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USER'S MANUAL FOR
STEADY-STATE COMPUTER SIMULATION FOR
AIR-TO-AIR HEAT PUMPS WITH SELECTED EXAMPLES

*FINAL*Topical Report Task 3

CONSUMER PRODUCTS MINIMUM ENERGY
EFFICIENCY STANDARDS PROGRAM

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1. INTRODUCTION

A steady state computer simulation model (CSM) has been developed for conventional, vapor compression cycle, electrically driven, air-to-air heat pumps based on an Oak Ridge National Laboratory (ORNL) design model (Reference 1). The CSM was developed with the objective of reducing the amount of equipment testing required by the Department of Energy's (DOE) test procedure by requiring a minimum amount of test data for a specific heat pump and using this data as input to the CSM to generate the complete spectrum of test procedure data points.

In order to simulate individual components, several component models were investigated (References 1, 2, 3, 4) for their applicability and availability. The ORNL Heat Pump Design Model (Reference 1) was chosen for its availability and comprehensive component modeling characteristics. The ORNL heat pump component models were used in the simulation model except for the compressor model, which was changed to utilize manufacturers' empirical performance curves (maps). (The original ORNL compressor model was based on component losses and efficiencies and intended for use in heat pump design studies.) The present compressor model was jointly developed by SAI and ORNL (Reference 5). The compressor model also contains built-in corrections to adjust for levels of refrigerant superheat in reciprocating compressors that differ from those levels at which the maps were generated.

Comparison between the heat pump simulation model predictions and available data from three heat pump experiments indicate that the predictions generally are within accepted tolerances of ARI standards and experimental measurements.

A sensitivity analysis was made to assess the effect of possible variations in some of the input parameters on the system's thermal performance. Results verify that the heat pump CSM, provides a useful and powerful tool for predicting thermal performance.

Appendix A of this report reproduces Reference 5, documenting the addition of compressor maps to the ORNL model.

Appendix B reproduces Reference 6 which illustrates the simulation capability of the CSM compared to heating mode data for three heat pumps.

Appendix C presents a description and tabulation of how experimental data is used to generate CSM input.

A listing of the CSM computer program is presented in a companion volume, Reference 7.

2. MODEL DESCRIPTION

The SAI CSM is a FORTRAN computer program for prediction of steady-state thermal performance of vapor compression electrically-driven, air-to-air heat pumps in both heating and cooling modes. This model can be used to reduce the amount of testing currently required by DOE test procedures or to predict performance changes of heat pumps due to component design improvements.

The Oak Ridge National Laboratory (ORNL) Heat Pump Design Model (Reference 1), served as the starting point for the simulation model. Modification and additions made to this model are the subject of this section. A brief introduction is also presented on the base ORNL Heat Pump Design Model. Information presented herein complements the more complete exposition of Reference 1.

The ORNL Heat Pump Design Model, which was derived from the Massachusetts Institute of Technology (MIT) model (Reference 3), predicts the steady-state performance of conventional vapor compression, electrically-driven, air-to-air heat pumps in the heating and cooling modes. The model serves as an analytical design tool in studies aimed at improving heat pump performance. Steady-state capacity and coefficient of performance (COP) (or energy efficiency ratio (EER)) are computed as a function of system operating conditions, compressor characteristics, refrigerant flow control device design, and heat exchanger parameters.

The heat pump design model's main elements include a compressor, a condenser, an evaporator, and a flow control device. Two approaches are possible for modeling the compressor; one based on compressor manufacturers' data and the other on loss and efficiency terms impacting compressor performance.

The condenser model calculates the heat transfer rate as the sum of the rates from the three flow regions in the heat exchanger, i.e., desuperheating, two-phase, and subcooled. The average temperature of the outlet air

is calculated as the weighted average of the temperature of the air exiting each region. The evaporator model is similar to that of the condenser, except that the amount of air dehumidification can also be computed.

The ORNL flow control device model can simulate either a capillary tube, thermostatic expansion valve, or short tube orifice. The model is also capable of bypassing the flow control device if the value of refrigerant subcooling at the condenser exit is input.

2.2 SAI HEAT PUMP COMPUTER SIMULATION MODEL (HEATING MODE)

Since the ORNL model was developed primarily for design purposes and our intention was to simulate the performance of a given heat pump, a few modifications and additions were necessary. These include:

- o Changes in indoor and outdoor fan input power algorithms,
- o Addition of a reversing valve model,
- o Addition of a refrigerant line heat loss model,
- o Changes in the compressor model.

Each of these alterations are discussed below.

2.2.1 Indoor and Outdoor Fan Input Power Algorithms

Power input to the indoor and outdoor fans, which were calculated as outputs in the ORNL model, are required as inputs to the SAI simulation model. These data are easily measured by heat pump manufacturers. This modification was necessary since fan efficiencies, which are usually not known exactly, are required to calculate fan power from air flow rates.

2.2.2 Reversing Valve Model

The pressure drops across the reversing valve in the discharge and suction lines were modeled. These values, obtainable from valve manufacturers, are normally in the 1 to 3 psi range, depending upon the valve size and refrigerant flow rate.

The heat transferred from the discharge line to the suction gas inside the reversing valve was also modeled. This transferred heat, which can be expressed in terms of the reversing valve factor (the ratio of suction line to discharge line heat losses), has not been measured by heat pump manufacturers. According to the only available source of information, this factor ranges from .17 to .25 for split-system heat pumps, depending upon the outdoor temperature (Reference 8).

These three parameters (suction and discharge pressure drops, and reversing valve factor) were assumed to be constant over the range of operation.

2.2.3 Refrigerant Line Heat Loss Model

The heat losses (gains) of the discharge, liquid and suction lines are functions of tube thermal insulation and surrounding temperature. Table 2-1 shows discharge line heat losses, which were calculated on the basis of values measured experimentally, for four split system heat pumps. For all heat pumps except "No. 3," the average discharge line heat loss was about 1700 Btu/hr, averaged over all outdoor temperatures. The loss for heat pump No. 3 was approximately half that of the other units because, according to the manufacturer (Reference 9), the section of discharge line located in the outdoor unit was insulated.

In order to simulate individual heat pumps, discharge line heat loss must therefore be estimated in advance and entered as an input to the program. For such cases where this information is not available, the program has been accomodated with a simple correlation for the discharge line heat loss. The following equation represents the best curve fit of calculated discharge line heat losses for three of the split system heat pumps in Table 2-1. This was used in the simulation model.

$$Q_{DISL} = 10(T_R - T_o) \text{ Btu/hr} \quad (1)$$

where: T_R (F) is the refrigerant temperature leaving the compressor,
 T_o (F) is the outdoor air temperature.

Table 2-1. Comparison of Calculated Value of Discharge Line Heat Loss
With the Correlation (Watts [Btu/hr])

HEAT PUMP NO. 1			HEAT PUMP NO. 2		
OUTDOOR TEMP °C (F)	CALC.	MODEL	OUTDOOR TEMP °C (F)	CALC.	MODEL
-11.7 (12.0)	142 (486)	360 (1230)	-5.1 (22.9)	420 (1433)	431 (1471)
-3 (26.5)	340 (1159)	444 (1515)	-2.9 (26.7)	441 (1505)	429 (1464)
3 (37.5)	354 (1209)	529 (1805)	3.6 (38.4)	448 (1530)	422 (1439)
5.4 (41.7)	494 (1684)	534 (1823)	9 (48.2)	534 (1821)	426 (1452)
10.6 (51.0)	555 (1892)	522 (1780)	13.3 (56.0)	515 (1757)	449 (1531)
17.9 (64.3)	671 (2291)	547 (1867)			
HEAT PUMP NO. 3			HEAT PUMP NO 4		
-8.3 (17)	156 (532)	208 (708)	-8.3 (17.1)	488 (1666)	572 (1952)
1.7 (35)	202 (690)	215 (733)	-6.8 (19.8)	484 (1650)	558 (1903)
8.3 (47)	239 (817)	239 (815)	.7 (30.8)	479 (1635)	509 (1738)
			1.3 (34.3)	461 (1572)	493 (1683)
			3.6 (38.5)	445 (1518)	478 (1630)
			8.3 (47.0)	460 (1570)	446 (1522)

The liquid line heat loss varies from 200 to 700 Btu/hr according to the outdoor temperature (see Tables 1-4, 2-4, 3-4 and 4-4 in Appendix C) as calculated from experimental data. Although it has been shown in Section 5 of Appendix B that a wide variation in liquid line heat loss has only a minute effect on thermal performance, an additional correction in the form of Equation (1) has been used in the CSM to predict this loss. In this case T_R is the refrigerant temperature leaving the condenser. This correlation has been compared with values derived from experimental data with results presented in the tables of Appendix C.

Since the length of the suction line is small compared to either the discharge or liquid line, and the temperature difference between the refrigerant and the outdoor temperature is also rather small, the heat gain or loss in the suction line (other than the heat gained in the section inside the reversing valve, as described in Section 2.2.2) has been neglected in the simulation model.

2.2.4 Compressor Model

SAI chose to base its compressor model on manufacturers' empirical performance curves for reciprocating compressors obtained from compressor calorimeter measurements. (This approach does not require estimation or measurement of efficiency terms that are difficult to evaluate precisely.) These performance curves provide compressor motor power input and refrigerating capacity and/or refrigerant mass flow rate as a function of evaporator saturation temperature for different condenser saturation temperatures.

Corrections for the effect of superheat level on refrigerant mass flow rate and compressor power were developed based on the average of the effects reported by manufacturers of two to four ton, R22, reciprocating hermetic compressors. The corrections can be applied easily to cases where the refrigerant at the compressor shell inlet is in a two-phase state; however, more experimental verification of the predictions for such conditions is needed.

Comparisons were made between the compressor model predictions and available data from four heat pump experiments in the heating mode. For two of the four heat pumps, agreement with the experimental data was excellent. For the other two heat pumps, measured compressor power was underpredicted by average differences of 7.2 and 9.5%; refrigerant mass flow rates were overpredicted by an average of 3.1% in one case and underpredicted by an average of 13.8% in the other case. In two of the four heat pumps, the differences in predicted and measured compressor power became larger at evaporator saturation temperatures below 24F. For the heat pump with the largest differences between modeled and experimental data, additional test data in both the heating and cooling modes after a "run-in" period, showed significant changes in compressor performance. However, remaining differences between model and experiment resulted in an average overprediction of COP by 9%. Appendix A gives detailed information on the compressor simulation model.

The method of incorporating the compressor map into the heat pump model is different from the method used in the ORNL design model. In the ORNL model, the map-based routine used curves, fit to the compressor motor power input and the refrigerant mass flow rate, to model the performance data. The user was required to provide sets of coefficients for bi-quadratic equations for power input and mass flow rate as functions of inlet and outlet saturation temperatures. These coefficients were separately calculated using a least squares algorithm.

In order to simplify the ORNL procedure, a mathematical routine has been implemented in the main program; the user must now only read from the compressor map. The user reads power input and capacity or mass flow rate in each interval of evaporating temperature (10F or less) for each condensing temperature curve on the compressor map. This data is used in the program via a three dimensional table look up subroutine (DIM3) that takes the input evaporator and condenser temperatures, and enters the compressor map, using a three point LaGrange interpolation scheme, to obtain the power and capacity.

2.3 HEAT PUMP SIMULATION MODEL (COOLING MODE)

All modifications that were mentioned in Section 2.2 will generally apply to the heat pump cooling mode simulations as well. However, a few similarities and differences are noted below:

- o The discharge line will be shorter in the cooling mode since both compressor and condenser are in the outdoor unit. The heat loss will consequently be smaller and the precise value of this input will not be important, since discharge side parameters should not significantly affect cooling performance.
- o The liquid line heat loss will be the same for both modes of operation since the line length is the same in both cases.
- o Although the suction line will be longer in the cooling mode, the temperature difference between the refrigerant and the outdoor temperature is rather small compared with the temperature difference in the discharge line. The heat gain in the suction line (other than the heat gained in the section inside the reversing valve described in Section 2.2.2) has been neglected in the simulation model.
- o Although the compressor map corrections were made for the heating mode (Reference 5), they are assumed to be the same for the cooling mode.
- o Since the compressor shell heat loss is on the discharge side of the system, wide variations should not cause any major change in cooling performance.

2.4 SIMULATION MODEL INPUT DATA

Table 2-2 shows the input variables to the heat pump simulation model for each of the heat pump components. The data consist of physical specifications of the unit (independent of changes in the system's operating conditions), which are grouped separately from variables that depend upon operating conditions.

