heat transfer coefficient can be improved by using higher performance heat transfer surfaces for the fins. One example of this application was the switch by the HVAC industry from straight to wavy fin designs. Wavy fins help break up the boundary layer of the air flowing through the heat exchanger, which improves the heat transfer coefficient. Wavy fins also increase the surface area. Because wavy fin designs are commonly used in heat pump heat exchangers, all the baseline designs used in this report start with wavy fins.

Many manufacturers use other enhanced heat transfer surfaces besides wavy fins. These include perforated plate fins and high density spine fins to increase the energy transfer/unit frontal area. Another heat transfer enhancement is the use of internal fins in the refrigerant tubes. These are commonly used on many commercial sized chillers.

Switching to higher performance heat transfer surfaces allow the manufacturer to decrease the amount of aluminum(and copper) used in the heat exchangers because it is possible to obtain the same heat transfer with a smaller surface area.

2) Decreased Compressor Size

In conjunction with Design Option #1(increased heat exchanger performance), the compressor size must be reduced to maintain the rated capacity. This is accomplished by installing a lower capacity compressor into the unit. The compressor piston displacement is decreased by shortening the length of the stroke until the rated capacity is reached at 95 degrees outdoor temperature. The computer model simply decreases the compressor mass flow rates specified by the compressor curve fit coefficients while maintaining the same efficiency characteristics. Once the proper displacement is determined, the computer runs are made for outdoor temperatures of 82, 47, 35 and 17 degrees. Using a smaller compressor provides a decrease in power consumption and a boost in efficiency.

3) Increased Combined Fan and Motor Efficiency

Using a permanent split capacitor motor with an efficiency of 55% and a centrifugal forward curved fan with an efficiency of 35% results in a combined fan and fan motor efficiency of 21%. The indoor units now being manufactured typically have a combined efficiency of 20% to 30%. The combined fan and fan motor efficiency is increased to 25% and 30% for the medium efficiency line and to 35% for the high efficiency line. A combined fan and fan motor efficiency of 34% has been used in other studies as a possibly attainable for the future [1]. Motors with efficiencies of 70% and centrifugal fans with efficiencies of 45 to 55% can now be purchased. This combination of fan and motor will give a combined efficiency of 30 to 35%.

Permanent split capacitor motors are also used on the outdoor unit. The fan is usually of the propeller type. Propeller fans are not as efficient as the centrifugal forward curved fans, and typically have efficiencies from 20% to 30%. The resulting combined fan and fan motor efficiency ranges from 10% to 20%. All baseline units used a combined efficiency of 10% on the outdoor section. The combined fan and fan motor efficiency was increased to 15% for the medium efficiency line and to 20% for the high efficiency line. A motor efficiency of 70% and fan efficiency of 30% result in the combined efficiency of 20% that was used on the high efficiency line.

4) Demand Defrost Control Systems

Many of the early heat pumps employed a simple timer to control the defrost cycle. Usually every 60 to 90 minutes of run-time, the heat pump would initiate a defrost cycle. Performance of these units suffered due to the inappropriate timing of the defrost cycles. Demand defrost control systems can now be found on many of the high performance heat pump units. These units defrost only when enough frost buildup is detected. There are two different procedures being used to initiate the defrost cycle. One is to measure the air-side pressure drop across the outdoor coil and initiate the defrost cycle once the pressure drop reaches a specified level. Another procedure measures the temperature between the outdoor coil tube surface and the outdoor air and initiates the defrost cycle once the temperature difference exceeds a preset level. Microprocessors are currently being placed in some heat pumps to implement these demand defrost strategies.

In the DOE test procedure, units with demand defrost receive a boost in capacity of 7% for the 35 F rating point[2]. Because the COP is defined as the capacity divided by the power, a 7% improvement in capacity at 35 F also means a 7% improvement in COP at 35 F. This procedure is followed in the analysis.

5) High Efficiency Compressors

Most residential sized heat pumps manufactured in the United States use reciprocating compressors to compress the refrigerant vapor in the heat pump. At least one U.S. and several of the Japanese manufacturers also use rotary compressors in their heat pump designs. For both compressors (reciprocating and rotary), the compressor is hermetically sealed in a pressure vessel with the compressor motor. The combination is called a hermetic compressor. Refrigerant is allowed to come into direct contact with the compressor motor before entering the compressor. This helps cool the compressor motor and ensure that superheated refrigerant enters the compressor.

Reciprocating technology is a mature technology. Recent improvements in compressor efficiency have centered on better valving and higer efficiency compressor motors. The ASHRAE Equipment handbook states that: "The most important components in the reciprocating compressor are the suction and discharge valves."[3] Proper valve design is necessary both for proper performance and long life.

For lower capacity heat pumps (under 2 tons), the most immediate promise for better performing compressors are the higher efficiency rotary compressors. These are used in many of the Japanese heat pumps. They offer EERs of approximately 5% better than reciprocating compressors of the same capacity. Because rotary compressors have historically had reliability problems due to contaminants in the refrigerant, they will more likely be limited to package systems.

The efficiency of the better compressor motors being used by the manufacturers as high as 87%. It should be possible to improve the combined motor/compressor efficiency another 5% with current technology. A 5% improvement in the motor/compressor efficiency should translate to approximately a 5% improvement in heating and cooling efficiency.

6) Two Speed Compressors

The two speed compressor has several advantages over the single speed compressor. The unit operates at low speed when the building load is low, resulting in a substantially reduced power requirement. When the building load is high, the compressor is switched over to high speed mode and the capacity is increased. The mode of these compressors is usually controlled by an outdoor temperature sensitive microprocessor.

The performance of a two speed compressor is shown in Figure 5.1 and 5.2 for a superheat of 15 F and subcooling of 10 F. The capacity of the two speed unit increases as the speed goes from low speed to high speed. However, the COP for the high speed is lower than that for the low speed in the cooling mode. The system is more efficient at low speed because the compressor is operating with heat exchangers that are sized for high speed operation. This means that the heat exchangers are oversized for low speed operation. With the oversized heat exchangers, the evaporating temperature is higher and the condensing temperatuer is lower, which provides for more efficient operation in the compressor. The fast dropoff in efficiency in the cooling mode for at low speed is a characteristic of this particular compressor.

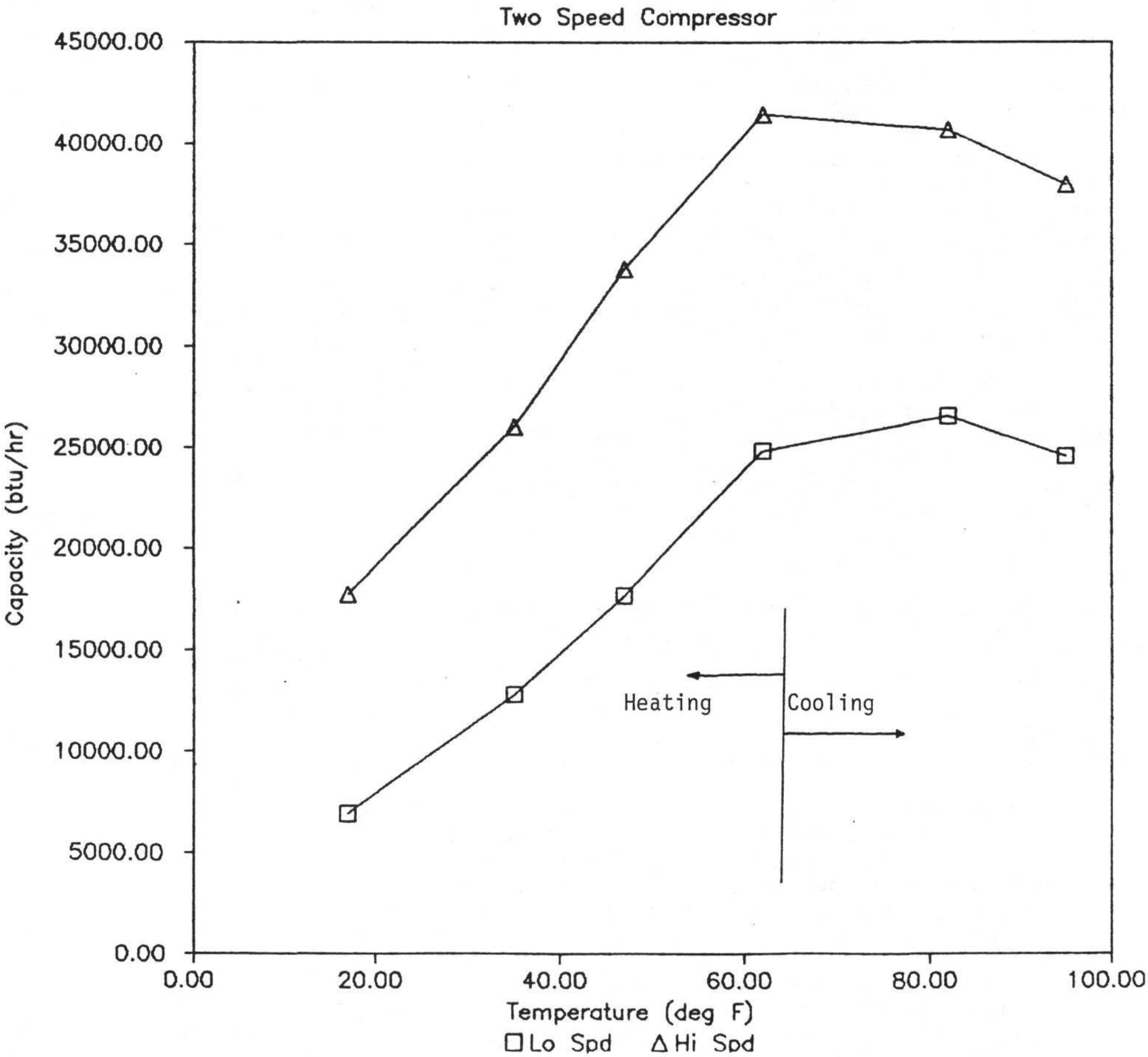

Figure 5.1 - Typical capacity performance to a two speed compressor

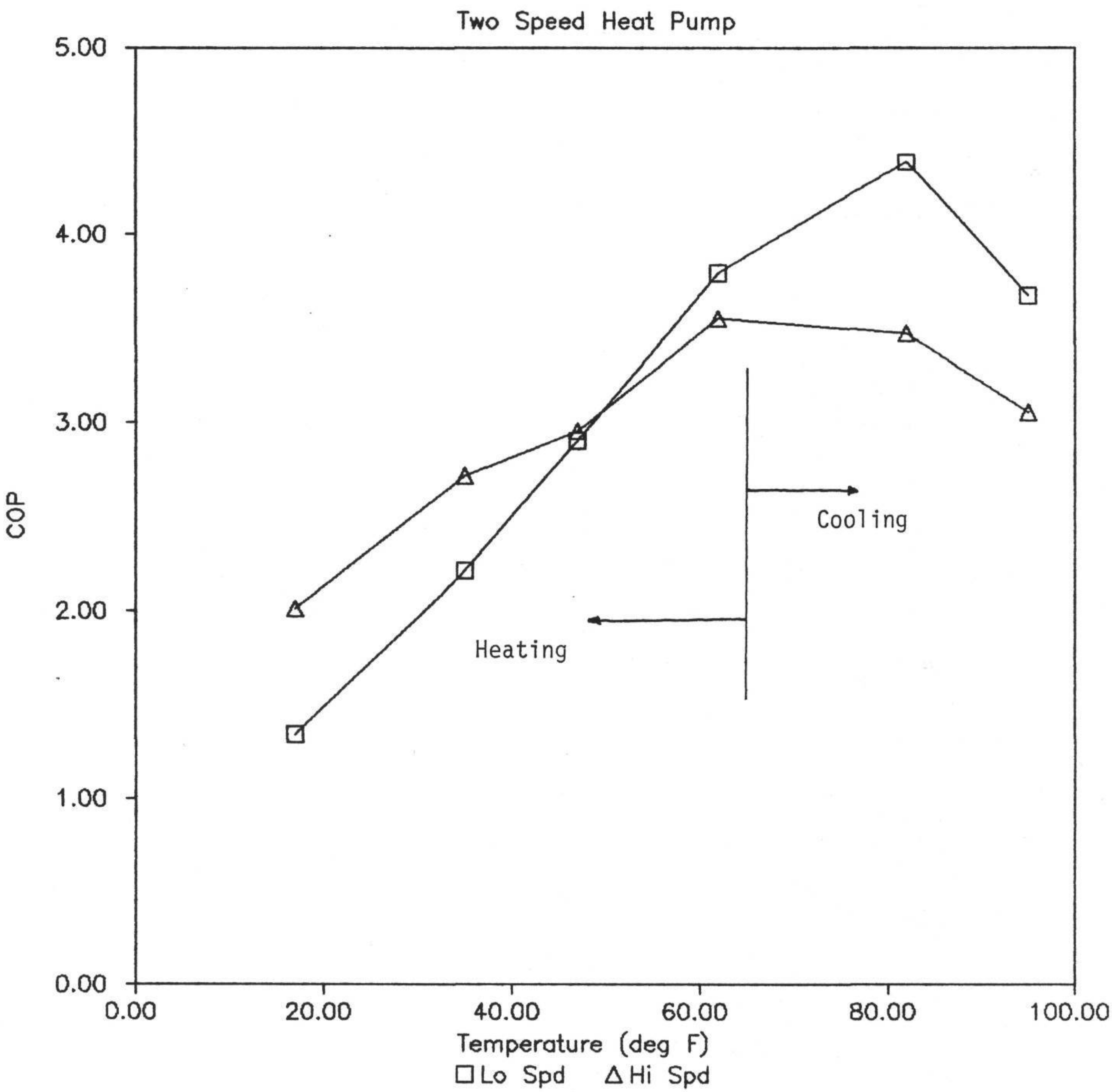

Figure 5.2 - Two speed COP for a two speed compressor

7) Variable speed compressors.

One of the advanced technology design options that will be incorporated into heat pump designs are variable speed electric motors to drive the compressor. Proponents claim an efficiency improvement of from 10% to 40% over a single-speed compressor system[4,5,6,7,8]. An electronic inverter is used to convert a standard 60 Hz power source to one ranging in frequency from 30 to 110 Hz. Since compressor motor speed is dependent on the power source frequency, adjusting frequency directly affects the motor speed. There is a small penalty in the overall efficieny of the motor with the losses in the electronic inverter. This technology has already been widely applied in the Japanese residential heat pump market. While other forms of capacity modulation are possible (variable stroke length or cylinder unloading), variable speed motors appear should be implemented quickest.

Figures 5.3 and 5.4 demonstrate the effect on capacity and COP of varying the compressor motor speed from 1580 to 4250 revolutions per minute for a Japanese manufactured variable speed compressor[4]. The compressor speed is dependent on the building load. At low building load conditions, the compressor is run at a lower speed and at a resulting lower capacity. When this is done, the need for the unit to cycle on and off is reduced and an increase in efficiency is realized. At higher building loads the compressor runs at higher speeds to meet the load.

One of the difficulties with evaluating the improvement due to the variable speed compressor is that DOE has not finalized a test procedure for these units. Thus, the savings attributed to a variable speed heat pump by various investigators may differ substantially from what will actually be determined through the test procedure.

Another consideration relating to the variable speed performance is the latent capacity of the unit at lower speeds. If the heat exchangers are designed for high speed operation, they will be considerably oversized for lower speeds. This usually implies a smaller latent capacity. Thus, it may be necessary to redesign the evaporator (or condenser) so that only a portion of it is used during the lower speeds. A variable (or two) speed fan could also be used to better control both the latent capacity and energy use.

For this analysis, it was assumed that the variable speed heat pump would perform 15% better for both the heating and cooling than the best conventional single speed reciprocating compressor.

8) Scroll compressor.

The concept of the scroll compressor has a long history. It first appeared in a U.S. patent in the early 1900's. The main elements of the scroll compressor are two identical involute spiral scrolls[4]. One of the scrolls is fixed and the other orbits around the center of the fixed scroll. A hermetic package of the

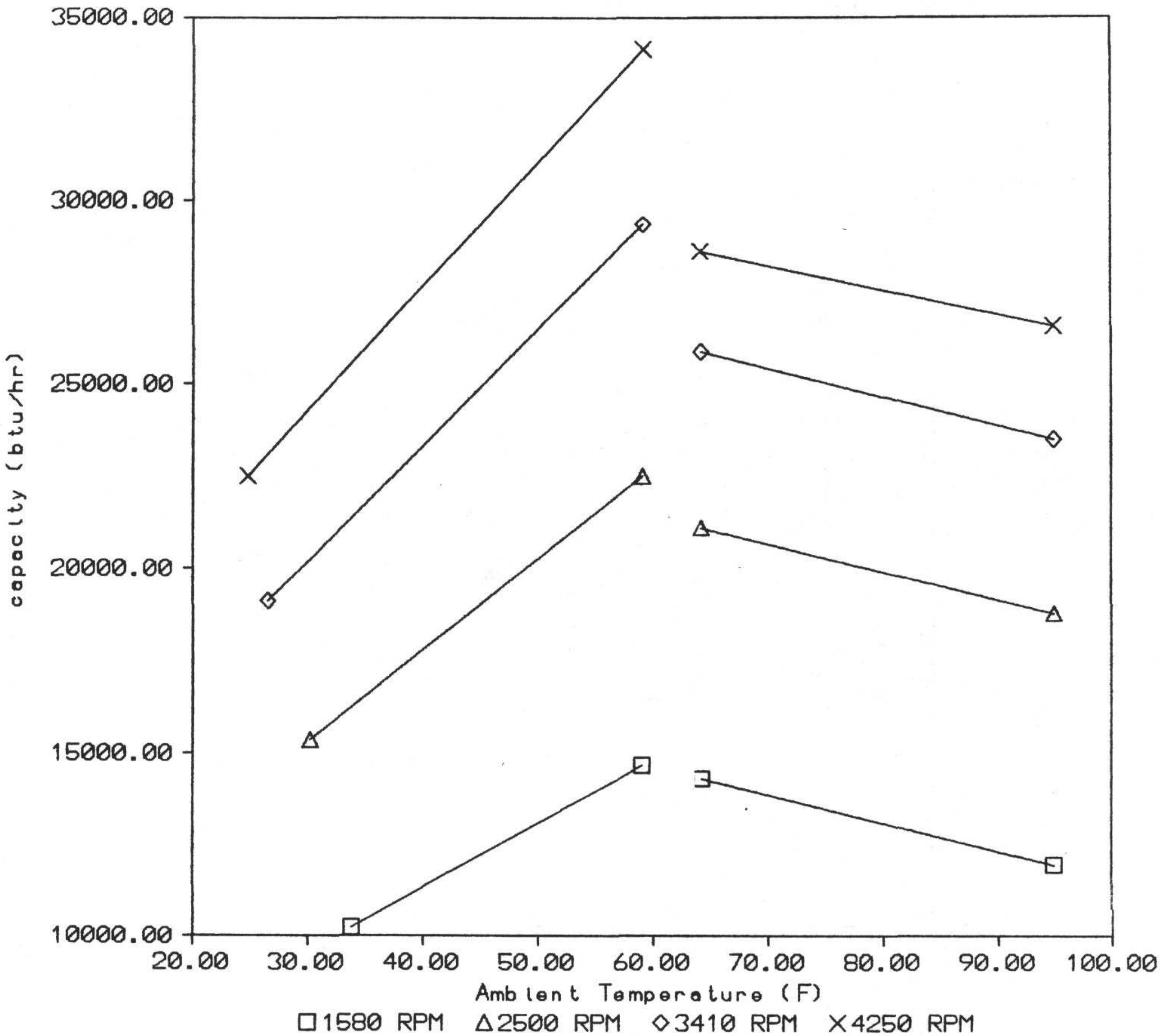

Figure 5.3 - Capacity versus outdoor temperature for a heat pump with a variable speed motor (Source: Ref. 13)

