ESL-TR-88/10-02

An Analysis of Efficiency Improvements In Room Air Conditioners

Final Report

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ESL/88-04

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Prepared Under Contract No. 4541410
For
University of California
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GLOSSARY OF TERMS

ADL	AUTHOR D. LITTLE, INC.
ARI	AIR CONDITIONING AND REFRIGERATION INSTITUTE
BTU	BRITISH THERMAL UNIT
CFM	CUBIC FEET PER MINUTE
COP	COEFFICIENT OF PERFORMANCE
DOE	DEPARTMENT OF ENERGY
EER	ENERGY EFFICIENCY RATIO
FPI	FINS PER INCH
HR	HOUR
LBL	LAWRENCE BERKELEY LABORATORY
MGT	MULTI-GROOVED TUBES
NAECA	NATIONAL APPLIANCE ENERGY CONSERVATION ACT
NBS	NATIONAL BUREAU OF STANDARDS
NECPA	NATIONAL ENERGY CONSERVATION POLICY ACT
NTU	NUMBER OF TRANSFER UNITS
OEM	ORIGINAL EQUIPMENT MANUFACTURER
ORNL	OAK RIDGE NATIONAL LABORATORY
RAC	ROOM AIR CONDITIONER
SF	SQUARE FEET
SHF	SENSIBLE HEAT FACTOR
TDB	DRY BULB TEMPERATURE
TON	12000 BTU/HR
TXV	THERMAL EXPANSION VALUE
TWB	WET BULB TEMPERATURE

CHAPTER 1

INTRODUCTION

In 1976, the U.S. Congress passed the National Energy Conservation Policy Act (NECPA) P.L. 95-619, which requires the imposition of minimum efficiency standards on eight major household appliances. The law required that the proposed standards be both technologically feasible and economically justifiable. One of the appliances for standards consideration was the room air conditioner (RAC). In 1980 the Department of Energy first proposed minimum efficiency standards on new room air conditioners on seven other appliances[1]. In 1983, "no standard" standards were issued by DOE for all eight appliances because the energy savings of standards were not significant enough to justify minimum efficiency standards.[2] The "no standard" standards were challenged in court ion 1984. In 1985, the "no standard" standards were ruled unlawful [3]. In 1983 However, due to a court challenge of the 1982 standards, they were not allowed.

In 1987, Congress passed the National Appliance Energy Conservation Act (NAECA), P.L. 100-12 which specified minimum efficiency levels for major appliances, including RACs (Table 1.1). The minimum efficiency standards for RACs must be met by January 1, 1990. Periodically, the Department of Energy can publish amendments to the standards after analyses have been performed to determine their technical and economic feasibility. This report summarizes the results of an engineering analysis used to evaluate the technical feasibility of improving the efficiency of RACs.

The objectives of this study included: (1) evaluation and selection of a suitable RAC design model, (2) selection of design options that can be used to improve RAC performance, and (3) development of high efficiency RAC designs.

Chapter 2 provides background material on RAC shipments and classes. It also provides information on the current range of efficiencies in RACs and the recent improvements in efficiency levels.

Computer analyses of the possible design models can be useful for determining possible efficiency improvements. A steady state design model is used for this analysis. Chapter 3 discusses the selection of this steady state model. Improvements to this model were made to adequately model RACs. These improvements are discussed in Chapter 4. Chapter 5 discusses the validation of the computer model with manufacturers data.

The four major design options available for improving the efficiency of RACs are discussed in Chapter 6. The design options were implemented in a systematic way to improve the efficiency of "typical" RACs in each class. The description of the improved units are found in Chapter 7. Major conclusions and recommendations are provided in Chapter 8.

REFERENCES

1. "Notice of Proposed Rulemaking", Federal Register, Vol. 4, No. 127, June 30, 1980.

2. "Final Rule", Federal Register, Vol. 48, No. 169, August 30, 1983.

3. National Resources Defense Council, Inc.,V. John S. Herrington, Secretary, U.S. Department of Energy, No. 83-1195, U.S. Court of Appeals, July 16, 1985.

CHAPTER 2

ROOM AIR CONDITIONER MARKET

The room air conditioner market is discussed in this chapter. The discussion includes shipments, market shares, efficiencies, type of units, etc. Factors influencing the analysis of standards are also highlighted.

Historical Background and Shipments

The first patent for the room air conditioner was applied for in 1931 but was not granted until 1936 [1]. The earlier room air conditioners employed water cooled condensers and required extensive plumbing. This detracted from the appeal of the unit as an appliance. The air-cooled model was seen as a major development because it could be marketed as a household appliance and mass produced by industry. In 1936, the average unit weighed over 450 pounds and occupied 15 cubic feet [2]. The early air conditioners were all console units with the window unit appearing in the late 1930's. The room air conditioner heat pump also made a debut in the early thirties.

Growth in room air conditioner sales was slow until the end of World War II. The room air conditioner was then marketed as the appliance that no home could be without. The initial sale of units was about 3,000 in 1936. This increased to 13,000 in 1938 and up to 230,000 in 1949. [1] In the fifties and sixties the market shipments increased dramatically as the price of the units fell and household incomes rose. The shipments reached a peak of over 5 million units in the seventies. In recent years, their popularity has decreased with the prevalence of central air conditioners and heat pumps. Fluctuations in sales coincide with weather fluctuations. RAC sales depend heavily on how hot the summer is, but a steady decline is evident. Sales in 1979 were 4 million while in 1983 they were less than 2 million [3].

Exports of RACs increased through the late seventies, reaching a peak of 873,000 in 1978 (Figure 2.1) which accounted for 21% of U.S. manufactured units. Exports have declined in recent years to 64,200 in 1986.

U.S. manufacturers have lost ground steadily to foreign competition (Japan, Brazil, etc.). In 1983, only 98,045 units were imported. In 1984, the U.S. became a net importer of room air conditioners. In 1986, over 375,000 units were imported. Imports made up 14% of all units sold in 1986.

Classes

The Department of Energy originally defined six classes of room air conditioners [4]:

(1) With outdoor side louvers 8,000 Btu/hr and under
(2) With outdoor side louvers 8,001 - 14,000 Btu/hr
(3) With outdoor side louvers 14,001 - 20,000 Btu/hr
(4) With outdoor side louvers over 20,000 Btu/hr
(5) Without outdoor side louvers or with reverse cycle 8,000 Btu/hr and under
(6) Without outdoor side louvers or with reverse cycle over 8,000 Btu/hr

These classes were expanded in 1987 under the National Appliance Energy Conservation Act (NAECA) (Table 2.1) [5].

Table 2.1 - Room Air Conditioner Classes

Type	Capacity (Btu/hr)
Louvered	6,000 and less
Louvered	6,000 - 7,999
Louvered	8,000 - 13,999
Louvered	14,999 - 19,999
Louvered	greater than 20,000
Non-louvered	6,000 and less
Non-louvered	6,000 - 7,999
Non-louvered	8,000 - 13,999
Non-louvered	14,000 - 19,999
Non-louvered	greater than 20,000
Reverse Cycle Louvered Sides	
Reverse Cycle Non-louvered Sides	

To create any new classes, a RAC must satisfy the following criteria stated by the DOE [6]:

(1) have a different primary energy source i.e., fossil fuel (oil or gas), or electricity,

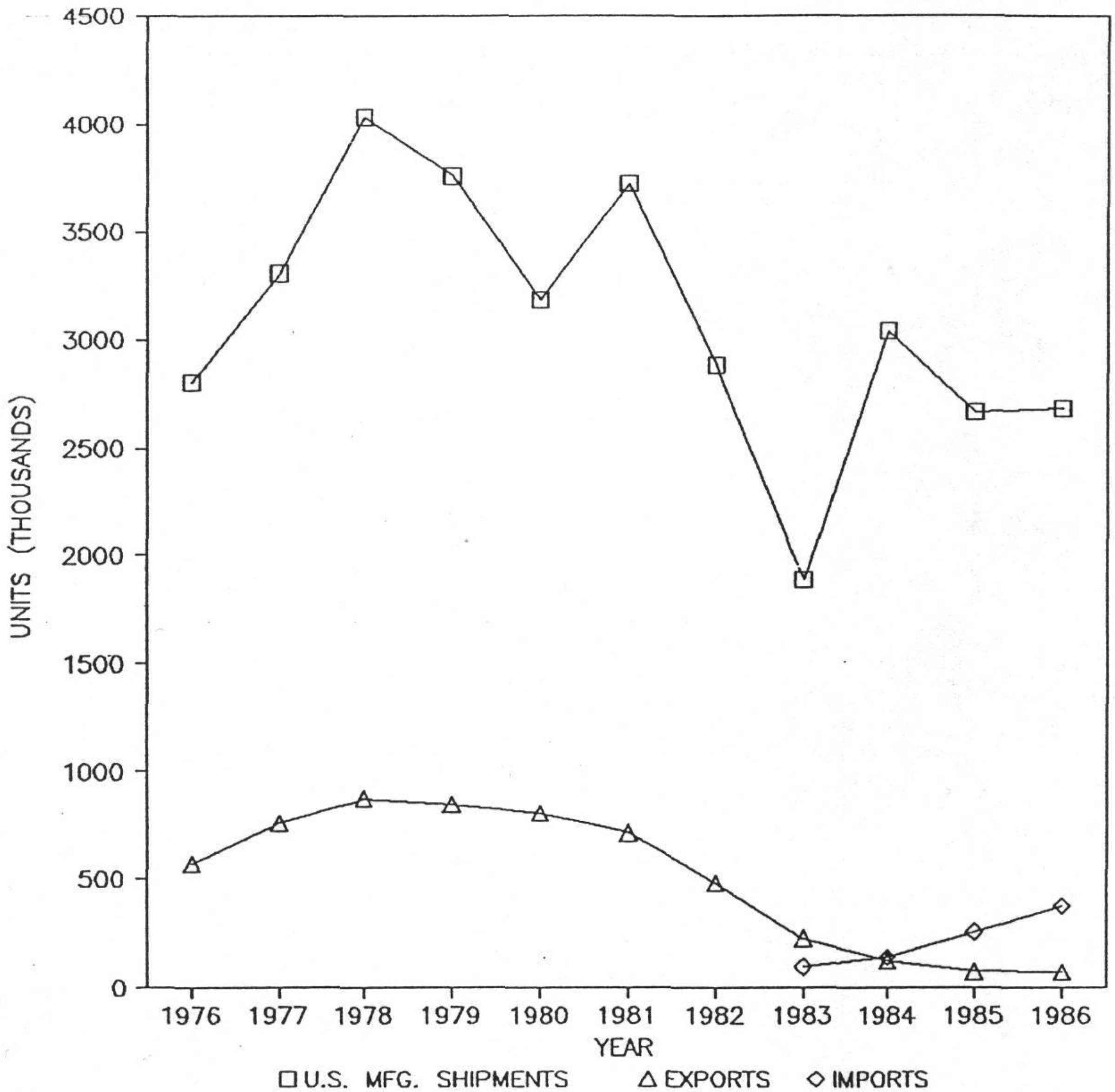


Figure 2.1 - RAC Shipments 1930 - 1985 [1,3,8,9]

(2) have a different capacity or other performance related feature which affects efficiency and utility, or

(3) ensure the availability of features providing utility which also affect efficiency of the model.

The new classes cover many of the concerns manufacturers had with the old classes. For instance, the previous break at 8,000 Btu/hr ignored the utility of the portable units whose capacity usually does not exceed 5,000 Btu/hr [7]. The new break at 6,000 Btu/hr and less recognizes the utility of portable units.

A test procedure covering the heating capability of the reverse cycle classes is under development. For this reason the reverse cycle classes are not covered in this analysis. If standard test conditions for heating capacity and electric input are published in the future, the reverse cycle classes should be analyzed for efficiency improvements.

The efficiency of the units being sold has seen a dramatic increase the last ten years. (Figures 2.2 and 2.3). The bulk of the units shipped in 1974 had an EER under 6.5 while in 1985 the majority of the units had an EER over 8.5. The number of units manufactured that had an EER over 7.5 have seen an increase of over 250% from the 1974 levels. In Figure 2.4, the rise in shipment weighted efficiency of the average unit can be seen. In 1972 the average EER for a RAC was 5.98. This has steadily climbed to the 1985 level of 7.70, an increase of 28.8 percent [7,8,9].

The lowest and highest efficiency systems on the market in 1987 are listed in Table 2.2 [10]. The strategy of the Engineering Analysis is to start with a low efficiency unit. The efficiency should then be incrementally improved until it is better than any unit on the market. Thus, the starting point of the Engineering Analysis will be near the lowest EER units shown in Table 2.2 for each class.

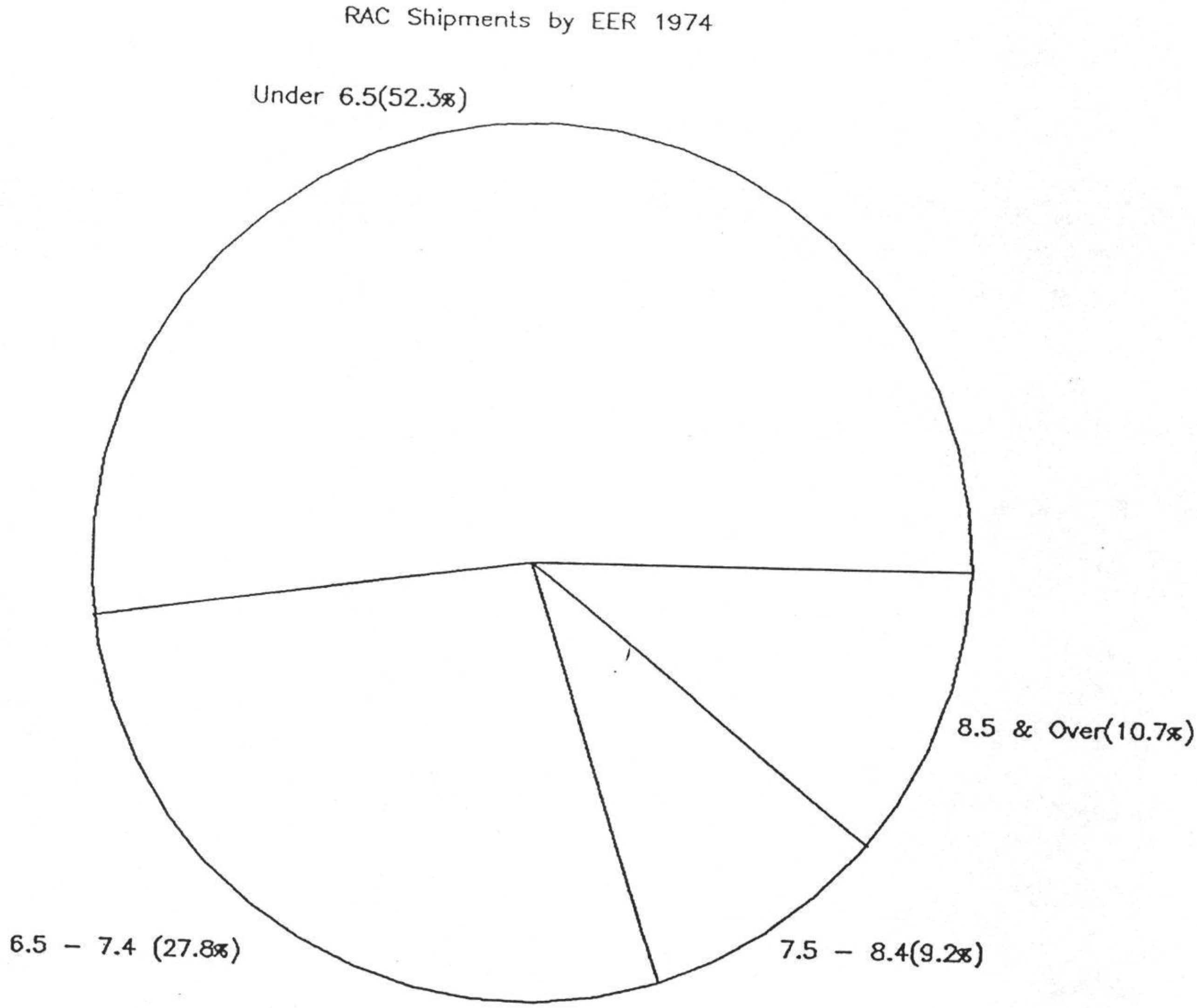


Figure 2.2 - EER of Room Air Conditioner Shipments in 1974 [8]

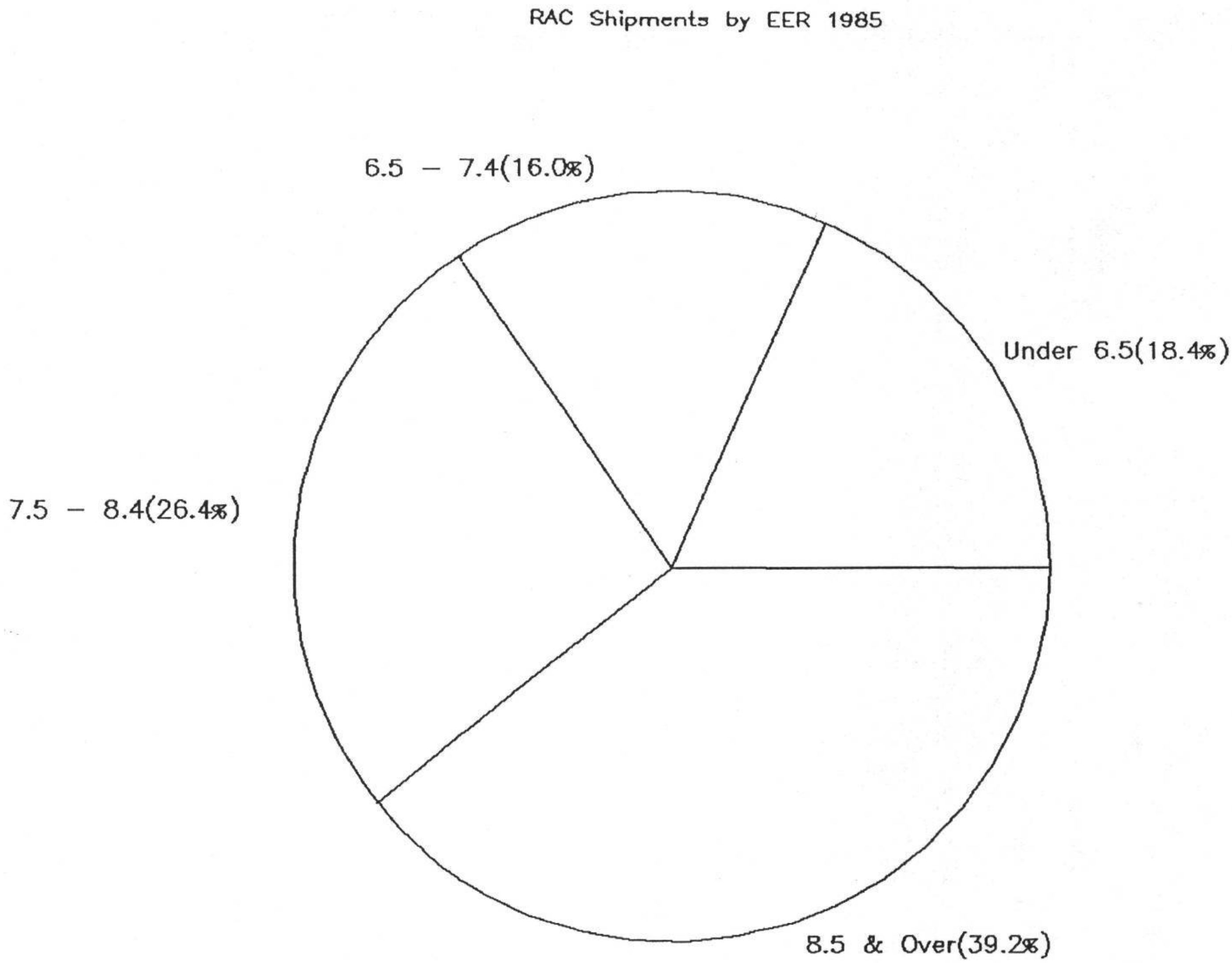


Figure 2.3 - EER of Room Air conditioner Shipments in 1985 [8]