As noted in the table, the input variables (except compressor shell heat loss which is discussed in Section 3.1 of Appendix B), are readily

Table 2-2. SAI Heat Pump Simulation Model Input Data

<p><u>INDEPENDENT OF OPERATING CONDITION</u></p> <p><u>Condenser and Evaporator</u></p> <p>Outside diameter of tubes</p> <p>Inside diameter of tubes</p> <p>Fin thickness</p> <p>Fin pitch and configuration</p> <p>Thermal conductivity of the fins</p> <p>Heat exchanger frontal area</p> <p>Number of tubes in direction of airflow</p> <p>Number of parallel circuits in heat exchanger</p> <p>Spacing of tube passes perpendicular to airflow</p> <p>Spacing of tube rows in direction of airflow</p> <p>Total number of return bends in heat exchanger</p> <p>Contact conductance between fins and tubes</p> <p><u>Reversing Valve and Interconnecting Tubes</u></p> <p>Pressure drops in discharge and suction lines in the reversing valve</p> <p>Inside diameter of line from compressor to reversing valve</p> <p>Length of line from compressor to reversing valve</p> <p>Inside diameter of line from reversing valve to condenser</p> <p>Length of line from reversing valve to condenser</p> <p>Inside diameter of liquid line (from condenser to expansion valve)</p> <p>Length of liquid line</p>	<p>Inside diameter of line from evaporator to reversing valve</p> <p>Length of line from evaporator to reversing valve</p> <p>Inside diameter of line from reversing valve to compressor</p> <p>Length of line from reversing valve to compressor</p> <p><u>Compressor</u></p> <p>Compressor performance map (mass flow rate or capacity and power input to compressor as a function of evaporator temperature for different values of condenser temperature)</p> <p><u>Refrigerant Flow Control Device</u></p> <p>Number of capillary tubes and capillary tube flow factor if the refrigerant flow control device is capillary tube</p> <p>Rating capacity and rating superheat of thermal expansion valve if the expansion device is thermal expansion valve</p> <p>Diameter of the orifice if the refrigerant flow control device is short tube orifice</p> <p><u>DEPENDENT ON OPERATING CONDITION</u></p> <p>Indoor and outdoor air dry-bulb temperatures</p> <p>Indoor and outdoor air relative humidities</p> <p>Indoor and outdoor fan airflow rate</p> <p>Indoor and outdoor fan power input</p> <p>Compressor shell heat loss</p> <p>Superheat or quality at the evaporator exit</p> <p>Subcooling only if the refrigerant flow control device model is not used</p>
---	---

available from heat pump manufacturers. The superheat level at the evaporator exit, which is one of the required input variables, is also usually known at the rating points by heat pump manufacturers. As discussed in Section 5 of Appendix B, some variations in the level of superheat will not affect the heating capacity and COP of the heat pump. A complete description of the makeup of the input file is given in Section 3.

2.5 SIMULATION MODEL OUTPUT

The model outputs include heating capacity, COP, mass flow rate through the compressor, refrigerant pressure and temperature at every point of the circuit, air pressure drop across the heat exchangers, and air temperature at the heat exchanger exits. A sample output listing is discussed in Section 4.

2.6 MODEL VALIDATION

2.6.1 Heating Mode

The model was applied to three heat pumps to demonstrate potential applicability for predicting rating point performance. The compressor shell heat loss and shell heat loss factor (the ratio of compressor shell heat loss to the compressor power input) for all heat pumps are shown in Figure 1 of Appendix B. A reversing valve factor of .25 was assumed for all heat pumps. The results of model validation are fully described in Section 4 of Appendix B. These results indicate that the heat pump simulation model provides a useful and powerful tool for predicting heating performance since the predictions generally fall within the accepted tolerances of the ARI standards and within the possible errors existing in the experimental measurements.

2.6.2 Cooling Mode

Since cooling mode data was available for only one of the heat pumps studied in the heating mode, the model was applied to only that unit at three

different operating conditions. Table 2-3 shows measured data of different parameters at ambient temperatures of 95F and 82F. There are two sets of data at 82F ambient temperature, one at 50% indoor relative humidity and the other one at 11% indoor relative humidity. The indoor dry bulb temperature for all cases is reported to be 80F. The compressor shell heat loss factor and reversing valve factor have been assumed to have the same values as for the heating mode. The other input data; superheat level at evaporator output and subcooling level at condenser output (the flow control device has been bypassed for this simulation), can be derived from Table 2-4.

The results of the model validation are presented in Table 2-5. The results indicate that the heat pump simulation model could also provide a useful tool for predicting cooling performance since the predictions generally fall within the accepted tolerances of the ARI standards and within the possible errors existing in the experimental measurements.

Table 2-3. Steady-State Experimental Performance Data for Heat Pump No. 3 in the Cooling Mode

	OUTDOOR TEMPERATURE (F) DB/WB		
	<u>95/63</u>	<u>82/57</u>	<u>82/56</u>
Air Temperature (F) DB/WB			
Evaporator Inlet	80/67	80/67	80/53
Evaporator Outlet	60/57.4	59.5/57	53.8/41
Condenser Outlet (DB)	116.4	102.8	103.4
Refrigerant Temperature (F)			
Evaporator Outlet*	59.9	68.1	47.0
Condenser Outlet	109.3	94.6	94.0
Refrigerant Pressure (psia)			
Evaporator Outlet*	90.7	82.5	78.5
Condenser Outlet	291	235	240
Air Flow Rate (CFM)			
Outdoor Fan	2291	2321	2320
Indoor Fan	1196	1196	1208
Power Input (watts)			
Outdoor Fan	450	450	460
Indoor Fan + Compressor ⁺	4620	4160	4880
Cooling Capacity (Btu/hr)	36560	37770	34410
Energy Efficiency Ratio (Btu/watt-hr)	7.21	8.19	7.57

* This point is at the end of a 25 foot suction line and before the reversing valve.

+ It is estimated that the outdoor fan power is about 350 watts.

Table 2-4. Estimates of Evaporator Superheat Level and Condenser Subcool Level From Experimental Data for Heat Pump No. 3 in the Cooling Mode

	OUTDOOR TEMPERATURE (F) DB/WB		
	<u>95/63</u>	<u>82/57</u>	<u>82/56</u>
Evaporator Conditions			
Outlet Pressure (psia)	90.7	82.5	78.5
Outlet Saturation Temperature (F)	45.0	39.5	36.8
Outlet Temperature	59.9	68.1	47.0
Outlet Superheat (F)	14.9	28.6	10.2
Condenser Outlet Pressure	291	235	240
Condenser Outlet Temperature	109.3	94.6	94.0
Condenser Outlet Saturation Temperature	124.4	108.2	109.4
Condenser Outlet Subcooling Level (F)	15.1	13.6	15.4

Table 2-5. Comparison of Experimental Data With the Model for Heat Pump No. 3 in the Cooling Mode

	OUTDOOR AND (INDOOR) AIR TEMPERATURES (F), DRY BULB/WET BULB								
	95/63(80/67)			82/57(80/67)			82/56(80/53)		
	EXPERI- MENTAL	MODEL	% DIFFER- ENCE	EXPERI- MENTAL	MODEL	% DIFFER- ENCE	EXPERI- MENTAL	MODEL	% DIFFER- ENCE
Cooling Capacity (Air Method) (Btu/hr)	36560	36204	1	37770	36427	3.5	34410	34836	-1.2
EER(Btu/watt-hr)	7.21	7.81	-8.3	8.19	8	2.3	7.57	7.83	-3.4
<u>Refrigerant Temperature (F)</u>									
Evaporator Outlet ^(a)	60	60.4	-.7	68	69.8	-2.6	47	49.8	-6
Condenser Outlet	109.3	102.9	5.8	94.6	96.4	-1.9	94.0	94.6	-.6
<u>Refrigerant Pressure (psia)</u>									
Evaporator Outlet ^(a)	90.7	94.2	-3.9	82.5	84.9	-2.9	78.5	82.6	-5.2
Condenser Outlet	291	304	-4.5	235	241	2.6	239.5	241	-.6
<u>Air Temperature (F)</u>									
Evaporator Outlet	60	59.3	1.2	59.5	60	.8	53.8	52.8	1.8
Condenser Outlet	116.4	116.6	.2	102.8	102.8	0	101.5	102.1	-.6

(a) After 25 feet suction line before reversing valve.

3. INPUT DATA

The CSM input data is similar to the ORNL heat pump design model input data file except for the compressor data and a few other inputs. The compressor data is based on compressor maps furnished by compressor manufacturers. The heating capacities and power inputs should be read from the compressor map at 10 degree (or less) evaporating temperature intervals for the whole set of condensing temperatures. These values are used as part of the compressor data in the input data file. A sample input listing (shown in Figure 3-1), follows the definitions of the input data file variables.

<u>OPERATING MODE</u>		<u>EXAMPLE VALUE</u>
Card 1	FORMAT (20A4)	
ITITLE	=Descriptive title for system defined by this data set.	SAMPLE
Card 2	FORMAT(5I10)	
NCORH	=Switch to specify cooling or heating mode: =1, for cooling mode, =2, for heating mode.	2
LPRINT	=Output switch to control the detail of printed results: =0, only an energy input and output summary is printed, =1, a summary of the compressor and heat exchanger operating conditions is printed as well as the energy summary, =2, the summaries are printed after each iteration converges, =3, calculated parameters are printed during each iteration.	1
MCMPOP	=Switch to control whether or not heat rejected from the compressor is added to the air flow for outdoor coil: =0, heat rejected from the compressor is not added to the air flow for the outdoor coil, =1, heat rejected from the compressor is added to the air flow before it crosses the outdoor coil, =2, heat rejected from the compressor is added to the air flow after it crosses the outdoor coil.	1

MFANIN	=Switch to control whether or not heat rejected from the indoor fan is added to the air flow, settings are similar to those for MCMPOP.	2
MFANOU	=Switch to control whether or not heat rejected from the outdoor fan is added to the air flow, settings are similar to those for MCMPOP.	1

FLOW CONTROL DEVICE DATA - the variables on this card depend on the type of flow control device selected.

Fixed Condenser Subcooling:

Card 3	FORMAT (I10,6F10.0)	
IREFC	=0, for specified refrigerant subcooling at condenser exit,	0
DTROC	=Desired refrigerant subcooling at condenser exit (F), use the negative of the desired quality for incomplete condensation, i.e. -0.05 for 5% quality.	34.5

Thermostatic Expansion Valve:

Card 3	FORMAT (I10,6F10.0)	
IREFC	=1 for the thermal expansion valve.	1
TXVRAT	=The capacity rating of the TXV (tons).	1.995
STATIC	=The static superheat for the TXV (F).	6.00
SUPRAT	=The rated operating superheat for the TXV (F).	11.00
BLEEDF	=A leakage or bleed factor for the TXV.	1.15
NZTBOP	=A switch to bypass the TXV nozzle and tube pressure drop calculations, =1.0, calculate the tube and nozzle pressure drops =0.0, bypass the tube and nozzle pressure drops.	0.0

Capillary Tube:

Card 3	FORMAT(I10,6F10.0)	
IREFC	=2 for a capillary tube expansion device.	2
CAPFLO	=The capillary tube flow factor, see ASHRAE Guide and Data Book, Equipment Vol. (1975), Fig. 41, p. 20.25.	2.24
NCAP	=The number of capillary tubes.	1.0

Short-Tube Orifice:

Card 3	FORMAT (I10,F10.0)	
IREFC	=3 for a short-tube orifice	3
ORIFD	=Diameter of the orifice (in).	0.0544

REFRIGERANT LINES DATA

Card 4	FORMAT (3F10.5)	
DLL	=Inside diameter of liquid line from indoor coil to outdoor coil (in).	.347
XLEQLL	=Equivalent length of liquid line from indoor coil to outdoor coil (ft).	38.4

QDISLN =Discharge line heat loss (Btu/hr). If set to zero, the loss will be calculated internally.

Card 5 FORMAT (4F10.5)

DLRVOC	=Inside diameter of refrigerant line from the reversing valve to the outdoor coil (in).	0.68
XLRVOC	=Equivalent length of the refrigerant line from the reversing valve to the outdoor coil (ft).	10.0
DLVRIC	=Inside diameter of refrigerant line from the reversing valve to the indoor coil (in).	.68
XLRVIC	=Equivalent length of the refrigerant line from the reversing valve to the indoor coil (ft.)	40.0

Card 6 FORMAT (5F10.5)

DSLRY	=Inside diameter of the refrigerant line from the compressor shell inlet to the reversing valve (in).	0.681
XLEQLP	=Equivalent length of the refrigerant line from the compressor suction port to the reversing valve (ft).	10.00
DDLRY	=Inside diameter of the refrigerant line from the compressor shell outlet to the reversing valve (in).	.318
XLEQHP	=Equivalent length of the refrigerant line from the compressor discharge port to the reversing valve (ft).	4.40
DPREV	=Reversing valve pressure drop (psi)	2.00

INITIAL ESTIMATES TO BEGIN ITERATIONS

Card 7 FORMAT (3F10.2)

TSATEO	=Estimate of the saturation temperature at the outlet of the evaporator (F).	30.0
TSATCI	=Estimate of the saturation temperature at the inlet of the condenser (F).	130.0
XMR	=Estimate of the refrigerant mass flow rate (lbm/h).	400.0

COMPRESSOR DATA

The compressor data is read using nested loops where data for a set of evaporating temperatures is read for each condensing temperature. Variable NTCC establishes the size of the outer loop. NTEC must be defined NTCC times.