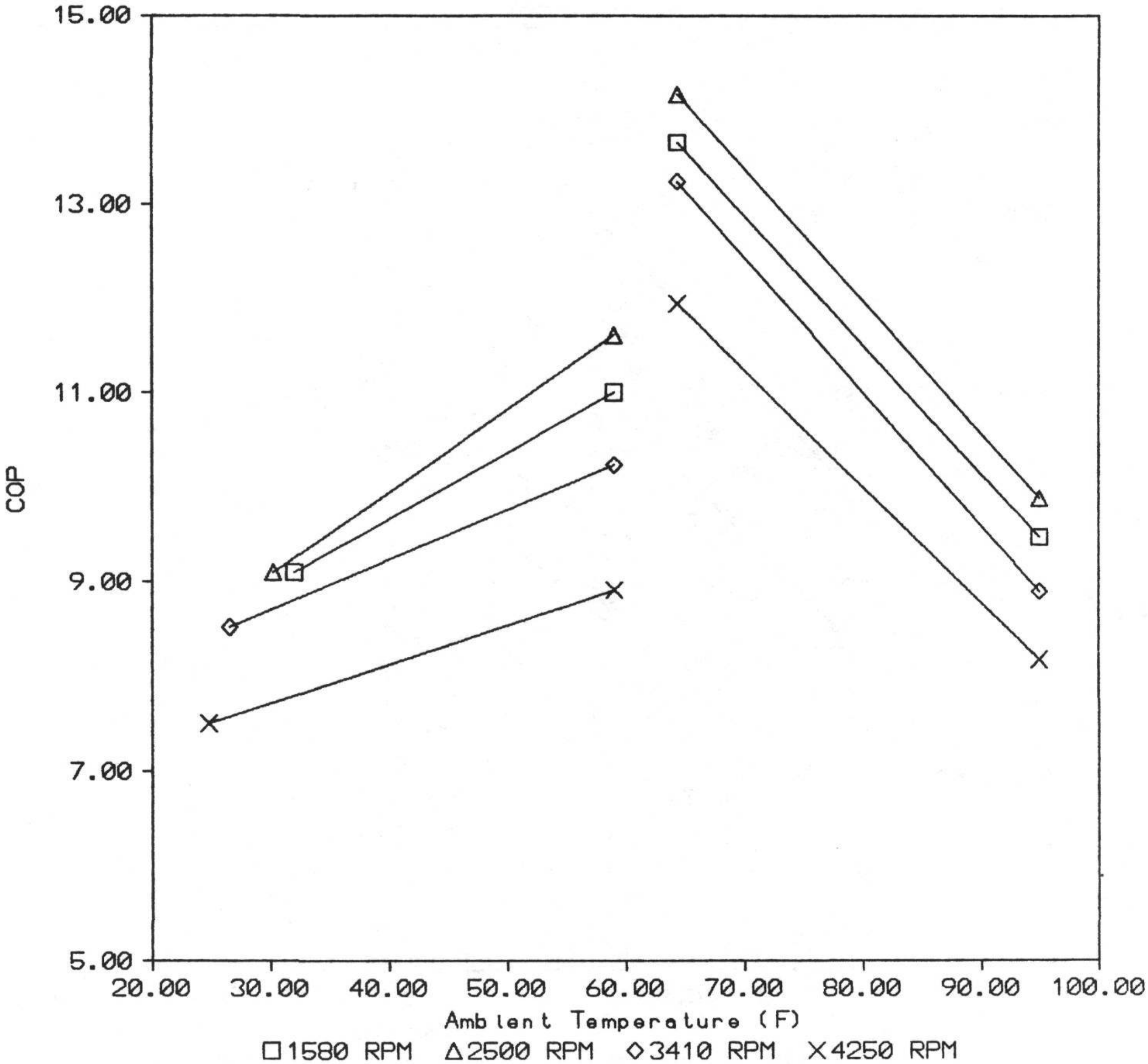

Figure 5.4 - COP versus outdoor temperature for a heat pump with a variable speed motor (Source: Ref. 13)

scroll compressor would probably be used for residential applications. Expected efficiency improvements of the scroll compressor over the best conventional reciprocating compressor should be about 10%.

The scroll compressor is still primarily in the development stages. Several prototypes have been built and tested[4]. Preliminary information from one U.S. compressor manufacturer indicates that the scroll compressor should be available to OEMs in 1987. It could appear in heat pump lines by 1988 or 1989. The developers of this compressor claim that it has several advantages over a conventional compressor: (1) lower leakage during the compression process resulting in high efficiency; (2) higher reliability because this compressor uses no suction or discharge valves; (3) a smaller torque change compared to a conventional compressor resulting in low noise and low vibrations.

9) Two-speed and variable speed fan motors.

With the introduction of variable speed compressor motors in heat pumps, the next logical step for improved capacity control would be the use of multi-speed or variable speed fan motors. The two-speed fan motors would be similiar to the two-speed motors used in compressors, having two sets of poles(one for high speed and the other for low speed). The variable speed fan motors would utilize inverter technology similar to variable speed motors for compressors.

Two speed fan motors are already in use on larger commercial air cooled chillers and heat pumps. In these units, the savings in fan energy costs are able to quickly offset the added costs of the two speed motors. Two speed fans have also been used in residential sized air conditioners to vary the outdoor airflow. These fans were thermostatically controlled to increase speed at a fixed outdoor temperature. Two speed or variable speed motors make the most sense when the compressor is capacity modulated. Variable speed motors could be used for the indoor coil to better control latent capacity and reduce fan energy in a unit with a variable speed compressor. Variable speed fans are only considered an option when used in conjunction with option #9, variable speed compressors. The expected savings due to either two speed or variable speed fans should not exceed five percent.

10) Electronic Expansion Valves.

The electronic expansion valve should make it possible to obtain better control of flow conditions (degree of superheat at the outlet of the evaporator, subcooling at the outlet of the condenser, etc.) than that which can be obtained with either a thermostatic expansion valve or capillary tubes[9,10]. Some electronic valves have an electric motor whose rotational motion is converted into vertical movement within the valve via gears[10]. Another design uses a solenoid to drive a plunger which controls the flow opening[9]. Either design would employ an electronic microprocessor would send a signal to the valve to either open or close more, depending on conditions being sensed.

This technology is currently employed on room air conditioners and heat pumps in Japan[7,8]. Claims of efficiency improvements from 5 to 10% in single speed systems have been made for both heating and cooling. If manufacturers are already optimizing their systems for 82 F, the electronic valve may not provide any improvement in cooling SEER. Another benefit of this valve is that should provide for a shorter defrost time. Because the defrost time is not measured in the test procedure, the savings of the valve would not be counted in the HSPF. The valve could provide better control of flow conditions at the more extreme temperatures(17 F in heating and 95 F cooling). The wide range of control offered by these valves may make them good fits for control in variable speed systems. The valves should also improve reliability over conventional expansion devices because it responds faster and can maintain superheat into the compressor over a wider range than conventional expansion devices. For this analysis, an electronic expansion valve was assumed to have no effect on SEER and a 5% improvement in HSPF.

11) <u>New/mixed refrigerants</u>.

New or combinations of existing refrigerants offer the potential of improving heat pump performance with minor changes in hardware. The use of nonazeotropic refrigerant mixtures appears to be the most promising alternative to existing refrigerants. They offer: (1) reduced compressor power for the same refrigeration capacity, and (2) capacity modulation, even though compressor displacement remains constant. For air-to-air heat pumps, a means must be provided to change the concentration of the mixture so that more of the denser refrigerant in the mixture is in active circulation at lower outdoor temperatures. This shift in concentration can increase capacity and COP. For instance, a nonazeotropic mixture of 65% R-13B1 and 35% R-152 improved heating COP 11% and 28% at -8.3 C and -17.8C, respectively, compared to using R-22[11]. New azeotropic mixtures may also offer improved performance for both the cooling and heating.

Application of non-azeotropic mixtures in heat pumps is still probably a decade away[12,13]. Quoting from one industry source at a recent heat pump conference: " ...the use of non-azeotropic mixtures is not apt to be a factor of consequence ... for some years"[12]. There remains much research on the best mixtures to use. Their primary benefit should be in the lower temperature heating capacity and COP. Because this option is not expected to be in the market by 1991, it was not evaluated.

REFERENCES

1. "Design Optimization and the Limits of Steady-State Heating Efficiency for Conventional Single-Speed Air-Source Heat Pumps", ORNL/CON-63, Oak Ridge National Laboratory, Oak Ridge, TN.

2. "Test Procedure for Central Air Conditioners, Including Heat Pumps", Federal Register, Dec. 27, 1979, pp. 76700-76723.

3. ASHRAE Handbook, 1983 Equipment Volume, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, GA, 1983.

4. T. Senshu, A. Arai, K. Oguni, and F. Harada, "Annual Energy-Saving Effect of Capacity-Modulated Air Conditioner Equipped with Inverter-Driven Scroll Compressor", ASHRAE Transactions, V. 91, Pt. 2, 1985.

5. F. A. Creswick, "New and Future Heat Pump Technologies", Proceedings of "Meeting Customer Needs with Heat Pumps", Electric Power Research Institute Seminar, April 7-9, 1986.

6. "Present and Projected Efficiency Improvements in Central Air Conditioners and Heat Pumps 1985-1992", DHR C5133-23, DHR, Inc., McLean, Virginia, Dec. 1985.

7. E.W. Grald, "The Impact of Electronics Innovation on Electric Heat Pump Performance", Honeywell Technology Strategy Center, Report prepared for the Gas Research Institute, Dec. 1985.

8. M. Hori, I. Akamine, and T. Sakai, "Seasonal Efficiencies of Residential Heat Pump Air Conditioners with Inverter-Driven Compressors", ASHRAE Transactions, V. 91, Pt. 2, 1985.

9. Y. Nakashima, et al, "Reversible-Flow-Type Linear Expansion Valves for Heat Pumps, ASHRAE Transactions, V. 91, Pt. 2, 1985.

10. H. Itoh, "Improvement of a Heat Pump Room Air-Conditioner by Use of Pulse-Motor-Driven Expansion Valve", ASHRAE Transactions, V. 92, Pt. 1, 1986.

11. W.D. Cooper, "The Use of Mixed Refrigerants in Air-to-Air Heat Pumps", ASHRAE Transactions, V. 88, Pt. 2, 1982.

12. J. Crawford, "Heat Pump Technology Overview", Proceedings of "Meeting Customer Needs with Heat Pumps", Electric Power Research Institute Seminar, April 7-9, 1986.

13. D. Didion, "Refrigerant Mixtures for Heat Pumps", Proceedings of "Meeting Customer Needs with Heat Pumps", Electric Power Research Institute Seminar, April 7-9, 1986.

CHAPTER 6

HEAT PUMP DESIGNS

The final heat pump designs and the methodology to arrive at the final designs for the four classes are discussed below. This chapter has three major sections: (1) design approach, (2) baseline units, and (3) final designs. Both air source split and package systems are discussed.

Design Approach

The general design approach consisted of developing a line of heat pumps for each class similar to what a larger manufacturer might do. Each line included the efficiency spectrum from the lowest efficiency in the class to the unit having an efficiency that was considered the maximum technology feasible with advanced technology. Each line consisted of enough heat pump designs to provide DOE with the data to evaluate, in small efficiency increments, the imposition of standards over the whole range of possible efficiencies.

The first step in the design process was to collect data from manufacturers on typical designs, the performance of compressors, heat exchangers, etc. These data provided a basis for designing units from currently available technology. Advanced technology options that should be on the market soon were also considered.

The next step was to choose what capacity units would be used for the efficiency analysis to represent each class. For both classes under 39000 Btu/hr capacity, heat pumps with 36000 Btu/hr capacity were chosen. For both classes over 39000 Btu/hr, heat pumps of 60000 Btu/hr capacity were chosen. These capacities are the same used for the central air conditioners for the previous standards analysis[1]. Another reason for using the 36000 and 60000 Btu/hr capacities was due to the effect of capacity on efficiency. As was shown in Figure 3.1 in Chapter 3, the maximum achievable SEER for heat pumps is dependent on capacity. Because the class split was at 39000, the representative unit for that class should be near the higher capacities in that class. A 36000 Btu/hr unit satisfies that criteria. Similarly, in the larger class, the 60000 unit is close to the largest capacity(65000 Btu/hr) in that class.

Baseline units were developed near the bottom of the efficiencies available in 1985 for each class. Starting with a low effiency unit allowed for design changes to be applied such that the whole range in heat pump efficiencies could be examined.

The next step was a test of the influence of important variables such as heat exchanger size, tube rows, degree of superheat, etc. on the performance of the baseline unit. These analyses allowed for the optimization of the overall performance of various heat pump designs. A more complete discussion of this process is provided in Appendix C.

The last step in the design process was the design of all the units and making the performance calculations. Ideally, to develop the most cost-effective lines, this portion of the process should have been done interactively with a group providing costing information on the designs. While this was not done for this analysis, the designs can be updated when costing data are developed.

Several design restrictions were used throughout the analysis(Table 6.1). First, the sensible heating factor(SHF) for the heat pumps in the cooling model had to be maintained below 0.80. Values above 0.80 do not produce sufficient latent cooling to properly dehumidify the air. A second restriction was limiting the size of the indoor coil to 5.5 sf for the 3 ton systems and 8 sf for the 5 ton systems. These limits are slightly higher than those used in the analysis for standards on central air conditioners[1]. However, they reflect limits that are consistent with units currently available on the market. Limiting the size of the indoor coil directly affects the maximum attainable efficiency for the heat pumps. The coil size limitation is necessary because of the physical constraints imposed by duct sizes in residential systems. A third restriction was the maximum size of the outdoor coil. In theory, an outdoor coil could be as large as a manufacturer wanted to make. However, our studies indicate that the incremental performance improvements with additional frontal area are marginal after reaching 25 sf and 30 sf in the 3 and 5 ton units, respectively. No distinction between split and package systems was made in regards to coil size.

Table 6.1 - Restrictions used for the heat pump designs.

Item	Value
Sensible Heating Factor	≤ 0.80
Indoor Coil Size	
3 ton	≤ 5 sf
5 ton	≤ 8 sf
Outdoor Coil Size	
3 ton	≤ 25 sf
5 ton	≤ 30 sf
Fin Density	
Indoor Coil	≤ 13 fpi
Outdoor Coil	≤ 19 fpi
Cooling CD	≥ 0.15
Heating CD	≥ 0.20

The design philosophy included beginning the analysis with baseline designs near the lowest efficiency units on the market in 1985. These baseline units were then either incrementally improved or redesigned to reach the maximum efficiency feasible for conventional design options. Advanced design options were then implemented. Conventional designs were applied first because it was felt that there were much larger uncertainties associated with both the costs and performance of many of the advanced design options.

Baseline Units

Baseline units were selected for the four heat pump classes. The baseline units are typical of the lower efficiency and lower priced units sold in 1985. These units are constructed using the less costly and less efficient compressors and fan motors along with smaller indoor and outdoor coils. A detailed description of the components and performance of the heat pumps is provided in Tables 6.2 and 6.3. Key features of the units included:

* Capacity is based on the standard 95 deg outdoor test.

* Unit SEER and HSPF are calculated using DOE test procedure.

* The compressors are currently available from compressor manufacturers.

* Evaporator and condenser size are specified by the frontal area and number of tube rows. All coils are of a wavy fin construction with a thickness of 0.0052 inches.

* The fans are assumed to use permanent split capacitor motors with efficiencies of 55%. The efficiency is the ratio of shaft output to the electrical input.

* A propeller fan is used in the outdoor unit and a centrifugal forward curved fan in the indoor unit.

* An additional penalty due to the heat pump cabinet was included for the package systems. This penalty was in the form of a reduced efficiency for fans in package systems. For example, the efficiency of the indoor fan/motor combination was 20% for split sytems versus 17% for the package systems.

The first two items in Table 6.2 are the superheat in the evaporator and subcooling in the condenser, specified in degrees F. For example, all the units had 25 F superheat at the outlet of the evaporator and 15 F subcooling at the outlet of the condenser in the cooling mode. In the heating mode, the units had 5 F superheat at the outlet of the evaporator and 10 F subcooling at the outlet of the condenser. It was assumed that all the heat pumps used a thermal expansion valve.

Table 6.2 - Hardware data on baseline systems.

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** BaseLine Systems **
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MODEL	3 Ton Units Split	3 Ton Units Packaged	5 Ton Units Split	5 Ton Units Packaged
SUPERHEAT	25 CL/5 HT	25 CL/5 HT	25 CL/5 HT	25 CL/5 HT
SUBCOOL	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT
COMPRESSOR	AD 4.800	AD 3.799	BG 9.400	BG 9.637
EER	10.00	10.00	9.21	9.21
OUTDOOR COIL				
Face Area (ft^2)	10.0000	6.0000	15.0000	14.0000
# of Rows	1.0000	4.0000	2.0000	3.0000
# of Parallel Ckts	2.0000	3.0000	4.0000	4.0000
Fins/Inch	15.0000	15.0000	15.0000	15.0000
Fin Thickness	0.0052	0.0052	0.0052	0.0052
O.D. of Tubes	0.3880	0.3880	0.3880	0.3880
I.D. of Tubes	0.3620	0.3620	0.3620	0.3620
Vert Space (in)	1.2500	1.2500	1.2500	1.2500
Hor Space (in)	1.0830	1.0830	1.0830	1.0830
# of Return Bends	26.0000	104.0000	52.0000	78.0000
Refrig. Control	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY
INDOOR COIL				
Face Area (ft^2)	3.8000	3.5000	5.0000	5.0000
# of Rows	4.0000	4.0000	4.0000	4.0000
# of Parallel Ckts	4.0000	4.0000	6.0000	6.0000
Fins/Inch	13.0000	13.0000	13.0000	13.0000
Fin Thickness	0.0052	0.0052	0.0052	0.0052
O.D. of Tubes	0.3250	0.3250	0.3250	0.3250
I.D. of Tubes	0.3030	0.3030	0.3030	0.3030
Vert Space (in)	1.0000	1.0000	1.0000	1.0000
Hor Space (in)	0.6250	0.6250	0.6250	0.6250
# of Return Bends	72.0000	72.0000	72.0000	72.0000
Refrig. Control	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY
OUTDOOR FAN				
CFM	2800.00	2800.00	4500.00	3500.00
Fan & Motor eff.	0.10	0.09	0.10	0.09
INDOOR FAN				
CFM	1100.00	1100.00	1400.00	1200.00
Fan & Motor eff.	0.20	0.17	0.20	0.17
Ref. Lines (feet)	30.00	6.00	30.00	6.00
Liquid Line O.D.	3/8	3/8	3/8	3/8
Suction Line O.D.	5/8	7/8	1 1/8	1 1/8

Table 6.3 - Performance data on baseline systems.

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*********************
** BaseLine Systems **
*********************
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	3 Ton Units Split	3 Ton Units Packaged	5 Ton Units Split	5 Ton Units Packaged
SYSTEM RATINGS				
CD Cooling/Heating	0.25/0.25	0.25/0.25	0.25/0.25	0.25/0.25
SEER	7.05	7.21	6.76	6.91
HSPF	6.50	5.83	6.18	5.99
SHF 95 deg	0.74	0.73	0.67	0.65
SHF 82 deg	0.72	0.71	0.66	0.64
95 deg F COP	2.07	2.15	2.02	2.07
CAPACITY	35900.00	35900.00	60000.00	60300.00
EER	7.07	7.34	6.90	7.06
82 deg F COP	2.36	2.42	2.26	2.32
CAPACITY	38400.00	38300.00	63600.00	64100.00
EER	8.06	8.25	7.72	7.90
47 deg F COP	2.60	2.25	2.29	2.20
CAPACITY	36300.00	37000.00	65100.00	61500.00
35 deg F COP	2.43	2.19	2.26	2.19
CAPACITY	29600.00	30900.00	54100.00	51300.00
17 deg F COP	2.02	1.80	2.03	1.99
CAPACITY	22200.00	22900.00	41700.00	39500.00

The next items in Table 6.2 provide a description of the compressor: model designation of the compressor, its displacement, and its rated energy efficiency ratio. The first letter in the model designation of the compressor is a code for the compressor manufacturer. The second letter is a code for the manufacturer's own model number (or line). The numbers in the model designation provide the displacement in cubic inches. The compressors for both the three and five ton units were available on the market in 1985. The EERs of the compressors are at the rating conditions shown in Table 6.4.

Table 6.4 - Rating conditions for compressor EER and capacity.

Condition	Value
Evaporator Temp. (F)	45
Gas Leaving Temp. (F)	45
Gas Entering Temp. (F)	95
Condensing Temp. (F)	130
Liquid Entering Temp. (F)	115
Ambient (F)	95

The next two major sections in Table 6.2 include descriptions of the indoor and outdoor heat exchangers. Major items include: the face area of the heat exchangers, fin thickness and spacing, and tube descriptions. The base three ton split and package systems had outdoor face areas of 10.0 and 6.0 sf, respectively. The base five ton units had larger outdoor face areas: 15 and 14 sf for the split and package system, respectively. Indoor heat exchanger areas were 3.8 and 3.5 sf for the three ton split and package systems, respectively, and 5 sf for both five ton systems. Fin spacing on all baseline units was 15 fpi and 13 fpi for the outdoor and indoor heat exchangers, respectively. These heat exchanger areas and fin spacings are comparable to those in lower efficiency systems found on the market.