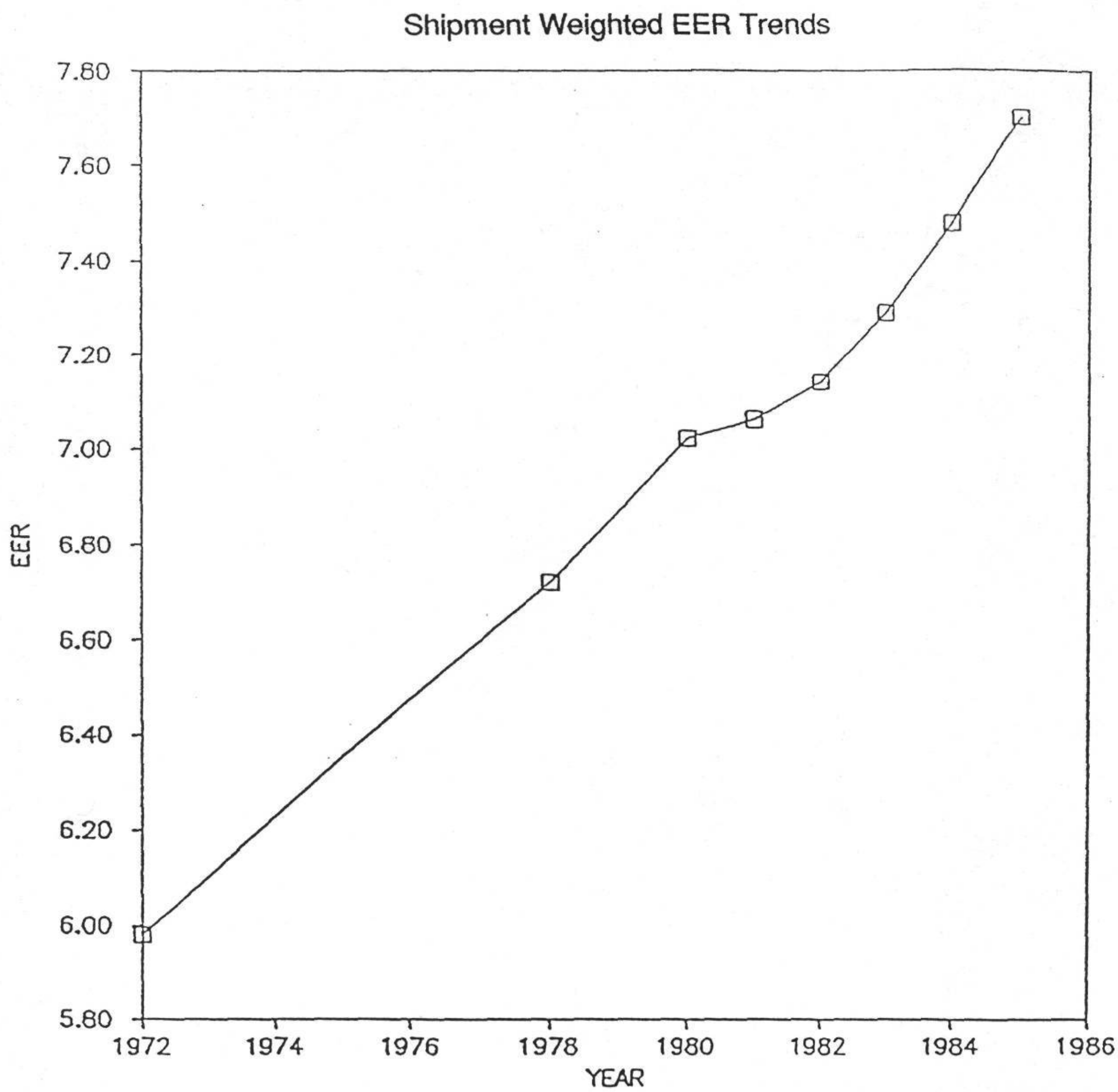


Figure 2.4 - Shipment Weighted EER Trend
from 1972 to 1986 [4]

Table 2.2 - Maximum and Minimum Efficiency RAC's Available in 1987

Size (Btu/hr)	Type	EER Minimum	EER Maximum
< 6,000	Louvered	5.6	9.0
6,000 - 7,999	Louvered	5.4	10.2
8,000 -13,999	Louvered	6.1	12.0
14,000 -19,999	Louvered	5.9	10.2
20,000 +	Louvered	6.5	9.3
< 6,000	Non-louvered	7.5	8.7
6,000 - 7,999	Non-louvered	7.2	9.0
8,000 -13,999	Non-louvered	5.9	10.0
14,000 -19,999	Non-louvered	6.5	8.0
20,000 +	Non-louvered	-	-
	Reverse Cycle Louvered	6.1	9.5
	Reverse Cycle Non-louvered	5.8	9.4

The major source of RAC performance data is the Directory of Certified Room Air Conditioners. This directory lists all the RAC's certified by AHAM testing, this testing is provided on a voluntary basis. All models manufactured or marketed by a participant must be certified. Certification covers cooling and heating capacity and amperes for both residential and built-in models.

Table 2.3 shows the minimum efficiency standards for the different RAC classes that was passed by Congress[5]. These standards go into effect on January 1, 1990. Comparing Tables 2.2 and 2.3, it is evident that in many of the classes, there were already units sold in 1987 that far surpassed the minimum standard to be imposed in 1990. For instance, in 1987, there was a 12.0 EER side louvered unit available in the 8000 to 13999 Btu/hr class while the standard only calls for a minimum EER of 9.0. Only the non-louvered 14000-19999 Btu/hr class did not have any units with efficiencies higher than the minimums set for 1990.

Table 2.3 - RAC efficiency standards for 1990[5].

Size (Btu/hr)	EER Standard Louvered	Non-louvered
< 6000	8.0	8.0
6000 - 7999	8.5	8.5
8000 - 13999	9.0	8.5
14000 - 19999	8.8	8.5
20000 +	8.2	8.0

References

1. Roger Thevenot, A History of Refrigeration Through The World, International Institute of Regrigeration, 1979.

2. Grant, W.A., "From '36 to '56 - Air Conditioning Comes of Age", ASHRAE Transactions, Vol. 63, 1957.

3. Moyers, J., "The Room Air Conditioner as an Energy Consumer", ORNL/CON-2, Oak Ridge National Laboratory.

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

5. 100th Congress, "National Appliance Energy Conservation Act", Public Law 100-12, March 17, 1987.

6. "Advanced Notice of Proposed Rule Making", Federal Register, Vol. 44, No. 1, January 2, 1979.

7. "G.E. Response to Proposed Rule Making", General Electric, December 15, 1980.

8. "Room Air Conditioners Energy Efficiency and Consumption", Association of Home Appliance Manufacturers, November 24, 1986.

9. Statistical Panorama, Refrigerating, Heating, and Air Conditioning News, 1980.

10. Statistical Panorama, Refrigerating, Heating, and Air Conditioning News, 1987

11. Directory of Certified Room Air Conditioners, issued quarterly, Association of Home Appliance Manufacturers, Chicago.

CHAPTER 3

PERFORMANCE MODEL SELECTION

To determine the possible improvements in the efficiency of RACs, it is necessary to model the RAC as a system. The components used in a RAC can represent current technology or advances in the technology possible within the next few years. The state-of-the-art technology can be modeled with existing computer models. For products not yet on the market, we must rely on engineering judgement to determine their impact on efficiency improvements.

There are two public domain models that we have considered using for this analysis: the Oak Ridge National Laboratory (ORNL) heat pump model [1] and the Arthur D. Little (ADL) room air conditioner model [2]. The ORNL model was completed in 1981. Although specifically designed to model heat pumps, it is adaptable for use on RAC's. The ADL model was previously used in the 1980 and 1982 Engineering Analysis for the Appliances Standards [3,4].

After a thorough review of both the ORNL and ADL models, we chose the ORNL model. The primary reasons for using the ORNL model centered on: (1) model purpose, (2) compressor model, (3) condenser and evaporator model, (4) detailed output. These items are discussed below.

Model Purpose

The ORNL model was developed as a tool to aide in the design of heat pumps. It allows evaluation of system performance using actual hardware. The performance of components (heat exchangers, compressors, expansion devices, etc.) is determined using empirical methods. The ORNL model is a well thought out design tool that incorporates many options to examine system performance with various "off-the-shelf" components. The ADL model was developed for the express purpose of performing the 1979 and 1982 engineering analysis. It's methodology relies on curve fits of data from systems available in 1982. This methodology severely restricts it's ability to use current or future technology in the determination of efficiency improvements. The model simply was not designed to be used as a tool to aide designers, but was intended to perform the analysis of a limited number of RACs. Consideration of the purpose for which each model was developed gives a clear edge to the ORNL model for use in this analysis.

Compressor Models

The Oak Ridge model provides two choices of compressor models. One uses curve fit data from manufacturer's compressor maps. The data is for both flow rates and power consumption. The model also includes software to curve fit the polynomials used as input for the compressor model. For these analysis the curve fit data are provided exogenous to the model.

The second compressor model performs an energy balance on the compressor, but requires motor efficiency, mechanical efficiency, isentropic compression efficiency from suction to discharge port, and internal and external heat loss measurements. These data, not readily available from most manufacturers, are intended to allow the designer to model possible developments in compressor technology. The map based model would be more accurate in measuring actual hardware output conditions.

The ADL model has only one compressor model, also based on curve fit data. Generally, the manufacturer's compressor maps relate mass flow rate as a function of condensing and evaporating temperature. The ADL model requires that the curve fit coefficients be determined as input the ADL model stores them internally and provides space for 42 models. This makes access to the coefficients a difficult and time consuming process, since the model has to be recompiled every time new compressor data are entered.

The ease of changing compressors by modifying the curve fit constants and the two models provided in the ORNL model make it more acceptable than the ADL model.

Condenser and Evaporator Models

The ORNL model uses the NTU (Number of Transfer Units) method to determine the heat exchanger effectiveness for dry coils. A modified NTU method is used for wet coils. Heat transfer correlations are calculated with existing correlations [5,6,7].

The ADL model uses curvefits determined from manufacturers' data on complete heat exchangers to determine heat transfer for the two phase condensing or evaporating region. It then uses an exogenous multiplier to determine coefficients for the subcooled and superheat regions.

The Oak Ridge model allows construction of various types of coils using different tube diameters, wall thicknesses, tube spacing, and fin thicknesses. The ORNL model allows use of wavy, louvered, or other fins if data are available. In contrast, the ADL model is limited on the coils that can be used. Coils must by constructed from staggered copper tubes, 3/8 inch outer diameter, 0.016 inch thick walls, with corrugated aluminum fins of 0.006 inch thickness. Vertical spacing, center-to-center, is 1.00 inches and horizontal spacing is 0.866 inches. The ADL model has only 3 fin pitches: 14, 17, 19 fins per inch. This limits the modeling of different coil characteristics to improve the performance.

Some manufacturers of RAC's also employ wet condensers to increase heat transfer on the condensing side. This is accomplished by funneling condensate from the evaporator to the condenser and using the condenser fan to sling the condensate on the coil. Neither the ORNL or ADL model had the capability to model these wet condensers. However, the ORNL model does employ a wet coil factor on the evaporator. A method similar to this can be used in the determination of the heat transfer for the wet condenser coils. Another component used by manufacturers to increase performance is a small heat exchanger that increases the subcooling exiting the condenser and increases the superheat entering the compressor. This subcooler passes the refrigerant line leaving the condenser near the evaporator allowing heat transfer to occur between the two. Thus, the temperature of the condensed refrigerant is reduced and the evaporating vapor is superheated (explained in more detail in Chapter 4). Neither model accounts for this heat exchanger, but an algorithm can be incorporated into ORNL model.

In comparing the ADL and ORNL heat exchanger models, the ORNL model emphasizes the total system effect while the ADL model concerns itself with ease of calculation. The ADL model uses simple algorithms (multiplying factors) to calculate performance of the heat exchangers. The ORNL model, by using empirical methods to calculate the performance of the coils, is much more exact in the determination of heat transfer characteristics. The RAC due to its limited size, depends heavily on the maximum heat transfer from the coils. Any model used to examine RAC's should pay strict attention to this most important detail. The ORNL model again is preferred on this account.

Model Output

A sample of the output provided by both models is shown in Appendix A. The ORNL model output in its shortened form includes detailed state point data at the inlet and outlet of each component. Compressor power, flow rate and efficiency are shown. Coil heat transfer data are shown for each region (superheated, saturated, subcooled) along with pressure drop data, both air and refrigerant side. A system summary concludes the output showing capacities, power consumption, and COP.

The ADL model output shows the necessary data of EER, capacity, power consumption, but for detailed output only the enthalpies at the inlet and exit points for all components are shown. Refrigerant pressure drops are ignored and little data on the heat transfer characteristics are shown.

Conclusions of Model Selection

Based on the previous comparison of the two models and what is required to best evaluate the performance of RAC's, the ORNL model is the better choice. The ADL model, while compact and easy to operate, tends to be restrictive on the heat transfer characteristics that can be varied. In a room air conditioner where the maximum performance from the coils is required, a model that performs detailed calculations on these coils is preferable.

Besides the better heat exchanger modeling, other strong points of the ORNL model include its use of available compressor maps, quick execution, and detailed output. While we chose the ORNL model for use in this project, several significant changes had to be made in the model so it could better simulate the performance of RACs. These changes are described in Chapter 4.

References

1. S.K. Fischer and C.K. Rice, "The Oak Ridge Heat Pump Models: A Steady-State Computer Design Model for Air-to-Air Heat Pumps", ORNL/CON-80/R1, Oak Ridge National Laboratory, August 1983.

2. "Engineering Computer Models for Refrigerators, Freezers, Furnaces, Water Heaters, Room and Central Air Conditioners", Arthur D. Little, Inc., No. 83528-20, Nov. 1982.

3. "Consumer Product Efficiency Standards Engineering Analysis Document", DOE/CS-0166, U.S. Department of Energy, June 1980.

4. "Consumer Product Efficiency Standards Engineering Analysis Document", DOE/CS-0030, U.S. Department of Energy, June 1980.

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

6. 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.

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

CHAPTER 4

MODEL DEVELOPMENT

Developing a RAC design model from the ORNL heat pump model involved three distinct steps. The first step was to identify the differences between heat pumps and RACs. Manufacturers of RACs and heat pumps were contacted. The manufacturers provided data on the components used to construct the units and the differences between the two were compiled. The second step was to identify the heat and mass transfer correlations that would be used to model the RAC components not found in heat pumps. These correlations were taken from current literature and represent the latest technology in heat transfer. The third step involved the coding of the changes into the ORNL model.

Identification of Changes

The primary differences between RACs and heat pumps evolve from the size differences between the units. Residential sized heat pump outdoor coils may have frontal areas of 30 square feet. RAC condensing coils rarely exceed three square feet. There are two factors which limit the size of RAC coils. First, all RACs are packaged units (both condenser and evaporator are in the same cabinet). Most heat pumps are split systems with the outdoor coil and indoor coil in seperate cabinets. Second, most RACs are window mounted units which limits cabinet dimensions and unit weights because of window frame size and structure. The limited coil sizes also restrict the amount of heat the unit is able to transfer; therefore other means to increase heat transfer must be evaluated.

Five modifications to the ORNL model have been identified to allow it to model RACs. These include: (1) the addition of a subcooler to the condenser, (2) the spraying of condensate on the condenser, (3) the use of internally roughened tubes in the coils, (4) the removal of the reversing valve, and (5) the improvement of the ORNL capillary tube model. The first three changes arise because of design options used in RACs that are not used in heat pumps. The fourth change was necessary because reversing valves are not used in RACs. The last change provides more precise calculation of the capillary tube response in the system. Each change is explained in greater detail below.

Subcooler

A pressure-enthalpy diagram of an idealized vapor compression cycle is shown in Figure 4.1. The purpose of the subcooler is to increase the amount of subcooling exiting the condenser (segment 1-1a) which extends the two phase region in the evaporator (segment 2-2a). There are two subcoolers developed for this purpose. One consists of a tube that takes the refrigerant leaving the condenser and passes it through a pool of condensate at the bottom of the condenser coil (Figure 4.2). The condensate temperature is between ambient temperature and the evaporator temperature (approximately 70 F) while the refrigerant leaving the condenser is at approximately 130 F. The heat transfer between the two allows the refrigerant to subcool while heating the condensate a few degrees. The length and diameter of the subcooler and temperature of the condensate determine the outlet temperature and subcooling of the refrigerant.

The second subcooler is a tube-in-tube heat exchanger (Figure 4.3). Liquid refrigerant exiting the condenser flows through the annulus of the two tubes while superheated vapor exiting the evaporator flows through the inside tube. The heat transfer between the two flows subcools the liquid and further superheats the vapor. The annulus of this subcooler can be either smooth or have internal ridges to increase heat transfer. This subcooler also causes an extra pressure drop compared to the first subcooler. The vapor flowing through the inside tube will increase the pressure drop along with the liquid flowing through the annulus. If internal ridges are used, an even greater pressure drop occurs. This subcooler provides a larger evaporation capacity at the cost of a greater pressure drop. A major disadvantage is the cost. This subcooler is more difficult to construct and, therefore, more expensive than the first.

Condensate Spray

The condensate pool mentioned above is also used to spray the condensing coil. The condensate is funneled to the condensing side (Figure 4.4) of the RAC and the condenser fan is used to sling water on the condenser. This serves two purposes: first, it removes the condensate and second, it increases heat transfer on the condenser. The increased heat