Card 8 FORMAT (I10, 4F10.0)

NTCC	=No. of constant condensing temperature curves to be read from the compressor map (15 maximum).	4
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CANFAC	=Compressor shell heat loss factor.	.25
SUPER	=Specified superheat at the outlet from the evaporator (F), use the negative of the desired quality for incomplete evaporation i.e., - 0.99 for 99% quality.	5
SUPERB	=Base superheat for the compressor map (F), use the negative of the return gas temperature to specify a fixed suction temperature, e.g., -65.0(F).	10
TSUB	=Base subcooling level for the compressor map (F)	15
Card 9	FORMAT (15I10)	
NTEC(1 to NTCC)	=No. of desired evaporating temperatures, in 10F increments (or less), at each condensing temperature. Repeated for each different condensing temperature. Limited to fifteen evaporating temperatures for each condensing temperature.	8
Card 10	FORMAT (15F10.0)	
TC(1 to NTCC)	=Desired condensing temperatures as read from compressor map (F). Limited to fifteen condensing temperatures.	80,90,100, 110
Card 11	FORMAT (15F10.0)	
TE(1 to NTEC)	=Desired evaporating temperatures as read from compressor map (F). Note: This card should be repeated for each condensing temperature curve (NTCC).	0,5,10,15, 20,25,30,35
Card 12	FORMAT (15F10.0)	
POWER (1to NTEC)	=Compressor power input at corresponding evaporating temperatures for each condensing temperature curve to be read from compressor map (watts). Note: Compressor power inputs for each condensing temperature curve should be typed on a separate card.	see input file
Card 13	FORMAT (15F10.0)	
COMCAP (1 to NTEC)	=Compressor capacity at corresponding evaporating temperatures for each condensing temperature curve to be read from compressor map (Btu/hr). Note: Compressor capacity for each condensing temperature curve should be typed on a separate card.	see input file

INDOOR COIL DATA

Card 14	FORMAT (7F10.6)	
FINTY	=Switch to specify the type of fins, =1.0, for smooth fins, =2.0, for wavy fins, =3.0, for louvered fins.	2.0
DEA	=Outside diameter of the tubes in the indoor coil (in).	.37
DER	=Inside diameter of the tubes in the indoor coil (in).	.347
DELTA	=Thickness of the fins in the indoor coil (in).	0.00636
FP	=Fin pitch for the indoor coil (fins/in).	14.00
XKF	=Thermal conductivity of the fins in the indoor coil (Btu/h-ft-F).	128.00
AAF	=Frontal area of the indoor coil (sq ft).	3.47
Card 15	FORMAT (2F10.0)	
NT	=Number of row of tubes in the indoor coil in the direction of the air flow.	3.0
NSECT	=Number of parallel refrigerant circuits in the indoor coil.	5.0
Card 16	FORMAT (14F10.5)	
HCONT	=Contact conductance between fins and tubes for the indoor coil (Btu/h-sq ft-F).	30000.0
ST	=The vertical spacing of tube passes (in).	1.00
WT	=The spacing of tube rows in the direction of air flow (in).	1.00
RTB	=The number of return bends in the indoor coil.	40.0
Card 17	FORMAT (4F10.4)	
QA	=Air flow rate for the indoor coil (cfm).	1109
TAI	=Inlet air temperature for the indoor coil (F).	70.0
RH	=Relative humidity of the inlet air for the indoor coil.	0.50
EINDF	=Power input to the indoor fan (watt).	420.

OUTDOOR COIL DATA

Card 18	FORMAT (7F10.6)	
FINTY	=Switch to specify the type of fins, =1.0, for smooth fins, =2.0, for wavy fins, =3.0, for louvered fins.	2.0
DEA	=Outside diameter of tubes in the outdoor coil (in).	0.375
DER	=Inside diameter of the tubes in the outdoor coil (in).	0.347

DELTA	=The thickness of the fins in the outdoor coil (in).	0.00636
FP	=The fin pitch for the outdoor coil (fins/in).	14.00
XKF	=The thermal conductivity of the fins in the outdoor coil (Btu/h-ft-F).	128.00
AAF	=The frontal area of the outdoor coil (sq ft).	7.14
Card 19	FORMAT (2F10.0)	
NT	=The number of rows of tubes in the outdoor coil in the direction of air flow.	3.0
NSECT	=The number of parallel refrigerant circuits in the outdoor coil.	4.0
Card 20	FORMAT (4F10.5)	
HCONT	=The contact conductance between the fins and tubes for the outdoor coil (Btu/h-sq ft-F).	30000.0
ST	=The vertical spacing of tube passes for the outdoor coil (in).	1.00
WT	=The spacing of tube rows in the direction of air flow for the outdoor coil (in).	1.00
RTB	=The number of return bends in the outdoor coil.	38.0
Card 21	FORMAT (4F10.5)	
QA	=The air flow rate for the outdoor coil (cfm).	2311
TAI	=The inlet air temperature for the outdoor coil (F).	47.0
RH	=The relative humidity of the air entering the outdoor coil.	0.70
EOUTF	=Power input to the outdoor fan (watt)	360.

SAMPLE DATA SET FOR SAMPLE HEAT PUMP DESIGN MODEL -- @ 47.0F AMBIENT

2	1	1	2	1			
0	34.5						
0.3470	38.40	817.0					
0.6800	10.0	0.6800	40.00				
0.6800	10.00	0.3180	4.40	2.00			
30.00	130.00	400.0					
4	0.25	5.0	10.0	15.0			
8	8	8	8				
80.0	90.0	100.0	110.0				
0.0	5.0	10.0	15.0	20.0	25.0	30.0	35.0
0.0	5.0	10.0	15.0	20.0	25.0	30.0	35.0
0.0	5.0	10.0	15.0	20.0	25.0	30.0	35.0
0.0	5.0	10.0	15.0	20.0	25.0	30.0	35.0
2390.0	2500.0	2630.0	2740.0	2840.0	2930.0	3000.0	3080.0
2420.0	2560.0	2700.0	2840.0	2960.0	3070.0	3170.0	3240.0
2450.0	2600.0	2770.0	2920.0	3080.0	3200.0	3320.0	3430.0
2460.0	2640.0	2840.0	3030.0	3190.0	3350.0	3480.0	3600.0
17339.0	20181.0	23214.0	26439.0	30176.0	34352.0	38974.0	44208.0
15930.0	18513.0	21505.0	24533.0	27901.0	31691.0	36219.0	40872.0
14569.0	17119.0	19847.0	22755.0	25991.0	29560.0	33690.0	38164.0
13258.0	15630.0	18241.0	21096.0	24128.0	27550.0	31440.0	35661.0
2.0	0.3750	0.3470	0.00636	14.0	128.0	3.4700	
3.000	5.000						
30000.00	1.00	1.00	40.00				
1109.0	70.0	0.500	420.0				
2.0	0.3750	0.3470	0.00636	14.0	128.0	7.1400	
3.000	4.000						
30000.00	1.0	1.000	38.00				
2311.0	47.00	0.700	360.0				

Figure 3-1. Sample Input

4. OUTPUT FILE

The CSM output file shown in Figure 4-1 is similar to the ORNL Heat Pump Design Model output file. The first section of the output file contains a printout of the input file data for verification purposes. This is followed by a summary of the compressor and heat exchanger operating conditions including energy consumption. The program has three other output options as defined in the input data section. The first option limits the output to only COP and capacity of the heat pump and a brief energy consumption summary. The other two options are for diagnostic purposes which are described in detail in Reference 1.

INPUT FILE NAME: ORNCBM.SAM

***** INPUT DATA *****

SAMPLE DATA SET FOR SAMPLE HEAT PUMP DESIGN MODEL -- @ 47.0F AMBIENT
HEATING MODE OF OPERATION (RCORH-2)
COMPRESSOR CAN HEAT LOSS ADDED TO AIR BEFORE CROSSING THE OUTDOOR COIL.
POWER TO THE INDOOR FAN ADDED TO AIR AFTER CROSSING THE COIL.
POWER TO THE OUTDOOR FAN ADDED TO AIR BEFORE CROSSING THE COIL.

DESCRIPTION OF CONNECTING TUBING:

LIQUID LINE FROM INDOOR TO OUTDOOR HEAT EXCHANGER

ID 0.34700 IN
EQUIVALENT LENGTH 30.40 FT

FROM INDOOR COIL TO REVERSING VALVE

ID 0.60000 IN
EQUIVALENT LENGTH 40.00 FT

FROM REVERSING VALVE TO COMPRESSOR INLET

ID 0.60000 IN
EQUIVALENT LENGTH 10.00 FT

FROM OUTDOOR COIL TO REVERSING VALVE

ID 0.68000 IN
EQUIVALENT LENGTH 10.00 FT

FROM REVERSING VALVE TO COMPRESSOR OUTLET

ID 0.31000 IN
EQUIVALENT LENGTH 4.40 FT

PRESSURE DROP ACROSS REVERSING VALVE 2.0 PSI

ESTIMATE OF:

REFRIGERANT MASS FLOW RATE 400.000 LBM/H
SATURATION TEMPERATURE INTO CONDENSER 130.000 F
SATURATION TEMPERATURE OUT OF EVAPORATOR 30.000 F

CORRECTION FACTOR FOR SUCTION GAS 0.330
BASE SUPERHEAT FOR COMPRESSOR MAP 10.000 F
BASE SUBCOOLING FOR COMPRESSOR MAP 15.000 F
EVAPORATOR SUPERHEAT IS HELD FIXED AT 5.000 F

COMPRESSOR SHELL HEAT LOSS FACTOR 0.250

CONDENSER SUBCOOLING IS HELD FIXED AT 34.500 F

***** COMPRESSOR MAP DATA *****

CONDENSING TEMPERATURE 80. F

T EVAP	0.	5.	10.	15.	20.	25.	30.	35.
POWER	2390.	2500.	2630.	2740.	2840.	2930.	3000.	3080.
CAPACITY	17339.	20181.	23214.	26439.	30176.	34352.	38974.	44208.

CONDENSING TEMPERATURE 90. F

T EVAP	0.	5.	10.	15.	20.	25.	30.	35.
POWER	2420.	2560.	2700.	2840.	2960.	3070.	3170.	3240.
CAPACITY	15930.	18513.	21503.	24533.	27901.	31691.	36219.	40872.

CONDENSING TEMPERATURE 100. F

T EVAP	0.	5.	10.	15.	20.	25.	30.	35.
POWER	2450.	2600.	2770.	2920.	3080.	3200.	3320.	3430.
CAPACITY	14569.	17119.	19847.	22755.	25991.	29560.	33690.	38164.

CONDENSING TEMPERATURE 110. F

T EVAP	0.	5.	10.	15.	20.	25.	30.	35.
POWER	2460.	2640.	2840.	3030.	3190.	3350.	3480.	3600.
CAPACITY	13258.	15630.	18241.	21096.	24128.	27550.	31440.	35661.

INDOOR UNIT: CONDENSER
OD OF TUBES IN HX

0.37500 IN

FIN THICKNESS

0.00636 IN

Figure 4-1. CSM Sample Output (Page 1 of 5)

ID OF TUBES IN HX 0.34700 IN
 FRONTAL AREA OF HX 3.470 SQ FT
 NUMBER OF PARALLEL CIRCUITS 5.00
 NUMBER OF TUBES IN DIRECTION OF AIR FLOW 3.00
 NUMBER OF RETURN BENDS 40.00
 AIR FLOW RATE 1109.00 CFM
 RELATIVE HUMIDITY 0.50000
 WAVY FINS

FIN PITCH
 THERMAL CONDUCTIVITY OF FINS
 CONTACT CONDUCTANCE
 HORIZONTAL TUBE SPACING
 VERTICAL TUBE SPACING
 INLET AIR TEMPERATURE
 POWER TO FAN

14.00 FINS/IN
 120.000 BTU/H-FT-F
 30000.0 BTU/H-SQ FT-F
 1.000 IN
 1.000 IN
 70.000 F
 420.000

OUTDOOR UNIT: EVAPORATOR
 OD OF TUBES IN HX 0.37500 IN
 ID OF TUBES IN HX 0.34700 IN
 FRONTAL AREA OF HX 7.140 SQ FT
 NUMBER OF PARALLEL CIRCUITS 4.00
 NUMBER OF TUBES IN DIRECTION OF AIR FLOW 3.00
 NUMBER OF RETURN BENDS 38.00
 AIR FLOW RATE 2311.00 CFM
 RELATIVE HUMIDITY 0.70000
 WAVY FINS

FIN THICKNESS
 FIN PITCH
 THERMAL CONDUCTIVITY OF FINS
 CONTACT CONDUCTANCE
 HORIZONTAL TUBE SPACING
 VERTICAL TUBE SPACING
 INLET AIR TEMPERATURE
 POWER TO FAN

0.00636 IN
 14.00 FINS/IN
 120.000 BTU/H-FT-F
 30000.0 BTU/H-SQ FT-F
 1.000 IN
 1.000 IN
 47.000 F
 360.000

Figure 4-1. CSM Sample Output (Page 2 of 5)

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

COMPRESSOR POWER	3.762 KW	BASE COMP POWER	3.766 KW
BASE MASS FLOW RATE	418.333 LBM/H	REFRIGERANT MASS FLOW RATE	416.872 LBM/H
POWER PER UNIT MASS FLOW	30.72298 BTU/LBM	POWER CORRECTION FACTOR	0.9989
COMPRESSOR SHELL HEAT LOSS	3289.695 BTU/H	MASS FLOW RATE CORRECTION FACTOR	0.9965