The next two items in Table 6.2 provide information on the indoor and outdoor fans.* Both fan flow rate in cubic feet per minute (cfm) and combined fan/motor efficiency are given. As

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*The DOE test procedure specifies a fan power of 365 watts/1000 cfm for units that do not have a fan with the evaporator. All units used in this analysis assumed a fan/coil unit. This should provide a better estimate of the best efficiency units that are attainable.

stated earlier, the fan/motor efficiencies for the package systems are slightly lower than those for the split systems to account for added losses in the package system cabinets.

The first items in Table 6.3 are the degradation coefficients for both heating and cooling. For the baseline units, the degradation coefficients were assumed to be 0.25, which is the default value in the DOE test procedure[2].

The next items in Table 6.3 are the SEER, HSPF, and sensible heating factor (SHF) for the baseline units. The HSPF corresponds to the minimum design heating load specified in the DOE test procedure for region IV[2]. This heating load is equal to the heat pump capacity at 47 F, rounded off to the nearest 5000 Btu/hr or 10000 Btu/hr. The values used for rounding off the heating load depend on the size of the load[2]. The HSPF values in the ARI Directory are based on the HSPF for the minimum design heating load[3]. The SHF is the ratio of sensible cooling to total cooling for the unit. The higher the SHF, the less moisture the unit will extract from the return air entering the indoor coil when in the cooling mode. Units typically have SHFs between 0.6 and 0.8[4].

The lowest efficiency three ton package and split heat pumps listed in the ARI Directory in 1985 had SEERs of 6.55 and 5.80, and HSPFs of 5.85 and 5.95, respectively[4]. For five ton systems, the lowest package unit had a SEER of 7.60 and HSPF of 5.88, while the lowest split unit had a SEER of 6.80 and HSPF of 6.10[4]. The baseline systems listed in Tables 6.2 and 6.3 had comparable performances. The object of the baseline system was not to match exactly the performance of the lowest efficiency system on the market, but to provide a starting point for the analysis that was close to the poorest performers on the market.

The last five items in Table 6.3 are the steady state capacity, and energy efficiency ratio(or coefficient of performance) for the units at the five rating points discussed in Chapter 4. These were produced using the ORNL heat pump model[5].

Final Designs

A line of heat pumps for each heat pump class was developed. Fourteen heat pump designs are in each line. The large number of designs provides small incremental improvements in efficiency from the bottom of the line to the top. Each class is discussed seperately below.

3 Ton Split Systems

Tables 6.5 and 6.6 provide the detailed data on the 3 ton split system line. Each unit has an alpha-numeric designation. The first two numbers in the designation specify the capacity of the unit(36 is 3 ton and 60 is 5 ton). For the split sytems, the letter following the two numbers is used to specify the model within the line. The baseline unit has the letter "A" for its model specification. The next model in the line would have "B", etc. For package systems, the two numbers are followed by two letters. The first letter is a "P", indicating it is a package unit. The second letter serves the same purpose as the letter designation on the split sytems (i.e., it indicates the model).

The models are arranged in increasing SEER from left to right in the tables. Thus, unit 36F has a higher SEER than unit 36C.

Below the model designation are the list of design options used on the unit. These options are all relative to the baseline system. The list of design options is in a code that corresponds with the list in Chapter 6. For instance, unit 36D has design options 1A, 1B, 1C, 2, 3, and 4. It has a larger heat exchanger frontal area (option 1A), more tube rows (option 1B), higher fin density (option 1C), smaller compressor (option 2), and higher fan/motor efficiency (option 3) than the baseline unit. It also has a demand defrost system (option 4).

The rest of the data in Tables 6.5 and 6.6 is the same data provided in the same order for the baseline units in Tables 6.2 and 6.3. Thus, all the details on the superheat, subcooling, coils, fans, refrigerant lines, steady state performance, etc., is available on each unit.

For models employing conventional design improvements(36B through 36K), the best SEER and HSPF are 14.98 and 9.64, respectively, in unit 36K. This is a unit whose airflow, fin density, tube rows, etc., have been optimized given the constraints discussed earlier. In 1985, the best 3 ton split sytem listed in the ARI directory had a SEER of 13.2 and HSPF of 8.75[3]. Thus, unit 36K is 10 to 13% more efficient than the best unit in this class in 1985.

Models 36L through 36N use conventional and advanced technology options. Model 36L uses an improved compressor with an EER of 11.0. Model 36M uses a scroll compressor with an EER of 11.3. Model 36N uses a variable speed scroll compressor with variable speed fan motors, and electronic expansion valves. It was assumed that this combination of options would improve heating and cooling efficiency by 19% over Model 36L and reduce the heating degradation coefficient to 0.15. With the advanced options, the estimated maximum technologically efficiency for this class is a SEER of 17.8 and HSPF of 11.5. These efficiencies are over a 30% improvement in cooling and heating efficiency compared to the best conventional units available in 1985. The

Table 6.5 - Hardware data on 3 ton split systems.

****
** 3 Ton Systems **
** Split Units **
****

MODEL	36A	36B	36C	36D	36E	36F	36G
DESIGN OPTIONS		1B,2	1A,1C,2,3	1A,1B,1C,2 3,4	1A,1B,1C,2 3,4	1A,1B,1C,2 3,4,5	1A,1C,1D,2 3,4,5
SUPERHEAT	25 CL/5 HT	25 CL/5 HT	15 CL/5 HT	15 CL/5 HT	15 CL/5 HT	10 CL/5 HT	10 CL/5 HT
SUBCOOL	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT
COMPRESSOR	AD 4.800	AD 4.010	AD 3.770	AD 3.480	AD 3.410	BH 3.270	BH 3.300
EER	10.0	10.0	10.0	10.0	10.0	10.5	10.5
OUTDOOR COIL							
Face Area (ft^2)	10.0000	10.0000	15.0000	15.0000	15.0000	15.0000	20.0000
# of Rows	1.0000	2.0000	1.0000	2.0000	2.0000	2.0000	1.0000
# of Parallel Ckts	2.0000	2.0000	2.0000	2.0000	2.0000	2.0000	2.0000
Fins/Inch	15.0000	15.0000	17.0000	17.0000	17.0000	17.0000	19.0000
Fin Thickness	0.0052	0.0052	0.0045	0.0045	0.0045	0.0045	0.0045
O.D. of Tubes	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880
I.D. of Tubes	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620
Vert Space (in)	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500
Hor Space (in)	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830
# of Return Bends	26.0000	52.0000	26.0000	52.0000	52.0000	52.0000	26.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	LOUVERED
INDOOR COIL							
Face Area (ft^2)	3.8000	3.8000	3.8000	3.8000	4.5000	4.5000	4.5000
# of Rows	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
# of Parallel Ckts	4.0000	4.0000	4.0000	4.0000	4.0000	6.0000	6.0000
Fins/Inch	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000
Fin Thickness	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052
O.D. of Tubes	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250
I.D. of Tubes	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030
Vert Space (in)	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Hor Space (in)	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250
# of Return Bends	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
OUTDOOR FAN							
CFM	2800.00	2800.00	3000.00	3000.00	3000.00	3000.00	3300.00
Fan & Motor eff.	0.10	0.10	0.15	0.15	0.20	0.20	0.20
INDOOR FAN							
CFM	1100.00	1100.00	1100.00	1100.00	1100.00	1100.00	1100.00
Fan & Motor eff.	0.20	0.20	0.25	0.25	0.30	0.30	0.30
Ref. Lines (30 ft)							
Liquid Line O.D.	3/8	3/8	3/8	3/8	3/8	3/8	3/8
Suction Line O.D.	5/8	5/8	7/8	7/8	7/8	7/8	7/8

Table 6.5 (con't) - Hardware data on 3 ton split systems.

****
** 3 Ton Systems **
** Split Units **
****

MODEL	36H	36I	36J	36K	36L	36M	36N
DESIGN OPTIONS	1A,1B,1C,1D 2,3,4,5	1A,1B,1C,1D 2,3,4,6	1A,1C,1D 2,3,4,5	1A,1B,1C,1D 2,3,4,5	1A,1B,1C,1D 2,3,4,5	1A,1B,1C,1D 2,3,4,8	1A,1B,1C,1D,2 3,4,7,8,9,10
SUPERHEAT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT
SUBCOOL	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT
COMPRESSOR	BH 3.330	CF 3.160	BH 3.130	BH 3.060	BI	AS	AS
EER	10.5	9.8	10.5	10.5	11.0	11.3	11.3
OUTDOOR COIL							
Face Area (ft^2)	20.0000	25.0000	25.0000	25.0000	25.0000	25.0000	25.0000
# of Rows	2.0000	2.0000	1.0000	2.0000	2.0000	2.0000	2.0000
# of Parallel Ckts	2.0000	3 CL/5 HT	3 CL/5 HT	3 CL/5 HT	3 CL/5 HT	3 CL/5 HT	3 CL/5 HT
Fins/Inch	19.0000	19.0000	19.0000	19.0000	19.0000	19.0000	19.0000
Fin Thickness	0.0045	0.0045	0.0045	0.0045	0.0045	0.0045	0.0045
O.D. of Tubes	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880
I.D. of Tubes	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620
Vert Space (in)	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500
Hor Space (in)	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830
# of Return Bends	52.0000	52.0000	26.0000	52.0000	52.0000	52.0000	52.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED
INDOOR COIL							
Face Area (ft^2)	4.5000	5.5000	5.5000	5.5000	5.5000	5.5000	5.5000
# of Rows	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
# of Parallel Ckts	6.0000	7 CL/ 3 HT	7 CL/ 3 HT	7 CL/ 3 HT	7 CL/ 3 HT	7 CL/ 3 HT	7 CL/ 3 HT
Fins/Inch	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000
Fin Thickness	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052
O.D. of Tubes	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250
I.D. of Tubes	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030
Vert Space (in)	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Hor Space (in)	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250
# of Return Bends	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
OUTDOOR FAN							
CFM	3300.00	3500.00	3500.00	3500.00	3500.00	3500.00	3500.00
Fan & Motor eff.	0.20	0.25	0.25	0.25	0.25	0.25	0.25
INDOOR FAN							
CFM	1100.00	1100.00	1100.00	1100.00	1100.00	1100.00	1100.00
Fan & Motor eff.	0.30	0.35	0.35	0.35	0.35	0.35	0.35
Ref. Lines (30 ft)							
Liquid Line O.D.	3/8	3/8	3/8	3/8	3/8	3/8	3/8
Suction Line O.D.	7/8	7/8	7/8	7/8	7/8	7/8	7/8

Table 6.6 - Performance data on 3 ton split systems.

****
** 3 Ton Systems **
** Split Units **
****

	36A	36B	36C	36D	36E	36F	36G
SYSTEM RATINGS							
Cd Cooling/Heating	0.25/0.25	0.25/0.25	0.20/0.20	0.20/0.20	0.20/0.20	0.20/0.20	0.15/0.20
SEER	7.05	8.42	9.43	10.28	10.82	12.81	13.09
HSPF	6.50	6.70	7.56	7.67	7.90	8.20	6.97
SHF 95 deg	0.74	0.74	0.75	0.75	0.75	0.75	0.75
SHF 82 deg	0.72	0.72	0.73	0.73	0.73	0.73	0.73
95 deg F COP	2.07	2.49	2.68	2.94	3.07	3.53	3.50
CAPACITY	35900.00	35900.00	36000.00	35800.00	36000.00	36100.00	36000.00
EER	7.06	8.49	9.15	10.02	10.48	12.03	11.95
82 deg F COP	2.36	2.82	3.07	3.35	3.52	4.17	4.15
CAPACITY	38400.00	38100.00	38400.00	38200.00	38400.00	38600.00	38500.00
EER	8.04	9.63	10.48	11.43	12.02	14.24	14.15
47 deg F COP	2.60	2.69	3.02	3.03	3.16	3.37	3.50
CAPACITY	36300.00	32500.00	32300.00	29600.00	26300.00	27900.00	29400.00
35 deg F COP	2.43	2.52	2.79	2.74	2.87	3.02	3.15
CAPACITY	29600.00	27000.00	26500.00	24400.00	24100.00	22800.00	24000.00
17 deg F COP	2.02	2.09	2.33	2.26	2.35	2.45	2.57
CAPACITY	22200.00	19800.00	19300.00	17600.00	17300.00	16100.00	17000.00

Table 6.6 (con't) - Performance data on 3 ton split systems.

****
** 3 Ton Systems **
** Split Units **
****

	36H	36I High Spd (2 Speed Compressor)	36I Low Spd (2 Speed Compressor)	36J	36K	36L	36M	36N
SYSTEM RATINGS								
CD Cooling/Heating	0.15/0.20	0.20/0.20		0.15/0.20	0.15/0.20	0.15/0.20	0.15/0.20	0.15/0.15
SEER	13.44	13.77		14.45	14.98	15.72	16.16	17.83
HSPF	8.25	9.78		9.50	9.64	10.03	10.27	11.34
SHF 95 deg	0.75	0.76	0.97	0.76	0.76	0.76	0.76	0.76
SHF 82 deg	0.73	0.73	0.93	0.73	0.73	0.73	0.73	0.73
95 deg F COP	3.62	3.84	4.40	3.81	3.95	4.14	4.26	4.70
CAPACITY	36100.00	36000.00	21200.00	35900.00	36000.00	36000.00	36000.00	36000.00
EER	12.34	13.11	15.02	13.00	13.47	14.15	14.55	16.03
82 deg F COP	4.26	4.41	5.38	4.58	4.75	4.98	5.12	5.65
CAPACITY	38500.00	38500.00	22900.00	38500.00	38500.00	38500.00	38500.00	38500.00
EER	14.54	15.06	18.36	15.63	16.20	17.01	17.49	19.27
47 deg F COP	3.37	3.92	4.64	4.02	4.08	4.28	4.41	4.86
CAPACITY	27100.00	32800.00	18400.00	32800.00	33300.00	33300.00	33300.00	33300.00
35 deg F COP	3.00	3.46	3.74	3.61	3.67	3.85	3.96	4.37
CAPACITY	22100.00	26300.00	14200.00	26800.00	27200.00	27200.00	27200.00	27200.00
17 deg F COP	2.43	2.78	2.49	2.92	2.97	3.11	3.20	3.53
CAPACITY	15500.00	18400.00	87000.00	18700.00	18900.00	18900.00	18900.00	18900.00

technological limit of 17.8 SEER is close to an 18.0 limit recently proposed by one industry expert[6].

Because it is uncertain how the final DOE test procedure will account for some of the advanced design options (variable speed compressors, variable speed fans, and electronic expansion valves), the actual value of the technological limit may vary by a few percent from what is provided here. Model 36N is based on our best engineering judgement. Once data are available on variable speed and scroll compressors and the test procedure is finalized, model runs should be made to better estimate the technological limit.

Even though the efficiencies of the maximum technological unit are significantly above currently available units, the industry is moving quickly to provide units with efficiencies approaching this performance. At a recent heat pump seminar, an Electric Power Research Institute representative stated that one major U.S. manufacturer will introduce a heat pump with a SEER of 16.7 and HSPF of 11.0 in 1987[7]. At a January 1986 trade show, another manufacturer was showing a 3 ton split heat pump with a SEER of 15.0 and HSPF over 10 that would be introduced in 1987.

3 Ton Package Systems

Tables 6.7 and 6.8 list the units in the line of 3 ton package sytems. Unit 36PA is the baseline unit, while unit 36PN has the maximum efficiency that is technologically feasible for 3 ton package units. Other data about the systems are in the same order as that for the 3 ton split systems. The efficiencies for the package systems are slightly lower than for the split because of assumptions about fan losses due to the cabinets in package sytems.

Model 36PK is the optimized system with conventional design options. It has a SEER of 14.59 and HSPF of 9.73. The highest listing for this class in the ARI Directory had a a SEER of 9.6 and HSPF of 7.60[3]. Thus, optimizing with conventional options offers large improvements over the best available unit on the market. With the addition of advanced options, the SEER of the 3 ton package system could improve to 17.56.

5 Ton Split Systems

Tables 6.9 and 6.10 list the units in the 5 ton split system line. Unit 60A is the baseline unit and 60N is the maximum technologically feasible unit. The maximum efficiency for the 5 ton split system is a SEER of 13.16 and HSPF of 10.21. The best 5 ton split system listed in the ARI directory had a SEER of 10.8 and HSPF of 8.05[3].

Table 6.7 - Hardware data on 3 ton package systems.

```
*******************
**  3 Ton Systems **
** Packaged Units **
*******************
```

MODEL	36PA	36PB	36PC	36PD	36PE	36PF	36PG
DESIGN OPTIONS		1A,2	1A	1A,1B,2	1A,1C,2,3	1A,1B,1C,2,3	1A,1C,2,3,4
SUPERHEAT	25 CL/5 HT	25 CL/5 HT	15 CL/5 HT	15 CL/5 HT	15 CL/5 HT	15 CL/5 HT	10 CL/5 HT
SUBCOOL	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT
COMPRESSOR	AD 3.799	AD 3.715	AD 3.809	AD 3.606	AD 3.473	AD 3.390	AD 3.404
EER	10.0	10.0	10.0	10.0	10.0	10.0	10.0
OUTDOOR COIL							
Face Area (ft^2)	6.0000	8.0000	10.0000	10.0000	14.0000	14.0000	17.0000
# of Rows	4.0000	4.0000	2.0000	4.0000	2.0000	4.0000	2.0000
# of Parallel Ckts	3.0000	3.0000	3.0000	3.0000	3.0000	3.0000	3.0000
Fins/Inch	15.0000	15.0000	15.0000	15.0000	17.0000	17.0000	19.0000
Fin Thickness	0.0052	0.0052	0.0052	0.0052	0.0045	0.0045	0.0045
O.D. of Tubes	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880
I.D. of Tubes	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620
Vert Space (in)	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500
Hor Space (in)	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830
# of Return Bends	104.0000	104.0000	52.0000	104.0000	52.0000	104.0000	104.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
INDOOR COIL							
Face Area (ft^2)	3.5000	3.5000	3.5000	3.5000	4.5000	4.5000	4.5000
# of Rows	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
# of Parallel Ckts	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
Fins/Inch	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000
Fin Thickness	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052
O.D. of Tubes	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250
I.D. of Tubes	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030
Vert Space (in)	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Hor Space (in)	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250
# of Return Bends	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
OUTDOOR FAN							
CFM	2800.00	2800.00	3000.00	2800.00	3000.00	2800.00	3000.00
Fan & Motor eff.	0.17	0.17	0.17	0.17	0.24	0.24	0.26
INDOOR FAN							
CFM	1100.00	1100.00	1100.00	1100.00	1100.00	1100.00	1100.00
Fan & Motor eff.	0.09	0.09	0.09	0.09	0.14	0.14	0.18
Ref. Lines (6 ft)							
Liquid Line O.D.	3/8	3/8	3/8	3/8	3/8	3/8	3/8
Suction Line O.D.	7/8	7/8	7/8	7/8	7/8	7/8	7/8

Table 6.7 (con't) - Hardware data on 3 ton package systems.

```
*******************
**  3 Ton Systems **
** Packaged Units **
*******************
```