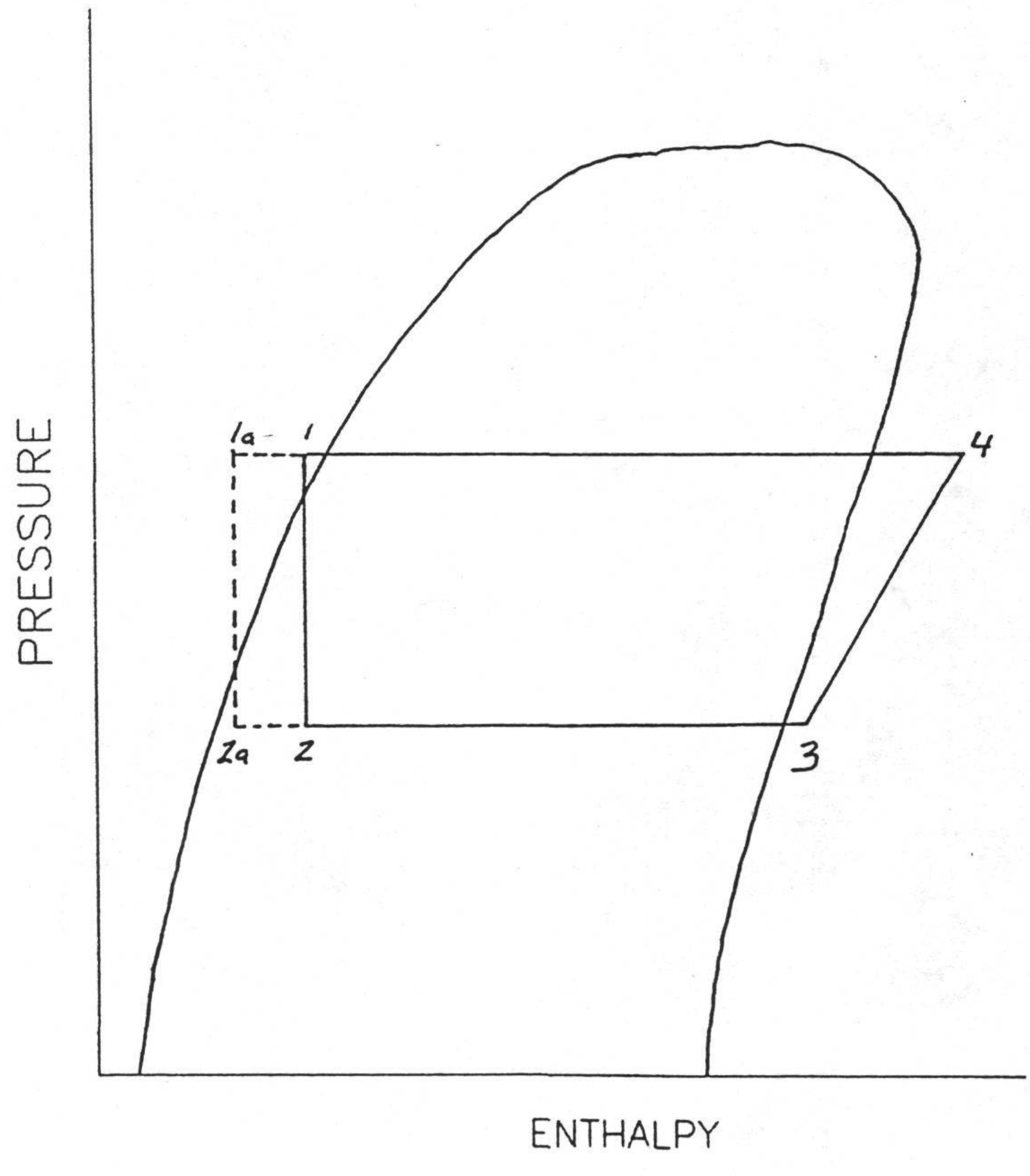
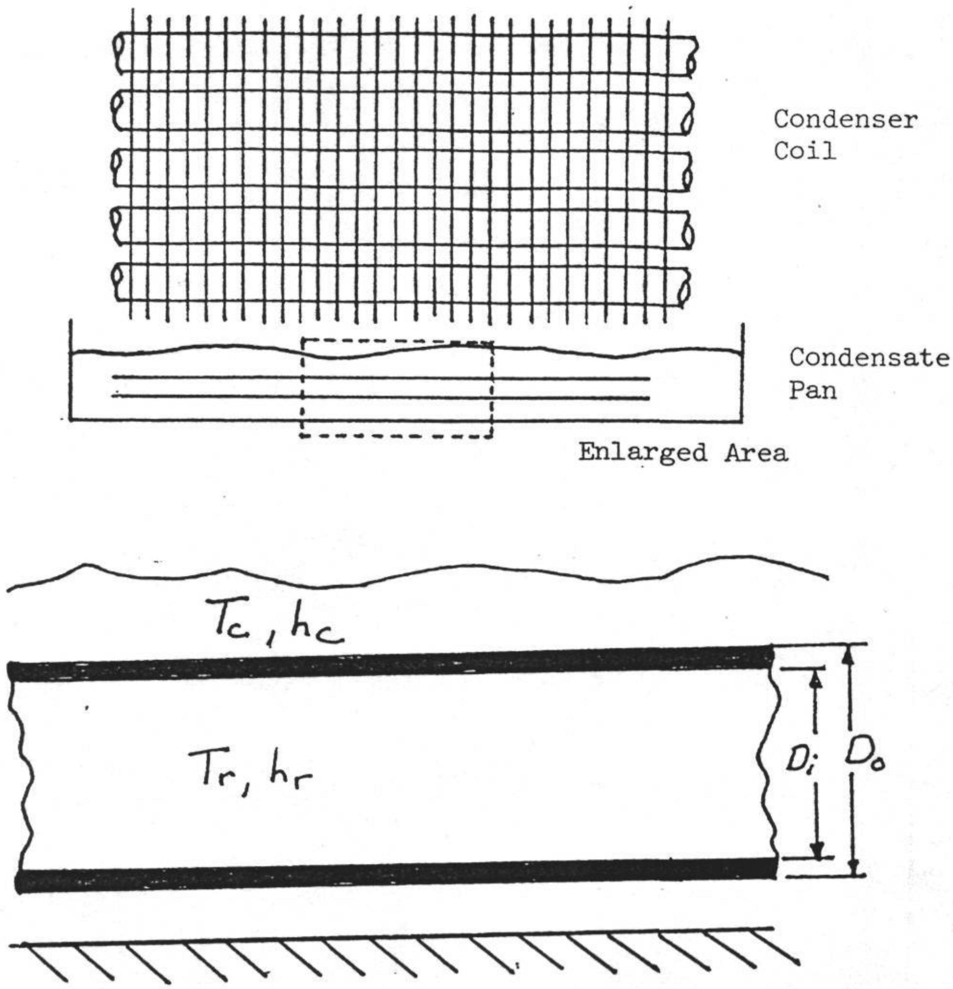


Figure 4.1 - Idealized Refrigeration Cycle on a Pressure-Enthalpy Diagram



Where,

D_i = Inside diameter

D_o = Outside diameter

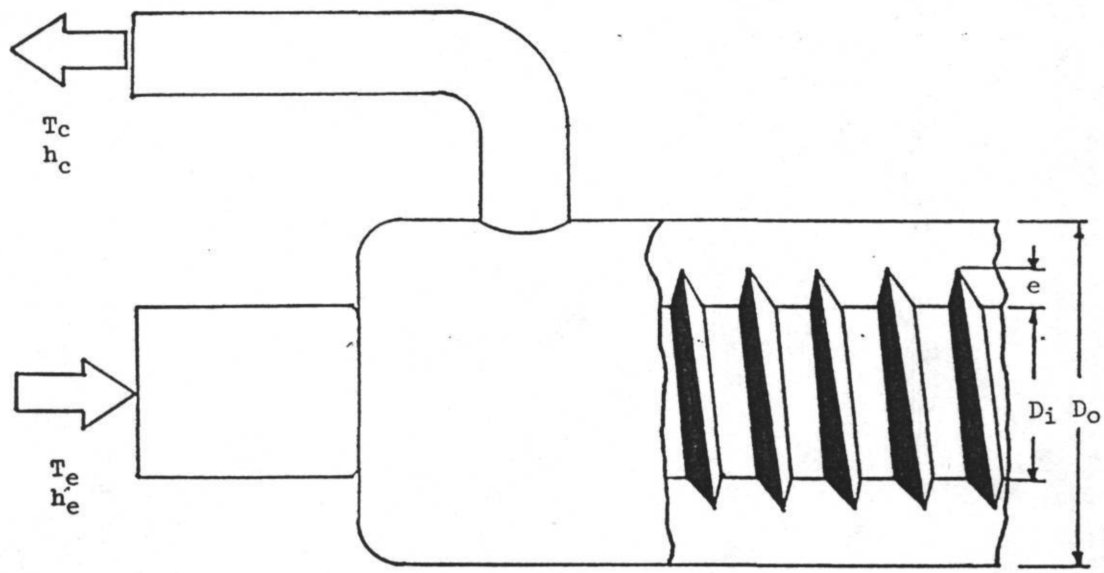
T_c = Temperature of the condensate

T_r = Temperature of the refrigerant

h_c = Heat transfer coefficient for condensate

h_r = Heat transfer coefficient for refrigerant

Figure 4.2 - Tube in Condensate Subcooler



where,

T_c = Temperature of refrigerant out of the condenser

T_e = Temperature of refrigerant out of the evaporator

h_c = Heat transfer coefficient out of the condenser

h_e = Heat transfer coefficient out of the evaporator

D_o = Outside diameter

D_i = Inside diameter

e = Ridge height

Figure 4.3 - Tube-in-Tube Subcooler

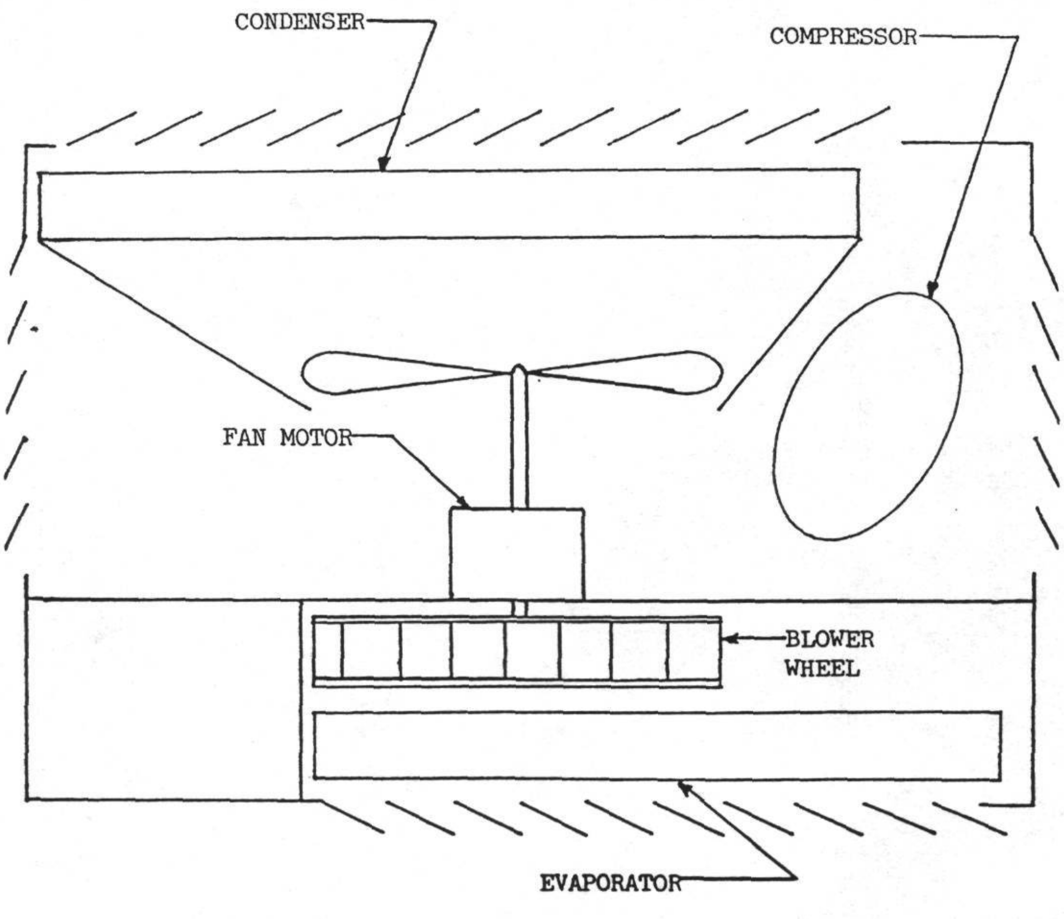


Figure 4.4 - Simplified Diagram of a Room Air Conditioner

transfer is accomplished by the water evaporating off the coil. Typically, only a small percentage (less than 25%) of the condenser coil is actually wetted by the spray because the fan can only sling water on the first few rows of the heat exchanger [1].

Roughened Tubes

Internally grooved tubes have become a popular method to increase heat transfer without increasing heat exchanger size [2]. RACs use a multi-grooved tube (MGT) which has a higher inside heat transfer coefficient compared to the conventional smooth tube.

Removal of the Reversing Valve

Because the ORNL model is intended to model heat pumps, it has extra tubing and pressure drops associated with the reversing valve. The reversing valve and tubing are not needed in the RAC model.

Capillary Tube Model Modification

The ORNL model, as it currently exists, has four refrigerant expansion device models (capillary tube, thermal expansion valve, orifice plate, or specified subcooling). RACs almost exclusively use capillary tubes for expansion. The ORNL capillary tube model consists of empirical fits to the curves given in the ASHRAE Equipment Handbook [3].

These curves are for standardized capillary tube flow rate as a function of inlet pressure and subcooling. The standardized flow rate must be modified for each particular length and diameter of capillary tube. These flow factor correction factors are also listed in the Handbook. As originally designed, the flow factor correction has to be manually input into the ORNL model. The Handbook also provides another correction factor for cases when the downstream pressure of the capillary tube is above the critical pressure. The current ORNL model does not have a correction factor for these conditions.

Correlations

After the necessary design changes were identified, correlations that described the new or changed components were needed. The FORTRAN coding of the ORNL program was modified to reflect these new correlations. The associated modifications and correlations for each design change are discussed below.

Subcooler

The correlation for both the subcoolers uses the Number of Transfer Units (NTU) method to determine the effectiveness of the heat exchanger. To determine the NTU, the heat transfer coefficients for all surfaces involved in the heat exchanger were determined. There were two coefficients needed for the tube-in-condensate subcooler, one for the refrigerant inside the tube and a natural convection correlation for the outside heat transfer coefficient (Figure 4.2). The inside correlation came from the Dittus-Boelter equation [4]:

$$h = 0.023\ k/d\ Re^{0.8}\ Pr^{0.3} \qquad (4.1)$$

where,
- h = heat transfer coefficient (Btu/hr-sf-F)
- k = thermal conductivity of the liquid (Btu/hr-ft-F)
- d = inside diameter of the tube (ft)
- Re = Reynolds number (dimensionless)
- Pr = Prandtl number (dimensionless).

This equation is valid for turbulent flow where the Reynolds number is greater than 10,000 and the Prandtl number is between 0.6 and 160. The length divided by the diameter must also be greater than 60. The outside correlation came from Churchill and Chu [5]:

$$h = k/d\{0.60+0.378Ra^{0.1667}[1+(0.559/Pr)^{9/16}]^{-8/27}\}^2 \qquad (4.2)$$

where,
- h = heat transfer coefficient (Btu/hr-sf-F)
- k = thermal conductivity of the liquid (Btu/hr-ft-F)
- d = inside diameter of the tube (ft)
- Ra = Rayleigh number (dimensionless)
- Pr = Prandtl number (dimensionless).

This correlation is valid for Rayleigh numbers from 10^{-5} to 10^{12} and Prandtl numbers greater than zero.

A series of tests were made to determine the effectiveness of the subcooler. An 8000 Btu/hr RAC was chosen to make the runs. The unit was similar to that of an medium efficiency model available on the market in 1987. The unit is described in more detail in Appendix C. The results of the runs are shown in Table 4.1. The only changes between the base case and the second run was the addition of the tube in condensate subcooler. The third run was done with a longer subcooler, while all other input parameters were held constant. The 24 inch subcooler increased the capacity by approximately 2 % over the base unit and increased the EER

Table 4.1 - Effect of tube-in-condensate subcooler

SUBCOOLER LENGTH (IN)	CAPACITY (BTU/HR)	EER (BTU/W-HR)
BASE (0)	8210	10.5
24	8350	11.0
48	8490	11.4

by 5 %. The 48 inch subcooler was even more effective at increasing performance. The capacity and EER increased by 3.5 % and 8.6 % respectively over the base case.

The second subcooler also needed two correlations (Figure 4.3). The internal heat transfer coefficient was determined with Equation 4.1. The annular heat transfer correlation came from Hsieh and Liauh [6]:

$$h = 5.23\ (k/D)\ Re^{0.51}\ (e/D)^{0.33} \tag{4.3}$$

where,

h = heat transfer coefficient (Btu/hr-sf-F)
k = thermal conductivity of the liquid (Btu/hr-ft-F)
Re = Hydraulic Reynolds number (dimensionless)
e = Ridge height (ft)
D = Hydraulic diameter (ft).

This equation is valid for Reynolds numbers between 3,500 and 30,000, when the roughness, e/D, is between 0.04 and 0.096.

Another effectiveness test was run on the 8,000 Btu/hr RAC with this subcooler. The input parameters were the same as in the previous test with the exception of the subcooler. The results are shown in Table 4.2 for the base case, a 12 inch, and a 24 inch subcooler. The capacity was increased by 1.3 % over the base case for the 12 inch subcooler while the

EER increased 10.4 %. For the 24 inch subcooler, the capacity and EER increased by 2.5 % and 12.4 %, respectively over the base case.

Table 4.2 - Effect of tube-in-tube subcooler

SUBCOOLER LENGTH (IN)	CAPACITY (BTU/HR)	EER (BTU/W-HR)
BASE (0)	8210	10.5
12	8320	11.6
24	8410	11.8

Wet condenser

The effect of water spray on fin and tube heat exchangers was examined by Tree, et al. [1] and they concluded that heat transfer could be enhanced by up to 40 percent. Heat transfer correlations were presented for the wet coils. Their correlation consists of a heat transfer coefficient for the wet coil as a function of the heat transfer coefficient for the dry coil

$$h_{wet}/h = 1 + [m_{eff} \ h_{fg} \ D]/[c_p \ dTX(mu \ A \ Re)] \quad (4.4)$$

where,

- h = heat transfer coefficient dry coil (Btu/hr-sf-F)
- h_{wet} = heat transfer coefficient wet coil (Btu/hr-sf-F)
- m_{eff} = effective mass flow (lbm/hr)
- h_{fg} = enthalpy of vaporization of water (Btu/lbm)
- D = hydraulic diameter (ft)
- c_p = specific heat of air (Btu/lbm-F)
- dT = temperature difference (F)
- mu = viscosity (lbm/ft-s)
- A = area (sf)
- Re = Reynolds number.

This equation is valid for flow rates of less than 7 lbm/hr and Reynolds numbers up to 1000.

A performance test was also made with the water spray on the condenser. The same baseline model was used and the water was sprayed at different rates on the condenser. The actual rate of water spray depends on the humidity of the indoor air. The results (Table 4.3) show that the increase in capacity is generally small and decreases with larger flow rates. Capacity increase is an added benefit from the water spray whose main purpose is the removal of the condensate from the unit. The initial effect was an increase of 40 Btu/hr in capacity and an increase of 0.1 in EER. Doubling the flow rate only increased the capacity another

Table 4.3 - Effect of water spray on the condenser

WATER SPRAY (LBM/HR)	CAPACITY (BTU/HR)	EER (BTU/W-HR)
BASE (0)	8210	10.5
1.0	8250	10.6
2.0	8270	10.7
3.0	8280	10.8

20 Btu/hr and EER another 0.1. Further increases produced similar results.

Roughened tubes

There has been extensive work done in the field of heat transfer in internally roughened tubes [2]. However, the main working fluid for most experiments was water. Refrigerant in two-phase flow has different characteristics than water. A paper by Schlager, et al. [7] examined various refrigerants under evaporating and condensing conditions in both smooth tubes and micro fin tubes. The literature surveyed provided various enhancement factors for the micro-fin tubes. Tojo, et al. [8] examined the performance of multi-grooved tubes in air conditioners under evaporating and condensing conditions. They determined that heat transfer could be enhanced by up to two times over smooth tubes, while the pressure drop remained approximately the same.

An enhancement factor similar to those shown in the Schlager paper was used to alter the heat transfer coefficients the ORNL program calculates. This enhancement factor is the ratio of the micro-fin heat transfer coefficient to that of the smooth tube. This enhancement factor can be supplied in the input to represent various tubes designs, if the grooved tube option is chosen.

A test of the effect of the enhancement factor was made on the same baseline model. The results (Table 4.4) suggest that the initial increase in the enhancement provide the greatest relative performance increase and that the performance increase is smaller for larger enhancement factors.

Table 4.4 - Effect of grooved-tube enhancement factor

ENHANCEMENT FACTOR	CAPACITY (BTU/HR)	EER (BTU/W-HR)
BASE (0)	8210	10.5
1.5	8270	10.6
2.0	8300	10.8
2.5	8310	10.9

Reversing valve

The tubing associated with the reversing valve was simply eliminated from the input as well as the pressure drop calculation subroutines. The previous drop in pressure for the refrigerant lines was 1 psi for the suction side of the compressor and 0.8 psi for the discharge side. After the removal of the extra tubing the suction side pressure drop fell to 0.4 psi and the discharge side pressure drop was 0.1 psi. The compressor power consumption decreased by 10 watts after the removal of the extra tubing.