SYSTEM SUMMARY	REFRIGERANT TEMPERATURE	SATURATION TEMPERATURE	REFRIGERANT ENTHALPY	REFRIGERANT QUALITY	REFRIGERANT PRESSURE	AIR TEMPERATURE
COMPRESSOR SUCTION LINE INLET	35.436 F	31.377 F	108.266 BTU/LBM	1.0000	71.359 PSIA	
SHELL INLET	41.146	29.046	109.267	1.0000	68.386	
SHELL OUTLET	212.296	127.315	132.365	1.0000	301.252	
CONDENSER INLET	192.688 F	123.861 F	120.360 BTU/LBM	1.0000	295.807 PSIA	70.000 F
OUTLET	91.328	123.793	36.567	.0000	295.556	102.630
EXPANSION DEVICE INLET	87.714 F	123.420 F	35.462 BTU/LBM	.0000	294.174 PSIA	
EVAPORATOR INLET	36.081 F	36.081 F	35.462 BTU/LBM	.1709	78.762 PSIA	47.034 F
OUTLET	36.456	31.377	108.266	1.0000	71.359	38.078

HEAT LOSS IN DISCHARGE LINE 1670.2 BTU/H
HEAT GAIN IN SUCTION LINE 417.4 BTU/H
HEAT LOSS IN LIQUID LINE 460.5 BTU/H

PERFORMANCE OF EACH CIRCUIT IN THE CONDENSER

INLET AIR TEMPERATURE 70.000 F
OUTLET AIR TEMPERATURE 102.630 F
HEAT LOSS FROM FAN 1433.0 BTU/H
AIR TEMPERATURE CROSSING COIL 101.452 F

TOTAL HEAT EXCHANGER EFFECTIVENESS 0.6682

	SUPERHEATED REGION	TWO-PHASE REGION	SUBCOOLED REGION
NTU	0.0000	1.1261	1.1849
HEAT EXCHANGER EFFECTIVENESS	1.0000	0.6757	0.6178
CR/CA	0.0000		0.4124
FRACTION OF HEAT EXCHANGER	0.0000	0.7332	0.2668
HEAT TRANSFER RATE	0.0 BTU/H	6730.2 BTU/H	923.0 BTU/H
OUTLET AIR TEMPERATURE	70.000 F	107.725 F	84.215 F

AIR SIDE:		REFRIGERANT SIDE:	
MASS FLOW RATE	997.0 LBM/H	MASS FLOW RATE	83.4 LBM/H
PRESSURE DROP	0.4323 IN H2O	PRESSURE DROP	0.251 PSI
HEAT TRANSFER COEFFICIENT	11.406 BTU/H-SQ FT-F	HEAT TRANSFER COEFFICIENT	
		VAPOR REGION	62.874 BTU/H-SQ FT-F
		TWO PHASE REGION	282.701 BTU/H-SQ FT-F
		SUBCOOLED REGION	69.718 BTU/H-SQ FT-F

UA VALUES:		TWO PHASE REGION (BTU/H-F)	SUBCOOLED REGION (BTU/H-F)
VAPOR REGION (BTU/H-F)			
REFRIGERANT SIDE	0.000	REFRIGERANT SIDE	2352.108
AIR SIDE	0.000	AIR SIDE	1753.237
COMBINED	0.000	COMBINED	1004.496
		REFRIGERANT SIDE	211.116
		AIR SIDE	638.097
		COMBINED	158.632

FLOW CONTROL DEVICE - CONDENSER SUBCOOLING IS SPECIFIED AS 34.500 F

CORRESPONDING TXV RATING PARAMETERS:
RATED OPERATING SUPERHEAT 11.000 F
STATIC SUPERHEAT RATING 6.000 F
PERMANENT BLEED FACTOR 1.150

CORRESPONDING CAPILLARY TUBE PARAMETERS:
NUMBER OF CAPILLARY TUBES 1
CAPILLARY TUBE FLOW FACTOR 2.625

CORRESPONDING ORIFICE PARAMETER:
ORIFICE DIAMETER 0.0549 IN

Figure 4-1. CSM Sample Output (Page 3 of 5)

SUPERHEAT IS BELOW THE OPERATING RANGE

PERFORMANCE OF EACH CIRCUIT IN THE EVAPORATOR

INLET AIR TEMPERATURE	47.034 F
HEAT LOSS FROM COMPRESSOR	3209.7 BTU/H
HEAT LOSS FROM FAN	1220.3 BTU/H
AIR TEMPERATURE CROSSING COIL	40.720 F
OUTLET AIR TEMPERATURE	38.070 F

MOISTURE REMOVAL OCCURS

SUMMARY OF DEHUMIDIFICATION PERFORMANCE (TWO-PHASE REGION)

	LEADING EDGE OF COIL AIR	POINT WHERE MOISTURE REMOVAL BEGINS AIR	WALL	LEAVING EDGE OF COIL AIR	WALL
DRY BULB TEMPERATURE	40.720 F	42.674 F	37.773 F	37.919 F	36.075 F
HUMIDITY RATIO	.00475	.00476	.00476	.00455	.00444
ENTHALPY	16.860 BTU/LBM	15.394 BTU/LBM	14.205 BTU/LBM	14.019 BTU/LBM	13.460 BTU/LBM

RATE OF MOISTURE REMOVAL	.5433 LBM/H
FRACTION OF EVAPORATOR THAT IS WET	.6098
LATENT HEAT TRANSFER RATE IN TWO-PHASE REGION	883. BTU/H
SENSIBLE HEAT TRANSFER RATE IN TWO-PHASE REGION	6916. BTU/H
SENSIBLE TO TOTAL HEAT TRANSFER RATIO FOR TWO-PHASE REGION	.9222

OVERALL SENSIBLE TO TOTAL HEAT TRANSFER RATIO	.9232
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OVERALL CONDITIONS ACROSS COIL

	ENTERING AIR	EXITING AIR
DRY BULB TEMPERATURE	40.720 F	38.070 F
WET BULB TEMPERATURE	43.454 F	37.390 F
RELATIVE HUMIDITY	0.657	0.947
HUMIDITY RATIO	.00476	.00455

TOTAL HEAT EXCHANGER EFFECTIVENESS (SENSIBLE)	0.7336
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	SUPERHEATED REGION	TWO-PHASE REGION
NTU	0.4160	1.3001
HEAT EXCHANGER EFFECTIVENESS	0.2902	0.7404
CR/CA	1.0120	
FRACTION OF HEAT EXCHANGER	0.0273	0.9727
HEAT TRANSFER RATE	89.7 BTU/H	7499.0 BTU/H
AIR MASS FLOW RATE	74.15 LBM/H	2640.65 LBM/H
OUTLET AIR TEMPERATURE	43.729 F	37.919 F

AIR SIDE

MASS FLOW RATE	2714.0 LBM/H
PRESSURE DROP	0.213 IN H2O
HEAT TRANSFER COEFFICIENT	
DRY COIL	11.290 BTU/H-SQ FT-F
WET COIL	12.671 BTU/H-SQ FT-F

DRY FIN EFFICIENCY	0.777
WET FIN EFFICIENCY	0.745

REFRIGERANT SIDE

MASS FLOW RATE	104.2 LBM/H
PRESSURE DROP	7.404 PSI
HEAT TRANSFER COEFFICIENT	
VAPOR REGION	60.493 BTU/H-SQ FT-F
TWO PHASE REGION	609.136 BTU/H-SQ FT-F

UA VALUES:	VAPOR REGION	TWO PHASE REGION
REFRIGERANT SIDE	30.581	13035.239 BTU/H-F
AIR SIDE		
DRY COIL	133.320	4052.586 BTU/H-F
WET COIL		3115.785 BTU/H-F
COMBINED		
DRY COIL	29.922	1379.258 BTU/H-F

Figure 4-1. CSM Sample Output (Page 4 of 5)

WET COIL

2275.450 BTU/H-F

SUMMARY OF ENERGY INPUT AND OUTPUT:

SAMPLE DATA SET FOR TRANE HEAT PUMP DESIGN MODEL -- @ 47.0F AMBIENT

AIR TEMPERATURE INTO EVAPORATOR	47.03	F
HEAT FROM CONDENSER TO AIR	38266.	BTU/H
HEAT TO EVAPORATOR FROM AIR	38355.	BTU/H
POWER TO INDOOR FAN	1433.	BTU/H
POWER TO OUTDOOR FAN	1228.	BTU/H
POWER TO COMPRESSOR MOTOR	12839.	BTU/H
COMPRESSOR SHELL HEAT LOSS	3210.	BTU/H
TOTAL HEAT TO/FROM INDOOR AIR	39699.	BTU/H

SYSTEM EFFICIENCY:
COP (HEATING)

2.561

HEAT OUTPUT:
HEAT FROM HEAT PUMP

39698.90 BTU/H

Figure 4-1. CSM Sample Output (Page 5 of 5)

5. REFERENCES

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APPENDIX A

A Compressor Simulation Model With Corrections for the Level of Suction Gas Superheat

A. E. Dabiri

C. K. Rice

A COMPRESSOR SIMULATION MODEL WITH CORRECTIONS FOR THE LEVEL OF SUCTION GAS SUPERHEAT

DR. A.E. DABIRI C.K. RICE
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INTRODUCTION

The guidelines for the appliance energy efficiency standards (1) currently being implemented by the U.S. Department of Energy indicate that more extensive testing of heat pumps will be required. However, the guidelines also allow for the use of computer programs to generate rating information in some cases. A Heat Pump Simulation Model (HPSM), currently under development by the authors, has potential as a tool for reducing the amount of testing needed to meet the guidelines. Although the compressor simulation routine described in this paper is for this HPSM, it may be used with other heat pump models as well.

In order to generate steady-state heat pump rating information with a minimum of experimental data, the HPSM has a compressor routine that uses data available from manufacturers' compressor maps and applies corrections for operation at evaporator superheat levels that differ from those used in generating the maps. The performance of other components of the heat pump must be simulated by the model used with this compressor routine. The authors based the HPSM on the Oak Ridge Heat Pump Model (2,3,4) which was chosen for its availability and comprehensive component modeling characteristics. Compressor routines used in heat pump models described in the open literature, such as those given by Kirshbaum and Veyo (5) and by Hiller and Glicksman (6), also were considered. Since these compressor routines, as well as the original compressor routine of Refs. 2, 3, and 4, were not suitable for the present purpose, the decision was reached to develop a new compressor model for the HPSM.

In the following sections of this paper, the compressor model is described and compared with available experimental data. Methods are derived to correct the mass flow rate and compressor power consumption values obtained from compressor maps to account for variations in suction gas superheat. For most cases the corrections are not large; however, it was judged important to seek maximum accuracy for a model with the potential for use in lieu of actual testing. The form of these corrections allows the use of a map-based compressor model in cases when "wet" refrigerant reaches the compressor shell inlet. The magnitude of the corrections is compared with available information from four manufacturers of two- to four-ton, refrigerant 22, hermetic, reciprocating compressors; the agreement between compressor map data (as corrected) and available data on the performance of compressors installed in heat pumps is examined.

OVERVIEW OF THE HPSM COMPRESSOR MODEL

The HPSM compressor model is based on the use of empirical compressor performance curves obtained from compressor calorimeter measurements performed by the manufacturers. These performance curves, or maps, which in most cases are readily available from the manufacturers,

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provide compressor power input, refrigerating capacity and/or refrigerant mass flow rate as functions of evaporator saturation temperature (or pressure) for four to six condenser saturation temperatures (or pressures). Usually each set of maps is generated for fixed values of condenser subcooling and evaporator superheat, although for some maps the superheat is allowed to vary with changes in evaporator saturation temperature while the suction gas temperature at the compressor shell inlet is held constant.

Refrigerating capacity $\dot{Q}_{\text{refr}}$, as commonly defined, is given by

$$\dot{Q}_{\text{refr}} = \dot{m}(h_{\text{in,comp}} - h_{\text{in,evap}}) = \dot{m}\Delta h_{\text{refr}}, \quad (1)$$

where

$\dot{m}$ = refrigerant mass flow rate;

$h_{\text{in,comp}}$ = refrigerant enthalpy at compressor shell inlet;

$h_{\text{in,evap}}$ = refrigerant enthalpy at evaporator inlet [equal to the enthalpy of saturated liquid refrigerant at $(T_{\text{sat,cond}} - \text{subcooling})$].

For given values of condenser subcooling, evaporator superheat, and evaporator and condenser saturation temperatures, Δh_{refr} can be calculated from refrigerant property routines. Thus, given either $\dot{Q}_{\text{refr}}$ or $\dot{m}$, the other quantity can be evaluated from Eq 1 (provided that oil circulation effects have been included in the given quantity). Refrigerant mass flow rate can be considered a more basic compressor performance characteristic than $\dot{Q}_{\text{refr}}$ since the latter is dependent on the value of condenser subcooling chosen for the map.

The main task in preparing compressor maps for use in heat pump simulations is to account for operating conditions that differ from those conditions for which the maps were generated. Since the level of condenser subcooling affects only the value of refrigerating capacity, this parameter is needed only when $\dot{m}$ must be calculated from given map values of $\dot{Q}_{\text{refr}}$. Changes in the levels of evaporator superheat, however, potentially may affect both the refrigerant mass flow rate and the required compressor power input. The effect of various superheat levels will be considered. Other possible factors affecting compressor performance include the fraction of oil circulating with the refrigerant and the temperature and quantity of air passing over the compressor shell ("forced air temperature"). Since the fraction of oil circulating with the refrigerant is not generally measured in heat pump experiments, this effect is not considered here. Possible effects due to forced air temperatures over the compressor shell that are lower than those used in the calorimeter tests are briefly discussed.