MODEL	36PH	36PI	36PJ	36PK	36PL	36PM	36PN	36PO
DESIGN OPTIONS	1A,1C,2,3,4 5	1A,1C,2,3,4 5	1A,1D,1C 2,3,4,6	1A,1D,1C 2,3,4,5	1A,1D,1C 2,3,4,5	1A,1D,1C 2,3,4,5	1A,1D,1C,2 3,4,5,8	1A,1D,1C,2 3,4,5,7,8,10
SUPERHEAT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT
SUBCOOL	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT
COMPRESSOR	BH 3.412	BH 3.348	BH 3.127	BH 3.094	BH 3.058	BI	AS	AS
EER	10.5	10.5	9.8	10.5	10.5	11.0	11.3	11.3
OUTDOOR COIL								
Face Area (ft^2)	17.0000	17.0000	25.0000	20.0000	25.0000	25.0000	25.0000	25.0000
# of Rows	2.0000	2.0000	2.0000	2.0000	2.0000	2.0000	2.0000	2.0000
# of Parallel Ckts	3.0000	3.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
Fins/Inch	19.0000	19.0000	19.0000	19.0000	19.0000	19.0000	19.0000	19.0000
Fin Thickness	0.0045	0.0045	0.0045	0.0045	0.0045	0.0045	0.0045	0.0045
O.D. of Tubes	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880
I.D. of Tubes	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620
Vert Space (in)	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500
Hor Space (in)	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830
# of Return Bends	52.0000	52.0000	52.0000	52.0000	52.0000	52.0000	52.0000	52.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED
INDOOR COIL								
Face Area (ft^2)	4.5000	5.5000	5.5000	5.5000	5.5000	5.5000	5.5000	5.5000
# of Rows	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
# of Parallel Ckts	4.0000	4.0000	6.0000	4.0000	6.0000	6.0000	6.0000	6.0000
Fins/Inch	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000
Fin Thickness	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052
O.D. of Tubes	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250
I.D. of Tubes	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030
Vert Space (in)	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Hor Space (in)	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250
# of Return Bends	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
OUTDOOR FAN								
CFM	3000.00	3000.00	3600.00	3300.00	3600.00	3600.00	3600.00	3600.00
Fan & Motor eff.	0.26	0.31	0.31	0.31	0.31	0.31	0.31	0.31
INDOOR FAN								
CFM	1100.00	1100.00	1000.00	1100.00	1000.00	1000.00	1000.00	1000.00
Fan & Motor eff.	0.18	0.23	0.23	0.23	0.23	0.23	0.23	0.23
Ref. Lines (6 ft)								
Liquid Line O.D.	3/8	3/8	3/8	3/8	3/8	3/8	3/8	3/8
Suction Line O.D.	7/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8

Table 6.8 - Performance data on 3 ton package systems.

```
*******************
** 3 Ton Systems **
** Package Units **
*******************
```

	36PA	36PB	36PC	36PD	36PE	36PF	36PG
SYSTEM RATINGS							
CD Cooling/Heating	0.25/0.25	0.25/0.25	0.25/0.25	0.25/0.25	0.20/0.20	0.20/0.20	0.15/0.20
SEER	7.21	8.06	8.29	8.66	10.32	10.55	11.20
HSPF	5.83	6.53	7.11	6.97	8.66	8.30	8.83
SHF 95 deg	0.73	0.73	0.74	0.74	0.75	0.75	0.75
SHF 82 deg	0.71	0.71	0.72	0.72	0.73	0.73	0.73
95 deg F COP	2.15	2.39	2.44	2.55	2.93	3.00	3.07
CAPACITY	35900.00	35900.00	35900.00	35900.00	36100.00	36100.00	36100.00
EER	7.34	8.15	8.34	8.71	10.00	10.25	10.48
82 deg F COP	2.42	2.70	2.78	2.90	3.36	3.43	3.55
CAPACITY	35300.00	38200.00	38400.00	38400.00	38400.00	38400.00	38600.00
EER	8.25	9.21	9.48	9.89	11.48	11.72	12.11
47 deg F COP	2.25	2.60	2.80	2.84	3.34	3.28	3.45
CAPACITY	37000.00	36200.00	37200.00	35400.00	35300.00	33500.00	34900.00
35 deg F COP	2.19	2.44	2.69	2.60	3.10	2.99	3.19
CAPACITY	30900.00	30300.00	31000.00	29500.00	29400.00	27900.00	29000.00
17 deg F COP	1.80	2.03	2.28	2.18	2.62	2.52	2.70
CAPACITY	22900.00	22500.00	23000.00	21900.00	21500.00	20400.00	21200.00

Table 6.8 - Performance data on 3 ton package systems.

```
*******************
** 3 Ton Systems **
** Package Units **
*******************
```

	36PH	36PI	36PJ 2 Speed Compressor High Spd	36PJ 2 Speed Compressor Low Spd	36PK	36PL	36PM	36PN	36PO
SYSTEM RATINGS									
CD Cooling/Heating	0.15/0.20	0.15/0.20	0.20/0.20		0.15/0.20	0.15/0.20	0.15/0.20	0.15/0.20	0.15/0.15
SEER	12.86	13.45	13.71		14.59	15.01	15.81	16.66	17.56
HSPF	9.21	9.53	10.39		9.73	9.81	10.26	10.74	11.22
SHF 95 deg	0.75	0.76	0.75	0.97	0.75	0.76	0.76	0.76	0.76
SHF 82 deg	0.73	0.73	0.73	0.94	0.73	0.73	0.73	0.73	0.73
95 deg F COP	3.42	3.57	3.84	4.34	3.84	3.94	4.15	4.38	4.61
CAPACITY	36100.00	36100.00	36000.00	21100.00	36100.00	36000.00	36000.00	36000.00	36000.00
EER	11.68	12.18	13.12	14.80	13.11	13.44	14.15	14.94	15.72
82 deg F COP	4.07	4.26	4.42	5.27	4.62	4.76	5.01	5.28	5.56
CAPACITY	38700.00	38800.00	38600.00	22700.00	38700.00	38600.00	38600.00	38600.00	38600.00
EER	13.91	14.55	15.08	17.98	15.77	16.23	17.09	18.04	18.99
47 deg F COP	3.69	3.86	4.05	4.85	4.01	4.06	4.28	4.52	4.75
CAPACITY	34200.00	33900.00	34000.00	19700.00	33800.00	33700.00	33700.00	33700.00	33700.00
35 deg F COP	3.38	3.53	3.62	4.00	3.63	3.67	3.87	4.08	4.30
CAPACITY	28300.00	28100.00	27600.00	15500.00	27900.00	27800.00	27800.00	27800.00	27800.00
17 deg F COP	2.83	2.94	2.97	2.83	3.00	3.03	3.19	3.37	3.55
CAPACITY	20400.00	20200.00	19700.00	10100.00	19900.00	19800.00	19800.00	19800.00	19800.00

Table 6.9 - Hardware data on 5 ton split systems.

```
******************
** 5 Ton Systems **
**  Split Units  **
******************
```

MODEL	60A	60B	60C	60D	60E	60F	60G
DESIGN OPTIONS		1C	1A,1C,2,3	1A,1C,2,3 4	1A,1C,2,3 4,5	1A,1C,1D,2 3,4,5	1A,1C,1D,2 3,4,5
SUPERHEAT	25 CL/5 HT	25 CL/5 HT	25 CL/5 HT	15 CL/5 HT	15 CL/5 HT	15 CL/5 HT	15 CL/5 HT
SUBCOOL	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT
COMPRESSOR	BG 9.538	BG 9.538	BG 8.746	BG 7.909	BJ 7.840	BJ 7.357	BJ 7.168
EER	9.21	9.21	9.21	9.21	9.51	9.51	9.51
OUTDOOR COIL							
Face Area (ft^2)	15.0000	15.0000	20.0000	20.0000	20.0000	25.0000	25.0000
# of Rows	2.0000	2.0000	2.0000	2.0000	2.0000	2.0000	2.0000
# of Parallel Ckts	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
Fins/Inch	15.0000	17.0000	17.0000	17.0000	17.0000	19.0000	19.0000
Fin Thickness	0.0052	0.0045	0.0045	0.0045	0.0045	0.0045	0.0045
O.D. of Tubes	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880
I.D. of Tubes	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620
Vert Space (in)	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500
Hor Space (in)	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830
# of Return Bends	52.0000	52.0000	52.0000	52.0000	52.0000	52.0000	52.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	LOUVERED	LOUVERED
INDOOR COIL							
Face Area (ft^2)	5.0000	5.0000	5.0000	6.0000	6.0000	6.0000	7.0000
# of Rows	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
# of Parallel Ckts	6.0000	6.0000	6.0000	6.0000	6.0000	6.0000	6.0000
Fins/Inch	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000
Fin Thickness	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052
O.D. of Tubes	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250
I.D. of Tubes	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030
Vert Space (in)	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Hor Space (in)	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250
# of Return Bends	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
OUTDOOR FAN							
CFM	4000.00	4000.00	4300.00	4300.00	4300.00	5000.00	5500.00
Fan & Motor eff.	0.10	0.10	0.15	0.15	0.15	0.20	0.20
INDOOR FAN							
CFM	1400.00	1400.00	1400.00	1400.00	1400.00	1400.00	1400.00
Fan & Motor eff.	0.20	0.20	0.25	0.25	0.25	0.30	0.30
Ref. Lines (30 ft)							
Liquid Line O.D.	3/8	3/8	3/8	3/8	3/8	3/8	3/8
Suction Line O.D.	7/8	7/8	7/8	1 1/8	1 1/8	1 1/8	1 1/8

Table 6.9 (con't) - Hardware data on 5 ton split systems.

```
******************
** 5 Ton Systems **
**  Split Units  **
******************
```

MODEL	60H	60I	60J	60K	60L	60M	60N
DESIGN OPTIONS	1A,1C,1D,2 3,4,5	1A,1C,1D,2 3,4,5	1A,1C,1D,2 3,4,5	1A,1C,1D 2,3,4,5	1A,1C,1D 2,3,4,8	1A,1C,1D,2 3,4,6	1A,1C,1D,2,3 4,7,8,9,10
SUPERHEAT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT
SUBCOOL	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT
COMPRESSOR	BJ 7.037	BJ 6.883	BJ 6.510	BI	AS	CF 6.134	AS
EER	9.51	9.51	9.51	11.00	11.30	9.80	11.30
OUTDOOR COIL							
Face Area (ft^2)	30.0000	30.0000	30.0000	30.0000	30.0000	30.0000	30.0000
# of Rows	2.0000	2.0000	1.0000	1.0000	1.0000	2.0000	1.0000
# of Parallel Ckts	4.0000	4.0000	4 CL/6 HT	4 CL/6 HT	4 CL/6 HT	4 CL/6 HT	4 CL/6 HT
Fins/Inch	19.0000	19.0000	19.0000	19.0000	19.0000	19.0000	19.0000
Fin Thickness	0.0045	0.0045	0.0045	0.0045	0.0045	0.0045	0.0045
O.D. of Tubes	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880
I.D. of Tubes	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620
Vert Space (in)	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500
Hor Space (in)	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830
# of Return Bends	52.0000	52.0000	26.0000	26.0000	26.0000	26.0000	26.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED
INDOOR COIL							
Face Area (ft^2)	7.0000	8.0000	8.0000	8.0000	8.0000	8.0000	8.0000
# of Rows	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
# of Parallel Ckts	6.0000	6.0000	8 CL/4 HT	8 CL/4 HT	8 CL/4 HT	8 CL/4 HT	8 CL/4 HT
Fins/Inch	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000
Fin Thickness	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052
O.D. of Tubes	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250
I.D. of Tubes	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030
Vert Space (in)	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Hor Space (in)	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250
# of Return Bends	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
OUTDOOR FAN							
CFM	6000.00	6000.00	6500.00	6500.00	6500.00	6000.00	6500.00
Fan & Motor eff.	0.20	0.25	0.25	0.25	0.25	0.25	0.25
INDOOR FAN							
CFM	1400.00	1400.00	1400.00	1400.00	1400.00	1400.00	1400.00
Fan & Motor eff.	0.30	0.35	0.35	0.35	0.35	0.35	0.35
Ref. Lines (30 ft)							
Liquid Line O.D.	3/8	3/8	3/8	3/8	3/8	3/8	3/8
Suction Line O.D.	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8

Table 6.10 - Performance data on 5 ton split systems.

```
*******************
** 5 Ton Systems **
**  Split Units  **
*******************
```

	60A	60B	60C	60D	60E	60F	60G
SYSTEM RATINGS							
CD Cooling/Heating	0.25/0.25	0.25/0.25	0.25/0.25	0.20/0.20	0.20/0.20	0.20/0.20	0.20/0.20
SEER	6.73	6.73	7.53	8.30	9.17	9.95	10.13
HSPF	6.19	6.21	6.59	7.25	7.58	7.77	7.85
SHF 95 deg	0.67	0.67	0.68	0.68	0.68	0.68	0.68
SHF 82 deg	0.66	0.67	0.67	0.67	0.67	0.67	0.67
95 deg F COP	2.01	2.01	2.25	2.40	2.63	2.85	2.89
CAPACITY	59800.00	60000.00	59700.00	60200.00	60000.00	59700.00	59800.00
EER	6.85	6.87	7.67	8.19	8.96	9.71	9.87
82 deg F COP	2.25	2.25	2.52	2.70	2.99	3.24	3.30
CAPACITY	63500.00	63400.00	63100.00	63800.00	63900.00	63600.00	63600.00
EER	7.69	7.69	8.60	9.23	10.19	11.06	11.26
47 deg F COP	2.32	2.30	2.51	2.64	2.80	2.95	3.01
CAPACITY	64900.00	65200.00	61700.00	59400.00	57700.00	63600.00	56100.00
35 deg F COP	2.28	2.27	2.42	2.52	2.68	2.76	2.80
CAPACITY	53900.00	54200.00	51500.00	49300.00	47800.00	46200.00	46400.00
17 deg F COP	2.03	2.03	2.17	2.26	2.36	2.43	2.45
CAPACITY	41500.00	41600.00	39200.00	37400.00	35500.00	34000.00	33900.00

Table 6.10 (con't) - Performance data on 5 ton split systems.

****
** 5 Ton Systems **
** Split Units **
****

	60H	60I	60J	60K	60L	60M (2 Speed Compressor) Hi Spd	60M (2 Speed Compressor) Lo Spd	60N
SYSTEM RATINGS								
CD Cooling/Heating	0.15/0.20	0.15/0.20	0.15/0.20	0.15/0.20	0.15/0.20	0.20/0.20		0.15/0.15
SEER	10.66	11.00	11.06	11.61	11.93	12.06		13.16
HSPF	7.86	7.98	8.56	8.94	9.54	9.26		10.21
SHF 95 deg	0.68	0.68	0.69	0.69	0.69	0.68	0.83	0.69
SHF 82 deg	0.67	0.67	0.67	0.67	0.67	0.67	0.79	0.67
95 deg F COP	2.96	3.04	3.04	3.45	3.28	3.36	3.91	3.62
CAPACITY	59900.00	59700.00	59000.00	59000.00	59000.00	59800.00	40500.00	59000.00
EER	10.09	10.39	10.38	11.77	11.21	11.47	13.35	12.35
82 deg F COP	3.38	3.48	3.50	3.97	3.78	3.82	4.74	4.17
CAPACITY	63700.00	63500.00	63000.00	63000.00	63000.00	63800.00	40500.00	63800.00
EER	11.53	11.88	11.96	13.56	12.91	13.05	16.17	14.23
47 deg F COP	3.03	3.13	3.32	3.76	3.58	3.40	4.12	3.94
CAPACITY	55300.00	54900.00	57600.00	57600.00	57600.00	57600.00	34800.00	57600.00
35 deg F COP	2.82	2.91	3.10	3.51	3.34	3.12	3.46	3.68
CAPACITY	45600.00	45200.00	47400.00	47400.00	47400.00	47200.00	27500.00	47200.00
17 deg F COP	2.45	2.52	2.69	3.05	2.91	2.66	2.51	3.20
CAPACITY	33200.00	32800.00	34300.00	34300.00	34300.00	34000.00	17900.00	34000.00

The maximum efficiencies achievable for the 5 ton systems were 30 to 35% smaller than 3 ton systems. The primary reason for this lower efficiency was the constraints on the coil sizes for the 5 ton system. The efficiencies of the 5 ton systems could have been higher with larger coils, but this would have led to unacceptably large cabinet sizes. Another contributing factor to the smaller efficiencies for the 5 ton systems was that the compressors available for this size system were about 10% less efficient than those for the 3 ton systems.

5 Ton Package Systems

Tables 6.11 and 6.12 list the units in the 5 ton package line. Unit 60PA is the baseline system and 60PN is the maximum technologically feasible unit. The maximum efficiencies estimated for the 5 ton package system are a SEER of 13.19 and HSPF of 9.97. The best unit in the ARI directory had a SEER of 9.0 and HSPF of 6.8[3].

REFERENCES

1. Consumer Products Efficiency Standards Engineering Analysis Document, DOE/CE-0030, U.S. Department of Energy, March 1982.

2. "Test Procedure for Central Air Conditioners, Including Heat Pumps", Federal Register, Dec. 27, 1979, pp. 76700-76723.

3. "Directory of Certified Unitary Air Conditioners, Air Source Heat Pumps, and Sound-Rated Outdoor Unitary Equipment", Air Conditioning and Refrigeration Institute, Jan. 1 - June 30, 1985, Arlington, VA.

4. ASHRAE Handbook, 1985 Fundamentals, American Society of Heating, Refrigerating, and Air Conditioning Engineers, Atlanta, GA, 1985.

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

6. J. Crawford, "Heat Pump Technology Overview", Proceedings of the Electric Power Research Institute's seminar: Meeting Customer Needs with Heat Pumps, Kansas City, April 7-9, 1986.

7. A. Lannus, "Current Heat Pump Development", Presented at the Electric Power Research Institute's seminar, Meeting Customer Needs With Heat Pumps, Kansas City, April 7-9, 1986.

Table 6.11 - Hardware data on 5 ton package systems.

```
*******************
** 5 Ton Systems **
** Package  Units **
*******************
```

MODEL	60PA	60PB	60PC	60PD	60PE	60PF	60PG
DESIGN OPTIONS		1A,1B,2	1A,1C,2,3 4	1A,1C,2,3 4,5	1A,1C,2,3 4,5	1A,1C,2,3 4,5	1A,1C,1D,2 3,4,5
SUPERHEAT	25 CL/5 HT	25 CL/5 HT	15 CL/5 HT	15 CL/5 HT	15 CL/5 HT	15 CL/5 HT	10 CL/5 HT
SUBCOOL	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT
COMPRESSOR	BG 9.637	BG 9.213	BG 9.637	BJ 9.087	BJ 8.375	BJ 8.375	BJ 8.001
EER	9.21	9.21	9.21	9.51	9.51	9.51	9.51
OUTDOOR COIL							
Face Area (ft^2)	14.0000	14.0000	17.0000	17.0000	20.0000	20.0000	25.0000
# of Rows	3.0000	5.0000	2.0000	2.0000	2.0000	2.0000	2.0000
# of Parallel Ckts	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
Fins/Inch	15.0000	15.0000	17.0000	17.0000	19.0000	19.0000	19.0000
Fin Thickness	0.0052	0.0052	0.0045	0.0045	0.0045	0.0045	0.0045
O.D. of Tubes	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880
I.D. of Tubes	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620
Vert Space (in)	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500
Hor Space (in)	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830
# of Return Bends	78.0000	130.0000	52.0000	52.0000	52.0000	52.0000	52.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	LOUVERED
INDOOR COIL							
Face Area (ft^2)	5.0000	5.0000	6.0000	6.0000	6.0000	7.0000	7.0000
# of Rows	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
# of Parallel Ckts	6.0000	6.0000	6.0000	6.0000	6.0000	6.0000	6.0000
Fins/Inch	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000
Fin Thickness	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052
O.D. of Tubes	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250
I.D. of Tubes	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030
Vert Space (in)	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Hor Space (in)	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250
# of Return Bends	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
OUTDOOR FAN							
CFM	3500.00	3500.00	3500.00	3500.00	4000.00	4000.00	4000.00
Fan & Motor eff.	0.09	0.09	0.14	0.14	0.18	0.18	0.23
INDOOR FAN							
CFM	1200.00	1200.00	1200.00	1200.00	1200.00	1200.00	1200.00
Fan & Motor eff.	0.17	0.17	0.24	0.24	0.27	0.27	0.31
Ref. Lines (6 ft)							
Liquid Line O.D.	3/8	3/8	3/8	3/8	3/8	3/8	3/8
Suction Line O.D.	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8

Table 6.11 (con't) - Hardware data on 5 ton package systems.

```
*******************
** 5 Ton Systems **
** Package Units **
*******************
```