Capillary tube model

The ORNL program models capillary tubes with a standardized mass flow rate that is modified by a flow factor (Equation 4.5)

$$m_r = phi_1 N_{cap} m_s phi_2 \quad (4.5)$$

where,

m_r = actual mass flow rate

phi_1 = flow factor

N_{cap} = number of capillary tubes

m_s = standard mass flow rate.

phi_2 = pressure correction factor

The original model accepted the flow factor as input, which is a function of length and diameter of the capillary (Figure 4.5). The current model was modified to accept length and diameter as input rather than the flow factor. The curves presented for the critical pressure (Figure 4.6) calculation are also based on length and diameter. The SAS statistical package was used to fit these curves [9]. The equations for the flow factor and pressure correction factor are:

$$phi_1 = e^{(A + B + C + E + F)} \quad (4.6)$$

where,

$A = 7.136$ $\qquad$ $B = 2.209 \ln(D)$

$C = 0.162 \ln(L)$ $\qquad$ $E = 0.081 \ln(L) \ln(D)$

$F = -0.042 \ln(L)^2$

phi_1 = flow factor

L = length of capillary tube (in)

D = inside diameter (in)

The pressure correction, phi_2, is equal to one when the outlet pressure of the capillary tube is less than the critical pressure. When the outlet pressure is greater than the critical pressure, the presure correction is :

$$phi_2 = A + B\,dP^2 + C\,dP^3 + D\,dP^4 + E\,dP^5 \quad (4.7)$$

where,

$A = 1.00$ $\qquad$ $B = -6.25$

$C = -7.29$ $\qquad$ $D = 9.64$

$E = -5.21$

$dP = (P_{evap} - P_{crit}) / (P_{cond} - P_{crit})$

P_{evap} = pressure at evaporator entrance

P_{cond} = pressure at condenser entrance

P_{crit} = critical pressure

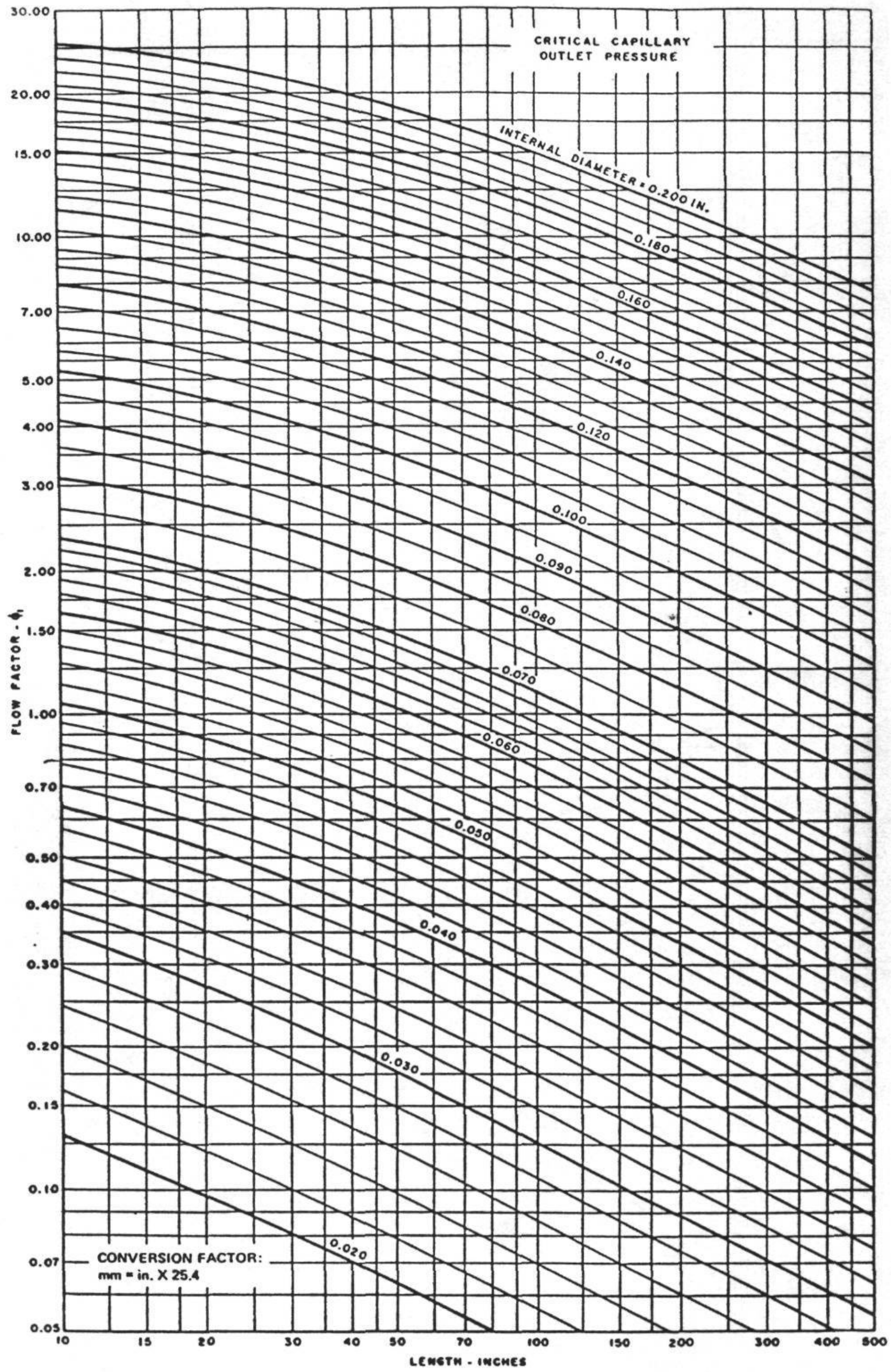
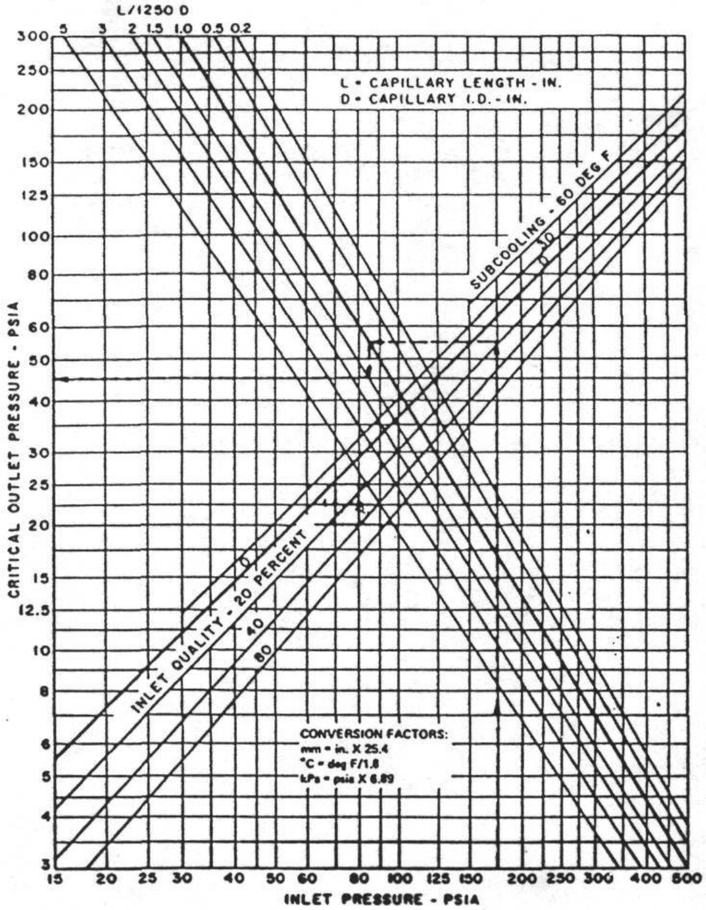


Figure 4.5 - Capillary Flow Factors



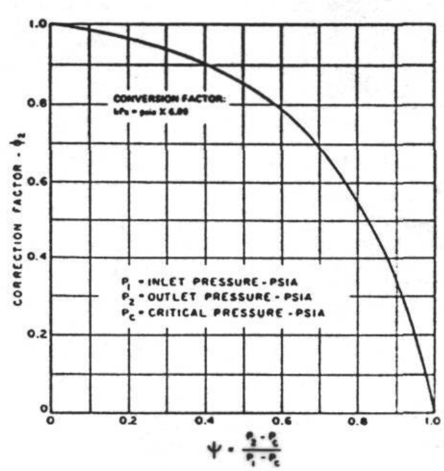


Figure 4.6 - Capillary Critical Pressure and Correction Factor

The critical pressure is determined with Equations (4.8) and (4.9). Equation (4.8a) is used if subcooling exists at the capillary inlet and Equation (4.8b) is used if saturated mixture exists at the entrance.

$$P_{temp} = e^{(A + B + C)} \tag{4.8a}$$

$$P_{temp}=e^{(D + E + F + G + H)} \tag{4.8b}$$

where,

$A = -0.903$ $\quad$ $B = 0.979 \ln(P)$

$C = -0.0000624 \ln(S)^2$ $\quad$ $D = -1.00$

$E = 1.00 \ln(P)$ $\quad$ $F = -2.27 \ln(Q)$

$G = 0.623 \ln(Q)^2$ $\quad$ $H = 0.235 \ln(Q)\ln(P)$

P_{temp} = temporary pressure

P = inlet pressure
S = inlet subcooling (F)
Q = inlet quality (fraction)

$$P_{crit}=e^{(A + B + C + D + E)} \tag{4.9}$$

where,

$A = 11.529$ $\quad$ $B = -0.492 \ln(lod)$

$C = -1.631 \ln(P_{temp})$ $\quad$ $D = 0.032 \ln(lod)^2$

$E = 0.026 \ln(lod)\ln(P_{temp})$

P_{crit} = critical pressure

lod = length/1250 diameter

P_{temp} = temporary pressure

All of the equations had an r-squared value greater than 0.99. Equation (4.8) is valid for inlet pressures from 15 to 500 psia, subcooling from 1 to 60 F, and quality from 0 to 0.8. Equation (4.9) is valid for lod from 0.2 to 5.

Coding of Changes

The ORNL program consists of 65 subroutines that are used to model each of the components in the unit. The primary changes are in the capillary, condenser, and evaporator subroutines. The subcoolers required the addition of a separate subroutine. Each coding problem is discussed below.

Subcoolers

An additional subroutine modeling the effectiveness of these two different subcooler models was added to the code. To determine the effectiveness of the first subcooler, this subroutine takes into account the length, diameter, and physical properties of the copper tube. It also accepts as input the approximate temperature of the condensate pool. The model currently determines the properties of the refrigerant exiting the condenser (temperature, quality, enthalpy, etc.). A check is made to determine if the refrigerant is fully condensed at the condenser outlet. If quality exists at the condenser exit, an energy balance is performed to determine the length necessary for full condensation. The subcooler length is shortened to account for that portion needed for the refrigerant to reach a saturated liquid state. The correlations mentioned above are used to determine the heat transferred from the liquid into the condensate pool and determine the outlet temperature. The outlet temperature of the subcooler is then input into the capillary tube subroutine to determine the temperature difference between saturated fluid and the actual bulk temperature. This temperature difference is the degree of subcooling.

The second subcooler model accepts the inside and outside diameters of the two tubes, the length of the heat exchanger, and the height of the ridges as input. Once again a check is performed for full condensation. This subcooler's outer surface is assumed to be insulated and heat transfer occurs only between the two fluids. The correlations are used to determine outlet temperatures. This portion of the subroutine also determines the exiting enthalpy and pressure for both the vapor and liquid.

Water Spray

The condenser subroutine of the ORNL model was changed to allow water spray on the condenser as an option. If this option was selected, the input consisted of the approximate percentage of the coil that can be considered wet. The mass flow rate of the water is the second input factor. A large mass flow rate would increase the heat transfer; however, the mass flow of the water cannot be larger than that removed by the evaporator from the air. A small subroutine was added to calculate the new air side heat transfer with the influence of water on the coil. The air side heat transfer coefficient was modified using the wet heat transfer coefficient for that percentage of the coil which was considered wet.

Roughened Tubes

The average enhancement factor for the studies surveyed in Schlager's paper is 2.0 for the condensation side and 2.1 for the evaporator. These are the values that are used to run the model. The only coding changes necessary were the multiplication of the heat transfer coefficients determined by ORNL and the enhancement factor.

Reversing Valve

The program originally had extra tubing associated with the reversing valve. The variables for these extra lengths were removed and new variables were created for lengths and diameters of the suction, discharge, and liquid lines.

Capillary Tube

The capillary tube subroutine originally accepted the flow factor as input to the program. The flow factor was determined from the length and diameter of the capillary tube. The input was changed to accept the length and diameter. The curves generated from the regression package were input to the capillary tube subroutine.

With these corrections and the new correlations the model could be verified as being accurate. This verification can be found in the next chapter.

References

1. D.R. Tree, V.W. Goldschmidt, R.W. Garrett, and E. Kach, "Effect of Water Sprays on Heat Transfer of a Fin and Tube Heat Exchanger", *Sixth International Heat Transfer Conference*, Vol. 4, Aug 7-11, 1978

2. A.E. Bergles, Techniques to Augment Heat Transfer, *Handbook Of Heat Transfer*, 1973.

3. Refrigerant Control Devices, ASHRAE Equipment Handbook, 1983.

4. F.W. Dittus and L.M.K. Boelter, Univ. Calif., Berkeley, Publ. Eng., Vol. 2, 1930, p.443.

5. S.W. Churchill and H.H.S. Chu, "Correlating Equations for Laminar and Turbulent Free Convection from a Horizontal Cylinder", *Int. J. Heat Mass Transfer*, 18, 1049, 1975.

6. S.S. Hsieh and C.T. Lisuh, "Thermal Analysis of Double Pipe Heat Exchanger with Helical Type Roughened Surface", *1987 ASME-JSME Thermal Engineering Joint Conference*, Vol. 3, 1987.

7. L.M. Schlager, M.B. Pate, A.E. Bergles, "Evaporation and Condensation of Refrigerant-Oil Mixtures in a Smooth Tube and a Micro-Fin Tube", *ASHRAE Transactions*, V. 94, Pt. 1, 1988.

8. S. Tojo, K. Mosokawa, T. Arimoto, H. Yamada, and T. Ohta, "Performance Characteristics of Multi-Grooved Tubes for Air Conditioners", *Australian Refrigeration, Air Conditioning and Heating*, August 1984.

9. SAS User's Guide: Statistics, 1982 Edition, Cary, N.C., SAS Institute Inc., 1982.

CHAPTER 5

MODEL VALIDATION

To examine the accuracy of the RAC design model, a comparison with several units currently on the market was performed. Detailed hardware data (coil size, fin density, compressor maps, etc.) were obtained from several manufacturers of RACs. The manufacturer also supplied performance data such as capacity, efficiency, and the refrigerant properties throughout the system.

The manufacturer supplied the compressor performance maps for all of the units used in the validation. Subcoolers were used in all three units. The length and diameter of the capillary tubes were also given.

Three units were used to validate the model. These included a 8000, 12000, and 19000 Btu/hr unit. Hardware data for the three units are shown in Tables 5.1, 5.2, and 5.3 for the 8000, 12000, and 19000 Btu/hr units, respectively. These three units represent a wide range in capacity and all three are above average in EER: 9.1 for the 8000 Btu/hr unit, 8.9 for the 12000 Btu/hr unit and 9.2 for the 19000 Btu/hr unit. All three use condensate subcoolers, wavy fin patterns, and higher efficiency Japanese rotary compressors. Because manufacturers could not provide us with fan and fan motor efficiencies, these values were estimated for the fans on both the evaporator and condenser.

Performance of the unit was calculated for steady state conditions at an outdoor temperature of 95 F and an indoor dry bulb temperature of 80 F and wet bulb temperature of 67 F. Unlike central air conditioners and heat pumps, RACs only undergo one steady state test and no cycling tests. Both the rated capacity and EER of RACs are determined at the one steady state condition.

Table 5.1 - Hardware data for an 8000 Btu/hr RAC

COMPRESSOR	
Displacement (cu in)	0.720
EER	10.7
CONDENSER	
Frontal Area (sq ft)	1.46
No. of Rows	3
Outside Diameter (in)	5/16
Wall Thickness (in)	0.012
Fins/Inch	14
Fin Design	WAVY
Fin Thickness (in)	0.0060
Fan & Fan Motor Eff.	0.10
No. Parallel Ckts.	1
Airflow (cfm)	400
EVAPORATOR	
Frontal Area (sq ft)	0.70
No. of Rows	2
Outside Diameter (in)	3/8
Wall Thickness (in)	0.012
Fins/Inch	12
Fin Design	WAVY
Fin Thickness (in)	0.0055
Fan & Fan Motor Eff.	0.15
No. Parallel Ckts.	1
Airflow (cfm)	150
OTHER DATA	
Subcooler Length (in)	52
Subcooler Diameter	5/16
Capillary Length (in)	36
Capillary Diameter	0.054
No. of Capillaries	1

Table 5.2 - Hardware data for an 12000 Btu/hr RAC

COMPRESSOR	
Displacement (cu in)	1.120
EER	10.7
CONDENSER	
Frontal Area (sq ft)	2.16
No. of Rows	3
Outside Diameter (in)	5/16
Wall Thickness (in)	0.012
Fins/Inch	14
Fin Design	WAVY
Fin Thickness (in)	0.0060
Fan & Fan Motor Eff.	0.20
No. Parallel Ckts.	1
Airflow (cfm)	600
EVAPORATOR	
Frontal Area (sq ft)	1.125
No. of Rows	4
Outside Diameter (in)	3/8
Wall Thickness (in)	0.012
Fins/Inch	12
Fin Design	WAVY
Fin Thickness (in)	0.0055
Fan & Fan Motor Eff.	0.25
No. Parallel Ckts.	2
Airflow (cfm)	320
OTHER DATA	
Subcooler Length (in)	53
Subcooler Diameter	5/16
Capillary Length (in)	30.25
Capillary Diameter	0.049
No. of Capillaries	2

Table 5.3 - Hardware data for an 19000 Btu/hr RAC

COMPRESSOR	
Displacement (cu in)	1.720
EER	10.7
CONDENSER	
Frontal Area (sq ft)	2.00
No. of Rows	3
Outside Diameter (in)	5/16
Wall Thickness (in)	0.012
Fins/Inch	18
Fin Design	WAVY
Fin Thickness (in)	0.0060
Fan & Fan Motor Eff.	0.20
No. Parallel Ckts.	1.5
Airflow (cfm)	800
EVAPORATOR	
Frontal Area (sq ft)	1.46
No. of Rows	5
Outside Diameter (in)	3/8
Wall Thickness (in)	0.012
Fins/Inch	12
Fin Design	WAVY
Fin Thickness (in)	0.0055
Fan & Fan Motor Eff.	0.25
No. Parallel Ckts.	1
Airflow (cfm)	450
OTHER DATA	
Subcooler Length (in)	117
Subcooler Diameter	5/16
Capillary Length (in)	24.5
Capillary Diameter	0.046
No. of Capillaries	3

Tables 5.4, 5.5, and 5.6 present the comparison between the manufacturers' data and the RAC model. Comparisons are made between important temperatures and pressures in the system as well as capacity and EER.

The model seems to overpredict the temperature out of the compressor in all three of the units. This in turn causes the pressure at the compressor exit to be high except for the 8,000 Btu/hr unit. All three units had subcoolers and the temperature exiting the condensers were highly accurate. The 12,000 Btu/hr unit's compressor suction temperature was very close to the actual temperature; however, the other two units' values were under predicted by the model. This again affected the pressure at the suction line.

The bottom line in all of these units is the prediction of capacity and EER. The predicted values of the larger model's capacity was too high by approximately 3 %. The EER for these two units differed by about 1 %. Overall, the capacity and efficiency comparisons were generally accurate to within 5 %. We were somewhat surprised at the close agreement between manufacturers' published EER data and that estimated by the RAC model. Manufacturers' compressor maps are usually stated to be accurate to +/- 5 %. Thus, one would expect that the a 5% error should be the best agreement between the data and model should be +/- 5%. It appears that the RAC model provides sufficiently accurate prediction of performance for use in this analysis.

Table 5.4 - Performance data for an 8000 Btu/hr RAC

Criteria	Actual	Modeled
Temperature (F)		
Compressor Inlet	58	46
Compressor Outlet	180	200
Condenser Outlet	105	180
Evaporator Outlet	46	42
Pressure (PSI)		
Compressor Inlet	78	84
Compressor Outlet	307	297
Performance		
Capacity (BTU/HR)	8390	8330
EER (BTU/W-HR)	9.1	9.2

Table 5.5 - Performance data for a 12000 Btu/hr RAC

Criteria	Actual	Modeled
Temperature (F)		
Compressor Inlet	50	51
Compressor Outlet	199	204
Condenser Outlet	99	100
Evaporator Outlet	47	45
Pressure (PSI)		
Compressor Inlet	75	85
Compressor Outlet	258	288
Performance		
Capacity (BTU/HR)	12910	13252
EER (BTU/W-HR)	8.9	9.0

Table 5.6 - Performance data for a 19000 Btu/hr RAC

Criteria	Actual	Modeled
Temperature (F)		
Compressor Inlet	54	47
Compressor Outlet	176	207
Condenser Outlet	108	108
Evaporator Outlet	43	47
Pressure (PSI)		
Compressor Inlet	70	86
Compressor Outlet	280	308
Performance		
Capacity (BTU/HR)	19160	19363
EER (BTU/W-HR)	9.2	9.3

CHAPTER 6

DESIGN OPTIONS

Design options used to improve the RAC are discussed below. Many of the design options used in the analysis of central air conditioners and heat pumps can also be used for RACs [1,2]. The design options considered include: (1) Increased condenser and heat exchanger performance, (2) Decreased compressor size, (3) Increased combined fan and motor efficiency, (4) High efficiency compressors.