The effect of the level of superheat at the compressor shell inlet on compressor performance has been discussed in the open literature by only a few investigators (7,8,9). Generally, these investigators focused on the effects of superheat on refrigerating capacity and EER. At the higher levels of superheat, refrigerating capacity and compressor EER were found to increase.

Further information was obtained from the compressor manufacturers' catalogs and from discussions with the manufacturers. Two of the manufacturers (10,11) provide refrigerating capacity and power consumption at two air-conditioning rating points that differ only in superheat level. Both rating points have 7.2°C (45°F) evaporator saturation temperature, 54.4°C (130°F) condenser saturation temperature, and 8.3°C (15°F) subcooling. Rating point A has a superheat value of 28°C (50°F) while rating point B has 11.1°C (20°F). For the 17°C (30°F) increase in superheat from condition B to condition A, tabulated refrigerating capacity values typically increase by 2.5 to 4%, and compressor power remains constant or increases 1 to 3%, depending upon the model.

A third manufacturer (12) provides capacity conversion graphs to correct for various superheat and subcooling levels. The correction curves show a 1.7% increase in refrigerating capacity for the increase in superheat from rating point B to A. In conversations with this manufacturer (13), it was noted that the basis for the correction curves is experimental data which indicate that, on the average, the refrigerant mass flow rate was found to increase by about 1% for each 2.8°C (5°F) drop in superheat.

Finally, another manufacturer (14) adjusts refrigerating capacity for superheat level by the equation

$$\dot{Q}_{\text{new}} = (1 + F_{\text{SH}})\dot{Q}_{\text{B}}, \quad (2)$$

where

$\dot{Q}_{\text{new}}$ = capacity at the new superheat level,

$\dot{Q}_B$ = capacity from the map at rating point B (11.1 C° superheat),

and

$$F_{\text{SH}} = 0.00144 \cdot (\text{superheat} - 11.1 \text{ C}^\circ). \quad (3)$$

For the 17 C° (30 F°) increase in superheat from rating point B to A, use of Eqs 2 and 3 yields a 2.4% increase in refrigerating capacity.

Thus the data from four compressor manufacturers indicate that the refrigerating capacity increases from 1.7 to 4% for the 17 C° (30 F°) increase in superheat from rating points B to A. Since Δh_{refr} increases by 7.8% from condition B to condition A, the 1.7 to 4% capacity corrections given by the manufacturers imply (from Eq 1) 5.6 to 3.5% decreases in mass flow rate, respectively, for the 17 C° (30 F°) change in superheat between the rating conditions.

The effect of superheat on compressor power was generally assumed by the manufacturers to be zero; however, the limited data available show that in some cases the power increased 1 to 3% from rating point B to rating point A. No information was found describing the effect on compressor performance due to wet suction gas at the compressor inlet.

Obviously, for cases where the actual superheat differs 5 to 10 C° from the map value, the magnitude of the corrections to mass flow rate and power are rather small. However, since a potential use of the compressor model is for heat pump simulation in lieu of testing, any corrections which can reduce potential simulation errors are worthy of consideration. Furthermore, for some compressors, the only maps available were generated at a 35°C (95F) suction gas temperature; in these cases the difference in superheat between the map and actual heat pump application could be 28 to 44 C° (50 to 80 F°). Even in cases where the suction gas temperature for the map is held constant at 18°C (65F), the difference in the superheat level could be 11 to 28 C° (20 to 50 F°).

Since the range of mass flow rate and power changes reported by the various manufacturers were within a reasonably narrow band, a general correction procedure was developed based on refrigerant density changes and the assumption of constant compression efficiency. One advantage of the general correction procedure developed herein over corrections based directly on superheat change (such as Eq 3) is that the former is easily applied to cases where the refrigerant at the compressor shell inlet is in the two-phase region.

ANALYSIS OF THE EFFECT OF SUPERHEAT ON REFRIGERANT MASS FLOW RATE

The refrigerant mass flow rate, $\dot{m}$, can be related to compressor parameters in the following manner:

$$\dot{m} = \eta_{\text{vol}} \rho V_D N \quad (4)$$

where

η_{vol} = volumetric efficiency,

ρ = refrigerant density,

V_D = compressor displacement (total),

N = compressor motor speed.

The volumetric efficiency can be based on either compressor shell inlet conditions or suction port conditions provided that ρ is evaluated accordingly.

A straightforward mass flow rate correction for various superheat levels results from the assumption that the compressor speed, N , and η_{vol} evaluated at the compressor shell inlet ($\eta_{vol,i}$) remain constant. The resulting mass flow rate correction, from Eq. 4, is given by

$$\frac{\dot{m}_{new}}{\dot{m}_{map}} = \frac{\rho_{new}}{\rho_{map}}, \quad (5)$$

where the ρ 's are evaluated at shell inlet conditions.

Use of Eq 5 yields a 7.5% decrease in mass flow rate and an 0.3% decrease in refrigerating capacity for the 17 C° (30 F°) increase in superheat from rating point B to rating point A. These results are an overprediction of the decrease in mass flow rate and a prediction in the wrong direction concerning the refrigerating capacity change. Thus the straightforward mass flow rate correction is inadequate. Since compressor power is generally observed to remain constant within a few percent as superheat is varied and since the compressor motor speed variation would be more than an order of magnitude smaller, the assumption of constant motor speed is justified. Therefore, the assumption of constant η_{vol} at the shell inlet must be inadequate. To provide a basis for a better assumption, the implications of the assumption of a constant η_{vol} at the shell inlet are examined.

Upon entrance to the compressor shell, the refrigerant gas comes in contact with the compressor motor windings, the compressor body, the discharge muffler, a section of the discharge line, the oil sump, and finally the suction muffler. This contact results in additional superheating of the suction gas before it reaches the suction port. Assuming that the refrigerant pressure, P , remains constant while the suction gas is being heated within the shell, and assuming an average value of the specific heat of the refrigerant at constant pressure, c_p , the heat transfer to the suction gas as it moves from the shell inlet to the suction port, $\dot{Q}_{i,p}$, may be calculated by the equation

$$\dot{Q}_{i,p} = \dot{m} c_p (T_p - T_i) = \dot{m} c_p \Delta T_{i,p}, \quad (6)$$

where the subscripts "p" and "i" refer to compressor suction port and shell inlet, respectively. Assuming the perfect gas relationship ($Pv = RT$), Eq 4 can be rewritten as

$$\dot{m} = \eta_{vol,p} V_D N \frac{P}{RT_p}. \quad (7)$$

Using Eq 7 to eliminate T_p in Eq 6 and solving for $\dot{m}$ yields

$$\dot{m} = \frac{1}{T_i} \left(\frac{\eta_{vol,p} P V_D N}{R} - \frac{\dot{Q}_{i,p}}{c_p} \right). \quad (8)$$

If $\eta_{vol,p}$ and $\dot{Q}_{i,p}$ are assumed to remain constant as the superheat level is varied, the expression in parentheses in Eq 8 remains constant. The ratio of mass flow rates for a change in superheat is then given from Eq 8 by

$$\frac{\dot{m}_{new}}{\dot{m}_{map}} = \frac{T_{i,map}}{T_{i,new}}, \quad (9)$$

where the temperatures are given in absolute degrees. Application of the perfect gas relation to Eq 9 gives Eq 5. Therefore, the original assumption of a constant $\eta_{vol,i}$ as superheat is varied implies that $\eta_{vol,p}$ and $\dot{Q}_{i,p}$ also remain constant. Further, since $\dot{m}$ decreases as T_i is increased (from Eq 9), a constant $\dot{Q}_{i,p}$ implies (from Eq 6) that $\Delta T_{i,p}$, the suction gas temperature rise from shell inlet to suction port, increases at higher evaporator superheat levels.

If $\dot{Q}_{i,p}$ instead is assumed to decrease in some manner as evaporator superheat is increased, $\eta_{vol,i}$ would increase at the higher superheat levels. The assumption will be made that $\dot{Q}_{i,p}$ varies with superheat in direct proportion to the refrigerant mass flow rate. From Eq 6, this assumption implies that $\Delta T_{i,p}$ or more exactly, $\Delta h_{i,p}$ remains constant as superheat is

varied. To incorporate this assumption, some estimate must be made of the magnitude of the suction gas heating. From the experimental data of Jacobs (15), an increase of about 28 C° (50 F°) was observed at rating point B. Temperature increases of this order have been reported by Hiller and Glicksman (5) as well. A temperature increase of this magnitude corresponds to a $\Delta h_{i,p}$ of about 21 kJ/kg (9 Btu/lbm). Assuming that $\Delta h_{i,p}$ is constant as all inlet superheat is varied and of the magnitude just given, refrigerant density values at the compressor suction port can be evaluated from known compressor shell inlet conditions. Thus Eq 5 can be used with the refrigerant density values evaluated at the suction port. For the 17 C° increase in superheat from rating point B, $\dot{m}$ decreases by 6.3% and the refrigerating capacity now increases by 1.1%. Since this refrigerating capacity change is still below the values given by the four compressor manufacturers, further corrections were investigated.

Jacobs (15) reported that when the suction gas superheating inside the shell of a hermetic compressor was reduced 24 C° (43 F°) by the use of a heat exchanger, the mass flow rate increase was only 75% of the theoretical change based on suction port densities. He attributed the difference to the unaccounted flow losses and valve dynamic changes as the mass flow rate increases. Thus, based on Jacob's measurements, η_{vol} evaluated at the suction port conditions decreases at the lower superheat levels.

To account for this effect, the following linear correction was incorporated into the compressor model:

$$\frac{\dot{m}_{new}}{\dot{m}_{map}} = 1 + F \left(\frac{\rho_{new}}{\rho_{map}} - 1 \right), \quad (10)$$

where F is a chosen percentage of the theoretical mass flow rate increase and where the densities are evaluated based on suction port conditions. A reanalysis of Jacobs' data showed that for his experiment, F was 0.62 rather than the reported 0.75. With F at a value of 0.62, Eq 10 yields a 3.9% decrease in $\dot{m}$ and a 3.6% increase in refrigerating capacity for the 17 C° (30 F°) increase in superheat from rating point B to A. Since this capacity increase is higher than the average of the values reported by the compressor manufacturers and since possible experimental error could have exaggerated the difference between the observed and predicted flow rates in Jacobs' experiment, a value for F of 0.75 was selected to obtain results closer to the average of compressor manufacturer's values. With F set at the value 0.75, Eq 10 yields a 4.8% decrease in $\dot{m}$ and a 2.7% increase in refrigerating capacity for the increase in superheat from rating point B to A.

ANALYSIS OF THE EFFECT OF SUPERHEAT ON COMPRESSOR POWER

On a theoretical basis, Jacobs (15) has shown that as superheat at the compressor suction port is increased, the work per unit mass of refrigerant increases. An equation can be obtained from Stoecker et al. (16) for calculating this effect (based on shell inlet conditions) using compressor performance curves. They calculated the work of compression, Δh_{new} , for a superheat level different from that used in generating the compressor map using the equation

$$\Delta h_{new} = \Delta h_{map} \left(\frac{\Delta h_{isen, new}}{\Delta h_{isen, map}} \right), \quad (11)$$

where

$$\Delta h_{map} = \frac{\dot{W}_{map}}{\dot{m}_{map}}. \quad (12)$$

The Δh_{isen} terms in Eq 11 represent the isentropic compression work from shell inlet conditions to discharge pressure; $\dot{W}$ represents compressor power. Equation 11 contains the implicit assumption that isentropic efficiency remains constant as superheat level is varied. With the formulation of Δh_{new} given by Eq 11, the compressor power correction at the new superheat level is given by

$$\frac{\dot{W}_{new}}{\dot{W}_{map}} = \frac{\dot{m}_{new}}{\dot{m}_{map}} \cdot \frac{\Delta h_{isen, new}}{\Delta h_{isen, map}} \quad (13)$$

For the 17 C° (30 F°) increase in superheat from rating point B to rating point A, use of Eq 13 in conjunction with Eq 10 gives a power increase of 3.3%, which is larger than the increase observed on the average by compressor manufacturers.

Equation 13 can be improved by using Δh_{isen} values calculated from the estimated suction port conditions rather than from shell inlet conditions. With this change, use of Eqs 13 and 10 results in only a 1.7% increase in power. Since the latter method gives relative power changes nearer to the average effect found in manufacturers' data, this method was adopted for use in the HPSM compressor model.

QUALIFICATION OF THE SUPERHEAT CORRECTION FACTORS

The assumptions of a constant $\Delta h_{i,p}$ of 21 KJ/Kg (9 Btu/lbm) from compressor shell inlet to suction port and a constant value of F of 0.75 as shell inlet superheat is varied are based on minimal experimental data. Faced with this lack of data and the variety of hermetic compressor designs, it should be understood that the assumptions are presented as only one of a possible number of physical explanations for the effects of superheat level on compressor performance. The chosen superheat corrections do, however, give results which agree with the average of the effects reported by four manufacturers of two- to four-ton, R-22, hermetic reciprocating compressors intended for air conditioning and heat pump duty. Modification of the corrections may be warranted in the future to account for changing levels of $\Delta h_{i,p}$ as a function of evaporating temperature, high efficiency compressor designs where different methods of motor cooling may be used, and possible differences in superheat effects between compressors with ringed and plug pistons. The factors $\Delta h_{i,p}$ and F could be determined experimentally for each compressor or each compressor line.