MODEL	60PH	60PI	60PJ	60PK	60PL	60PM	60PN
DESIGN OPTIONS	1A,1C,1D,2 3,4,5	1A,1C,1D,2 3,4,5	1A,1C,1D,2 3,4,5	1A,1C,1D,2 3,4,5	1A,1C,1D,2 3,4,7	1A,1C,1D,2 3,4,6	1A,1C,1D,2 3,4,7,8,9,10
SUPERHEAT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT	10 CL/5 HT
SUBCOOL	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT	15 CL/10 HT
COMPRESSOR	BJ 7.233	BJ 7.200	BJ 7.001	BI	AS	CF 6.746	AS
EER	9.51	9.51	9.51	10.00	10.30	9.80	10.30
OUTDOOR COIL							
Face Area (ft^2)	25.0000	25.0000	30.0000	30.0000	30.0000	30.0000	30.0000
# of Rows	2.0000	2.0000	2.0000	2.0000	2.0000	2.0000	2.0000
# of Parallel Ckts	4 CL/6 HT	4 CL/6 HT	4 CL/6 HT	4 CL/6 HT	4 CL/6 HT	4 CL/6 HT	4 CL/6 HT
Fins/Inch	19.0000	19.0000	19.0000	19.0000	19.0000	19.0000	19.0000
Fin Thickness	0.0045	0.0045	0.0045	0.0045	0.0045	0.0045	0.0045
O.D. of Tubes	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880	0.3880
I.D. of Tubes	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620	0.3620
Vert Space (in)	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500	1.2500
Hor Space (in)	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830	1.0830
# of Return Bends	52.0000	52.0000	52.0000	52.0000	52.0000	52.0000	52.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED
INDOOR COIL							
Face Area (ft^2)	7.0000	8.0000	8.0000	8.0000	8.0000	8.0000	8.0000
# of Rows	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000	4.0000
# of Parallel Ckts	8 CL/4 HT	8 CL/4 HT	8 CL/4 HT	8 CL/4 HT	8 CL/4 HT	8 CL/4 HT	8 CL/4 HT
Fins/Inch	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000	13.0000
Fin Thickness	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052	0.0052
O.D. of Tubes	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250	0.3250
I.D. of Tubes	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030	0.3030
Vert Space (in)	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Hor Space (in)	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250	0.6250
# of Return Bends	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000	72.0000
Refrig. Control	TXV	TXV	TXV	TXV	TXV	TXV	TXV
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
OUTDOOR FAN							
CFM	4000.00	4000.00	4500.00	4500.00	4500.00	5000.00	4500.00
Fan & Motor eff.	0.23	0.23	0.23	0.23	0.23	0.23	0.23
INDOOR FAN							
CFM	1200.00	1200.00	1200.00	1200.00	1200.00	1200.00	1200.00
Fan & Motor eff.	0.31	0.31	0.31	0.31	0.31	0.31	0.31
Ref. Lines (6 ft)							
Liquid Line O.D.	3/8	3/8	3/8	3/8	3/8	3/8	3/8
Suction Line O.D.	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8

Table 6.12 - Performance data on 5 ton package systems.

```
*******************
** 5 Ton Systems **
** Package Units **
*******************
```

	60PA	60PB	60PC	60PD	60PE	60PF	60PG
SYSTEM RATINGS							
CD Cooling/Heating	0.25/0.25	0.25/0.25	0.25/0.25	0.25/0.25	0.20/0.20	0.20/0.20	0.15/0.20
SEER	6.91	7.10	7.52	8.12	9.13	9.15	9.95
HSPF	5.99	5.98	6.73	6.97	7.59	7.69	8.14
SHF 95 deg	0.65	0.65	0.65	0.65	0.65	0.65	0.65
SHF 82 deg	0.64	0.64	0.64	0.64	0.64	0.64	0.64
95 deg F COP	2.07	2.12	2.24	2.40	2.61	2.62	2.78
CAPACITY	60300.00	60100.00	60000.00	60200.00	59700.00	60100.00	59900.00
EER	7.07	7.24	7.64	8.20	8.90	8.96	9.48
82 deg F COP	2.32	2.38	2.52	2.72	2.97	2.98	3.15
CAPACITY	64100.00	63800.00	63400.00	64100.00	63600.00	63900.00	63600.00
EER	7.92	8.11	8.60	9.28	10.14	10.17	10.76
47 deg F COP	2.20	2.18	2.52	2.64	2.76	2.80	2.91
CAPACITY	61500.00	58100.00	64500.00	60300.00	58500.00	59000.00	56900.00
35 deg F COP	2.19	2.18	2.47	2.54	2.63	2.66	2.73
CAPACITY	51300.00	49100.00	51200.00	49200.00	47800.00	48100.00	46600.00
17 deg F COP	1.99	1.98	2.24	2.35	2.43	2.47	2.53
CAPACITY	39500.00	38000.00	39500.00	38100.00	36600.00	36900.00	35600.00

Table 6.12 (con't) - Performance data on 5 ton package systems.

****
** 5 Ton Systems **
** Package Units **
****

	60PH	60PI	60PJ	60PK	60PL	60PM 2 Speed Compressor High Spd	60PM 2 Speed Compressor Low Spd	60PN
SYSTEM RATINGS								
CD Cooling/Heating	0.15/0.20	0.15/0.20	0.15/0.20	0.15/0.20	0.15/0.20	0.20/0.20		0.15/0.15
SEER	10.59	10.70	11.10	11.65	11.99	12.20		13.19
HSPF	8.36	8.45	8.52	8.91	9.14	9.76		9.97
SHF 95 deg	0.65	0.65	0.65	0.65	0.65	0.65	0.76	0.65
SHF 82 deg	0.64	0.64	0.64	0.64	0.64	0.64	0.74	0.64
95 deg F COP	2.95	2.97	3.07	3.23	3.32	3.22	3.94	3.66
CAPACITY	59800.00	60200.00	60100.00	60100.00	60100.00	60300.00	39100.00	60100.00
EER	10.06	10.13	10.49	11.02	11.33	10.99	13.44	12.49
82 deg F COP	3.35	3.39	3.52	3.69	3.80	3.66	4.77	4.18
CAPACITY	63500.00	63900.00	63800.00	63800.00	63800.00	64100.00	42200.00	63800.00
EER	11.45	11.57	12.00	12.60	12.96	12.49	16.28	14.28
47 deg F COP	3.11	3.15	3.20	3.35	3.45	3.23	4.11	3.80
CAPACITY	59700.00	59900.00	59500.00	59500.00	59500.00	59400.00	37200.00	59500.00
35 deg F COP	2.93	2.97	3.00	3.15	3.24	3.06	3.57	3.57
CAPACITY	45800.00	48600.00	48400.00	48400.00	48400.00	49200.00	30000.00	48400.00
17 deg F COP	2.70	2.72	2.74	2.88	2.96	2.72	2.74	3.26
CAPACITY	36500.00	36600.00	36400.00	36400.00	36400.00	36200.00	20100.00	36400.00

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

This study has considered the efficiency improvements that are possible in residential sized heat pumps. The major conclusions and recommendations are discussed below:

Conclusions

The major implications from the survey of the residential heat pump market centered on (1) the geographic bias in the installation of heat pumps and (2) the lack of definitive relationship between the cooling and heating efficiency of heat pumps. Because most heat pumps are installed in the Southern U.S., DOE should consider doing regional forecasts of heat pumps to determine the energy savings resulting from imposition of a standard. The lack of definitive relationships between the heating and cooling efficiencies of heat pumps could make it difficult to forecast the energy savings of a standard using the current residential modeling system. This issue needs to be resolved before the energy savings forecast are done.

Heat pumps should be divided into a minimum of nine classes. Three of these classes currently do not have a DOE test procedure. Two of the classes (heating only heat pumps) should only be evaluated if there are any being sold in the U.S.

The ORNL heat pump model along with the seasonal performance model provided adequate system simulation capabilities for this analysis. One important deficiency of the ORNL model was its lack of a charge inventory model. The version of the ORNL model with charge inventory being developed at ORNL should be used in any future analysis when it becomes available.

While major improvements in the efficiency of heat pumps have occured over the past ten years, this study has demonstrated that large efficiency improvements are still possible by optimizing performance with conventional design options and application of advanced technology options. While estimates were made of the maximum possible technological efficiencies with advanced options, these estimates were not based on performance simulations of real hardware. Instead, they were based on our best judgement from reviewing the current literature and discussion with industry experts. Once data are available on some of the options, performance simulations should be done to refine the estimates.

Recommendations

First, the ORNL heat pump model should be modified to easily handle variable speed compressors. Currently, it is set up for only single speed compressors. The modifications would primarily consist of an external program that would make multiple runs of the ORNL model to obtain performance of the unit as a function of temperature and compressor speed. This modification will be necessary to better quantify the effect of variable speed compressors on system performance when variable speed compressor data become available.

Second, the ORNL heat pump model should be modified to handle water source heat pumps. Two of the classes mentioned in Chapter Three are water source heat pumps. Once a test procedure is developed for this class, a regulatory analysis for these classes will need to be performed. With modifications to the outdoor heat exchanger model, the ORNL heat pump model could be used to evaluate the effect of design changes on water source heat pumps. The primary modification would be the addition of several subroutines that could simulate the performance of water-to-refrigerant heat exchangers. This work should be started immediately to allow enough time for the development of the necessary data, implementation of the subroutines, and validation of the modified ORNL model.

Third, cost data must be developed for the components and systems used in this report. This study was without the aid of cost data. These data are needed to evaluate the cost-effectiveness of setting standards at a particular efficiency level. These data can also be used to find the most cost-effective path to the highest efficiency systems.

APPENDIX A

ORNL MODEL OUTPUT

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

COMPRESSOR OPERATING CONDITIONS:

		EFFICIENCY	
COMPRESSOR POWER	3.145 KW	VOLUMETRIC	0.7494
MOTOR SPEED	3500.000 RPM	OVERALL	0.6184

REFRIGERANT MASS FLOW RATE	523.578 LBM/H	POWER PER UNIT MASS FLOW	19.84734 BTU/LBM
COMPRESSOR SHELL HEAT LOSS	3757.163 BTU/H	POWER CORRECTION FACTOR	1.0116
		MASS FLOW RATE CORRECTION FACTOR	0.9792

SYSTEM SUMMARY	REFRIGERANT TEMPERATURE	SATURATION TEMPERATURE	REFRIGERANT ENTHALPY	REFRIGERANT QUALITY	REFRIGERANT PRESSURE	AIR TEMPERATURE
COMPRESSOR SUCTION LINE INLET	74.270 F	46.389 F	113.683 BTU/LBM	1.0000	92.895 PSIA	
SHELL INLET	76.959	44.947	114.256	1.0000	90.638	
SHELL OUTLET	184.080	117.732	127.583	1.0000	266.713	
CONDENSER INLET	166.101 F	117.663 F	123.763 BTU/LBM	1.0000	266.470 PSIA	95.000 F
OUTLET	101.783	111.823	39.828	0.0000	246.925	112.607
EXPANSION DEVICE	100.566 F	110.956 F	39.446 BTU/LBM	0.0000	244.116 PSIA	
EVAPORATOR INLET	48.627 F	48.627 F	39.446 BTU/LBM	0.1832	96.480 PSIA	80.000 F
OUTLET	74.270	46.389	113.683	1.0000	92.895	58.835

PERFORMANCE OF EACH CIRCUIT IN THE CONDENSER

INLET AIR TEMPERATURE	95.000 F
OUTLET AIR TEMPERATURE	112.607 F
HEAT LOSS FROM COMPRESSOR	3757.2 BTU/H
HEAT LOSS FROM FAN	817.2 BTU/H
AIR TEMPERATURE CROSSING COIL	110.947 F

TOTAL HEAT EXCHANGER EFFECTIVENESS 0.7804

	SUPERHEATED REGION	TWO-PHASE REGION	SUBCOOLED REGION
NTU	1.3988	1.6937	1.3874
HEAT EXCHANGER EFFECTIVENESS	0.5412	0.8162	0.5985
CR/CA	1.6156		0.7441
FRACTION OF HEAT EXCHANGER	0.0264	0.8920	0.0815
HEAT TRANSFER RATE	1375.0 BTU/H	19752.8 BTU/H	845.3 BTU/H
OUTLET AIR TEMPERATURE	133.510 F	111.071 F	102.523 F

AIR SIDE:		REFRIGERANT SIDE:	
MASS FLOW RATE	5580.4 LBM/H	MASS FLOW RATE	261.8 LBM/H
PRESSURE DROP	0.0784 IN H2O	PRESSURE DROP	19.546 PSI
HEAT TRANSFER COEFFICIENT	10.343 BTU/H-SQ FT-F	HEAT TRANSFER COEFFICIENT	
		VAPOR REGION	153.113 BTU/H-SQ FT-F
		TWO PHASE REGION	693.815 BTU/H-SQ FT-F
		SUBCOOLED REGION	162.284 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	139.107	REFRIGERANT SIDE	21289.062	REFRIGERANT SIDE	455.222
AIR SIDE	153.249	AIR SIDE	5175.750	AIR SIDE	473.161
COMBINED	72.918	COMBINED	4163.523	COMBINED	232.009

FLOW CONTROL DEVICE - CONDENSER SUBCOOLING IS SPECIFIED AS 10.000 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	0.0761 IN
STATIC SUPERHEAT RATING	6.000 F	CAPILLARY TUBE FLOW FACTOR	4.929		
PERMANENT BLEED FACTOR	1.150				

TXV CAPACITY RATING 1.848 TONS
INCLUDING NOZZLE AND TUBES

PERFORMANCE OF EACH CIRCUIT IN THE EVAPORATOR

INLET AIR TEMPERATURE	80.000 F
OUTLET AIR TEMPERATURE	58.835 F
HEAT LOSS FROM FAN	882.2 BTU/H
AIR TEMPERATURE CROSSING COIL	58.094 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	80.000 F	80.000 F	61.281 F	53.800 F	51.329 F
HUMIDITY RATIO	0.01203	0.01203	0.01156	0.00873	0.00802
ENTHALPY	32.428 BTU/LBM	32.428 BTU/LBM	27.314 BTU/LBM	22.405 BTU/LBM	21.031 BTU/LBM

RATE OF MOISTURE REMOVAL	4.0153 LBM/H
FRACTION OF EVAPORATOR THAT IS WET	1.0000
LATENT HEAT TRANSFER RATE IN TWO-PHASE REGION	4259. BTU/H
SENSIBLE HEAT TRANSFER RATE IN TWO-PHASE REGION	7828. BTU/H
SENSIBLE TO TOTAL HEAT TRANSFER RATIO FOR TWO-PHASE REGION	0.6476

OVERALL SENSIBLE TO TOTAL HEAT TRANSFER RATIO 0.6712

OVERALL CONDITIONS ACROSS COIL

	ENTERING AIR	EXITING AIR
DRY BULB TEMPERATURE	80.000 F	58.094 F
WET BULB TEMPERATURE	67.979 F	56.814 F
RELATIVE HUMIDITY	0.550	0.928
HUMIDITY RATIO	0.01203	0.00954

TOTAL HEAT EXCHANGER EFFECTIVENESS (SENSIBLE) 0.8692

	SUPERHEATED REGION	TWO-PHASE REGION
NTU	2.3061	2.0724
HEAT EXCHANGER EFFECTIVENESS	0.8270	0.8741
CR/CA	0.3222	
FRACTION OF HEAT EXCHANGER	0.2471	0.7529
HEAT TRANSFER RATE	865.0 BTU/H	12087.6 BTU/H
AIR MASS FLOW RATE	399.01 LBM/H	1215.75 LBM/H
OUTLET AIR TEMPERATURE	71.179 F	53.800 F

AIR SIDE		REFRIGERANT SIDE	
MASS FLOW RATE	1614.8 LBM/H	MASS FLOW RATE	174.5 LBM/H
PRESSURE DROP	0.401 IN H2O	PRESSURE DROP	3.589 PSI
HEAT TRANSFER COEFFICIENT		HEAT TRANSFER COEFFICIENT	
DRY COIL	7.242 BTU/H-SQ FT-F	VAPOR REGION	52.589 BTU/H-SQ FT-F
WET COIL	7.624 BTU/H-SQ FT-F	TWO PHASE REGION	475.833 BTU/H-SQ FT-F

DRY FIN EFFICIENCY	0.794
WET FIN EFFICIENCY	0.692

UA VALUES:	VAPOR REGION	TWO PHASE REGION
REFRIGERANT SIDE	303.024	8354.078 BTU/H-F

AIR SIDE		
DRY COIL	783.985	0.000 BTU/H-F
WET COIL		2191.716 BTU/H-F
COMBINED		
DRY COIL	218.550	0.000 BTU/H-F
WET COIL		1736.216 BTU/H-F

SUMMARY OF ENERGY INPUT AND OUTPUT:

LENNOX HP14-261/411 HI-SPD 95 TEST SUBCOOL 10 2B034AP1== 3.81 DISP

AIR TEMPERATURE INTO EVAPORATOR	80.00	F
HEAT FROM CONDENSER TO AIR	43946.	BTU/H
HEAT TO EVAPORATOR FROM AIR	38858.	BTU/H
POWER TO INDOOR FAN	882.	BTU/H
POWER TO OUTDOOR FAN	817.	BTU/H
POWER TO COMPRESSOR MOTOR	10735.	BTU/H
COMPRESSOR SHELL HEAT LOSS	3757.	BTU/H
TOTAL HEAT TO/FROM INDOOR AIR	37976.	BTU/H

COP (COOLING)	3.054
COOLING CAPACITY	37975.539 BTU/H

APPENDIX B

SEASONAL PERFORMANCE MODEL DESCRIPTION

The seasonal performance model(SPM) estimates the heating seasonal performance factor(HSPF) and seasonal energy efficiency ration(SEER) for single and two speed air source heat pumps. The following is an outline and description of the SPM.

The SPM calculates the HSPF for the minimum and maximum design heating requirements specified in the DOE test procedure[1]. The equations shown below are only for the minimum heating requirements. The maximum heating requirements proceed the same except that the corresponding variables end with an x. The procedure for units with single speed compressors is addressed first followed by that for units with two speed compressors.

The required user input for the SPM is shown Table B.1 for single speed heat pumps. Similar data are required for two speed units for both the high and low speed. In addition, the low speed capacity and COP for 62 F are required for the two speed model.

Table B.1 - Required input data for the SPM

Input Data	Source
Steady State Capacities and COPs(EERs)	ORNL Heat Pump Model
95 F	
82 F	
47 F	
35 F	
17 F	
Compressor on/off temperatures	Manufacturer, user specified
Demand defrost on unit	Manufacturer, user specified
Degradation Coefficient	Manufacturer, user specified

SINGLE SPEED COMPRESSOR UNITS

HSPF

The minimum heating seasonal performance factor, HSPFN was calculated as:

$$HSPFN = \frac{\sum FH(Tj)*BLN(Tj)}{[\sum FH(Tj)*(XN(Tj)/PLFN(X))*DEL(Tj)*E(Tj) + \sum RHN(Tj)]} \qquad (B.1)$$

where,

j = 1,2,3...,n corresponds to the jth temperature bin,
Tj = the representative temperature in the jth bin(F),
$\sum$ = indicates a summation over all temperature bins,
RHN(Tj) = supplementary resistance heat at Tj required in those cases where the heat pump automatically turns off(Tj<TOFF) or when it is needed to meet the balance of the heating requirements(watts),
FH(Tj) = fractional hours in the jth temperature bin,
3.413 = conversion factor to convert watt hours to BTUs,
BLN(Tj) = building load at temperature t (BTU/hr),
DEL(Tj) = heat pump low temperature cut out factor,
XN(Tj) = heat pump heating load factor, and
PLFN(X) = heat pump part load factor,

Data for temperature bins, fractional hours, and design temperatures used are shown in Table B.2 and B.3 for each climatic region.

Table B.2 - Design temperatures for each bin [1].

Region	I	II	III	IV	V	VI
TOD	37	27	17	5	-10	30

Table B.3 - Temperature bin and fractional hour data for the six DOE climate regions[1].

		Fractional hours by region					
Bin#	Tj	I	II	III	IV	V	VI
1	62	.291	.215	.153	.132	.106	.113
2	57	.239	.189	.142	.111	.092	.206
3	52	.194	.163	.138	.103	.086	.215
4	47	.129	.143	.137	.093	.076	.204
5	42	.081	.112	.135	.100	.078	.141
6	37	.041	.088	.118	.109	.087	.076
7	32	.019	.056	.092	.126	.102	.034
8	27	.005	.024	.047	.087	.094	.008
9	11	.001	.008	.021	.055	.074	.003
10	17	0	.002	.009	.036	.055	0
11	12	0	0	.005	.026	.047	0
12	7	0	0	.002	.013	.038	0
13	2	0	0	.001	.006	.029	0
14	-3	0	0	0	.002	.018	0
15	-8	0	0	0	.001	.010	0
16	-13	0	0	0	0	.005	0
17	-18	0	0	0	0	.002	0
18	-23	0	0	0	0	.001	0

The first calculations in the program are of the minimum and maximum design heating requirements, DMIN and DMAX, respectively, which the heat pump encounters when installed in a residence,. They are defined as:

$$DMIN=Q47*(65-TOD)/60 \quad (B.2)$$

for regions 1,2,3,4 and 6; and

$$DMIN=Q47 \quad (B.3)$$

for region 5. DMAX is defined as:

$$DMAX=2*Q47*(65-TOD)/60 \quad (B.4)$$

for regions 1,2,3,4 and 6; and

$$DMAX=2.2*Q47 \quad (B.5)$$

for region 5.