Other design options are available that could possibly increase the efficiency of RAC's. These include thermal or electronic expansion valves, new refrigerants, variable speed fan motors, etc. These options have the potential to increase efficiency if the test procedure for RAC's were changed. Because RAC performance is only determined from a steady state condition at 95 F, these options will not provide any measured improvement in performance. If a change in the rating system were made, these options should be examined.

1) Increased Condenser and Evaporator Heat Exchanger Performance

One of the easiest methods to increase RAC efficiency is to increase the heat transfer of the heat exchangers. The governing equation for heat transfer, in a simplified form is:

$$q = U*A*(Tref - Tair) \qquad (6.1)$$

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 heat transfer equations are more complicated than Equation (6.1), but it illustrates the major influences on heat transfer. Heat transfer can be improved by increasing the heat transfer coefficient or the surface area of the coil. However, to maintain a constant capacity, an increase in the surface area or heat transfer coefficient must be accompanied by a decrease in the temperature difference between the air and the refrigerant. The temperature in the condenser must be lowered and the average temperature in the evaporator must be raised. This in turn raises the efficiency of the compressor and subsequently the RAC. Increasing the surface area can be accomplished by adding more frontal area, adding tube rows, or increasing fin density. Each is discussed below.

1A) Increased Heat Exchanger Frontal Area

The addition of more frontal area increases the area for air to make contact with the fins and tubes of the heat exchanger. The added frontal area also increases the distance that the refrigerant must flow. This increases the pressure drop of the refrigerant through the coil unless the refrigerant tubes are recircuited.

Increasing the frontal area in a RAC is a compromise between performance increase and the weight and size of the RAC. In a small capacity unit a large coil would increase performance, but also increase the weight of the unit which would decrease its portability. In the higher capacity units, a larger coil is possible but a limit on the size is required due to window size constraints. Another practical limit on the coil size relates to the ability to maintain latent cooling capacity. Typically, as the coil size is increased, it is possible to raise the evaporating temperature of the refrigerant and maintain the same total cooling capacity. However, as the evaporating temperature is increased, the RAC's latent cooling capacity decreases.

Limits were imposed on the maximum size of both the evaporator and the condenser coils (see Chapter 6). The limits either equaled or slightly exceeded the maximum coil sizes for systems currently on the market.

1B) Increased Tube Rows

Another way to increase surface area is to add tube rows. This increases the amount of copper tubing and fin material but the overall dimension of the chassis remains small. However, the efficiency improvement of each added row is smaller than that of the previous row. Four row coils are common and upwards of five row coils have been used.

1C) Increased Fin Density

The last approach to increasing surface area is by increasing the fin density. Low to medium efficiency units typically have 12 fins per inch (fpi) on the coils. The higher efficiency units have up to 18 fpi on the condenser. A limit of 20 fpi was chosen on the condensing coil. Fin densities higher than these increase the fan power required to move air through the coils. The evaporator fin density has a limit of 14 to 16 fpi due to the need of space for condensation to form and drain from the evaporator.

1D) Increased Heat Transfer Coefficients

The overall heat transfer coefficient can be improved by using higher performance heat transfer surfaces for the fins. The move of the HVAC industry from straight to wavy fins is an example of this improvement. Wavy fins increase the turbulence of the air flowing over the coils thus increasing the heat transfer coefficient. The wavy fins also increase the surface area of the coil. All baseline units in this analysis have wavy fins.

Some manufacturers use other enhanced heat transfer surfaces, such as louvered or slit fins to further increase heat transfer. Another heat transfer enhancement is the use of internally rifled tubes in the coil. This increases the turbulence and surface area inside the tubes, again increasing heat transfer.

A number of the manufacturers of RACs also use wet condensers to increase the heat transfer coefficient on the condensing side. The condensate from the evaporator is sprayed on the condenser through the use of a slinger ring on the condenser fan. This ring on the fan picks up the condensate from a pan and sprays it over the coil. Evaporation of the water occurs from the condenser thus increasing the ability of the condenser coil to transfer heat.

The use of higher performance heat transfer surfaces allow the manufacturer to decrease the amount of material used in the coil because it is possible to obtain the same amount of heat transfer with a smaller surface area. This allows a reduction in weight and size of the cabinet.

1E) Subcooler

Another option employed to increase heat transfer is the use of a small subcooler. This is a small heat exchanger used to increase the amount of subcooling done by the condenser and boost the superheat of the vapor exiting the evaporator. The liquid refrigerant leaving the condenser is passed through the evaporator decreasing the temperature of the condensed refrigerant and increasing the temperature of the vapor. This is done at the cost of an added pressure drop through the extra line required to move the refrigerant. However, this increases the useful refrigerating effect for the system thereby increasing capacity and efficiency [3]. Subcoolers are commonly used on RACs. The use of larger subcoolers could be used to further increase performance.

2) Decreased Compressor Size

In conjunction with the increased heat exchanger performance, compressor size must be reduced to maintain the same capacity. This is accomplished by installing a lower capacity compressor into the unit. Using a smaller compressor provides a decrease in power consumption and a boost in efficiency.

3) Increased Combined Fan and Motor Efficiency

The same motor is used to operate both the condenser and the evaporator fans. Currently, motors of approximately 55% efficiency are used in most models. The evaporator fan typically has an efficiency of 35% giving a combined efficiency of 21%. The combined fan and fan motor efficiency for the evaporator coil in medium efficiency line is assumed to be 25 to 30% and 35% for the high efficiency line. Motors with efficiencies of 70% are currently available. Forward curved 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%.

The condenser fan is usually of propeller type. Propeller fans are not as efficient as centrifugal fans. The combined fan and motor efficiency range from 10 to 20%. The medium efficiency units will use an efficiency of 15% and 20% for the high efficiency units. With motor efficiencies of 70% and fan efficiency of 30% combined efficiencies of 20% result.

4) High Efficiency Compressors

Most of the RAC's currently sold employ rotary compressors. The majority of the compressors are manufactured in Japan. The motors currently used are of the permanent split capacitor type. The improvement in efficiency available comes in the form of higher efficiency motors used for the compressor. Current motor efficiencies are as high as 87%. Expected efficiency increases of approximately 5% are expected in the next few years. This relates to a 5% increase in cooling efficiency for the system.

Other types of compressors (two speed, scroll, variable speed) exist which might increase efficiency. The current test procedure of RACs would not recognize the contributions of two speed and variable speed compressors. The scroll compressor would be recognized; however, most applications of scroll compressors have been in units with capacities greater than two to three ton units because of excess leakage in the smaller units. A change in the test procedure of the RACs might warrant examination of these other compressors.

References

1. "An Analysis of Efficiency Improvements in Residential Sized Heat Pumps", ESL/85-24, Energy Systems Laboratory, Texas A&M University, College Station, Texas.

2. "An Analysis of Efficiency Improvements In Residential Sized Heat Pumps and Central Air Conditioners", ESL/86-08, Energy Systems Laboratory, Texas A&M University, College Station, Texas.

3. Cooper, W.D., "Refrigeration Compressor Performance as Affected by Suction Vapor Superheating", ASHRAE Transactions, Vol. 80, Pt. 1, 1974.

CHAPTER 7

RAC DESIGNS

The final room air conditioner designs and the methodology used to arrive at the final designs for the different classes are discussed below. This chapter has three major sections: (1) design approach, (2) baseline units, and (3) final designs.

Design Approach

The general design approach consisted of developing a line of room air conditioners for each class similar to what a larger manufacturer might do. The specific line chosen in this analysis for a given class includes a larger selection or range in efficiency than is currently offered by any single manufacturer. The larger selection is provided to allow DOE to evaluate the possible imposition of standards in small efficiency increments. The larger selection is also necessary because we must consider units that are more efficient than those currently offered. The high efficiency units must be technologically feasible.

Another general design consideration was the calculation of performance for RACs that did not have side louvers. Specific units were designed with the assumption that they had side louvers. The performance of these units with side louvers was then calculated. A 4% power penalty was imposed on the louvered RACs to estimate the performance of units without side louvers. The 4% power penalty should account for the increased airside pressure drop for the units without side louvers. Because the actual increase in pressure drop between a louvered and non-louvered unit will depend on cabinet design, air flow rates, heat exchanger placement, etc., it was felt that using a fixed percentage increase in power would adequately compensate for these differences in design. Because fan power typically represents from 10 to 20% of total power, a 4% power penalty represents a sizeable increase in fan power for the typical unit.

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 which could be on the market in the next few years were also considered.

The next step was to choose what capacity units would be used for the efficiency analysis to represent each class. For all classes, units with capacities near the top of the capacity range in each class were chosen (Table 7.1).

Table 7.1 - Capacity of units used for the analysis in each class.

Capacity Range of Class (Btu/hr)	Capacity of Unit Selected (Btu/hr)
< 6000	5950
6000 to 7999	7900
8000 to 13999	13900
14000 to 19999	19900
> 20000	28000

Because there are physical limitations on the size of heat exchanger coils that can be used to improve RACs, choosing a unit near the top of the capacity range for a particular class provides for the most conservative estimate of the improvements in efficiency that can be achieved in that class. For example, in the 6000 to 7999 Btu/hr class, any unit whose capacity is below 7900 Btu/hr should be able to exceed the maximum efficiency developed for the 7900 Btu/hr unit if heat exchanges and compressors of comparable performance are used for the smaller units. This same strategy was used in developing the maximum technologically feasible designs for central air conditioners and heat pumps in an earlier analysis. [1,2].

Baseline units were developed with efficiencies near the bottom of those available in 1987 for each class. Starting with a low efficiency unit allowed for design changes to be applied such that the whole range in RAC efficiencies could be examined for each class.

The next step was a test of the influence of important variables, such as heat exchanger frontal area, tube rows,air flow, etc. on the performance of the baseline unit. These analyses allowed for an optimization of the overall performance of the RAC designs. The process follows very closely that done for heat pumps[3].

The next step was to develop a set of "cabinets" in which to put the room air conditioners. While a given manufacturer may offer dozens of different RAC models, typically these models will all be packaged in three or four cabinets. Our approach to defining a cabinet was to specify the largest coils (evaporator and condenser) that could be placed into that cabinet. A total of four cabinets were chosen (Table 7.2). The maximum heat exchanger sizes are specified for each. From data

Table 7.2 - Maximum frontal area of heat exchangers for each cabinet.

Cabinet Model	Frontal Area (ft^2) Evaporator	Condenser
A	0.75	1.00
B	1.25	2.00
C	1.50	2.50
D	2.00	3.10

provided by RAC manufacturers, the ratio of the evaporator to condenser frontal area varied from 0.55 to 0.77. The relative sizes shown in Table 7.2 are within that range.

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 design. While this was not done for this analysis, the design can be updated when costing data are developed.

Besides the cabinet sizes, there were several design restrictions that were used throughout the analysis (Table 7.3) First, the sensible heating factor (SHF) for the RACs was maintained below 0.75 to provide adequate dehumidification. The larger the SHF, the smaller the portion of the cooling is dedicated to dehumidification. A second restriction was on the fin density. The evaporator coil was restricted to 14 fins per inch (fpi) while the condenser was restricted to 19 fpi. In theory, higher fin densities are possible on the evaporator, however, from discussions with manufacturers, 14 fpi was the smallest fin spacing which allowed the condensate to freely run off the coil.

Table 7.3 - Restrictions used for the RAC designs

Item	Value
Sensible Heating Factor	$\leq$0.80
Fin Density	
Evaporator	$\leq$14 fpi
Condenser	$\leq$19 fpi
Coil Rows	
Evaporator	5
Condenser	4

Baseline Units

Baseline units were selected for the RACs with side louvers. These are the most prevalant units sold on the market. The baseline units are typical of the lower efficiency and lower priced units sold in 1987. A detailed description of the components and performance of the RACs is provided in Tables 7.4 and 7.5. Key features of the units include:

* Capacity and EER are based on the standard 95 degree outdoor test.

* 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 50%. The efficiency is the ratio of the shaft output to the electrical input.

* A propeller fan is used in the condenser and a centrifugal forward curved fan in the evaporator.

* Because all RACs are packaged systems, a penalty in the form of reduced efficiency for fans was used to account for poorer airflow due to the cabinet.

* The units without side louvers had a 4% power penalty added to the power calculated for the louvered units.

Table 7.4 - Hardware data for the baseline RACs.

Capacity (Btu/hr)	5950	7900	13900	19900	28000
COMPRESSOR	B1	B1	A1	A1	A2
Displacement (in^3)	0.73	1.13	2.48	3.58	4.62
EER	9.2	9.2	8.9	8.9	8.9
CONDENSER					
Frontal Area (ft^2)	0.90	1.00	2.00	2.30	2.50
No. of Rows	2	2	2	3	3
Parallel Circuits	1.0	1.0	2.0	2.0	3.0
Fins/Inch	14	14	14	15	15
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	0.70
Hor. Space (in)	0.625	0.625	0.625	0.625	0.80
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow (CFM)	300	400	600	900	1100
Fan & Motor Eff.	0.10	0.10	0.10	0.10	0.10
EVAPORATOR					
Frontal Area (ft^2)	0.60	0.75	1.20	1.20	1.50
No. of Rows	2	2	3	4	3
Parallel Circuits	1.0	1.0	2.0	3.0	3.0
Fins/Inch	14	14	12	12	12
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	0.70
Hor. Space (in)	0.625	0.625	0.625	0.625	0.80
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow	150	200	340	445	600
Fan & Motor Eff.	0.15	0.15	0.15	0.15	0.15
MISCELLANEOUS					
Subcooler Size (in)	12	12	24	24	60
Cap. Tubes					
Number	1	1	2	2	2
Diameter (in)	0.044	0.054	0.049	0.054	0.064
Length (in)	40	40	30	30	35
Cabinet Model	A	A	B	B	C

Table 7.5 - Performance data for the side louvered and non-side louvered RACs.

Model	5950	7900	13900	19900	28000
WITH SIDE LOUVERS					
EER (Btu/W-hr)	6.2	6.0	6.0	5.9	5.9
SHR	0.66	0.65	0.67	0.66	0.67
W/OUT SIDE LOUVERS					
EER (Btu/W-hr)	6.0	5.8	5.8	5.7	5.7
SHR	0.66	0.65	0.67	0.66	0.67

The first item in Table 7.4 is the capacity of the unit used for a specific class. The units for the five classes range in size from 5950 Btu/hr to 28000 Btu/hr.

The next item in Table 7.4 is a description of the compressor: model designation, its displacement and its rated energy efficiency ratio. The letter in the model designation of the compressor is a code for the compressor manufacturer. The number represents a specific compressor manufactured by the manufacturer. The displacement of the compressor is the adjusted displacement to meet the specific requirements for the model line. The baseline compressors are units that were on the market in late 1987. The energy efficiency ratio (EER) of the compressors are for the conditions shown in Table 7.6.

Table 7.6 - Rating conditions for the compressor EER and capacity.

Condition	Value
Evaporating 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 7.4 include descriptions of the indoor and outdoor heat exchangers. Major items include: the face area of the heat exchangers, fin thickness and spacing, tube descriptions, and fans. The last section includes details on the subcooler, capillary tubes and cabinet designation (See Table 7.2).

The performance information in Table 7.5 includes both the rated energy efficiency ratio (EER) and the sensible heat ratio (SHR). The EER is at an outdoor temperature of 95 F and an indoor dry-bulb temperature of 80 F and wet-bulb temperature of 67 F. The SHR is the ratio of the sensible cooling to total cooling capacity of the unit at the rated conditions. The baseline units represent units that are close to those that are lowest in efficiency on the market. For example, in the louvered class for capacities less than 6000 Btu/hr, the baseline unit above has an EER of 6.2 compared to 5.6 for the minimum units available in 1987 in that class.

Final Designs

A line of RACs for each class was developed to cover the efficiency spectrum from the lowest to the highest value that was viewed to be technologically achievable in that class. Each class is discussed separately below.

6000 Btu/hr and Smaller Class

This class has the most severe physical size restrictions of all the classes. One of the reasons for creating a class of RACs with capacities less than 6000 Btu/hr is that these units are designed to be lightweight and portable. The portability adds utility to that class of units that is not available in the other classes. In order to retain portability in this class of units, only cabinet size A (Evaporator size of 0.75 sq. ft. and Condenser size of 1.0 sq. ft) was considered for these units. While larger cabinets could have been used in conjunction with larger heat exchangers to push up efficiencies, these would have removed the portability of this class. To limit the weight on these units, a maximum of 3 row coils was used for evaporators and 2 row coils for condensers.

Tables 7.7 and 7.8 provide the detailed hardware and performance data, respectively, for the units in this class. Each unit has an alpha-numberic designation. The number in the designation specifies the capacity rounded off to the nearest thousand Btu/hr. The letter following the number is used to specify the model within the line. The baseline units has the letter "A" for its model specification. The next model in the line has a "B", etc.

Table 7.7 - Hardware data for the 5950 Btu/hr units

Model	6A	6B	6C	6D	6E	6F
COMPRESSOR	B1	B1	B1	B1	B1	B1
Displacement (in^3)	0.73	0.69	0.65	0.58	0.59	0.57
EER	9.2	9.2	9.2	9.2	9.2	9.2
DESIGN OPTIONS		1B,2	1B,1C,1E 2	1B,1C,1E 2	1B,1C,1E 2,3	1B,1D,1E 2,3
CONDENSER						
Frontal Area (sq ft)	0.90	0.90	0.90	0.90	0.90	0.90
No. of Rows	2	2	2	2	2	2
Parallel Circuits	1.0	1.0	1.0	1.0	1.0	1.0
Fins/Inch	14	14	17	17	17	14
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow (CFM)	300	300	350	350	350	320
Fan & Motor Eff.	0.10	0.10	0.10	0.10	0.13	0.13
Internal Fins	NO	NO	NO	NO	NO	YES
EVAPORATOR						
Frontal Area (sq ft)	0.60	0.60	0.60	0.60	0.60	0.60
No. of Rows	2	3	3	3	3	3
Parallel Circuits	1.0	1.0	1.0	1.0	1.0	1.0
Fins/Inch	14	12	12	14	14	14
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow	150	150	150	175	175	175
Fan & Motor Eff.	0.15	0.15	0.15	0.15	0.18	0.18
MISCELLANEOUS						
Subcooler Length(in)	12	12	24	24	24	24
Cap. Tubes						
Number	1	1	1	1	1	1
Diameter (in)	0.044	0.044	0.044	0.046	0.048	0.048
Length (in)	40	40	40	40	40	40
Cabinet Model	A	A	A	A	A	A

Table 7.7 - Continued
Hardware data for the 5950 Btu/hr units.

Model	6G	6H	6I	6J	6K
COMPRESSOR	C1	C1	C1	C1	C2
Displacement (in^3)	0.57	0.54	0.53	0.50	0.50
EER	10.2	10.2	10.2	10.2	10.6
DESIGN OPTIONS	1B, 1D, 1E 2, 3, 4	1A, 1B, 1D 1E, 2, 3, 4	1A, 1B, 1D 1E, 2, 3, 4	1A, 1B, 1D 1E, 2, 3, 4	1A, 1B, 1D 1E, 2, 3, 4
CONDENSER					
Frontal Area (sq ft)	0.90	0.90	0.90	1.00	1.00
No. of Rows	2	2	2	2	2
Parallel Circuits	1.0	1.0	1.0	1.0	1.0
Fins/Inch	14	14	14	14	14
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	LOUVERED	LOUVERED	LOUVERED
Airflow (CFM)	320	320	340	375	375
Fan & Motor Eff.	0.13	0.13	0.15	0.15	0.15
Internal Fins	YES	YES	YES	YES	
EVAPORATOR					
Frontal Area (sq ft)	0.60	0.75	0.75	0.75	0.75
No. of Rows	3	3	3	4	4
Parallel Circuits	1.0	1.0	1.0	1.0	1.0
Fins/Inch	14	14	14	14	14
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow	175	175	185	175	175
Fan & Motor Eff.	0.18	0.18	0.20	0.20	0.20
MISCELLANEOUS					
Subcooler Length(in)	24	24	24	24	24
Cap. Tubes					
Number	1	1	1	1	1
Diameter (in)	0.052	0.052	0.052	0.052	0.052
Length (in)	40	40	40	40	40
Cabinet Model	A	A	A	A	A

Table 7.8 - Performance data on the 5950 Btu/hr units.