COMPARISON OF THE COMPRESSOR MODEL WITH EXPERIMENTAL DATA

The model was used to predict the performance of the compressors in four heat pumps for which experimental data (17,18,19,20) are available. For each heat pump, the compressor manufacturer's performance curves were reduced to equations for use in the model. Other input to the model consisted of the measured suction and discharge pressures, and the superheat at the compressor shell inlet.

The compressor performance equations, expressed as functions of the evaporating and condensing saturation temperatures, are accurate to within $\pm 2\%$ of the map values. The map values, however, generally have a tolerance of $\pm 5\%$ except at the ARI Standard 520-78 rating conditions (21) where a representative production unit, if tested, must have "a capacity of not less than 95 percent of rated capacity and a power input per unit capacity... of not more than 105 percent of the rated values." Therefore at the ARI rating conditions, the ratio of compressor power input to mass flow rate should not exceed 105 percent of the map value - a more stringent condition than simple $\pm 5\%$ tolerances on capacity and power input.

The superheat value and the temperature of the forced air flowing over the compressor shell for each manufacturer's calorimeter tests are as follows.

<u>Heat pump</u>	<u>Map superheat [C° (F°)]</u>	<u>Forced air temperature [°C (F)]</u>
1	11.1 (20)	35 (95)
2	11.1 (20)	27 (80)
3	5.6 (10)	8.3 (47)
4	13.9 (25)	24 (75)

The compressor in heat pump 1 has plug pistons and the compressors in heat pumps 2, 3, and 4 have ringed pistons.

In actual application, each of the compressors was located in the outdoor unit and thus exposed to lower ambient temperatures in heating mode operation than used in the calorimeter tests. All the compressors were uninsulated except for heat pump 4 which was surrounded by an insulating jacket.

The data from the heat pump tests were scanned for values of superheat (at the compressor shell inlet) close to zero. For these cases, the shell inlet conditions were checked for the possibility of wet refrigerant vapor by comparing the inlet enthalpy with that for saturated vapor at the measured suction pressure. The inlet enthalpy was calculated from an equation

$$h_i = h_o - (\dot{W}_{\text{comp}} - \dot{Q}_{\text{shell}})/\dot{m} , \quad (14)$$

where h represents enthalpy, the subscripts "i" and "o" represent compressor shell inlet and outlet respectively, and $\dot{Q}_{\text{shell}}$ represents the rate of heat loss from the compressor shell. Measured values of h_o , $\dot{W}_{\text{comp}}$, and $\dot{m}$ were used with values of $\dot{Q}_{\text{shell}}$ calculated for low levels of superheat to estimate h_i . If h_i , as calculated from Eq 14, was less than that for saturated vapor, the refrigerant quality was estimated for use in place of superheat in the model.

Some of the values of refrigerant mass flow rate and discharge pressure required adjustments before the results of the compressor model could be compared with them. For heat pumps 1 and 2 (17,18), the high-side pressures were not measured at the compressor discharge, but rather at a point immediately upstream of the refrigerant flow control device. In both cases, the discharge pressure was obtained by adding theoretically estimated values of pressure drops in the liquid line and condenser to the observed values. The error in condenser saturation temperature due to this procedure is estimated to be less than $\pm 1.1^\circ \text{C}$ (2°F). The refrigerant mass flow rates observed for heat pumps 1 and 2 were measured using rotameters calibrated for use with refrigerant 22. Since the data for heat pumps 1 and 2 were published (17,18), these rotameters were found to indicate values somewhat lower than those measured concurrently with turbine flow meters (22,23). Since turbine flow meters are generally considered more accurate, the mass flow rates for heat pumps 1 and 2 were corrected for use in this work. The refrigerant mass flow rate for heat pump 3 (19) was calculated from air-side measurements of capacity and measurements of refrigerant temperatures and pressures. In heat pump 4, a turbine meter was used for the refrigerant flow rate measurement (22).

The mass flow rates and the compressor power consumption predicted by the compressor model are compared with the observed values in Tables 1 and 2 for all four heat pumps. In both tables, the temperature of the ambient air surrounding the compressor and the operating conditions for the compressor are given; mass flow rates are compared in Table 1 and compressor power consumption in Table 2. The comparisons of mass flow rate in Table 1 show the uncorrected values from the compressor maps, the results from the compressor model with corrections for the observed value of superheat (or quality for wet gas), and the observed mass flow rate. The deviations of the values predicted by the model and by the compressor maps from the observations are expressed as percentages of the observed values. In Table 2, the compressor power consumption values are similarly tabulated and compared.

The mass flow rate predictions for heat pump 1 are, on the average, 3.1% higher than the observed values. For heat pump 2, the predicted mass flow rates average 13.8% lower than the measured values. The compressor power predictions for heat pump 1 average 7.2% lower than the measurements. For heat pump 2, the power predictions are an average of 9.5% lower than the measured values. Both mass flow rate and compressor power average within $\pm 5\%$ for heat pump 3 and are in excellent agreement for heat pump 4.

DISCUSSION

For heat pumps 1 and 2, the compressor power deviations are greatest at conditions of low evaporating saturation temperature. The air flow over the compressor shell was at a substantially lower temperature in each of these cases than in the calorimeter tests. While it cannot be concluded that the compressor power deviations are due to these differences, they may be a contributing effect. Other possible contributing factors are potential differences in oil circulation rate between the calorimeter tests and the actual installations and the effects of wet suction gas at the compressor inlet. More experimental data are needed to evaluate whether or not performance data taken at the conditions of standard calorimeter tests (21) can be used to adequately represent the performance of compressors operating in heat pumps at low evaporating temperatures.

For heat pump 2, the experimental mass flow rates and compressor power consumptions are substantially higher than model predictions. Since this heat pump has ringed pistons, it is possible that the compressor had not been adequately "run in" (that is, the piston rings had not properly sealed) when the heat pump tests were made. (Compressor manufacturers often

note on the compressor maps that particular compressors require "run in" to achieve rated performance.)

To further explore this possibility, heat pump 2 was retested after the unit had been subjected to numerous defrost, charge sensitivity, and other miscellaneous laboratory experiments over a period of two years. In Tables 3 and 4, the results of the retesting are summarized. The experimental mass flow rates were obtained using the same rotameter and turbine meter correction procedure as used in the previous experiment. The compressor discharge pressure was directly measured rather than calculated from the liquid line pressure as was done previously. Data from both the heating and cooling modes were obtained for a range of operating conditions in the new series of tests.

The refrigerant mass flow rate predictions shown in Table 3 are consistently higher than the experimental values (with one exception) with an average difference of +5.0%. This result is a reversal and reduction of differences from the previous model underpredictions averaging -13.8% for heat pump 2 in Table 1. The compressor model power predictions given in Table 4 differ from the experimental values by an average of -5.0% - differences in the same direction but smaller than the -9.5% average difference in Table 2. Overall, the new results support the hypothesis that this compressor was not adequately "run in" for the initial tests.

It should be noted that even though the newer experimental results for heat pump 2 average within $\pm 5\%$ of the model predictions, the predicted compressor power per unit capacity averages 11% lower than experimental results and the predicted COP's (or EER's) average 9% higher. The new compressor power differences also generally become larger at lower evaporating temperatures, although the lowest two evaporating temperatures of the previous experiment with heat pump 2 were not reached in the new tests.

For the various heat pumps tested, the mass flow rate corrections for superheat level usually decreased the differences between the compressor map values and the experimental results; however, the data are too scattered to verify the magnitude of the corrections applied. No definite trends were identified with regard to the significance of the compressor power corrections due to the uncertainties regarding the effect of low air temperatures over the compressor shell and the general tendency for experimental power consumption to vary from model predictions by a larger factor than the power correction term for superheat level.

SUMMARY AND CONCLUSIONS

A compressor simulation model has been developed based on manufacturers' empirical performance curves. The compressor model is a first step in the development of a heat pump simulation model for potential use in reducing the testing required by the rating procedures currently being implemented by the U.S. Department of Energy.

Corrections for the effect of superheat level on refrigerant mass flow rate and compressor power were developed to agree with the average of the effects reported by manufacturers of two- to four-ton, R-22, reciprocating hermetic compressors. The corrections can be applied easily to cases where the refrigerant at the compressor shell inlet is in a two-phase state; however, more experimental verification of the predictions for such conditions is needed.

Comparisons were made between the compressor model predictions and available data from four heat pump experiments in the heating mode. For two of the four heat pumps, agreement with the experimental data was generally excellent. For the other two heat pumps, measured compressor power was underpredicted by average differences of 7.2 and 9.5%; refrigerant mass flow rates were overpredicted by an average of 3.1% in one case and underpredicted by an average of 13.8% in the other case. In two of the four heat pumps, the differences in compressor power became larger at evaporator saturation temperatures below -4°C (24°F). For the heat pump with the largest differences between model and experiment, additional data were taken in both the heating and cooling modes after a "run-in" period and showed significant change in compressor performance. However, the remaining differences between model and experiment resulted in an average overprediction of the COP of 9%.

In view of the potential use of the model for rating purposes, more experimental data are needed to quantify the effects on compressor performance of wet suction gas and air flow over the compressor shell at temperatures more representative of eventual heat pump application. Refinements to the superheat correction factors may be warranted to account for the superheat characteristics of new high-efficiency compressor designs, different methods of motor cooling, and superheat effects at low evaporator temperature. In lieu of such information, our results suggest that each compressor map should be calibrated with steady-state heat pump test data (possibly at two heating mode and two cooling mode conditions) to ensure simulation accuracy comparable to the experimental testing program. Further, attention should be given to proper "run-in" of the test units so that experimental data reflect eventual field performance.

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Table 1

Comparison of Compressor Model Predictions for Refrigerant Mass Flow Rate With Experimental Results for Four Heat Pumps

HEAT PUMP NO.	AMBIENT TEMPERATURE °C(F)	T _{sat, evp} °C(F)	T _{sat, cond} °C(F)	COMPRESSOR SHELL INLET SUPERHEAT OR [QUALITY] °C(°F)	MASS FLOW RATE - kg/s(1bm/hr)				
					MAP	MODEL	EXPERIMENTAL	% DIFFERENCE	
								MODEL/EXP	MAP/EXP
1	- 3.1 (26.5)	- 9.2 (15.5)	47.2 (117)	[.94]	.037 (295)	.041 (325)	[Adjusted*] .041 (322)	.9	- 8.4
	3.1 (37.5)	- 5.8 (21.6)	50.0 (122)	8.9 (16)	.043 (340)	.043 (343)	.042 (336)	2.1	1.2
	5.4 (41.7)	- 4.5 (23.9)	51.7 (125)	10.6 (19)	.045 (358)	.045 (359)	.044 (351)	2.3	2.0
	10.6 (51.0)	- 2.3 (27.9)	55.0 (131)	14.4 (26)	.048 (383)	.048 (379)	.047 (371)	2.2	3.2
	17.9 (64.3)	1.3 (34.4)	58.9 (138)	18.3 (33)	.055 (434)	.054 (426)	.050 (395)	7.8	9.9
2	- 5.1 (22.9)	-12.7 (9.2)	38.9 (102)	[.98]	.025 (201)	.027 (212)	[Adjusted*] .033 (261)	-18.8	-23.1
	- 2.9 (26.7)	-10.9 (12.4)	40.0 (104)	[.98]	.028 (221)	.029 (233)	.036 (282)	-17.4	-21.7
	3.6 (38.4)	- 6.3 (20.6)	46.1 (115)	[.99]	.034 (271)	.036 (283)	.042 (330)	-14.2	-17.8
	9.0 (48.2)	- 1.2 (29.8)	50.5 (123)	[1.00]	.043 (344)	.045 (356)	.048 (381)	- 6.7	- 9.6
	13.3 (56.0)	- .3 (31.4)	52.8 (127)	2.2 (4)	.044 (351)	.046 (361)	.051 (405)	-10.9	-13.3
3	- 8.3 (17)	-14.1 (6.6)	35.6 (96)	[.97]	.032 (253)	.034 (266)	[Computed] .034 (267)	- .5	- 5.4
	1.7 (35)	- 5.7 (21.7)	48.3 (119)	[.97]	.041 (329)+	.046 (364)	.044 (347)	4.9	- 5.2
	8.3 (47)	- 1.2 (29.9)	52.8 (127)	5.0 (9)	.050 (396)+	.052 (413)	.052 (409)	1.0	- 3.2
4	- 8.3 (17.1)	-15.1 (4.8)	32.2 (90)	8.3 (15)	.024 (190)	.024 (193)	[Turbine Meter] .024 (192)	.9	- .8
	- 6.0 (19.8)	-14.0 (6.8)	32.8 (91)	7.8 (14)	.026 (202)	.026 (205)	.026 (204)	.7	- .9
	- .8 (30.5)	-10.0 (14.0)	35.6 (96)	7.2 (13)	.031 (245)	.032 (250)	.032 (251)	- .6	- 2.4
	1.4 (34.6)	- 8.4 (16.8)	36.7 (98)	7.2 (13)	.033 (264)	.034 (269)	.034 (271)	- .8	- 2.6
	3.6 (38.5)	- 6.6 (20.2)	38.3 (101)	6.7 (12)	.036 (287)	.037 (293)	.037 (292)	.4	- 1.7
	8.3 (47.0)	- 3.7 (25.3)	40.0 (104)	7.2 (13)	.041 (320)	.042 (334)	.042 (334)	0.0	- 1.9