For Equations (B.2) through (B.5), TOD is the outdoor design temperature given in Table B.2. Q47 is the estimated capacity (in Btu/hr) of the heat pump at 47 F. The DMIN and DMAX values are rounded to the nearest 5000 if less than or equal to 40000 or to the nearest 10000 if greater than 40000.

The building load, BLN(Tj) is calculated from:

$$BLN(Tj)=(65-Tj)/(65-TOD)*C*DMIN \qquad (B.6)$$

where,

C = 0.77 is a correction factor to improve agreement between calculated and measured building loads.

The numerator of the HSPFN equation, NUMN(Tj), is then calculated:

$$NUMN(Tj)= (FH(Tj)*BLN(Tj)) \qquad (B.7)$$

The heat pump capacity, Q(Tj)is calculated with:

$$Q(Tj) = \frac{Q17 + (Q47-Q17)*(Tj-17)}{30} \qquad (B.8)$$

for Tj≥45 F or Tj≤17 F, and

$$Q(Tj) = \frac{Q17 + (Q35*0.9*ANS-Q17)*(Tj-17)}{18} \qquad (B.9)$$

for 17 F<Tj<45 F. ANS is either 1.0 for timed defrost or 1.07 for demand defrost systems.

The heat pump power, E(Tj), is calculated with:

$$E(Tj) = \frac{E17 + (E47-E17)*(Tj-17)}{30} \qquad (B.10)$$

for Tj≥45 F or Tj≤17 F, and,

$$E(Tj) = \frac{E17 + (E35-E17)*(Tj-17)}{18} \qquad (B.11)$$

for 17 F< Tj <45 F.

The E17, E47 and E35 values are calculated knowing the relationship between COP, capacity and power as

$$E17=Q17/(3.413*C17) \tag{B.12}$$

$$E47=Q47/(3.413*C47) \tag{B.13}$$

$$E35=.99*Q35/(3.413*C35) \tag{B.14}$$

The E35 value and the Q35 value in the second Q(Tj) equation are adjusted by .99 and .9, respectively, to correct for frost accumulation at 35 F. These values were obtained by evaluating data similar to Figures B.1 and B.2 obtained using an SAI Heat Pump Seasonal Performance Model [2].

With the Q(Tj) and the E(Tj) values calculated the quantities DEL(Tj), XN(Tj), PLFN(X) and RHN(Tj) can be calculated. They are defined by the following equations:

$$DEL(Tj) = \begin{cases} 0; \ Tj \leq TOFF \text{ or } \dfrac{Q(Tj)}{(3.413*E(Tj))} < 1 \\ .5; \ TOFF < Tj \leq TON \text{ and } \dfrac{Q(Tj)}{(3.413*E(Tj))} \geq 1 \\ 1; \ Tj > TON \text{ and } \dfrac{Q(Tj)}{(3.413*E(Tj))} \geq 1 \end{cases} \tag{B.15}$$

If TON and TOFF are not applicable then Tj>TON.

$$XN(Tj) = \begin{cases} \dfrac{BLN(Tj)}{Q(Tj)} \ ; \ Q(Tj) \geq BLN(Tj) \\ 1; \ Q(Tj) < BLN(Tj) \end{cases} \tag{B.16}$$

$$PLFN(X)=1-CD(1-XN(Tj)) \tag{B.17}$$

$$RHN(Tj)=[BLN(Tj)-Q(Tj)*XN(Tj)*DEL(Tj)]FH(Tj) \tag{B.18}$$

Having all of the required values calculated, the two components of the denominator of the HSPFN equation may be calculated. (see Eq. B.1). The fraction is then solved giving the HSPF at minimum and maximum design heating requirements.

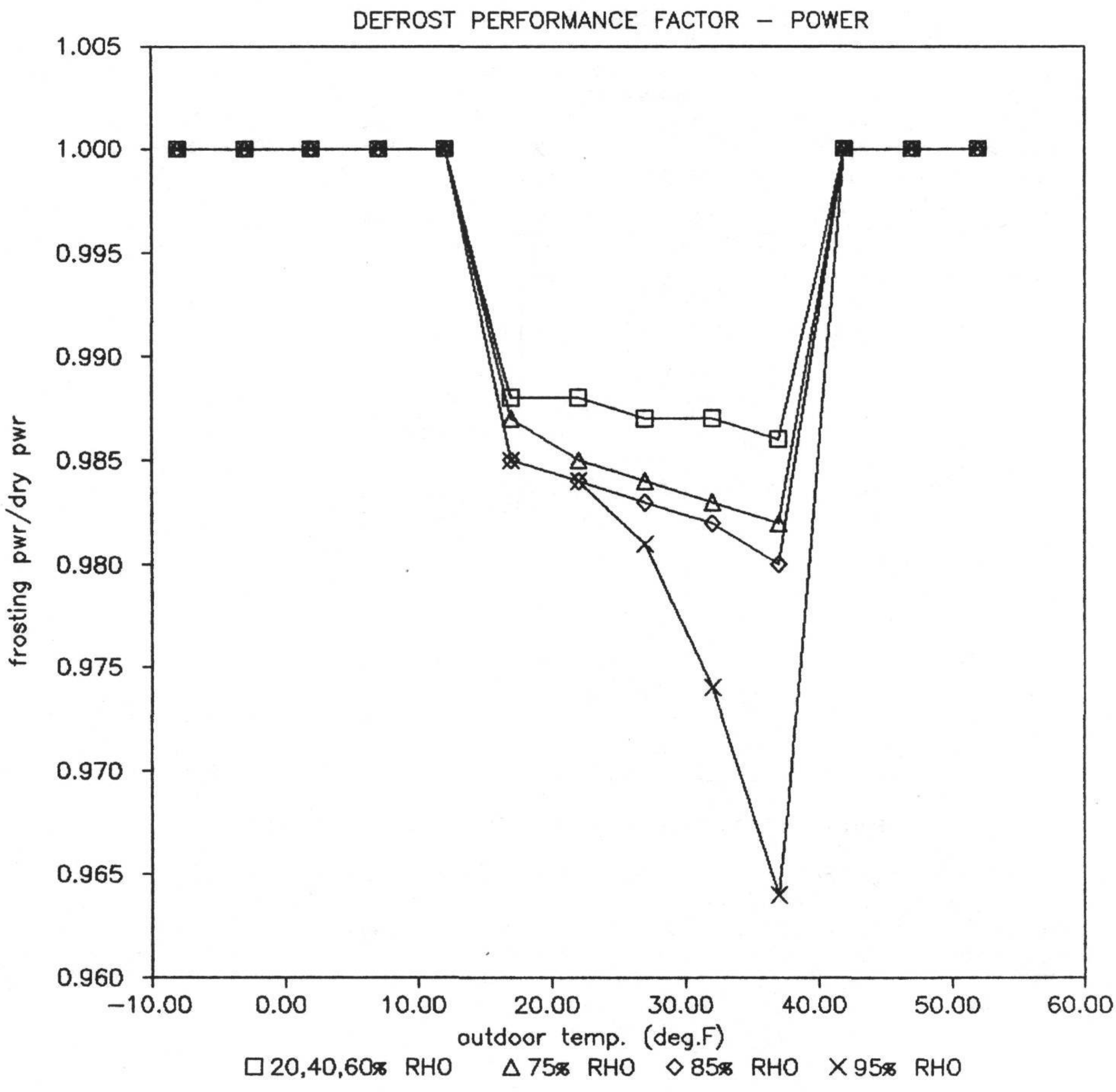

Figure B.1 - Effect of frosting on heat pump power (Source: Ref. 2)

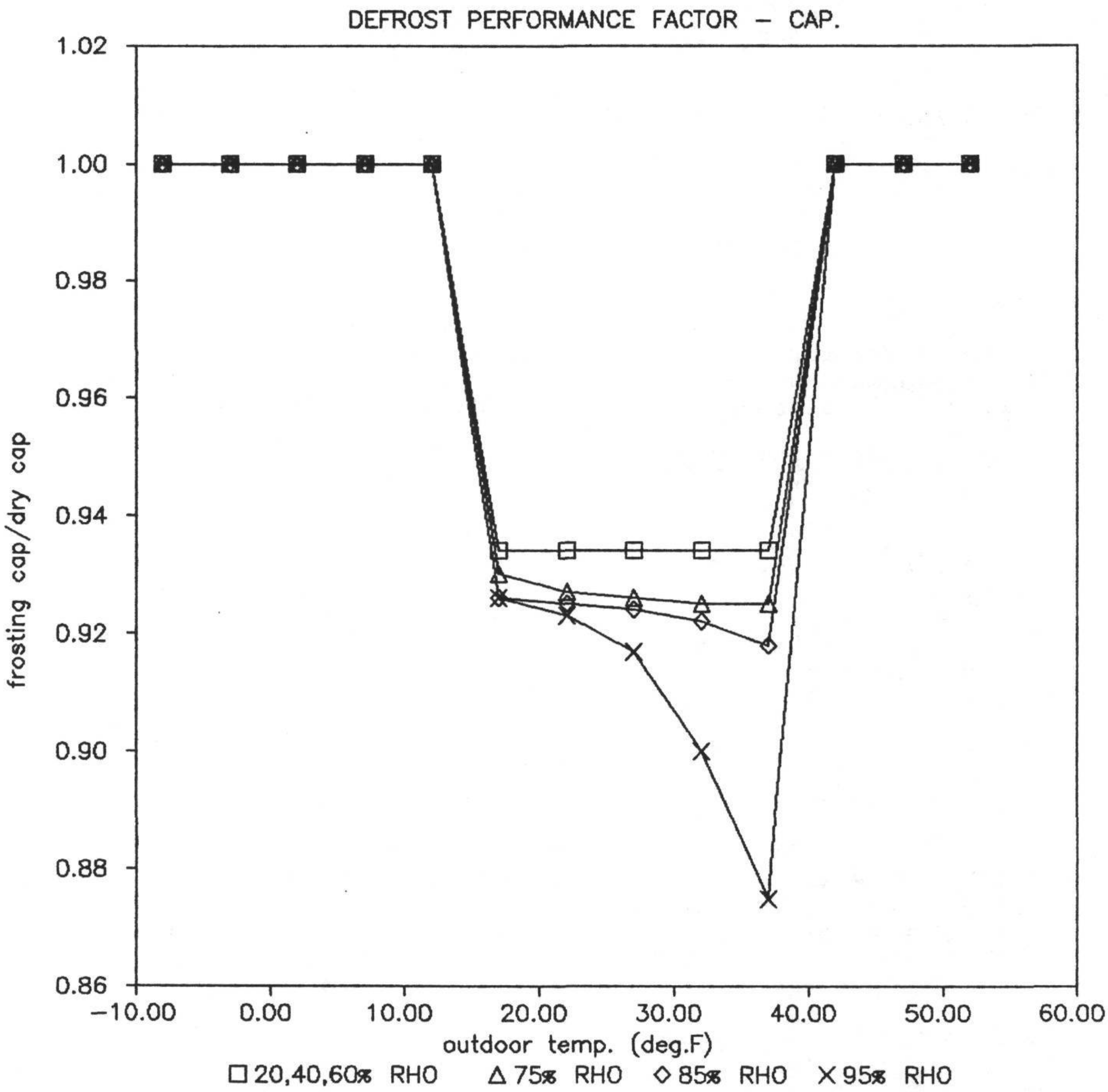

Figure B.2- Effect of frosting on heat pump capacity (Source: Ref. 2)

SEER

The SEER is calculated using the following equation

$$SEER=PLF(0.5)*EERb \tag{B.19}$$

where,

PLF(0.5) = is the part load performance factor when the cooling load factor=.5 as determined from:

$$PLF(0.5)=1-.5*CD \tag{B.20}$$

EERb = is the energy efficiency ratio determined at 82 F, and is calculated from

$$EERb=3.413*C82 \tag{B.21}$$

C82 = is the COP at 82F.

Output

The output for the units with single speed compressors is the HSPF at minimum and maximum design heating requirements for each of the six climatic regions and the SEER of the heat pump.

TWO SPEED COMPRESSOR UNITS

HSPF

The minimum heating seasonal performance factor, HSPFN, is calculated as

$$HSPFN = \frac{FH(Tj)*BLN(Tj)}{EN(Tj)+ RHN(Tj)} \tag{B.22}$$

where,

j = as defined for single speed compressor units.
Tj = as defined for single speed compressor units.
= as defined for single speed compressor units.
RHN(Tj) = defined for single speed compressor units.
FH(Tj) = as defined for single speed compressor units.
BLN(Tj) = as defined for single speed compressor units.
EN(tj) = the heat pump electrical usage in the j temperature bin divided by the total number of bin hours.

The data for each climatic region is the same as that shown in Table 1. Calculations of the minimum and the maximum heating requirements, building load and the numerator of the HSPF equation are the same as those for single speed compressor units with one change. In calculating the minimum and maximum design requirements the capacity at 47F, Q47, are replaced by the high speed capacity at 47F, Q47H. The pump capacities and powers at high and low speed operation are now calculated. The capacities and powers at high speed operation are calculated using the same equations used for the single speed compressor units. For these calculations the capacities and powers are replaced by the high speed capacities and powers for the respective temperatures. The low speed heat pump capacity, QL(Tj) is calculated as:

$$QL(Tj) = \frac{Q47L + (Q62-Q47L)*(Tj-47)}{15} \qquad (B.23)$$

for Tj≥40 F,

$$QL(Tj) = Q17L + \frac{(Q35L*.9*ANS-Q17)*(Tj-17)}{18} \qquad (B.24)$$

for 17 F≤Tj<40 F, and,

$$QL(Tj) = Q17L + \frac{(Q47L-Q17L)*(Tj-17)}{30} \qquad (B.25)$$

for Tj<17 F.

The heat pump power, EL(Tj), is calculated with:

$$EL(Tj) = E47L + \frac{(E62-E47L)*(Tj-47)}{15} \qquad (B.26)$$

for Tj≥40 F,

$$EL(Tj) = E17L + \frac{(E35L*.99-E17L)*(Tj-17)}{18} \qquad (B.27)$$

for 17 F≤Tj<40 F, and,

$$EL(Tj) = E17L + \frac{(E47L-E17L)*(Tj-17)}{30} \qquad (B.28)$$

for Tj<17 F.

The E values (E47L, etc.) are calculated using the following relationship between COP, capacity and power at the respective temperatures

Power (watts)=Capacity (BTU/hr)/(3.413*COP)

The E35L and Q35L values are adjusted by .99 and .9, respectively, to correct for frost accumulation. The correction values are the same as those used for the single speed compressor units obtained from the SAI model[2]. See figures B.1 and B.2 for the corresponding plots.

The remaining calculations are separated into three sections for three possible cases of heat pump operation to satisfy the building heating load at temperature Tj. The test procedure considers four such cases but the case when the unit alternates between low and high speed compressor operation will not be considered. The first section is when the unit operates with the compressor at low speed. In the second case the unit cycles on and off at high compressor speed. And the third case has the unit operating continuously at high compressor speed.

<u>Case 1: Low Speed</u>

The check for this case is

$$BLN(Tj) \leq QL(Tj) \qquad (B.29)$$

If this is satisfied, then,

$$XL(Tj) = BLN(Tj)/QL(Tj) \qquad (B.30)$$

$$PLFL = 1 - CDL*(1 - XL(Tj)) \qquad (B.31)$$

$$Del(Tj) = \begin{cases} 0.00; & Tj \leq TOFF \\ 0.05; & TOFF < Tj \leq TON \\ 1.00; & Tj > TON \end{cases} \qquad (B.32)$$

$$EN(Tj) = \frac{EL(Tj)*XL(Tj)*DEL(Tj)*FH(Tj)}{PLFL} \qquad (B.33)$$

$$RHN(Tj) = \frac{FH(Tj)*BLN(Tj)*(1-DEL(Tj))}{3.413} \qquad (B.34)$$

<u>Case 2: High Speed Cycle</u>

The check for this case is

$$QL(Tj) < BLN(Tj) < Q(Tj) \qquad (B.35)$$

If this is satisfied then

$$XH(Tj) = BLN(Tj)/Q(Tj) \qquad (B.36)$$

$$PLFH = 1 - CDH*(1 - XH(Tj)) \qquad (B.37)$$

The value of DEL (Tj) is determined as in Case 1.

$$EN(Tj) = \frac{E(Tj)*XH(Tj)*DEL(Tj)*FH(Tj)}{PLFH} \qquad (B.38)$$

$$RHN(Tj) = \frac{FH(Tj)*BLN(Tj)*(1-DEL(Tj)}{3.413} \quad (B.39)$$

Case 3: Continuous High Speed Operation

The check for this case is

$$BLN(Tj) \geq Q(Tj) \quad (B.40)$$

If this is satisfied then

$$XH(Tj) = 1.0 \quad (B.41)$$

DEL(Tj) is calculated using the same set of equations that is used to calculate it for units with a single speed compressor.

$$EN(Tj) = E(Tj)*DEL(Tj)*XH(Tj)*FH(Tj) \quad (B.42)$$

$$RHN(Tj) = \frac{(BLN(Tj)-O(Tj)*XH(Tj)*DEL(Tj))*FH(Tj)}{3.413} \quad (B.43)$$

where,

XL(Tj) = load factor at low compressor speed.
XH(Tj) = load factor at high compressor speed.
PLFL = part load factor at low compressor speed.
PLFH = part load factor at high compressor speed.
DEL(Tj) = low temperature cut out factor.

Having all of the required values calculated, the two components of the denominator of the HSPF equation may be calculated. The fraction is then solved giving the HSPF at minimum and maximum design heating requirements.

SEER

The seasonal energy efficiency ratio, SEER, is calculated using the following equation

$$SEER = \frac{QN(Tj)}{EN(Tj)} \quad (B.44)$$

where,

QN(Tj) = ratio of total cooling (BTU) in temperature bin j to the number of temperature bin hours.
EN(Tj) = ratio of energy usage (watt-hr) in temperature bin j to the number of temperature bin hours.

To determine the values of the denominator and numerator the following calculations must first be done.

$$QSSL(Tj) = Q95L + \frac{(Q82L-Q95L)}{(95-82)}*(33-(5*j)) \qquad (B.45)$$

$$ESSL(Tj) = E95L + \frac{(E82L-E95L)}{(95-82)}*(33-(5*j)) \qquad (B.46)$$

$$QSSH(Tj) = Q95H + \frac{(Q82H-Q95H)}{(95-82)}*(33-(5*j)) \qquad (B.47)$$

$$ESSH(Tj) = E95H + \frac{(E82H-E95H)}{(95-82)}*(33-(5*j)) \qquad (B.48)$$

$$BL(Tj) = \frac{((5*j)-3)*Q95H}{(95-65)*1.1} \qquad (B.49)$$

where,

QSSL(Tj) = the steady state capacity at Tj for low compressor speed.
QSSH(Tj) = the steady state capacity at Tj for high compressor speed.
ESSL(Tj) = the steady state power input at Tj for low compressor speed.
ESSH(Tj) = the steady state power input at Tj for high compressor speed.
BL(Tj) = building load at Tj.

The values of QN(Tj) and EN(Tj) are determined according to one of the following three cases.