Model	6A	6B	6C	6D	6E	6F	6G
WITH SIDE LOUVERS							
EER (Btu/W-hr)	6.2	6.9	7.7	8.2	8.6	9.2	9.7
SHR	0.66	0.67	0.67	0.71	0.71	0.71	0.71
W/OUT SIDE LOUVERS							
EER (Btu/W-hr)	6.0	6.6	7.4	7.9	8.2	8.8	9.3
SHR	0.66	0.67	0.67	0.71	0.71	0.71	0.71

Table 7.8 - Continued

Model	6H	6I	6J	6K
WITH SIDE LOUVERS				
EER (Btu/W-hr)	10.4	10.9	11.6	12.1
SHR	0.72	0.74	0.74	0.74
W/OUT SIDE LOUVERS				
EER (Btu/W-hr)	10.0	10.5	11.1	11.6
SHR	0.72	0.74	0.74	0.74

The models are arranged in increasing EER from left to right in the tables. Thus, unit 6D has a higher EER than 6C.

Below the model designation are the list of design options used on the unit. These options are all relative to the basline system. The list of design options is in a code that corresponds with the list in Chapter 6. For instance, unit 6D has design options 1B, 1C, 1E, and 2. It has increased tube rows (Option 1B) on the evaporator, increased fin density (Option 1C) on both the evaporator and condenser, larger subcooler (Option 1E), and smaller compressor (Option 2).

The rest of the data in Tables 7.7 and 7.8 are the same data provided in the same order for the baseline units in Tables 7.4 and 7.5. Thus, all the details on the coils, fans, steady state performance, etc. is available on each unit.

The highest efficiency unit in the louvered class in 1987 had an EER of 9.0. With the design options, such as high efficiency compressors, optimized air flow and heat exchanger circuiting, the best louvered unit designed had an EER of 12.1 in this class. This unit was the maximum technological feasible (MTF) efficiency of units in this class. For all classes, the unit with the highest efficiency should be considered as the MTF unit. The highest efficiency unit in the non-louvered class was 8.7 in 1987 compared to the 11.6 estimated in Table 7.8. The minimum efficiency standards for RACs include EERs of 8.0 for both louvered and non-louvered units in this class.

6000 to 7999 Btu/hr Class

Tables 7.9 and 7.10 summarize the hardware and performance data for both the louvered and non-louvered classes in the 6000 to 7999 Btu/hr capacity range. All units were designed to fit into either cabinet style A and B. Thus, the largest coil face areas for these classes of units is 1.25 square feet for the evaporator and 1.75 square feet for the condenser. While larger heat exchanger surface areas could produce higher efficiencies, the larger size would make the unit unreasonably large for its class.

Even with the limitations on coil size, it was still possible to attain a maximum EER of 13.8 for louvered and 13.3 for non-louvered units. These compare to maximums available in 1987 of 10.2 and 9.0 for louvered and non-louvered units, respectively. Thus, it appears that there is still a large potential for improvement in efficiencies in these classes.

Table 7.9 - Hardware data for the 7900 Btu/hr units

Model	8A	8B	8C	8D	8E	8F
COMPRESSOR	B1	B1	B1	B1	B1	B1
Displacement (in^3)	1.08	0.96	0.90	0.83	0.83	0.84
EER	9.2	9.2	9.2	9.2	10.1	10.1
DESIGN OPTIONS		1B,2E,2	1B,1C,1E 2	1B,1C,1E 2	1B,1C,1E 2,4	1B,1C,1D 1E,2,3,4
CONDENSER						
Frontal Area (sq ft)	1.00	1.00	1.00	1.00	1.00	1.00
No. of Rows	2	2	2	3	3	3
Parallel Circuits	1.0	1.0	1.0	1.0	1.0	1.0
Fins/Inch	14	14	17	17	17	17
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow (CFM)	400	400	400	400	400	400
Fan & Motor Eff.	0.10	0.10	0.10	0.10	0.10	0.13
Internal Fins	NO	NO	NO	NO	NO	YES
EVAPORATOR						
Frontal Area (sq ft)	0.75	0.75	0.75	0.75	0.75	0.75
No. of Rows	2	3	3	3	3	4
Parallel Circuits	1.0	1.0	1.0	2.0	2.0	2.0
Fins/Inch	14	12	12	14	14	14
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow	200	200	220	220	220	220
Fan & Motor Eff.	0.15	0.15	0.15	0.15	0.15	0.18
MISCELLANEOUS						
Subcooler Length(in)	12	24	24	24	24	24
Cap. Tubes						
Number	1	1	1	1	1	1
Diameter (in)	0.054	0.056	0.058	0.058	0.060	0.060
Length (in)	40	40	40	40	40	40
Cabinet Model	A	A	A	A	A	A

Table 7.9 - Continued
Hardware data for the 7900 Btu/hr units

Model	8G	8H	8I	8J	8K	8L
COMPRESSOR	B2	B2	B2	B3	B3	B3
Displacement (in^3)	0.75	0.74	0.69	0.69	0.72	0.62
EER	10.1	10.1	10.6	10.6	10.6	10.6
DESIGN OPTIONS	1B,1C,1D 1E,2,3,4	1B,1D,1E 2,3,4	1B,1D,1E 2,3,4	1A,1B,1D 1E,2,3,4	1A,1B,1D 1E,2,3,4	1A,1B,1D 1E,2,3,4
CONDENSER						
Frontal Area (sq ft)	1.00	1.00	1.00	1.00	1.00	1.00
No. of Rows	3	3	3	2	2	3
Parallel Circuits	1.0	1.0	1.0	1.0	1.0	1.5
Fins/Inch	17	14	14	14	14	14
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	LOUVERED	LOUVERED	LOUVERED	LOUVERED	LOUVERED
Airflow (CFM)	400	375	375	375	375	375
Fan & Motor Eff.	0.13	0.13	0.15	0.15	0.15	0.15
Internal Fins	YES	YES	YES	YES	YES	YES
EVAPORATOR						
Frontal Area (sq ft)	0.75	0.75	0.75	1.25	1.25	1.25
No. of Rows	4	4	5	3	3	4
Parallel Circuits	2.0	2.0	2.0	2.0	2.0	2.0
Fins/Inch	14	14	14	14	14	14
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow	220	220	220	220	220	230
Fan & Motor Eff.	0.18	0.18	0.20	0.20	0.20	0.20
MISCELLANEOUS						
Subcooler Length(in)	24	36	36	36	36	36
Cap. Tubes						
Number	1	1	1	1	1	1
Diameter (in)	0.060	0.060	0.060	0.060	0.060	0.060
Length (in)	40	40	40	40	40	40
Cabinet Model	A	A	A	A	A	A

Table 7.10 - Performance data on the 7900 Btu/hr units.

Model	8A	8B	8C	8D	8E	8F	8G
WITH SIDE LOUVERS							
EER (Btu/W-hr)	6.0	7.0	7.6	8.1	8.5	9.0	9.8
SHR	0.65	0.67	0.68	0.69	0.70	0.68	0.71
W/OUT SIDE LOUVERS							
EER (Btu/W-hr)	5.8	6.7	7.3	7.8	8.2	8.6	9.4
SHR	0.65	0.67	0.68	0.69	0.70	0.68	0.71

Table 7.10 - Continued

Model	8H	8I	8J	8K	8L
WITH SIDE LOUVERS					
EER (Btu/W-hr)	10.3	10.9	11.5	12.4	13.8
SHR	0.71	0.73	0.73	0.70	0.74
W/OUT SIDE LOUVERS					
EER (Btu/W-hr)	9.9	10.5	11.0	11.9	13.3
SHR	0.71	0.73	0.73	0.70	0.74

8000 to 13999 Btu/hr Classes

Tables 7.11 and 7.12 summarize the hardware and performance data for both the louvered and non-louvered classes in the 8000 to 13999 Btu/hr capacity range. All units were designed to fit into either cabinet style B and C. The largest coil face areas for these classes of units is 1.50 square feet for the evaporator and 2.5 square feet for the condenser. These heat exchanger sizes are compared to the higher efficiency units currently on the market.

The maximum EERs calculated for this capacity were 12.4 for louvered and 11.9 for non-louvered classes, respectively. These values compare to maximums available in 1987 of 12.0 and 10.0 for louvered and non-louvered units, respectively.

Table 7.11 - Hardware data for the 13900 Btu/hr units.

Model	14A	14B	14C	14D	14E	14F
COMPRESSOR	A1	A1	A1	A1	A3	A3
Displacement (in^3)	2.43	2.31	1.94	1.85	1.83	1.64
EER	8.9	8.9	8.9	8.9	9.8	9.8
DESIGN OPTIONS		1C, 1E 2	1B, 1C, 1E 2	1B, 1C, 1E 2, 3	1B, 1C, 1E 2, 3, 4	1B, 1C, 1E 2, 3, 4
CONDENSER						
Frontal Area (sq ft)	2.00	2.00	2.00	2.00	2.00	2.00
No. of Rows	2	2	2	3	3	3
Parallel Circuits	2.0	2.0	2.0	2.0	2.0	2.0
Fins/Inch	14	17	17	17	17	17
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow (CFM)	600	600	600	600	600	600
Fan & Motor Eff.	0.10	0.10	0.10	0.13	0.13	0.13
Internal Fins	NO	NO	NO	NO	NO	NO
EVAPORATOR						
Frontal Area (sq ft)	1.20	1.20	1.20	1.20	1.20	1.20
No. of Rows	3	3	4	4	4	5
Parallel Circuits	2.0	2.0	2.0	2.0	2.0	2.5
Fins/Inch	14	13	13	13	13	13
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow	340	340	340	360	360	370
Fan & Motor Eff.	0.15	0.15	0.15	0.18	0.18	0.18
MISCELLANEOUS						
Subcooler Length(in)	24	36	42	48	48	54
Cap. Tubes						
Number	2	2	2	2	2	2
Diameter (in)	0.049	0.049	0.049	0.049	0.049	0.050
Length (in)	30	30	30	30	30	30
Cabinet Model	B	B	B	B	B	B

Table 7.11 - Continued
Hardware data for the 5950 Btu/hr units.

Model	14G	14H	14I	14K	14L	14M
COMPRESSOR	A3	C1	C1	C1	C2	C2
Displacement (in^3)	1.56	1.24	1.20	1.22	1.17	1.12
EER	9.8	10.4	10.4	10.4	10.8	10.8
DESIGN OPTIONS	1B,1C,1D 1E,2,3,4	1B,1C,1D 1E,2,3,4	1B,1D,1E 2,3,4	1A,1B,1D 1E,2,3,4	1A,1B,1D 1E,2,3,4	1A,1B,1D 1E,2,3,4
CONDENSER						
Frontal Area (sq ft)	2.00	2.00	2.00	2.35	2.50	2.50
No. of Rows	4	4	3	3	3	3
Parallel Circuits	1.5	1.5	1.5	1.5	1.5	1.5
Fins/Inch	17	17	14	14	14	14
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	LOUVERED	LOUVERED	LOUVERED	LOUVERED
Airflow (CFM)	650	650	700	800	800	835
Fan & Motor Eff.	0.13	0.13	0.15	0.15	0.15	0.15
Internal Fins	YES	YES	YES	YES	YES	YES
EVAPORATOR						
Frontal Area (sq ft)	1.20	1.20	1.20	1.40	1.50	1.50
No. of Rows	5	5	5	4	4	5
Parallel Circuits	2.5	2.5	2.5	2.5	2.5	2.6
Fins/Inch	13	13	13	13	13	13
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow	370	370	385	395	405	395
Fan & Motor Eff.	0.18	0.18	0.20	0.20	0.20	0.20
MISCELLANEOUS						
Subcooler Length(in)	54	54	60	60	60	60
Cap. Tubes						
Number	2	2	2	2	2	2
Diameter (in)	0.051	0.050	0.050	0.050	0.050	0.050
Length (in)	30	30	30	30	30	30
Cabinet Model	B	B	B	B	B	B

Table 7.12 - Performance data on the 13900 Btu/hr units.

Model	14A	14B	14C	14D	14E	14F	14G
WITH SIDE LOUVERS							
EER (Btu/W-hr)	6.0	6.5	7.3	7.8	8.6	9.2	9.6
SHR	0.67	0.66	0.69	0.69	0.69	0.71	0.71
W/OUT SIDE LOUVERS							
EER (Btu/W-hr)	5.8	6.2	7.0	7.5	8.2	8.8	9.2
SHR	0.67	0.66	0.69	0.69	0.69	0.71	0.71

Table 7.12 - Continued

Model	14H	14I	14J	14K	14L	14M
WITH SIDE LOUVERS						
EER (Btu/W-hr)	10.1	10.5	11.0	11.5	12.0	12.4
SHR	0.71	0.73	0.71	0.73	0.74	0.74
W/OUT SIDE LOUVERS						
EER (Btu/W-hr)	9.7	10.1	10.6	11.1	11.5	11.9
SHR	0.71	0.73	0.71	0.73	0.74	0.74

14000 to 19999 Btu/hr Classes

Tables 7.13 and 7.14 summarize the hardware and performance data for both the louvered and non-louvered classes in the 14000 to 19999 Btu/hr capacity range. All units were designed to fit into either cabinet styles B, C or D. The largest coil face areas for these classes of units are 1.90 square feet for the evaporator and 3.10 square feet for the condenser.

Even with the limitations on coil size, it was still possible to attain a maximum EER of 11.9 for louvered and 11.4 for non-louvered units. These values compare to maximums available in 1987 of 10.2 and 10.0 for louvered and non-louvered units, respectively. Thus, it appears that there is still a potential for improvement in efficiencies in these classes.

Table 7.13 - Hardware data for the 19,900 Btu/hr units.

Model	20A	20B	20C	20D	20E	20F
COMPRESSOR	A1	A1	A1	A1	A3	A3
Displacement (in^3)	3.50	3.32	2.79	2.58	2.55	2.41
EER	8.9	8.9	8.9	8.9	9.8	9.8
DESIGN OPTIONS		1A,1E,2	1A,1B,1C 1E,2,3	1A,1B,1C 1E,2,3	1A,1B,1C 1E,2,3,4	1A,1B,1C 1E,2,3,4
CONDENSER						
Frontal Area (sq ft)	2.30	2.50	2.50	2.50	2.50	3.00
No. of Rows	3	3	3	3	3	2
Parallel Circuits	2.0	2.0	2.0	2.0	2.0	1.5
Fins/Inch	15	15	17	17	17	17
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow (CFM)	900	900	900	900	900	900
Fan & Motor Eff.	0.10	0.10	0.10	0.13	0.13	0.13
Internal Fins	NO	NO	NO	NO	NO	NO
EVAPORATOR						
Frontal Area (sq ft)	1.20	1.40	1.45	1.50	1.50	1.90
No. of Rows	4	4	5	5	5	4
Parallel Circuits	3.0	3.0	3.0	3.0	3.0	3.0
Fins/Inch	14	12	12	14	14	14
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow	445	445	480	490	490	535
Fan & Motor Eff.	0.15	0.15	0.15	0.18	0.18	0.18
MISCELLANEOUS						
Subcooler Length(in)	24	48	60	72	72	84
Cap. Tubes						
Number	2	2	2	2	2	2
Diameter (in)	0.054	0.054	0.054	0.054	0.054	0.054
Length (in)	30	30	30	30	30	30
Cabinet Model	C	C	C	C	C	D

Table 7.13 - Continued
Hardware data for the 19900 Btu/hr units.

Model	20G	20H	20I	20K	20L
COMPRESSOR	A3	C1	C1	C1	C2
Displacement (in^3)	2.33	1.79	1.78	1.67	1.67
EER	9.8	10.3	10.3	10.3	10.8
DESIGN OPTIONS	1A,1B,1C 1E,2,3,4	1A,1B,1C 1E,2,3,4	1A,1B,1D 1E,2,3,4	1A,1B,1D 1E,2,3,4	1A,1B,1C 1D,1E,2 3,4
CONDENSER					
Frontal Area (sq ft)	3.00	3.00	3.00	3.00	3.00
No. of Rows	3	3	3	3	2.5
Parallel Circuits	1.5	1.5	1.5	2.0	2.0
Fins/Inch	17	17	14	14	15
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	LOUVERED	LOUVERED	LOUVERED
Airflow (CFM)	1000	1000	1000	1000	950
Fan & Motor Eff.	0.15	0.15	0.15	0.15	0.15
Internal Fins	NO	NO	YES	YES	YES
EVAPORATOR					
Frontal Area (sq ft)	1.90	1.90	1.90	1.90	2.00
No. of Rows	4	4	4	5	5
Parallel Circuits	3.0	3.0	3.0	3.5	3.5
Fins/Inch	14	14	14	14	14
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	1.0	1.0	1.0	1.0	1.0
Hor. Space (in)	0.625	0.625	0.625	0.625	0.625
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow	535	535	535	545	535
Fan & Motor Eff.	0.20	0.20	0.20	0.20	0.20
MISCELLANEOUS					
Subcooler Length(in)	96	108	108	108	108
Cap. Tubes					
Number	2	2	2	2	2
Diameter (in)	0.054	0.058	0.059	0.060	0.059
Length (in)	30	30	30	30	30
Cabinet Model	D	D	D	D	D

Table 7.14 - Performance data on the 19,900 Btu/hr units.

Model	20A	20B	20C	20D	20E	20F	20G
WITH SIDE LOUVERS							
EER (Btu/W-hr)	5.9	6.4	7.0	7.6	8.4	9.0	9.4
SHR	0.66	0.65	0.69	0.68	0.69	0.69	0.70
W/OUT SIDE LOUVERS							
EER (Btu/W-hr)	5.7	6.1	6.8	7.3	8.0	8.7	9.1
SHR	0.66	0.65	0.69	0.68	0.69	0.69	0.70

Table 7.14 - Continued

Model	20H	20I	20J	20K
WITH SIDE LOUVERS				
EER (Btu/W-hr)	10.2	10.6	11.3	11.9
SHR	0.70	0.70	0.72	0.72
W/OUT SIDE LOUVERS				
EER (Btu/W-hr)	9.8	10.2	10.9	11.4
SHR	0.70	0.70	0.72	0.72

20000 Btu/hr and Greater Classes

Tables 7.15 and 7.16 summarize the hardware and performance data for both the louvered and non-louvered classes that are greater than 20000 Btu/hr in capacity. All units were designed to fit into either cabinet styles C or D. The largest coil face areas for these classes of units are 2.00 square feet for the evaporator and 3.10 square feet for the condenser.

The maximum EERs calculated for this capacity were 10.8 for louvered and 10.4 for non-louvered classes, respectively. These values compare to maximums available in 1987 of 9.3 and 8.0 for louvered and non-louvered units, respectively. As with the other classes, units with higher efficiencies could have been designed. However, using larger coils would have created heavy units that would be difficult to use in window applications.

Table 7.15 - Hardware data for the 28000 Btu/hr units.

Model	28A	28B	28C	28D	28E
COMPRESSOR	A2	A2	A3	A3	A3
Displacement (in^3)	4.62	3.65	3.60	3.45	3.34
EER	8.9	8.9	9.8	9.8	9.8
DESIGN OPTIONS		1B,1E,2	1B,1E,2 4	1A,1B,1E 2,4	1A,1B,1C 1E,2,4
CONDENSER					
Frontal Area (sq ft)	2.50	2.50	2.50	2.80	3.10
No. of Rows	3	4	4	4	4
Parallel Circuits	3.0	3.0	3.0	3.0	3.0
Fins/Inch	15	15	15	15	15
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	0.70	0.70	0.70	0.70	0.70
Hor. Space (in)	0.80	0.80	0.80	0.80	0.80
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow (CFM)	1100	1200	1200	1200	1200
Fan & Motor Eff.	0.10	0.10	0.10	0.10	0.10
Internal Fins	NO	NO	NO	NO	NO
EVAPORATOR					
Frontal Area (sq ft)	1.50	1.50	1.50	1.80	1.90
No. of Rows	3	4	4	4	4
Parallel Circuits	3.0	4.0	4.0	3.5	3.5
Fins/Inch	12	12	12	12	13
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	0.70	0.70	0.70	0.70	0.70
Hor. Space (in)	0.80	0.80	0.80	0.80	0.80
Fin Design	WAVY	WAVY	WAVY	WAVY	WAVY
Airflow	600	650	650	650	625
Fan & Motor Eff.	0.15	0.15	0.15	0.15	0.15
MISCELLANEOUS					
Subcooler Length(in)	60	72	72	72	96
Cap. Tubes					
Number	2	2	2	2	2
Diameter (in)	0.064	0.064	0.064	0.064	0.064
Length (in)	35	35	35	35	35
Cabinet Model	C	C	C	D	D

Table 7.15 - Continued
Hardware data for the 28000 Btu/hr units.