* Refer to text + Map values based on 18°C (65F) return gas temperature, 32°C (90F) ambient air over compressor

Table 2

Comparison of Compressor Model Predictions for Compressor Power With Experimental Results for Four Heat Pumps

HEAT PUMP NO.	AMBIENT TEMPERATURE °C(F)	T _{sat, evp} °C(F)	T _{sat, cond} °C(F)	COMPRESSOR SHELL INLET SUPERHEAT OR [QUALITY] °C(°F)	COMPRESSOR POWER INPUT - Watts				
					MAP	MODEL	EXP.	% DIFFERENCE	
								MODEL/EXP	MAP/EXP
1	- 3.1 (26.5)	- 9.2 (15.5)	47.2 (117)	[.94]	3318	3232	3670	-12.0	- 9.6
	3.1 (37.5)	- 5.8 (21.6)	50.0 (122)	8.9 (16)	3630	3627	3960	- 8.4	- 8.1
	5.4 (41.7)	- 4.5 (23.9)	51.7 (125)	10.6 (19)	3783	3779	4090	- 7.6	- 7.5
	10.6 (51.0)	- 2.3 (27.9)	55.0 (131)	14.4 (26)	4020	4011	4170	- 3.8	- 3.6
	17.9 (64.3)	1.3 (34.4)	58.9 (138)	18.3 (33)	4448	4475	4660	- 4.0	- 4.5
2	- 5.1 (22.9)	-12.7 (9.2)	38.9 (102)	[.98]	2345	2305	2610	-11.7	-10.1
	- 2.9 (26.7)	-10.9 (12.4)	40.0 (104)	[.98]	2461	2429	2730	-11.0	- 9.9
	3.6 (38.4)	- 6.3 (20.6)	46.1 (115)	[.99]	2810	2766	3100	-10.8	- 9.4
	9.0 (48.2)	- 1.2 (29.8)	50.5 (123)	[1.00]	3218	3188	3390	- 6.0	- 5.1
	13.3 (56.0)	- .3 (31.4)	52.8 (127)	2.2 (4)	3300	3337	3580	- 7.7	- 7.0
3	- 8.3 (17)	-14.1 (6.6)	35.6 (96)	[.97]	2637	2597	2700	- 3.8	- 2.3
	1.7 (35)	- 5.7 (21.7)	48.3 (119)	[.97]	3270+	3165	3290	- 3.8	0.6
	8.3 (47)	- 1.2 (29.9)	52.8 (127)	5.0 (9)	3720+	3668	3710	- 1.1	0.3
4	- 8.3 (17.1)	-15.1 (4.8)	32.2 (90)	8.3 (15)	1989	1976	1996	- 1.0	- .4
	- 6.8 (19.8)	-14.0 (6.8)	32.8 (91)	7.8 (14)	2052	2046	2053	- .3	0.0
	- .8 (30.5)	-10.0 (14.0)	35.6 (96)	7.2 (13)	2291	2282	2258	1.0	1.5
	1.4 (34.6)	- 8.4 (16.8)	36.7 (98)	7.2 (13)	2376	2364	2322	1.8	2.3
	3.6 (38.5)	- 6.6 (20.2)	38.3 (101)	6.7 (12)	2495	2475	2416	2.4	3.3
	8.3 (47.0)	- 3.7 (25.3)	40.0 (104)	7.2 (13)	2647	2631	2564	2.6	3.2

+ Map values based on 18°C (65F) return gas temperature, 32°C (90F) ambient air over compressor

Table 3

Additional Test Data and Model Comparisons of Refrigerant Mass Flow Rate for Heat Pump 2

AMBIENT TEMPERATURE °C(F)	T _{sat,evp} °C(F)	T _{sat.cond} °C(F)	COMPRESSOR SHELL INLET SUPERHEAT OR [QUALITY] C°(F°)	MASS FLOW RATE - kg/s (lbm/hr)				
				MAP	MODEL	EXP.	% DIFFERENCE	
							MODEL/EXP	MAP/EXP
Heating Mode						[Adjusted*]		
- 2.2 (28)	-6.9 (19.5)	43.6 (110.4)	[.98]	.035 (274)	.037 (291)	.038 (299)	-2.8	-8.3
- 1.1 (30)	-5.9 (21.4)	44.8 (112.6)	[.97]	.036 (287)	.039 (307)	.037 (294)	4.4	-2.5
1.7 (35)	-4.8 (23.3)	46.3 (115.3)	[.97]	.038 (299)	.040 (318)	.039 (309)	2.9	-3.4
3.9 (39)	-2.6 (27.3)	47.9 (118.2)	[1.00]	.042 (330)	.043 (342)	.041 (322)	6.1	2.5
6.1 (43)	-1.3 (29.7)	49.3 (120.8)	3.3 (6)	.044 (348)	.045 (356)	.042 (337)	5.7	3.3
7.2 (45)	- .4 (31.2)	50.2 (122.4)	5.0 (9)	.045 (360)	.046 (366)	.043 (345)	6.2	4.2
7.3 (46)	1.3 (34.4)	51.6 (124.9)	5.0 (9)	.049 (387)	.050 (395)	.047 (370)	6.7	4.6
10.0 (50)	2.5 (36.5)	52.7 (125.8)	9.4 (17)	.051 (405)	.051 (408)	.048 (382)	6.7	6.0
Cooling Mode								
22.8 (82)	2.9 (37.2)	43.7 (110.7)	15.0 (27)	.057 (450)	.056 (445)	.052 (410)	8.5	9.7
31.1 (88)	3.4 (38.1)	46.4 (115.6)	10.0 (18)	.057 (450)	.057 (452)	.053 (421)	7.4	6.9
34.4 (94)	4.1 (39.4)	49.2 (120.5)	8.9 (16)	.057 (454)	.058 (458)	.054 (429)	6.7	5.9
37.8 (100)	4.8 (40.7)	52.2 (126.0)	7.8 (14)	.057 (455)	.058 (460)	.055 (438)	5.0	4.0
41.1 (106)	5.6 (42.0)	55.6 (132.0)	7.8 (14)	.057 (454)	.058 (458)	.057 (453)	1.2	0.2

*Refer to text

Table 4

Additional Test Data and Model Comparisons of Compressor Power Input for Heat Pump 2.

AMBIENT TEMPERATURE °C(F)	T _{sat.evp} °C(F)	T _{sat.cond} °C(F)	COMPRESSOR SHELL INLET SUPERHEAT OR [QUALITY] C°(F°)	COMPRESSOR POWER INPUT - WATTS				
				MAP	MODEL	EXP.	% DIFFERENCE	
							MODEL/EXP	MAP/EXP
Heating Mode								
- 2.2 (28)	-6.9 (19.5)	43.6 (110.4)	[.98]	2741	2689	2905	-7.4	-5.6
- 1.1 (30)	-5.9 (21.4)	44.8 (112.6)	[.97]	2826	2761	2990	-7.7	-5.5
1.7 (35)	-4.8 (23.3)	46.3 (115.3)	[.97]	2917	2856	3090	-7.6	-5.6
3.9 (39)	-2.6 (27.3)	47.9 (118.2)	[1.00]	3084	3047	3220	-5.4	-4.2
6.1 (43)	-1.3 (29.7)	49.3 (120.8)	3.3 (6)	3197	3174	3330	-4.7	-4.0
7.2 (45)	- .4 (31.2)	50.2 (122.4)	5.0 (9)	3268	3249	3390	-4.2	-3.6
7.3 (46)	1.3 (34.4)	51.6 (124.9)	5.0 (9)	3412	3388	3535	-4.2	-3.5
10.0 (50)	2.5 (36.5)	52.7 (125.8)	9.4 (17)	3511	3504	3640	-3.7	-3.5
Cooling Mode								
22.8 (82)	2.9 (37.2)	43.7 (110.7)	15.0 (27)	3273	3286	3430	-4.2	-4.6
31.1 (88)	3.4 (38.1)	46.4 (115.6)	10.0 (18)	3389	3382	3530	-4.2	-4.0
34.4 (94)	4.1 (39.4)	49.2 (120.5)	8.9 (16)	3514	3503	3635	-3.6	-3.3
37.8 (100)	4.8 (40.7)	52.2 (126.0)	7.8 (14)	3646	3632	3770	-3.7	-3.3
41.1 (106)	5.6 (42.0)	55.6 (132.0)	7.8 (14)	3782	3771	3930	-4.0	-3.8

APPENDIX B

A Steady-State Computer Simulation Model for Air-to-Air Heat Pumps

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A STEADY-STATE COMPUTER SIMULATION MODEL FOR AIR-TO-AIR HEAT PUMPS

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INTRODUCTION

The U.S. Department of Energy's (DOE) appliance energy efficiency standard currently being implemented requires that extensive testing be conducted to obtain heat pump performance rating information. In some cases, the standard permits use of computer programs to generate these data. Therefore, a Heat Pump Simulation Model has been developed that is capable of reducing the amount of testing currently required by DOE. The simulation model, which generates steady state heat pump performance data, is based upon the minimum number of experimental data for the specific heat pump of interest.

The first step in developing a heat pump simulation model is to generate a compressor simulation model using the minimum amount of compressor data. This model, based on manufacturers' empirical performance curves and discussed in a previous paper, contains built-in corrections to adjust for levels of refrigerant superheat in the reciprocating² compressors, which differ from those levels for which the performance curves were generated.

In order to simulate other heat pump components, several component models were investigated for their applicability and availability.³⁻⁵ Heat Pump Design Model developed in Reference 3 was chosen for its availability and comprehensive component modeling characteristics.

The following sections of this paper will present a brief description of the component model, along with major modifications. The overall heat pump simulation model performance is then compared to heat pump experimental data and recommendations are given for improving and utilizing the model.

HEAT PUMP DESIGN MODEL AND MAJOR MODIFICATIONS

The Heat Pump Design Model,³ which was derived from Hiller and Glicksman model,⁵ predicts the steady state performance of conventional vapor-compression, electrically driven, air-to-air heat pumps in the heating and cooling modes. The model serves as an analytical design tool in studies aimed at improving heat pump performance. Steady state capacity and coefficient of performance (COP) or energy efficiency ratio (EER) are computed as a function of system operating conditions, compressor characteristics, the refrigerant flow control device, and heat exchanger parameters.

The heat pump design model's main elements include a compressor, a condenser, an evaporator, and a flow control device.

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Two approaches are possible for modeling the compressor -- one based on compressor manufacturers' data and the other on loss and efficiency terms impacting compressor performance. It was decided to base the compressor model on manufacturers' empirical performance curves for reciprocating compressors obtained from compressor calorimeter measurements. (This approach does not require estimation or measurement of efficiency terms that are usually difficult to evaluate precisely.) These performance curves provide compressor motor power input and refrigerating capacity and/or refrigerant mass flow rate as a function of the evaporator saturation temperature for different condenser saturation temperatures. Corrections have been built into the compressor model to adjust for levels of refrigerant superheat in reciprocating compressors, which differ from those levels for which the performance curves were generated.² The model could easily be applied to rotary compressors, although new corrections for the level of superheat would have to be determined.

The condenser model calculates the heat transfer rate as the sum of the rates from the three flow regions in the heat exchanger, i.e., desuperheating, two-phase, and subcooled. The average temperature of the outlet air is the weighted average of the temperature of the air exiting each region. The model of the evaporator is similar to that of the condenser, except that the amount of air dehumidification can be computed.

The flow control device model can handle either a capillary tube, thermostatic expansion valve, or short tube orifice. The model can bypass the flow control device, but an additional input -- the value of refrigerant subcooling at the condenser exit -- would then be required.

Since the original model was developed primarily for design purposes and the researcher's intention was to simulate the performance of a given heat pump, a few modifications were necessary. Power inputs to indoor and outdoor fans, which were calculated as outputs in the original model, are inputs to the present simulation model. These data are easily measured by heat pump manufacturers, and the modification is necessary since fan efficiencies, which are usually not known exactly, are required to calculate fan power. Other modifications -- a reversing valve model and refrigerant line losses will be discussed in the subsequent sections.

HEAT PUMP SIMULATION MODEL

Tab. 1 shows the input variables to the heat pump simulation model for each of the heat pump components. The data consist of physical specifications of the unit, independent of changes in the system's operating conditions. These data are grouped separately from variables that depend upon operating conditions. For improved accuracy, indoor and outdoor fan power input are given as input data instead of being calculated.

As noted in the table, the input variables (except compressor shell heat loss, which will be discussed below), are easily accessible and readily available from heat pump manufacturers. The superheat level at the evaporator exit, which is one of the dependent input variables, is usually known at the rating points by heat pump manufacturers. As will be shown later, some variations in the level of superheat will not affect the heating capacity and COP of the heat pump.

The model's outputs include heating capacity, COP, mass flow rate through the compressor, refrigerant pressure and temperature at every point of the circuit, air pressure drops across the heat exchangers, and air temperatures at the heat exchangers' exits.