Case 1

When

$$BL(Tj) \leq QSSL(Tj)$$

then,

$$XL(Tj)=BL(Tj)/QSSL(Tj) \qquad (B.50)$$

$$PLFL=1-CDL*(1-XL(Tj)) \qquad (B.51)$$

$$QN(Tj)=XL(Tj)*QSSL(Tj)*FH(Tj) \qquad (B.52)$$

$$EN(Tj)=\frac{XL(Tj)*ESSL(Tj)*FH(Tj)}{PLFL} \quad (B.53)$$

Case 2

When

$$QSSL(Tj)<BL(Tj)\leq QSSL(Tj)$$

then,

$$XL(Tj)=\frac{QSSH(Tj)-BL(Tj)}{QSSH(Tj)-QSSL(Tj)} \quad (B.54)$$

$$XH(Tj)=1-XL(Tj) \quad (B.55)$$

$$QN(Tj)=(XL(Tj)*QSSL(Tj)+XH(Tj)*QSSH(Tj))*FH(Tj) \quad (B.56)$$

$$EN(Tj)=(XL(Tj)*ESSL(Tj)+XH(Tj)*ESSH(Tj))*FH(Tj) \quad (B.57)$$

Case 3

When

$$BL(Tj)>QSSH(Tj)$$

then,

$$QN(Tj)=QSSH(Tj)*FH(Tj) \quad (B.58)$$

$$EN(Tj)=ESSH(Tj)*FH(Tj) \quad (B.59)$$

where,

XL(Tj) = load factor at low compressor speed.
XH(Tj) = load factor at high compressor speed.
PLFL = part load factor at low compressor speed.

The appropriate terms are then summed and the SEER fraction solved.

Output

The output for two speed compressor units is the HSPF at minimum and maximum design heating requirements for each of the six climatic regions. Also, the SEER values for low speed and high speed operation along with an overall SEER value will be reported.

REFERENCES

1. "Test Procedure for Central Air Conditioners, Including heat Pumps", Federal Register, December 27, 1979, pp. 76700-76723.

2. "User's Manual for Heat Pump Seasonal Performance Model (SPM) With Selected Parametric Examples," Science Appl. Inc., June 1982.

APPENDIX C

OPTIMIZATION PROCEDURE

The performance of heat pumps using conventional design options was optimized using the ORNL Heat Pump Model. The results of the optimization are a series of high efficiency heat pump designs. This appendix focuses on the three ton split systems. Similar analyses were performed for the other heat pump classes. The improvements in performance in these heat pumps are attributed to an optimization procedure, larger heat exchangers, and more efficient fans and motors. The efficiency improvements are also accompanied by significant reductions in compressor and fan motor size. No exotic components or radical design changes were used on these units.

Certain parameters such as coil size were limited by the space available within a given sized heat pump cabinet. The coils used are slightly larger than currently those found on most heat pumps. Fin spacing in the indoor coil was held at 13 fpi in order to maintain adequate latent capacity while the outdoor coil fin density was held at 19 fpi due to the manufacturing difficulties in increasing the fin spacing and reduced performance during frosting. There are however heat pumps available that have as many as 21 fpi. A major problem with higher fin densities is faster frost buildup which reduces coil performance. Fan power is also increased for coils with high fin densities due to increased air pressure drop across the coil.

The compressors used in the heat pump simulation are compressors available on the market. All compressors used are standard 230 volt single phase hermetic reciprocating compressors. The ORNL heat pump program allows the user to fit equations to the performance characteristics of a given compressor.

For the optimization below, the fan and fan motor combined efficiencies were held constant at 35 and 25 percent for the indoor and outdoor fans respectively. This combined efficiency is simply the electric motor efficiency multiplied by the fan static efficiency.

Methodology

All heat pump systems were optimized for an outdoor ambient temperature of 82 F and an indoor temperature of 80 F. This is the Air Conditioning and Refrigeration Institute (ARI) low temperature rating point for cooling mode operation. The optimum coil air flow rates and coil circuiting were determined at this temperature as well as the proper compressor displacement.

Once the unit was optimized at 82 F, it was then optimized at 47 F to determine the best coil circuiting for heating mode operation. The same compressor displacement and heat exchanger air flow rates determined for 82 F operation were used in the 47 F optimizations. Since a different optimum coil circuiting results in heating mode from cooling mode, we assumed that the heat pump has a valve within each coil to change the refrigerant circuiting for cooling and heating operation. The air flow rates through the coils were not adjusted because it is assumed that the heat pump is using single speed fans. Since higher air flow rates are required during cooling mode operation, the flow rates determined at 82 F were used for heating mode operation.

The COP in cooling mode was optimized while keeping the capacity of the unit constant. A high efficiency compressor with a capacity appropriate for the desired capacity of the heat pump was used in the optimization procedure. The capacity of the heat pump was calculated and compared to the desired capacity for the unit. If the calculated capacity was not close enough to the desired capacity, the compressor displacement was altered and the heat pump capacity was recalculated. This procedure continued until the desired capacity was obtained.

To ensure the accuracy of the compressor model, only relatively small deviations from the actual compressor displacement were allowed. If the calculated displacement was more than 25 percent from the actual displacement, an alternate compressor for the heat pump was selected. By using the correctly sized compressor for a heat pump, its performance and efficiency would be more typical for the sized unit being analyzed.

As the outdoor ambient temperature rises, the heat pump cooling capacity decreases. Typically, the cooling capacity decreases about 8 percent when going from 82 to 95 F outdoors. Therefore, the capacity of the unit at 82 F was held constant at approximately 8 percent greater than the nominal capacity at 95 F. Doing so ensures that the optimized unit had sufficient capacity at 95 F to meet its nominal rating.

Since a heat pump must operate in both heating and cooling mode, some compromises must be made so that good performance will be realized in both modes of operation. The optimum configuration can be selected by observing the results obtained while optimizing the heat pumps at 82 and 47 F outdoors. Sensitivity plots of COP as a function of the heat pump parameters in both heating and cooling mode are quite useful for selecting components for heating and cooling use. For example, if a heat pump manufacturer does not wish to install the valves in the coils to change the circuiting, he may decide to choose a fixed coil circuiting that performs well in both heating and cooling modes. This eliminates added costs and difficulties in heat exchanger manufacturing. The proper circuiting can be determined by observing the sensitivity plots of coil circuiting and choosing the circuiting that provides good performance in both modes with only a small loss in performance compared to a variable circuiting coil.

Thousands of runs of the ORNL model were made to develop the optimum selection of components. Once the optimum selection of components was made, the runs were made at all temperatures necessary to calculate the HSPF for the given heat pump.

Sensitivity Analysis

Sensitivity plots were generated showing contours of constant COP as a function of the design parameters while holding capacity constant. These plots allow visualizion of the interactions between the variables. Only two parameters were varied at a time while all other system variables were held fixed. Therefore, instead of designing a single "best" set of design parameters with a maximum COP, a family of design parameters clustered about the optimum is achieved. This gives the designer more flexibility in determining the most efficient heat pump.

Since all units listed in this appendix were designed using the same procedure, only the results for the 3 ton unit are discussed.

Sensitivity to Condenser and Evaporator Air Flow Rates

After selecting appropriate coil sizes for the indoor and outdoor coils based on reasonable heat pump cabinet dimensions, the proper air flow rates through the coils were then determined for optimum performance. The coil parameters such as fin density, number of rows, refrigerant line sizes and coil circuiting were specified, and then the condenser and evaporator air flow rates were varied while observing the effect on COP. The results of this procedure for the 3 ton unit at 82 degrees ambient are shown in Figure C.1.

For this unit, the outdoor coil was 25 square foot and had 1 row coil. The indoor coil was 5.5 square feet and had 4 tube rows. The capacity of the unit was held constant at 38500 btu/hr by varying the compressor displacements throughout the analysis. This unit is more sensitive to indoor coil cfm than outdoor coil cfm. This results from the increased fan power necessary to force the air through more tube rows. Also, the optimum indoor coil cfm is almost independent of the outdoor cfm. This may not be the case if a smaller outdoor coil was used. In this case the outdoor coil was sufficiently large that the outdoor cfm had only a small effect on performance.

The same procedure was performed in heating mode to determine the optimum air flow rates for the coils. In heating mode, slightly lower air flow rates outdoors and similar flow rates indoors compared to those at 82 F resulted. Since single speed fans are assumed to be used in these units, the air flow rates determined for optimum performance at 82 F were used for heating mode also.

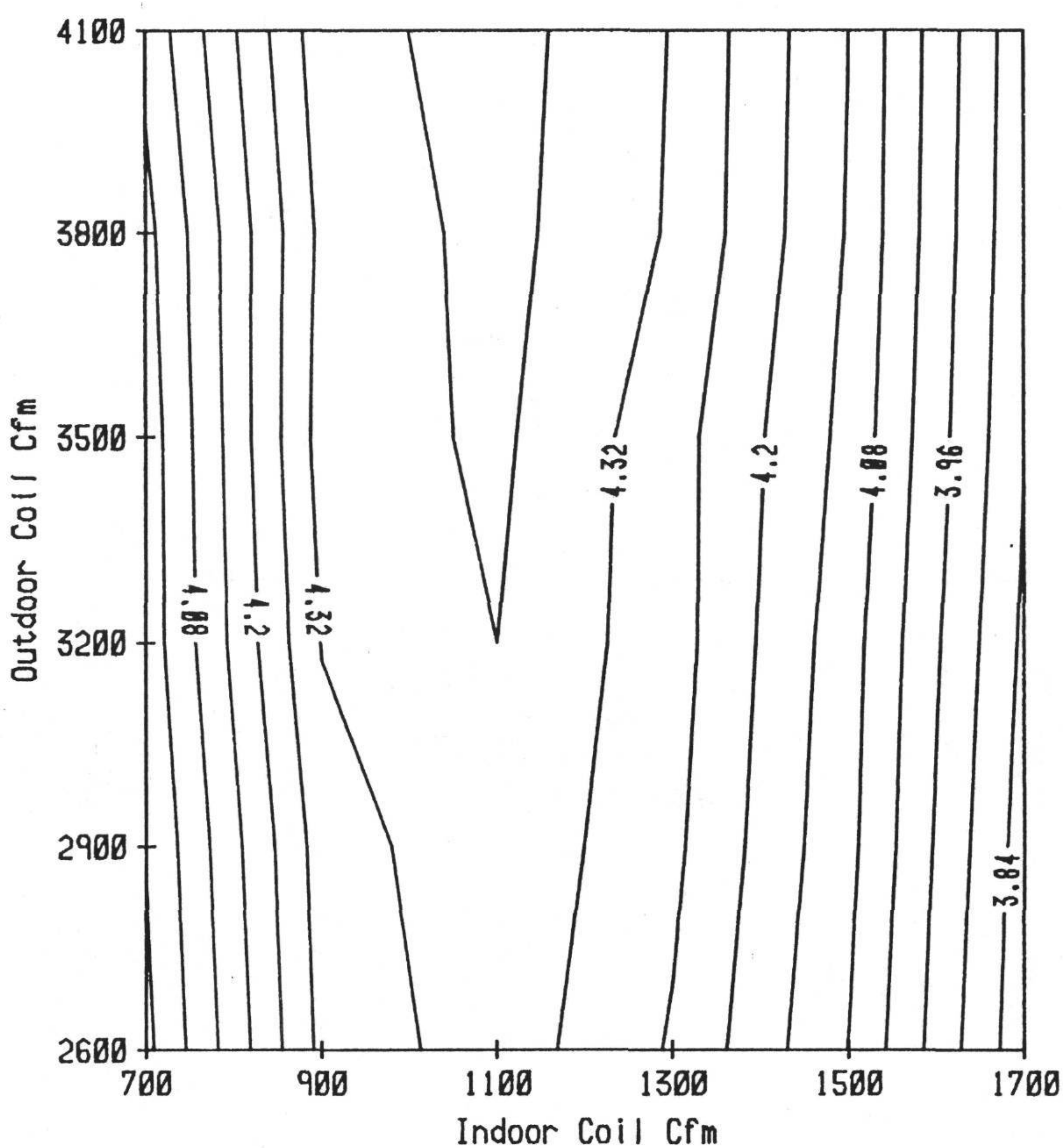

Figure C.1 - COP as a function of indoor and outdoor airflow (cfm) at 82F.

Sensitivity to Coil Circuiting

The circuiting in the evaporator and condenser coils also has a large effect on heat pump performance. Initially, a best guess at coil circuiting was made when searching for the optimum coil air flow rates. Then, the optimum circuiting was found.

The number of parallel circuits in the indoor and outdoor coils were varied while recording the effect on the COP. The air flow rate through the coils is the same as that found in the previous optimization procedure at 82 F. Figures C.2 and C.3 show the results of varying coil circuiting at 82 and 47 F respectively.

The optimimum coil circuiting is different between heating and cooling modes. Therefore, if a coil can be made to change its circuiting, it can operate closer to its optimum capability for the heating and cooling mode. If a fixed circuiting coil is used, these plots can assist in finding the number of circuits that perform well in both modes.

There is another difference in Figures C.2 and C.3. In Figure C.2, the capacity of the unit is held constant by varying the compressor displacement. In Figure C.3, a fixed compressor displacement was used and the capacity of the heat pump was allowed to reach its natural capacity. The compressor displacement was determined after optimizing the unit in cooling mode and then using the displacement that gave the maximum cooling COP in heating mode.

Sensitivity of Compressor Displacement

Because varying the design parameters changes the performance characteristics of the heat pump, a method of keeping the capacity of the unit constant was needed. The capacity was kept constant by varying the compressor displacement until the correct capacity was reached.

A relationship between the coil air flow rates and the compressor displacement exists as shown in Figure C.4. Increasing the air flow rate produces a decrease in the compressor displacement needed to produce a given capacity. However, there are tradeoffs that must be considered. Typically, as the compressor displacement decreases, the COP of the heat pump increases. However, if the displacement is decreased, the heat pump capacity also decreases. Thus, higher air flow rates are needed through the coils to keep a constant capacity. There is a point when the air flow rates continue to increase until the fan powers required outweigh the decreasing compressor power.

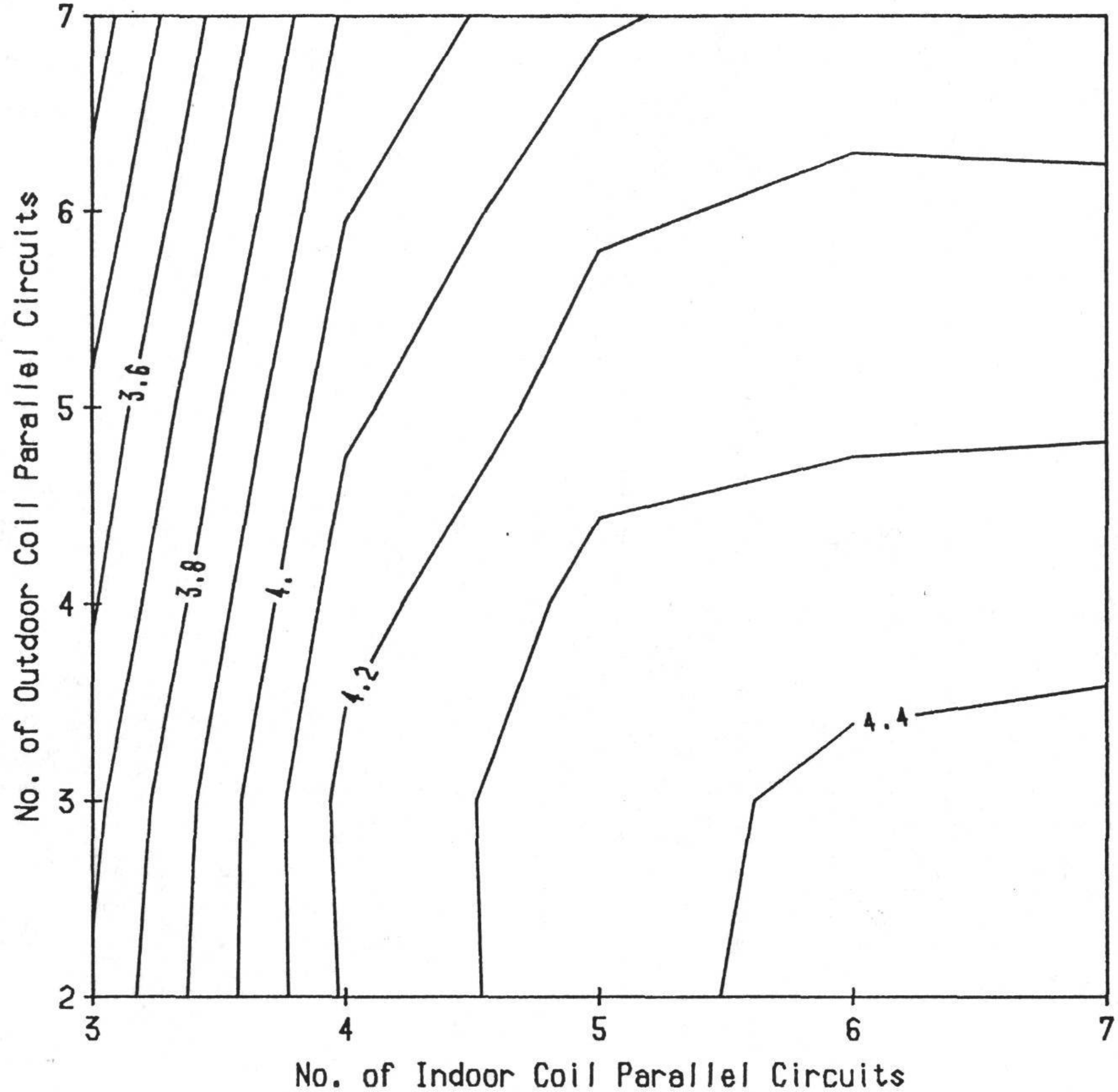

Figure C.2 - COP as a function of coil circuiting (82F).

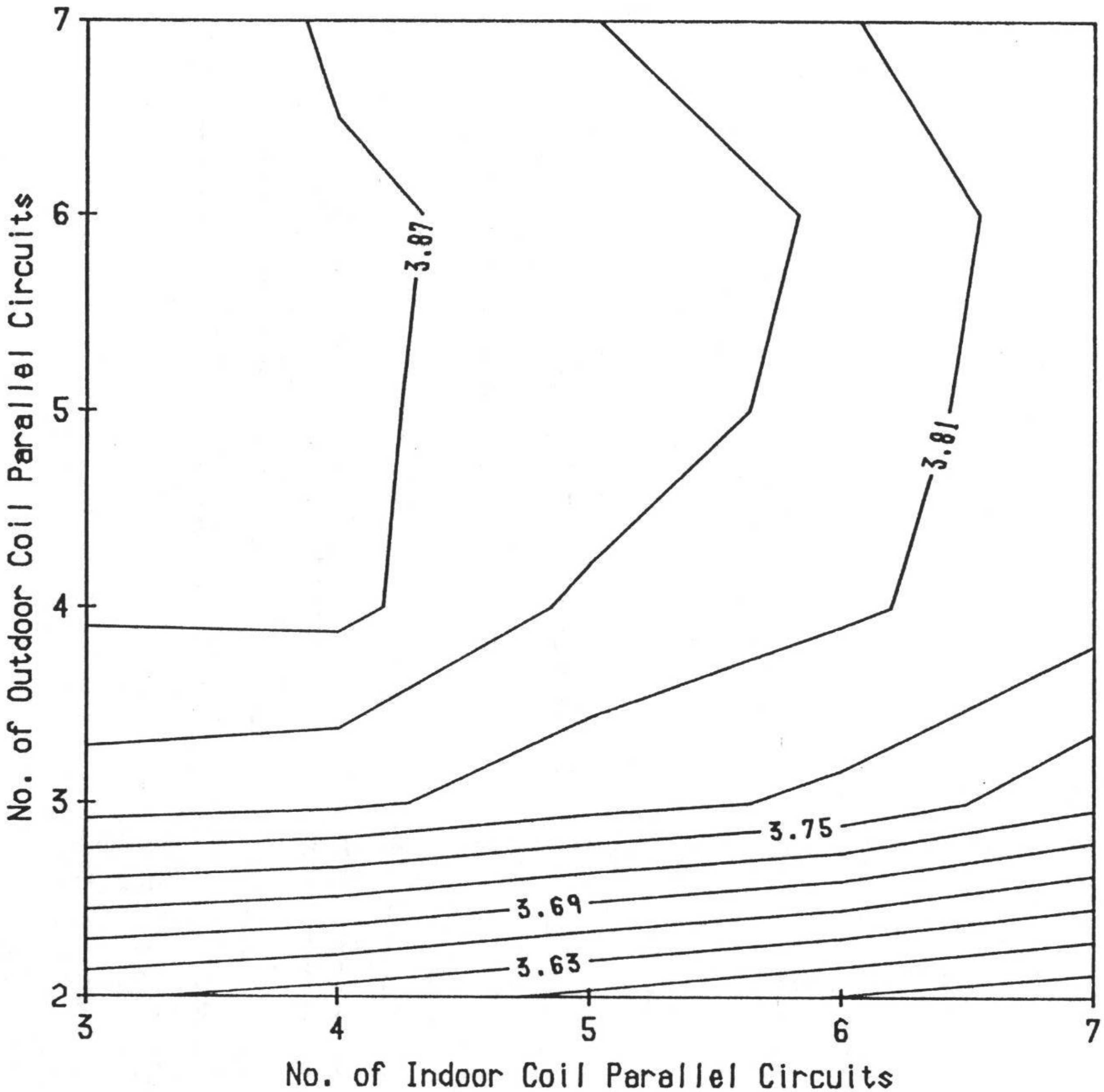

Figure C.3 - COP as a function of coil circuiting (47F).