Model	28F	28G	28H	28I
COMPRESSOR	A3	A4	A4	A4
Displacement (in^3)	3.17	3.14	3.10	3.04
EER	9.8	10.4	10.4	10.4
Design Options	1A, 1B, 1C 1E, 2, 3, 4	1A, 1B, 1C 1E, 2, 3, 4	1A, 1B, 1C 1E, 2, 3, 4	1A, 1B, 1C 1D, 1E, 2 3, 4
CONDENSER				
Frontal Area (sq ft)	3.10	3.10	3.10	3.10
No. of Rows	4	4	3.5	2.0
Parallel Circuits	3.0	3.0	3.0	2.0
Fins/Inch	15	15	15	14
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	0.70	0.70	0.70	0.70
Hor. Space (in)	0.80	0.80	0.80	0.80
Fin Design	WAVY	WAVY	WAVY	LOUVERED
Airflow (CFM)	1150	1150	1150	1200
Fan & Motor Eff.	0.13	0.13	0.15	0.15
Internal Fins	NO	NO	NO	YES
EVAPORATOR				
Frontal Area (sq ft)	2.0	2.0	2.0	2.0
No. of Rows	5	5	5	5
Parallel Circuits	4.0	4.0	4.5	4.5
Fins/Inch	13	13	13	13
Fin Thickness (in)	0.0055	0.0055	0.0055	0.0055
O.D. of Tubes	0.3125	0.3125	0.3125	0.3125
I.D. of Tubes	0.2865	0.2865	0.2865	0.2865
Vert. Space (in)	0.70	0.70	0.70	0.70
Hor. Space (in)	0.80	0.80	0.80	0.80
Fin Design	WAVY	WAVY	WAVY	WAVY
Airflow	625	625	625	630
Fan & Motor Eff.	0.18	0.18	0.20	0.20
MISCELLANEOUS				
Subcooler Length(in)	110	110	120	120
Cap. Tubes				
Number	2	2	2	2
Diameter (in)	0.064	0.064	0.064	0.064
Length (in)	35	35	35	35
Cabinet Model	D	D	D	D

Table 7.16 - Performance data on the 28,000 Btu/hr units.

Model	28A	28B	28C	28D	28E	28F	28G
WITH SIDE LOUVERS							
EER (Btu/W-hr)	5.9	6.6	7.5	8.1	8.6	9.2	9.8
SHR	0.67	0.66	0.69	0.69	0.69	0.71	0.71
W/OUT SIDE LOUVERS							
EER (Btu/W-hr)	5.7	6.3	7.2	7.8	8.3	8.9	9.4
SHR	0.67	0.66	0.69	0.69	0.69	0.71	0.71

Table 7.16 - Continued

Model	28H	28I
WITH SIDE LOUVERS		
EER (Btu/W-hr)	10.1	10.8
SHR	0.71	0.73
W/OUT SIDE LOUVERS		
EER (Btu/W-hr)	9.7	10.4
SHR	0.71	0.73

REFERENCES

1. D. O'Neal, C. Boecker, and S. Penson, "An Analysis of Efficiency Improvements in Residential Sized Heat Pump and Central Air Conditioners", ESL/87-03, Energy Systems Laboratory, Texas A & M University, August 1987.

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

3. D. O'Neal, C. Boecker, W. Murphy and J. Notman, "An Analysis of Efficiency Improvements in Residential Sized Heat Pumps", ESL/85-24, Energy Systems Laboratory, Texas A & M University, May 1986.

CHAPTER 8

CONCLUSIONS

The potential improvements that can be made in the efficiency of room air conditioners has been considered in this study. A performance model for RACs has been developed and used in estimating these efficiency improvements. Major conclusions are discussed below.

Conclusions

The changes made in the Oak Ridge Heat Pump Model allowed us to model the performance of RACs. These changes included the major characteristics of RACs that are different from heat pumps: subcoolers, water spray on the condenser, and capillary tubes. The new model appears to simulate system performance adequately as demonstrated from the limited validation done in this study.

Major improvements in the efficiency of room air conditioners have occurred over the past 15 years (Figure 2.4). These improvements have resulted in an average efficiency increase of almost 30%. While these improvements are substantial, this study has examined designs that should push efficiencies in all classes of room air conditioners beyond the best that are available today. The maximum technologically feasible units in each class are shown in Table 8.1. The strategy for developing the MTF units involves better heat transfer surfaces such as internally groved tubes and louvered fins, higher efficiency compressors, and optimized system design. Manufacturers are currently using some of these options in some of their units.

The minimum efficiency standards passed by Congress for the different RACs should pose no significant technological challenges to RAC manufacturers. It appears from the analysis of efficiency options available today, that manufacturers should readily be able to build units that will meet the efficiency standards in 1990.

Table 8.1 - Maximum Technologically Feasible (MTF) EERs for Room Air Conditioners by Class.

Size Class (Btu/hr)	MTF EER	
	Louvered	Non-louvered
< 6000	12.1	11.6
6000 - 7999	13.8	13.3
8000 - 13999	12.4	11.9
14000 - 19999	11.9	11.4
20000 +	10.8	10.4

APPENDIX A

ORNL AND ADL

MODEL OUTPUT

ORNL OUTPUT

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**** INPUT DATA *****

      A8670 btu/hr comp, 7900 RAC

      SUMMARY OUTPUT
      COOLING MODE OF OPERATION
      REFRIGERANT CHARGE IS NOT SPECIFIED

      COMPRESSOR INLET SUPERHEAT IS SPECIFIED AT   10.00 F
       1 CAPILLARY TUBE(S) USED TO REGULATE FLOW
      LENGTH OF THE CAPILLARY TUBE(S)        40.0000        DIAMETER OF THE CAPILLARY TUBE(S)       0.0600
       TUBE IN CONDENSATE SUBCOOLER
      INSIDE DIAMETER OF THE TUBE     0.2865IN          OUTSIDE DIAMETER OF THE TUBE           0.3125IN
      LENGTH OF SUBCOOLER                  48.000IN     TEMPERATURE OF THE CONDENSATE            90.0F
       ESTIMATE OF:
        SATURATION TEMPERATURE INTO  COMPRESSOR   36.00 F
        SATURATION TEMPERATURE OUT OF COMPRESSOR  120.00 F
       COMPRESSOR CHARACTERISTICS:
        TOTAL DISPLACEMENT                    0.629 CUBIC INCHES
        GIVEN MOTOR SPEED                  3450.000 RPM
       MAP-BASED COMPRESSOR INPUT:
        POWER CONSUMPTION=    -1.950E-05*CONDENSING  TEMPERATURE**2 +  9.774E-03*CONDENSING  TEMPERATURE
                        + -6.930E-05*EVAPORATING TEMPERATURE**2 + -2.861E-03*EVAPORATING TEMPERATURE
                        +  7.958E-05*CONDENSING  TEMPERATURE*EVAPORATING TEMPERATURE + -3.405E-01
         MASS FLOW RATE=        1.925E-03*CONDENSING  TEMPERATURE**2 + -7.705E-01*CONDENSING  TEMPERATURE
                        +  3.508E-02*EVAPORATING TEMPERATURE**2 + -5.052E-01*EVAPORATING TEMPERATURE
                        +  3.493E-04*CONDENSING  TEMPERATURE*EVAPORATING TEMPERATURE +  1.366E+02
         BASE SUPERHEAT FOR COMPRESSOR MAP         20.000 F
        BASE DISPLACEMENT FOR COMPRESSOR MAP      0.751 CU IN
         SUPERHEAT CORRECTION TERMS (SET IN BLOCK DATA):
          SUCTION GAS HEATING FACTOR              0.330
          VOLUMETRIC EFFICIENCY CORRECTION FACTOR  0.750
         HEAT REJECTED FROM COMPRESSOR SHELL IS 0.050 TIMES THE COMPRESSOR POWER
       INDOOR UNIT: EVAPORATOR
        INLET AIR TEMPERATURE                    80.000 F        RELATIVE HUMIDITY                      0.50000
        AIR FLOW RATE                            220.00 CFM      COMBINED FAN - FAN MOTOR EFFICIENCY    0.15000
        ID OF EACH OF 6 EQUIVALENT DUCTS           8.00 IN       HOUSE LOAD TO BE MET BY INDOOR UNIT    19000.0 BTU/H
        FRONTAL AREA OF HX                        0.750 SQ FT    WAVY FINS

        NUMBER OF TUBES IN DIRECTION OF AIR FLOW    3.00         FIN PITCH                                14.00 FINS/IN
        NUMBER OF PARALLEL CIRCUITS                 2.00         FIN THICKNESS                          0.00550 IN
        OD OF TUBES IN HX                        0.31250 IN      THERMAL CONDUCTIVITY: FINS              128.30 BTU/H-FT
        ID OF TUBES IN HX                        0.28650 IN      THERMAL CONDUCTIVITY: TUBES             225.00 BTU/H-FT
         HORIZONTAL TUBE SPACING                   0.625 IN       FRACTION OF COMPUTED CONTACT CONDUCTANCE 100.000
        VERTICAL TUBE SPACING                      1.000 IN      NUMBER OF RETURN BENDS                   23.00
        CABINET PRESSURE DROP MULTIPLIER

      OUTDOOR UNIT: CONDENSER
       CONDENSATE IS BEING SPRAYED ON THE CONDENSER
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THE FLOWRATE OF THE WATER IS                         0.267 LBM/HR

  INLET AIR TEMPERATURE                       92.000 F        RELATIVE HUMIDITY                         0.40000
  AIR FLOW RATE                               400.00 CFM      COMBINED FAN - FAN MOTOR EFFICIENCY       0.10000
  FRONTAL AREA OF HX                           1.000 SQ FT    WAVY FINS

  NUMBER OF TUBES IN DIRECTION OF AIR FLOW    3.00            FIN PITCH                                 17.00 FINS/IN
  NUMBER OF PARALLEL CIRCUITS                 1.00            FIN THICKNESS                           0.00550 IN
  OD OF TUBES IN HX                         0.31250 IN        THERMAL CONDUCTIVITY: FINS               128.30 BTU/H-FT
  ID OF TUBES IN HX                         0.28650 IN        THERMAL CONDUCTIVITY: TUBES              225.00 BTU/H-FT
  HORIZONTAL TUBE SPACING                     0.625 IN        FRACTION OF COMPUTED CONTACT CONDUCTANCE 100.000
  VERTICAL TUBE SPACING                       1.000 IN        NUMBER OF RETURN BENDS                    26.00
  CABINET PRESSURE DROP MULTIPLIER
COMPRESSOR CAN HEAT LOSS ADDED TO AIR BEFORE CROSSING THE OUTDOOR COIL.
POWER TO THE INDOOR  FAN ADDED TO AIR AFTER  CROSSING THE INDOOR  COIL.
POWER TO THE OUTDOOR FAN ADDED TO AIR BEFORE CROSSING THE OUTDOOR COIL.
 HEAT GAIN IN SUCTION LINE                  100.0 BTU/H
HEAT LOSS IN DISCHARGE LINE                 100.0 BTU/H
HEAT LOSS IN LIQUID LINE                    100.0 BTU/H
 DESCRIPTION OF CONNECTING TUBING:
  LIQUID LINE FROM INDOOR TO OUTDOOR HEAT EXCHANGER
     ID                          0.50000 IN
     EQUIVALENT LENGTH              1.00 FT
  THE COMPRESSOR SUCTION LINE                        THE COMPRESSOR DISCHARGE LINE
     ID                          0.31250 IN             ID                          0.37500 IN
     EQUIVALENT LENGTH              2.50 FT             EQUIVALENT LENGTH              2.50 FT
 ITERATION TOLERANCES:
  AMBCON   0.200 F     CMPCON   0.050 BTU/LBM    TOLH    0.00100 BTU/LBM
  CNDCON   0.200 F     FLOCON   0.400 LBM/H      TOLS    0.00005 BTU/LBM-R
  EVPCON   0.500 F     CONMST   0.003 F
```

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

SYSTEM SUMMARY	REFRIGERANT TEMPERATURE	SATURATION TEMPERATURE	REFRIGERANT ENTHALPY	REFRIGERANT QUALITY	REFRIGERANT PRESSURE	AIR TEMPERATU
COMPRESSOR SUCTION LINE INLET	44.999 F	39.781 F	109.008 BTU/LBM	1.0000	82.888 PSIA	
SHELL INLET	49.466	39.466	109.860	1.0000	82.433	
SHELL OUTLET	190.289	115.326	129.174	1.0000	258.516	
CONDENSER INLET	186.189 F	115.303 F	128.323 BTU/LBM	1.0000	258.439 PSIA	92.000
OUTLET	106.646	113.891	41.371	0.257	253.68	110.396
EXPANSION DEVICE	106.646 F	113.879 F	40.520 BTU/LBM	0.0000	253.680 PSIA	
EVAPORATOR INLET	40.978 F	40.978 F	40.520 BTU/LBM	0.2175	84.637 PSIA	80.000
OUTLET	44.999	39.781	109.008	1.0000	82.888	57.123

COMPRESSOR PERFORMANCE

		EFFICIENCY	
COMPRESSOR MOTOR POWER	0.699 KW	OVERALL ISENTROPIC	0.6217
REFRIGERANT MASS FLOW RATE	117.405 LBM/H	VOLUMETRIC	0.8482
MOTOR SPEED	3450.000 RPM	AT A PRESSURE RATIO OF	3.136
COMPRESSOR SHELL HEAT LOSS	119.350 BTU/H	SUPERHEAT CORRECTION TERMS	
		POWER	0.9944
		MASS FLOW RATE	1.0182

PERFORMANCE OF EACH CIRCUIT IN THE CONDENSER

INLET AIR TEMPERATURE	92.000 F
HEAT LOSS FROM COMPRESSOR	119.4 BTU/H
HEAT GENERATED FROM FAN	318.1 BTU/H
AIR TEMPERATURE ENTERING COIL	93.029 F
OUTLET AIR TEMPERATURE	110.396 F
TOTAL HEAT EXCHANGER EFFECTIVENESS	0.8049

	SUPERHEATED REGION	TWO-PHASE REGION	SUBCOOLED REGION
NTU	0.0000	1.6342	0.0000
HEAT EXCHANGER EFFECTIVENESS	1.0000	0.8049	1.0000
CR/CA	0.0000		0.0000
FRACTION OF HEAT EXCHANGER	0.0000	1.0000	0.0000
HEAT TRANSFER RATE	0.0 BTU/H	7386.0 BTU/H	0.0 BTU/H
OUTLET AIR TEMPERATURE	93.029 F	110.396 F	93.029 F

AIR SIDE:		REFRIGERANT SIDE:	
MASS FLOW RATE	1726.4 LBM/H	MASS FLOW RATE	117.4 LBM/H
PRESSURE DROP	0.1983 IN H2O	PRESSURE DROP	4.720 PSI
HEAT TRANSFER COEFFICIENT	26.676 BTU/H-SQ FT-F	HEAT TRANSFER COEFFICIENT	
		VAPOR REGION	118.771 BTU/H-SQ FT-F
		TWO PHASE REGION	707.623 BTU/H-SQ FT-F
		SUBCOOLED REGION	129.235 BTU/H-SQ FT-F

CONTACT INTERFACE:
CONTACT CONDUCTANCE 182100.9 BTU/H-SQ FT-F

UA VALUES PER CIRCUIT:

VAPOR REGION (BTU/H-F)		TWO PHASE REGION (BTU/H-F)		SUBCOOLED REGION (BTU/H-F)	
REFRIGERANT SIDE	0.000	REFRIGERANT SIDE	1910.724	REFRIGERANT SIDE	0.000
AIR SIDE	0.000	AIR SIDE	1116.731	AIR SIDE	0.000
CONTACT INTERFACE	0.000	CONTACT INTERFACE	49804.086	CONTACT INTERFACE	0.000

```
        COMBINED              0.000     COMBINED             694.970     COMBINED             0.000

PERFORMANCE OF EACH CIRCUIT IN THE EVAPORATOR
     INLET AIR TEMPERATURE              80.000 F
     AIR TEMPERATURE  LEAVING COIL      56.511 F
       HEAT GENERATED FROM FAN           145.7 BTU/H
     OUTLET AIR TEMPERATURE             57.123 F
      MOISTURE REMOVAL OCCURS
      SUMMARY OF DEHUMIDIFICATION PERFORMANCE (TWO-PHASE REGION)
                     LEADING EDGE         POINT WHERE MOISTURE
                      OF COIL                REMOVAL BEGINS
    LEAVING EDGE OF COIL
                        AIR              AIR              WALL
    AIR              WALL
     DRY BULB TEMPERATURE  80.000 F        80.000 F        56.336 F        55.972 F        47.987 F
     HUMIDITY RATIO       0.01092         0.01092         0.00965         0.00844         0.00707
     ENTHALPY             31.209 BTU/LBM  31.209 BTU/LBM  24.031 BTU/LBM  22.629 BTU/LBM  19.188 BTU/LBM
      RATE OF MOISTURE REMOVAL                              1.1507 LBM/H
     FRACTION OF EVAPORATOR THAT IS WET                     1.0000
     LATENT HEAT TRANSFER RATE IN TWO-PHASE REGION           1227. BTU/H
     SENSIBLE HEAT TRANSFER RATE IN TWO-PHASE REGION         2742. BTU/H
     SENSIBLE TO TOTAL HEAT TRANSFER RATIO FOR TWO-PHASE REGION  0.6910

     OVERALL SENSIBLE TO TOTAL HEAT TRANSFER RATIO          0.6951
      OVERALL CONDITIONS ACROSS COIL
                       ENTERING     EXITING
                         AIR          AIR
     DRY BULB TEMPERATURE   80.000 F     56.511 F
     WET BULB TEMPERATURE   66.460 F     54.463 F
     RELATIVE HUMIDITY       0.500        0.881
     HUMIDITY RATIO        0.01092      0.00855
      TOTAL HEAT EXCHANGER EFFECTIVENESS (SENSIBLE)    0.6236
                               SUPERHEATED      TWO-PHASE
                                 REGION          REGION
     NTU                         0.3445          1.0209
     HEAT EXCHANGER EFFECTIVENESS 0.2732         0.6397
     CR/CA                       2.1177
      FRACTION OF HEAT EXCHANGER  0.0413          0.9587
     HEAT TRANSFER RATE            53.9 BTU/H    3968.7 BTU/H
     AIR MASS FLOW RATE           20.03 LBM/H    465.28 LBM/H
     OUTLET AIR TEMPERATURE      69.022 F        55.972 F
      AIR SIDE:                                   REFRIGERANT SIDE:
       MASS FLOW RATE           485.3 LBM/H        MASS FLOW RATE               58.7 LBM/H
       PRESSURE DROP            0.248 IN H2O       PRESSURE DROP               1.749 PSI
       HEAT TRANSFER COEFFICIENT                   HEAT TRANSFER COEFFICIENT
         DRY COIL              9.994 BTU/H-SQ FT-F   VAPOR REGION             54.952 BTU/H-SQ FT-F
         WET COIL             11.104 BTU/H-SQ FT-F   TWO PHASE REGION        543.236 BTU/H-SQ FT-F
      CONTACT INTERFACE:
       CONTACT CONDUCTANCE    138659.1 BTU/H-SQ FT-F

     DRY FIN EFFICIENCY          0.867
     WET FIN EFFICIENCY (AVERAGE)  0.802
     WET CONTACT FACTOR (AVERAGE)  1.330
      UA VALUES PER CIRCUIT:VAPOR     TWO PHASE
                     REGION        REGION
       REFRIGERANT SIDE   2.297       527.362 BTU/H-F
       AIR SIDE
         DRY COIL         6.495         0.000 BTU/H-F
         WET COIL                     155.009 BTU/H-F
       CONTACT INTERFACE
         DRY COIL       749.116         0.000 BTU/H-F
         WET COIL                   17926.525 BTU/H-F
```

***** SUMMARY OF ENERGY INPUT AND OUTPUT *****

A8670 btu/hr comp, 7900 RAC

OPERATING CONDITIONS:

AIR TEMPERATURE INTO EVAPORATOR	80.00 F
AIR TEMPERATURE INTO CONDENSER	92.00 F
SATURATION TEMP INTO COMPRESSOR	39.47 F
SATURATION TEMP OUT OF COMPRESSOR	115.33 F

ENERGY INPUT SUMMARY:

HEAT PUMPED FROM AIR SOURCE	8045.3 BTU/H
POWER TO INDOOR FAN MOTOR	42.7 WATTS
POWER TO OUTDOOR FAN MOTOR	93.2 WATTS
TOTAL PARASITIC POWER	135.9 WATTS
POWER TO COMPRESSOR MOTOR	699.4 WATTS
TOTAL INPUT POWER	835.3 WATTS

REFRIGERANT-SIDE SUMMARY:

HEAT GAIN TO EVAPORATOR FROM AIR	8045.3 BTU/H
HEAT GAIN TO SUCTION LINE	0.0 BTU/H
ENERGY INPUT TO COMPRESSOR	2387.0 BTU/H
HEAT LOSS FROM COMPRESSOR SHELL	119.4 BTU/H
HEAT LOSS FROM DISCHARGE LINE	0.0 BTU/H
HEAT LOSS FROM CONDENSER TO AIR	7386.0 BTU/H
HEAT LOSS FROM LIQUID LINE	0.0 BTU/H

ENERGY OUTPUT SUMMARY:

HEAT RATE FROM REFRIGERANT TO INDOOR AIR	8045.3 BTU/H
HEAT RATE FROM FAN TO INDOOR AIR	145.7 BTU/H
TOTAL HEAT RATE TO/FROM INDOOR AIR	7899.6 BTU/H

COOLING PERFORMANCE:

COP	2.771
EER	9.457 BTU/H-W
CAPACITY	7899.6 BTU/H

ADL OUTPUT

ECHO OF INPUT DATA: RUN NO. 2 6/19

TITLE CARD: 8000 btu/hr rac

CARD NO. 2:	3	22	1 2	1	0.30	0.23			
CARD NO. 3:		100	0.010	0.001	8000.000				
CARD NO. 4:	2	1	4.000						
CARD NO. 5:	2	3	1.200	19.000	3.000	3.000	5.000	0.000	27.000
CARD NO. 6:	2	3	0.820	14.000	3.000	3.000	10.000	0.000	20.000
CARD NO. 7:		1	500.000	230.000	38.400	31.500			
CARD NO. 8:	80.000		67.000	95.000	75.000				
CARD NO. 9:	5.000		0.683	0.980					

SUMMARY OF INPUT DATA TO REFRIGERATION VAPOR COMPRESSION CYCLE ENGINEERING MODEL

TITLE:	8000 btu/hr rac		RUN NO. 2		6/19
TYPE OF SYSTEM:	ROOM AIR CONDITIONER		REFRIGERANT: FREON	22	
COMPRESSOR:	MANUFACTURER	COMPANY Y	RATED CAPACITY	8500.00 (BTU/HR)	
	MODEL NUMBER	AK8494E	COMP. HT. COEFF.	4.00(WATTS/F)	
CONDENSER:	MANUFACTURER	COMPANY Y	AIR FLOW AREA	1.200 (FT2)	
	MODEL NUMBER	3CZ 19 03 B	NO. OF ROWS	3	
	DEFINED SUBCOOLING	5.00 (F)	FINS PER INCH	19.00	
	EXTRA SUBCOOLING	5.00 (F)	NO. OF COIL CIRCUITS	3	
	FREON PRESSURE DROP	0.00 (PSIA)	SYSTEM PRESSURE DROP	0.23 (IN-H2O)	
	SUBCOOLED/2-PHASE HT	0.68	SUPHEAT/2-PHASE H.T.	0.98	
EVAPORATOR:	MANUFACTURER	COMPANY Y	AIR FLOW AREA	0.820 (FT2)	
	MODEL NUMBER	3EZ 14 03 B	NO. OF ROWS	3	
	DEFINED SUPERHEAT	10.00 (F)	FINS PER INCH	14.00	
	FREON PRESSURE DROP	0.00 (PSIA)	NO. OF COIL CIRCUITS	3	
	SYSTEM PRESSURE DROP	0.30 (IN-H2O)			
FANS:	CONDENSER AIR FLOW	500.00 (FT3/MIN)	EVAPORATOR AIR FLOW	230.00 (FT3/MIN)	
	CONDENSER FAN EFF.	38.40 (%)	EVAPORATOR FAN EFF.	31.50 (%)	
	MOTOR TYPE	PERM SPL CAP	MOTOR EFFICIENCY	67.00 (%)	
AIR TEMPERATURES:	ROOM DRY BULB	80.00 (F)	OUTDOOR DRY BULB	95.00 (F)	
	ROOM WET BULB	67.00 (F)	OUTDOOR WET BULB	75.00 (F)	
DESIGN GOAL:	SPECIFIED CAPACITY	8000.00 (BTU/HR)			

SUMMARY OF RESULTS FROM REFRIGERATION VAPOR COMPRESSION CYCLE MODEL

TITLE: 8000 btu/hr rac   RUN NO. 2   6/19

TYPE OF SYSTEM: ROOM AIR CONDITIONER   REFRIGERANT: FREON 22

SYSTEM PARAMETERS:	COMPRESSOR POWER	790.30 (WATTS)	FAN POWER	148.98 (WATTS)
	TOTAL POWER	939.28 (WATTS)	P(SAT) CONDENSER	276.64 (PSIA)
	TOTAL CAPACITY	8130.81 (BTU/HR)	P(SAT) EVAPORATOR	91.34 (PSIA)
	EER	8.66	MASS FLOW RATE	122.74 (LBM/HR)
	COMP DERATING FACTOR	1.00		
SYSTEM TEMPERATURES:	CONDENSER INLET	178.30 (F)	EVAPORATOR INLET	45.56 (F)
	CONDENSING	120.58 (F)	EVAPORATING	45.48 (F)
	CONDENSER EXIT	115.56 (F)	EVAPORATOR EXIT	55.40 (F)
	CONDENSER SUPERHEAT	57.72 (F)	EVAPORATOR SUPERHEAT	9.92 (F)
	CONDENSER SUBCOOLING	5.01 (F)	EVAPORATOR COIL	WET
	OUTDOOR AIR (DB)	95.00 (F)	ROOM AIR (DB)	80.00 (F)
	OUTDOOR AIR (WB)	75.00 (F)	ROOM AIR (WB)	67.00 (F)
	COND. AIR IN (DB)	97.12 (F)	EVAP. AIR IN (DB)	80.00 (F)
	COND. AIR OUT (DB)	116.65 (F)	EVAP. AIR OUT (DB)	56.12 (F)
	COND. AIR OUT (WB)	75.00 (F)	EVAP. AIR OUT (WB)	54.75 (F)
SYSTEM BALANCE:	CONDENSER HT COEF	327.83 (BTU/HR-F)	EVAPORATOR HT COEF	291.45 (BTU/HR-F)
	COND. FLOW AREA	1.20 (FT2)	EVAP. FLOW AREA	0.82 (FT2)
	CONDENSER Q(REF)	10031.64 (BTU/HR)	EVAPORATOR Q(REF)	8320.40 (BTU/HR)
	CONDENSER Q(AIR)	10002.85 (BTU/HR)	EVAPORATOR Q(AIR)	8250.04 (BTU/HR)
	CONDENSER IMBALANCE	28.80 (BTU/HR)	EVAPORATOR IMBALANCE	70.36 (BTU/HR)
	COMPRESSOR Q(REF)	1913.75 (BTU/HR)	SYSTEM IMBAL Q(REF)	202.51 (BTU/HR)
	COMPRESSOR HEAT	784.58 (BTU/HR)	EVAP COIL E1	0.98
AIR SIDE SYSTEM:	CONDENSER AIR FLOW	500.00 (CFM)	EVAPORATOR AIR FLOW	230.00 (CFM)
	CONDENSER AIR FLOW	475.61 (SCFM)	EVAPORATOR AIR FLOW	224.87 (SCFM)
	COND. AIR VELOCITY	396.34 (FT/MIN)	EVAP. AIR VELOCITY	274.24 (FT/MIN)
	CONDENSER PRESS DROP	0.447 (IN H2O)	EVAPORATOR PRES DROP	0.418 (IN H2O)
	CONDENSER AIR POWER	24.92 (WATTS)	EVAPORATOR AIR POWER	11.00 (WATTS)
	CONDENSER FAN POWER	96.86 (WATTS)	EVAPORATOR FAN POWER	52.12 (WATTS)
SYSTEM ENTHALPIES:	COMPRESSOR INLET	110.39 (BTU/LBM)	COMPRESSOR OUTLET	125.98 (BTU/LBM)
	CONDENSER INLET	125.98 (BTU/LBM)	EVAPORATOR INLET	42.63 (BTU/LBM)
	CONDENSER EXIT	44.25 (BTU/LBM)	EVAPORATOR EXIT	110.39 (BTU/LBM)
	EVAPORATOR AIR IN	31.45 (BTU/LBM)	EVAPORATOR AIR OUT	23.07 (BTU/LBM)
SYSTEM WEIGHTS:	COND. CU WEIGHT	3.50 (LBM)	COND. AL WEIGHT	5.45 (LBM)
	EVAP. CU WEIGHT	2.39 (LBM)	EVAP. AL WEIGHT	2.75 (LBM)
	TOTAL CU WEIGHT	5.90 (LBM)	TOTAL AL WEIGHT	8.20 (LBM)
COND. COIL:	TOTAL FLOW AREA	1.20 (FT2)	SUPERHEAT AREA	0.10 (FT2)
	SATURATED AREA	1.06 (FT2)	SUBCOOLING AREA	0.05 (FT2)
OTHER INFO:	PERCENT LATENT	29.15	REF MASS.	0.49 (LBS)
	CALC.2-PHASE H.T.	7709.17 (BTU/HR)	SUBT.2-PHASE H.T.	8170.38 (BTU/HR)

8000.000	8130.805	1.000	8.656	0.000

SUMMARY OF RESULTS FROM REFRIGERATION VAPOR COMPRESSION CYCLE MODEL

TITLE: 8000 btu/hr rac RUN NO. 2 6/19

TYPE OF SYSTEM: ROOM AIR CONDITIONER REFRIGERANT: FREON 22

SYSTEM PARAMETERS:	COMPRESSOR POWER	784.94 (WATTS)	FAN POWER	148.98 (WATTS)
	TOTAL POWER	933.93 (WATTS)	P(SAT) CONDENSER	277.09 (PSIA)
	TOTAL CAPACITY	8088.43 (BTU/HR)	P(SAT) EVAPORATOR	91.50 (PSIA)
	EER	8.66	MASS FLOW RATE	121.99 (LBM/HR)
	COMP DERATING FACTOR	0.99		
SYSTEM TEMPERATURES:	CONDENSER INLET	180.31 (F)	EVAPORATOR INLET	45.66 (F)
	CONDENSING	120.70 (F)	EVAPORATING	45.58 (F)
	CONDENSER EXIT	115.69 (F)	EVAPORATOR EXIT	55.42 (F)
	CONDENSER SUPERHEAT	59.61 (F)	EVAPORATOR SUPERHEAT	9.84 (F)
	CONDENSER SUBCOOLING	5.01 (F)	EVAPORATOR COIL	WET
	OUTDOOR AIR (DB)	95.00 (F)	ROOM AIR (DB)	80.00 (F)
	OUTDOOR AIR (WB)	75.00 (F)	ROOM AIR (WB)	67.00 (F)
	COND. AIR IN (DB)	97.22 (F)	EVAP. AIR IN (DB)	80.00 (F)
	COND. AIR OUT (DB)	116.72 (F)	EVAP. AIR OUT (DB)	56.19 (F)
	COND. AIR OUT (WB)	75.00 (F)	EVAP. AIR OUT (WB)	54.82 (F)
SYSTEM BALANCE:	CONDENSER HT COEF	327.83 (BTU/HR-F)	EVAPORATOR HT COEF	290.81 (BTU/HR-F)
	COND. FLOW AREA	1.20 (FT2)	EVAP. FLOW AREA	0.82 (FT2)
	CONDENSER Q(REF)	10016.20 (BTU/HR)	EVAPORATOR Q(REF)	8260.69 (BTU/HR)
	CONDENSER Q(AIR)	10012.60 (BTU/HR)	EVAPORATOR Q(AIR)	8207.66 (BTU/HR)
	CONDENSER IMBALANCE	3.60 (BTU/HR)	EVAPORATOR IMBALANCE	53.03 (BTU/HR)
	COMPRESSOR Q(REF)	1955.14 (BTU/HR)	SYSTEM IMBAL Q(REF)	199.63 (BTU/HR)
	COMPRESSOR HEAT	724.90 (BTU/HR)	EVAP COIL E1	0.97
AIR SIDE SYSTEM:	CONDENSER AIR FLOW	500.00 (CFM)	EVAPORATOR AIR FLOW	230.00 (CFM)
	CONDENSER AIR FLOW	475.61 (SCFM)	EVAPORATOR AIR FLOW	224.87 (SCFM)
	COND. AIR VELOCITY	396.34 (FT/MIN)	EVAP. AIR VELOCITY	274.24 (FT/MIN)
	CONDENSER PRESS DROP	0.447 (IN H2O)	EVAPORATOR PRES DROP	0.418 (IN H2O)
	CONDENSER AIR POWER	24.92 (WATTS)	EVAPORATOR AIR POWER	11.00 (WATTS)
	CONDENSER FAN POWER	96.86 (WATTS)	EVAPORATOR FAN POWER	52.12 (WATTS)
SYSTEM ENTHALPIES:	COMPRESSOR INLET	110.37 (BTU/LBM)	COMPRESSOR OUTLET	126.39 (BTU/LBM)
	CONDENSER INLET	126.39 (BTU/LBM)	EVAPORATOR INLET	42.67 (BTU/LBM)
	CONDENSER EXIT	44.29 (BTU/LBM)	EVAPORATOR EXIT	110.37 (BTU/LBM)
	EVAPORATOR AIR IN	31.45 (BTU/LBM)	EVAPORATOR AIR OUT	23.11 (BTU/LBM)
SYSTEM WEIGHTS:	COND. CU WEIGHT	3.50 (LBM)	COND. AL WEIGHT	5.45 (LBM)
	EVAP. CU WEIGHT	2.39 (LBM)	EVAP. AL WEIGHT	2.75 (LBM)
	TOTAL CU WEIGHT	5.90 (LBM)	TOTAL AL WEIGHT	8.20 (LBM)
COND. COIL:	TOTAL FLOW AREA	1.20 (FT2)	SUPERHEAT AREA	0.10 (FT2)
	SATURATED AREA	1.06 (FT2)	SUBCOOLING AREA	0.04 (FT2)
OTHER INFO:	PERCENT LATENT	28.98	REF MASS.	0.61 (LBS)
	CALC.2-PHASE H.T.	8096.93 (BTU/HR)	SUBT.2-PHASE H.T.	8174.55 (BTU/HR)

8000.000	8088.428	0.992	8.661	8130.805

SUMMARY OF RESULTS FROM REFRIGERATION VAPOR COMPRESSION CYCLE MODEL

TITLE: 8000 btu/hr rac RUN NO. 2 6/19

TYPE OF SYSTEM: ROOM AIR CONDITIONER REFRIGERANT: FREON 22

SYSTEM PARAMETERS:	COMPRESSOR POWER	771.21 (WATTS)	FAN POWER	148.98 (WATTS)
	TOTAL POWER	920.19 (WATTS)	P(SAT) CONDENSER	276.06 (PSIA)
	TOTAL CAPACITY	8015.84 (BTU/HR)	P(SAT) EVAPORATOR	91.77 (PSIA)
	EER	8.71	MASS FLOW RATE	120.59 (LBM/HR)
	COMP DERATING FACTOR	0.98		
SYSTEM TEMPERATURES:	CONDENSER INLET	179.34 (F)	EVAPORATOR INLET	45.83 (F)
	CONDENSING	120.41 (F)	EVAPORATING	45.75 (F)
	CONDENSER EXIT	115.40 (F)	EVAPORATOR EXIT	55.52 (F)
	CONDENSER SUPERHEAT	58.92 (F)	EVAPORATOR SUPERHEAT	9.76 (F)
	CONDENSER SUBCOOLING	5.01 (F)	EVAPORATOR COIL	WET
	OUTDOOR AIR (DB)	95.00 (F)	ROOM AIR (DB)	80.00 (F)
	OUTDOOR AIR (WB)	75.00 (F)	ROOM AIR (WB)	67.00 (F)
	COND. AIR IN (DB)	97.24 (F)	EVAP. AIR IN (DB)	80.00 (F)
	COND. AIR OUT (DB)	116.50 (F)	EVAP. AIR OUT (DB)	56.31 (F)
	COND. AIR OUT (WB)	75.00 (F)	EVAP. AIR OUT (WB)	54.94 (F)
SYSTEM BALANCE:	CONDENSER HT COEF	327.83 (BTU/HR-F)	EVAPORATOR HT COEF	289.70 (BTU/HR-F)
	COND. FLOW AREA	1.20 (FT2)	EVAP. FLOW AREA	0.82 (FT2)
	CONDENSER Q(REF)	9892.02 (BTU/HR)	EVAPORATOR Q(REF)	8176.18 (BTU/HR)
	CONDENSER Q(AIR)	9873.08 (BTU/HR)	EVAPORATOR Q(AIR)	8135.08 (BTU/HR)
	CONDENSER IMBALANCE	18.94 (BTU/HR)	EVAPORATOR IMBALANCE	41.11 (BTU/HR)
	COMPRESSOR Q(REF)	1911.86 (BTU/HR)	SYSTEM IMBAL Q(REF)	196.02 (BTU/HR)
	COMPRESSOR HEAT	721.29 (BTU/HR)	EVAP COIL E1	0.96
AIR SIDE SYSTEM:	CONDENSER AIR FLOW	500.00 (CFM)	EVAPORATOR AIR FLOW	230.00 (CFM)
	CONDENSER AIR FLOW	475.61 (SCFM)	EVAPORATOR AIR FLOW	224.87 (SCFM)
	COND. AIR VELOCITY	396.34 (FT/MIN)	EVAP. AIR VELOCITY	274.24 (FT/MIN)
	CONDENSER PRESS DROP	0.447 (IN H2O)	EVAPORATOR PRES DROP	0.418 (IN H2O)
	CONDENSER AIR POWER	24.92 (WATTS)	EVAPORATOR AIR POWER	11.00 (WATTS)
	CONDENSER FAN POWER	96.86 (WATTS)	EVAPORATOR FAN POWER	52.12 (WATTS)
SYSTEM ENTHALPIES:	COMPRESSOR INLET	110.37 (BTU/LBM)	COMPRESSOR OUTLET	126.22 (BTU/LBM)
	CONDENSER INLET	126.22 (BTU/LBM)	EVAPORATOR INLET	42.57 (BTU/LBM)
	CONDENSER EXIT	44.20 (BTU/LBM)	EVAPORATOR EXIT	110.37 (BTU/LBM)
	EVAPORATOR AIR IN	31.45 (BTU/LBM)	EVAPORATOR AIR OUT	23.19 (BTU/LBM)
SYSTEM WEIGHTS:	COND. CU WEIGHT	3.50 (LBM)	COND. AL WEIGHT	5.45 (LBM)
	EVAP. CU WEIGHT	2.39 (LBM)	EVAP. AL WEIGHT	2.75 (LBM)
	TOTAL CU WEIGHT	5.90 (LBM)	TOTAL AL WEIGHT	8.20 (LBM)
COND. COIL:	TOTAL FLOW AREA	1.20 (FT2)	SUPERHEAT AREA	0.10 (FT2)
	SATURATED AREA	1.06 (FT2)	SUBCOOLING AREA	0.04 (FT2)
OTHER INFO:	PERCENT LATENT	28.71	REF MASS.	0.57 (LBS)
	CALC.2-PHASE H.T.	7890.71 (BTU/HR)	SUBT.2-PHASE H.T.	8088.31 (BTU/HR)

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   8000.000    8015.841        0.975        8.711    8088.428
* SUCCESSFUL COMPLETION *  EER= 8.71
  920.19    0.98  148.98  120.41    45.75
EF WT: 0.57      PC.LATE:28.71           AL. WT:    8.20CU. WT:    5.90
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