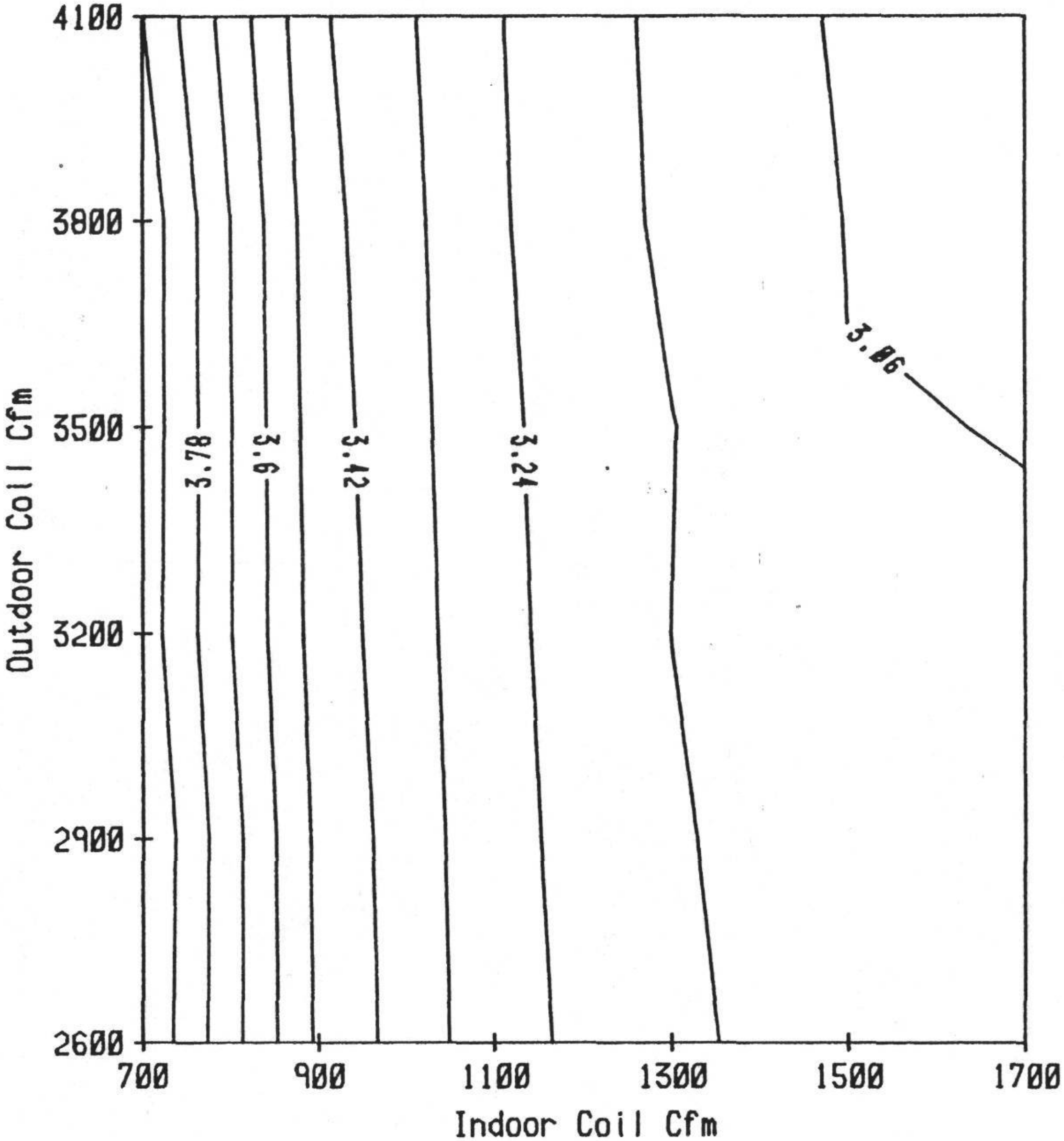

Figure C.4 - Compressor displacement (cubic inches) as a function of indoor and outdoor air flow (cfm) at 82F.

Sensistivity to Refrigerant Line Sizes

The sizes of the refrigerant lines connecting the indoor and outdoor sections also has an effect on the heat pump's performance. The best line sizes give a low pressure drop between the indoor and outdoor sections. To find the best line dimensions, both the liquid line and suction line dimensions were varied while keeping the heat pump capacity constant. This was done both for heating and cooling operation. The results of this analysis are shown in figures C.5 and C.6. The liquid line has very little effect on the heat pump's performance. Varying the suction line size did produce a noticable effect on COP at both temperatures. The improvements associated with using larger diameter lines begins to diminish with lines larger than 0.70 inches as can be seen by the wide spacing between the contour lines.

Sensitivity to Coil Area and Air Flow Rate

To find the best coil size for a given unit, a range of coil sizes were tried. Figure C.7 shows the effects of varying the coil sizes while also varying the air flow rate through the coils. The COP continues to increase as long as the coil surface are increases. However, the increase in performance begins to diminish once the coil sizes reach 28 sf as shown by the increasing distance between the contour lines.

Since the coil size is constrained by the heat pump cabinet, we decided to use only a 25 sf coil for the three ton units, even though using a larger coil would improve performanace slightly.

Also the fin spacing was varied while varying the air flow rate to find the best coil fin density. Figure C.8 shows the effects of varying these parameters for a 20 sf outdoor coil at 82 F. The fins used on this coil had a thickness of 0.0045 inches and were of the louvered type. Optimum performance occurs for a fin spacing of about 24 fpi at 4500 cfm. Since such a high fin density coil has problems with blockage due to frosting and other foreign materials such as leaves, we decided to use a more conventional spacing of 19 fpi.

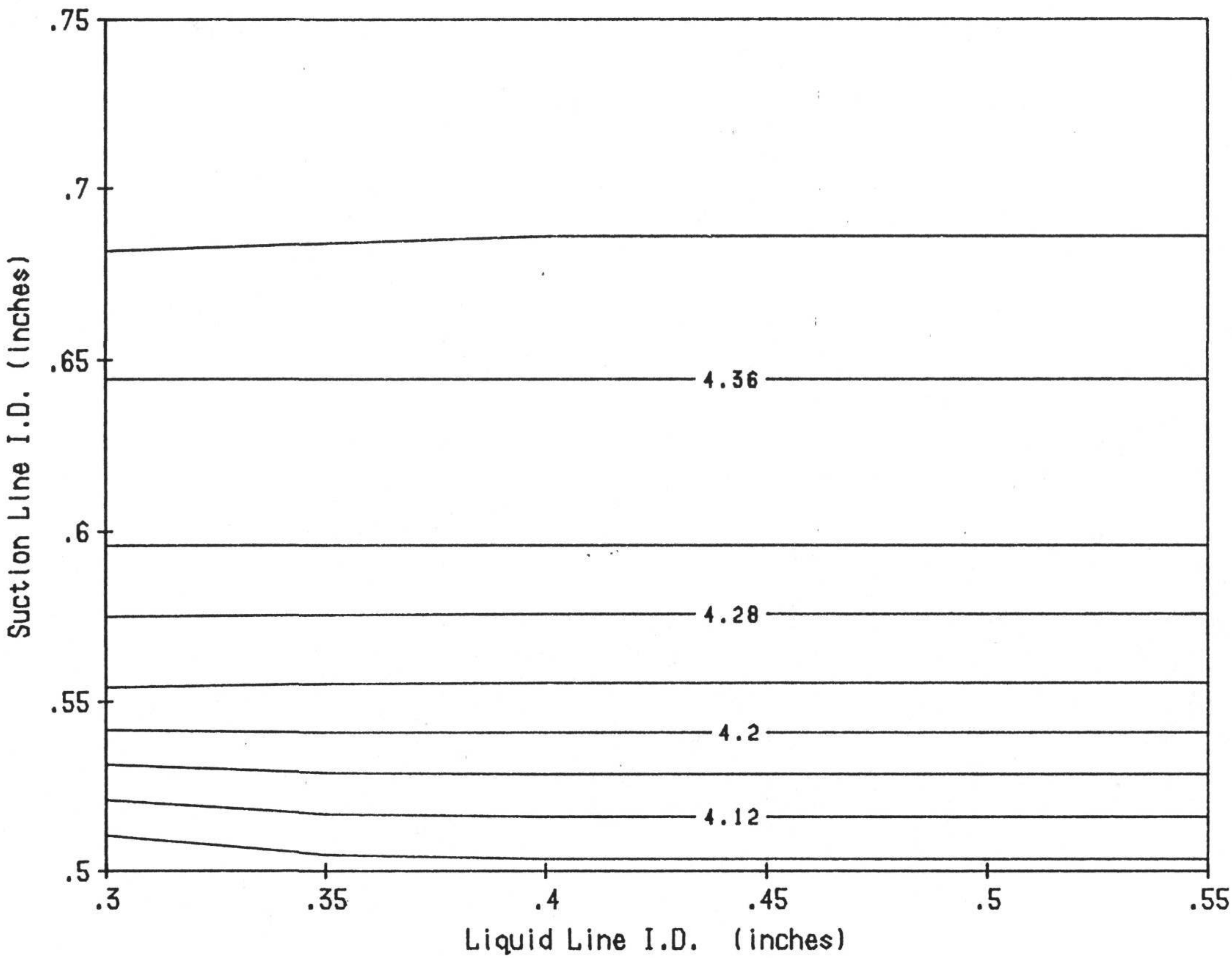

Figure C.5 - COP as a function of refrigerant line size (82F).

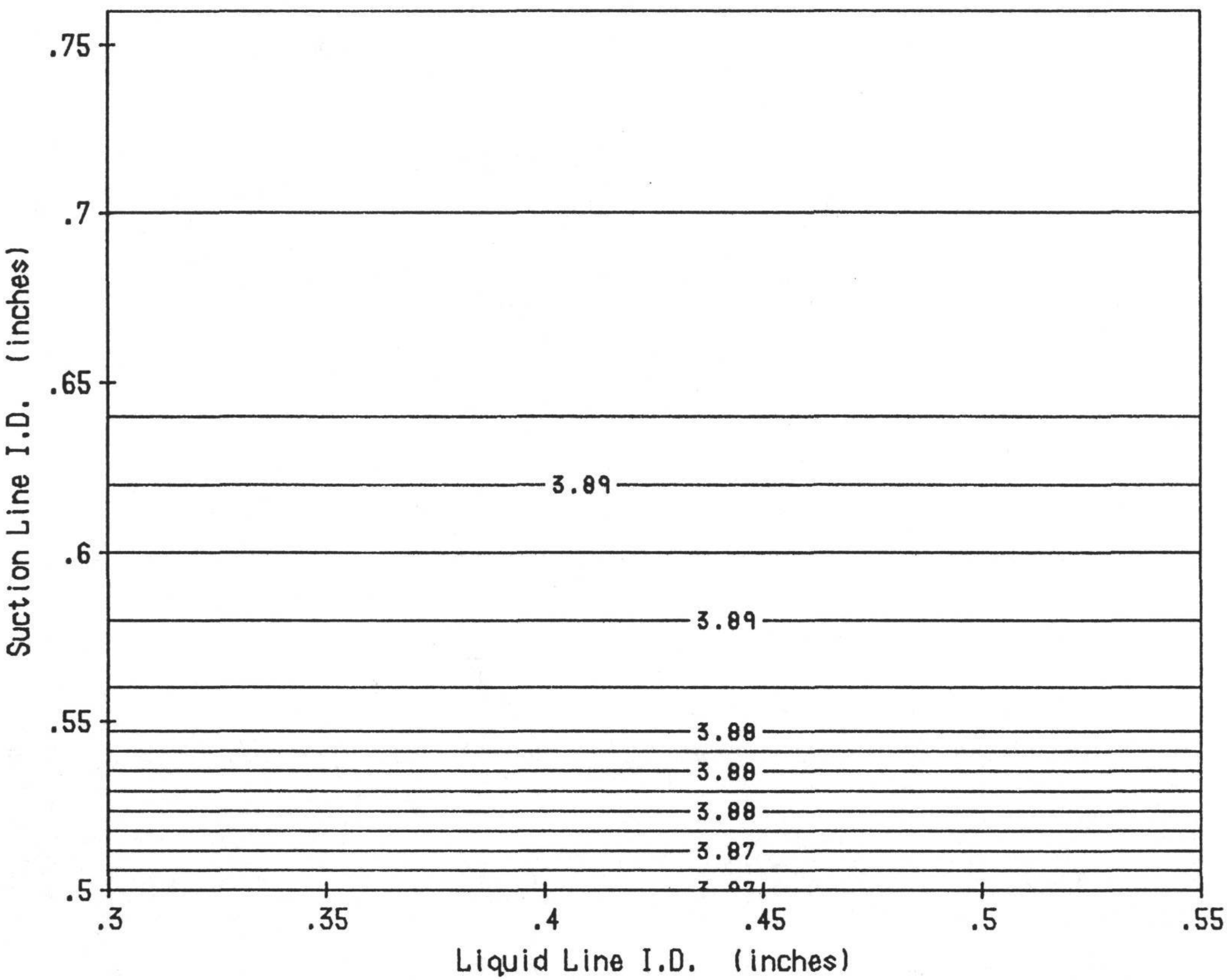

Figure C.6 - COP as a function of refrigerant line size (47F).

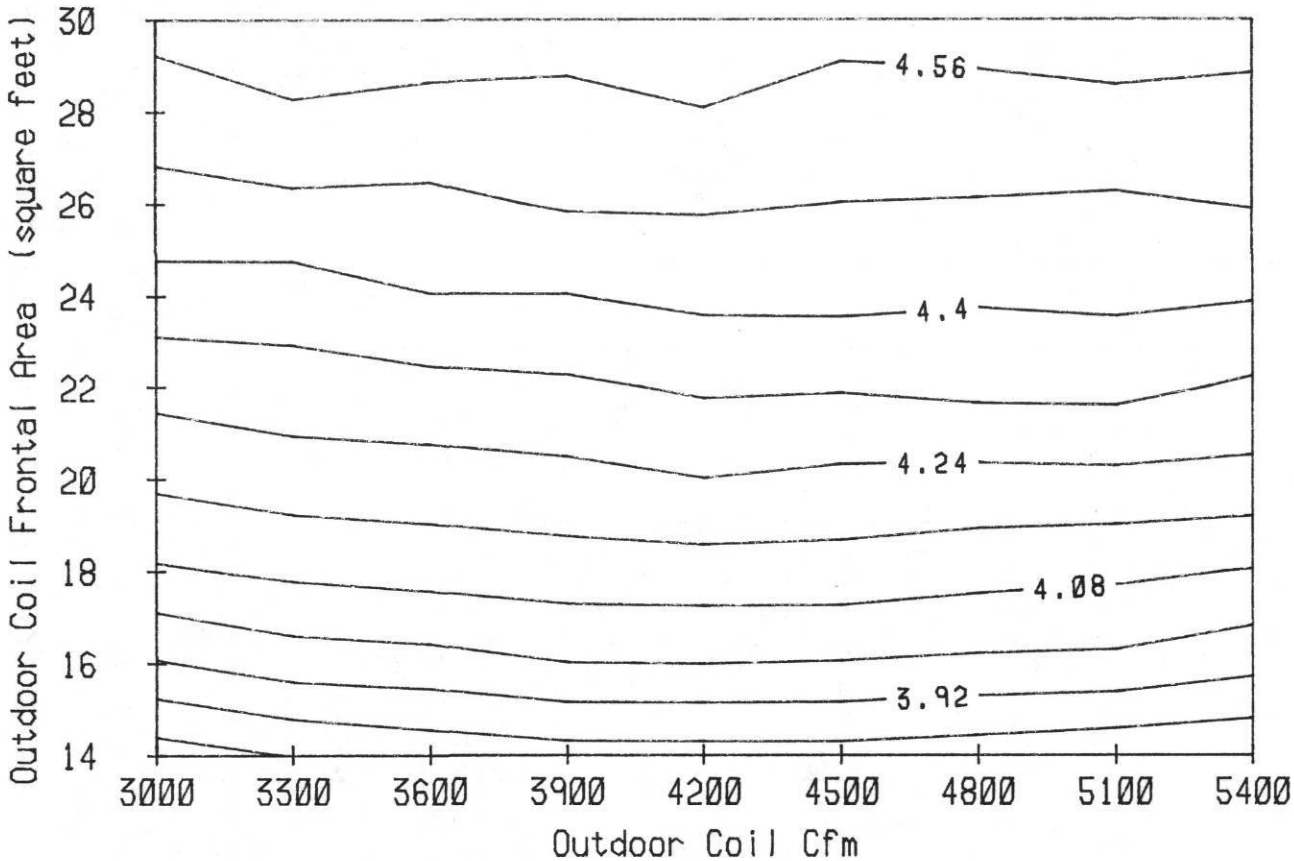

Figure C.7 - COP as a function of outdoor coil area and cfm (82F).

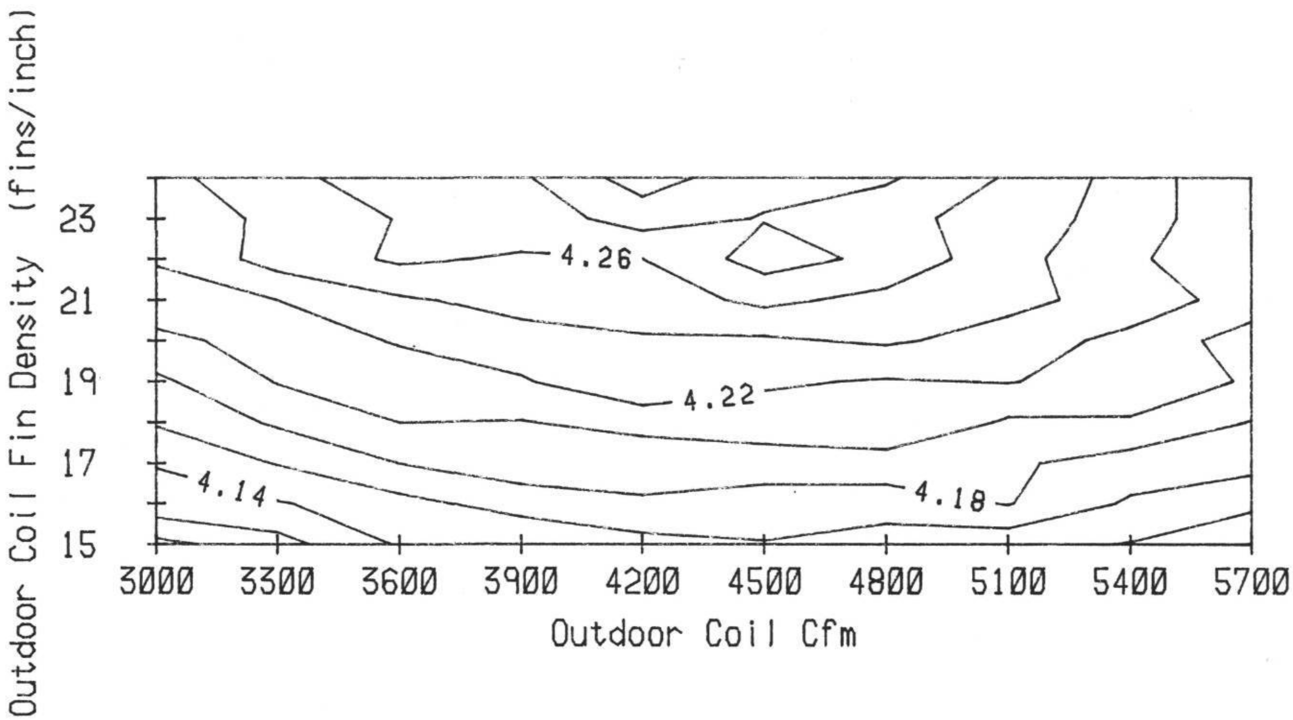

Figure C.8 - COP as a function of outdoor cfm and fin density (82F).