Compressor Shell Heat Loss

Although the compressor shell heat loss is a parameter that should be obtained experimentally, it is not measured by compressor manufacturers at the present time. Some indirect measurements have been made by other investigators.⁷⁻⁹

Compressor shell heat loss and the compressor shell heat loss factor (the ratio of compressor shell heat loss to the compressor power input) are illustrated in Fig. 1 with respect to the evaporating temperature for four different heat pumps. Calculations of the shell heat loss conducted by Bonne et al.¹⁰ are also plotted in Fig. 1 for condensing temperatures of 26.7°C (80°F) and 46.1°C (115°F). As can be seen, no specific pattern exists in the relationship between the shell heat loss and evaporating temperature. The compressor shell heat loss factor varies between 10% and 40%. Heat pump No. 4's low level of shell heat loss occurs because the compressor is surrounded by a sound-insulating jacket.

Reversing Valve Model

Two main reversing valve parameters were modeled. The first parameter was the pressure drop across the valve in the discharge and suction lines. These values, obtainable from valve manufacturers, are normally in the 1 to 3 psi range, depending upon the valve size and refrigerant flow rate.

The second parameter modeled was heat transferred from the discharge line to the suction gas inside the reversing valve. This transferred heat, which could be expressed in terms of the reversing valve factor (the ratio of transferred heat to discharge line heat loss), has not been measured by heat pump manufacturers. According to the only available source of information, this factor could range from .17 to .25 for split system heat pump, depending upon the outdoor temperature.¹¹ Results presented below indicate that the thermal performance predicted by the model is not very sensitive to the reversing valve factor.

Refrigerant Line Heat Loss Model

Discharge line heat loss is a function of tube thermal insulation and the surrounding temperature. Tab. 2 shows these losses, which were calculated on the basis of values measured experimentally, for four heat pumps. For all heat pumps except No. 3, the average discharge line heat loss was about 500 watts (~ 1700 Btu/hr), averaged over outdoor temperatures. The loss for heat pump No. 3 was approximately half that of the other units because, according to the manufacturer, the section of discharge line located in the outdoor unit was insulated.¹²

The liquid line heat loss varies from 60 to 200 watts (~ 200 to 700 Btu/hr) according to the outdoor temperature. Heat pump No. 4's heat loss, however, was about twice as much as the other units due to low thermal insulation around the tube.¹¹ As will be shown later, wide variation in liquid line heat loss has only a minute effect on thermal performance. Therefore, this heat loss was neglected in the simulation model.

In order to simulate individual heat pumps, discharge line heat loss must be estimated in advance. For this work, however, a simplistic approach to correlate the discharge line heat loss was used. The following equation was the best fit for the calculated discharge line heat losses (Btu/hr):

$$Q_{DISL} = 10 (T_R - T_O)$$

where

T_R = the refrigerant temperature leaving the compressor (°F) and
 T_O = the outdoor air temperature (°F).

For heat pump No. 3, the factor of 10 in the equation has been replaced by 5 to approximate the system discharge-line heat loss. The correlated values are also shown in Tab. 2 for comparison with the calculated heat losses.

Since the length of the suction line is small compared to either the discharge or liquid line, and the temperature difference between the refrigerant and the outdoor temperature is also rather small, the heat gain or loss in the suction line (other than the heat gained in the section inside the reversing valve described above) has been neglected in the simulation model.

HEAT PUMP SIMULATION MODEL RESULTS

The heat pump simulation model was applied to three different heat pumps. The results are presented in this section.

Heat Pump No. 1

The model was applied to this heat pump to demonstrate potential applicability for predicting rating point performance. The compressor shell heat loss and shell heat loss factor are shown in Fig. 1. The shell heat loss factor is approximately independent of evaporating temperature and, for simplicity, a constant shell heat loss factor of .35 was used as an input parameter. The reversing valve factor was assumed to be .25, which is within the range of values reported above. A comparison of experimental data and the model results for this heat pump appear in Tab. 3. The flow factor for the capillary tube was taken to be 2.24 for a tube of 38 inches long and 0.08 inches in diameter.¹³

Two experimental values for heating capacity are reported at each outdoor air temperature. One value is based on refrigerant enthalpy drop across the condenser (refrigerant method) and one value is based on the air enthalpy change as it flows through the indoor coil (air method). The average of these two values has been compared with the results of the model. At all outdoor temperatures, except -11.1°C and -3°C (12°F and 26.5°F), the agreement between the experimental and predicted heating capacity and mass flow rate is highly satisfactory. (Notice that if, for discharge line heat loss, calculated values were used instead of correlated values [see Tab. 2], the agreement would be even better.) The COP predicted by the model is generally within a few percent of the experimental results. At outdoor air temperatures of -11.1°C and -3°C (12°F and 26.5°F), however, the refrigerant leaving the evaporator is in the two-phase region. The difference between the model and the experimental results at these temperatures could be explained by the earlier finding that more experimental data are needed to quantify the effects of wet suction gas on compressor performance.²

At almost all outdoor air temperatures, the agreement between the measured refrigerant temperature at the compressor shell inlet and that predicted by the model is satisfactory. Since this temperature is a function of the reversing valve factor, no definite comparison between the experimental and predicted values is justified.

The agreement between the experimental data and the model prediction for refrigerant temperatures at the compressor outlet and condenser inlet is highly satisfactory at all outdoor air temperatures, except at -11.1°C and -3°C (12°F and 26.5°F) where the measured values are higher. These refrigerant temperatures are a function of compressor shell heat loss. For example, at an outdoor air temperature of -3°C (26.5°F), a change in compressor shell heat loss from a present value of 1.25 to 1 kW (4257 to 3500 Btu/h) will increase the compressor outlet temperature from 72.8°C to 79.4°C (163 to 175°F) and the condenser inlet temperature from 60°C to 67.8°C (140 to 154°F). The change will have no major impact on heating capacity, COP, or any other output parameters. The predicted refrigerant temperatures at the condenser exit are very close to the experimental values at all outdoor air temperatures.

The refrigerant temperature predicted by the model at the evaporator inlet is lower than the experimental values at all outdoor air temperatures. This discrepancy may occur because the reported values are not taken exactly at the evaporator inlet but, instead, further upstream where the refrigerant temperature is higher.

The agreement between the experimental data and the prediction of the refrigerant temperature at the evaporator outlet is satisfactory at all outdoor air temperatures, other than -11°C and -3°C (12°F and 26.5°F), where the refrigerant is in the two-phase region.

The values predicted by the model for capillary tube inlet pressure and for compressor suction pressure are in close agreement with the measured data at all outdoor air temperatures.

The outlet air temperatures of the outdoor and indoor coils predicted by the model are in agreement with the experimental values.

Heat Pump No. 3

The model was next applied to heat pump No. 3. The compressor shell heat loss factor for this heat pump could be calculated from experimental data only at 8.3°C (47°F). This value -- equal to .25 -- was used for other operating conditions; the reversing valve factor was assumed to be .25. Tab. 4 presents the model's output for this heat pump. The agreement between the experimental data and the predicted values for all parameters considered was highly satisfactory.

Heat Pump No. 4

Heat pump No. 4 was also analyzed. The compressor shell heat loss, shown in Fig. 1, and the average compressor shell heat loss factor, which is about .15, were used in the model. The reversing valve factor was assumed to be the same as for the other heat pumps.

This heat pump's flow control device was composed of a thermal expansion valve and a short capillary tube connected in series. The simulation model is not currently capable of modeling such an uncommon flow control device; therefore, it was bypassed in modeling the

Heat pump No. 4's outdoor coil was composed of spine fins, which are reported to have improved heat transfer characteristics as compared to wavy fins. Since, at present, the spine fin correlations are not yet included in the model, the air-side heat transfer coefficient was increased to make the predicted heating capacity at -8.3°C (17.1°F) equal the experimental heating capacity. This required that the air-side heat transfer coefficient be increased by 38% compared to the wavy fin. For reference, the wavy fin's air-side heat transfer coefficient is about 45% higher than that of the flat fin for the same condition.

The output of the model in terms of heating capacity and COP for heat pump No. 4 is tabulated in Tab. 5. The agreement between the experimental data and the predicted values at other outdoor temperatures is highly satisfactory. Reversing valves are reported to have leakage as high as 5% at higher outdoor air temperatures (and resulting higher compressor discharge pressures). In other words, the experimental heating capacity should have been higher at higher outdoor temperatures, which would eventually reduce the tabulated differences in heating capacities in Tab. 5.

Required Tolerance in Model Predictions

According to the most recently published Air Conditioning and Refrigeration Institute (ARI) standard, heating or cooling capacity rating and COP or seasonal energy efficiency ratio (SEER) should not be less than 95% of the values reported by heat pump manufacturers in accordance with the standard. Furthermore, some error often occurs in experimental measurements. For example, the experimental measurements of heating capacity and COP for heat pump No. 1 (see Tab. 2) differ for two measurement techniques (air-side and refrigeration-side) by as much as 9% for some outdoor air temperatures.

The results presented in Tab. 3 through 5 indicate that the simulation model predictions generally fall within the accepted tolerances of the ARI standards and within the possible errors existing in the experimental measurements.

SENSITIVITY ANALYSIS OF THE MODEL

A sensitivity analysis was made using heat pump No. 3 to assess the effect of possible variations of some of the input parameters on the system's thermal performance. The input parameters that could not be determined accurately are listed below:

- compressor shell heat loss
- reversing valve factor
- level of superheat at the evaporator exit
- liquid line heat loss
- discharge line heat loss

Tab. 6 presents the variations in heat pump heating capacity and COP caused by changes in the value of the compressor shell heat loss factor for rating points at -8.3°C and 8.3°C (17°F and 47°F) outdoor air temperatures. If the shell heat loss factor varies from .25 to .15, the heating capacity and COP increase approximately 2.5% at either -8.3°C and 8.3°C (17°F or 47°F). Therefore, relatively accurate information on compressor shell heat loss is necessary for the model.

A comparison of the experimental heat pump results and the model for three different reversing valve factors with a fixed shell heat loss factor of .25 is presented in Tab. 7. The results indicate that the heating capacity and COP will increase only about .81% if the reversing valve factor is increased from zero to its maximum likely value of .5. In other words, the thermal performance predicted by the model is not very sensitive to the reversing valve factor. Therefore, a reversing valve factor of .25 is recommended as a constant in the model rather than an input parameter.

The sensitivity analysis indicates that at an outdoor temperature of 8.3°C (47°F), the heating capacity and COP changes are about .2% and .15%, respectively, if the superheat level at the evaporator exit varies from -1.7°C to 1.7°C (-3°F to $+3^{\circ}\text{F}$) with respect to the nominal value. In other words, some variations in the level of superheat will not affect the heating capacity and COP of the heat pump.

When the heat pump's liquid line heat loss was changed from zero to its calculated value of about 145-175 watts (500-600 Btu/hr), the heating capacity changed only .6% and .25% at -8.3°C and 8.3°C (17°F and 47°F), respectively. The corresponding changes in COP were .6% and .15% at low and high outdoor temperatures. For this reason, the liquid line heat loss term was neglected in the simulation model.

Doubling the calculated value of the discharge line heat loss, which is about 175-235 watts (600-800 Btu/hr), resulted in a decrease in heating capacity of approximately the same amount. This is equivalent to a decrease in heating capacity of 3% and 2% at -8.3°C and 8.3°C (17°F and 47°F), respectively. The COPs are down about 2.4% and 1.3% at low and high outdoor air temperatures, respectively. In other words, relatively accurate information on discharge line heat loss is necessary for the model.

CONCLUSION

The heat pump computer simulation model, which is based on the use of compressor performance maps, provides a useful and powerful tool for predicting thermal performance. The inputs to the model are essentially available from heat pump manufacturers, except for the compressor shell heat loss factor and discharge line heat loss, which must be determined for each heat pump.

Comparison of the simulation results with experimental measurements on three heat pumps indicates that the simulation model predictions generally fall within the accepted tolerances of the ARI standards and within the possible errors existing in the experimental measurements.

RECOMMENDATIONS

A thorough study of compressor shell heat loss is warranted. The heat loss measurement could be included in the routine compressor rating test presently conducted by compressor manufacturers.

Further investigation into simulating heat pump thermal performance when the refrigerant in the suction line is in the two-phase region should be conducted. This research could be accomplished using more experimental data to quantify the effect of wet suction gas on compressor performance. Extension of this model to the heat pump's cooling mode and cycling operations, as well as operation in the frost region, is recommended.

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Heat Pump Simulation Model Input Data

INDEPENDENT OF OPERATING CONDITION

Condenser and Evaporator

Outside diameter of tubes

Inside diameter of tubes

Fin thickness

Fin pitch and configuration

Thermal conductivity of the fins

Heat exchanger frontal area

Number of tubes in direction of airflow

Number of parallel circuits in heat exchanger

Spacing of tube passes perpendicular to airflow

Spacing of tube rows in direction of airflow

Total number of return bends in heat exchanger

Contact conductance between fins and tubes

Reversing Valve and Interconnecting Tubes

Pressure drops in discharge and suction lines in the reversing valve

Inside diameter of line from compressor to reversing valve

Length of line from compressor to reversing valve

Inside diameter of line from reversing valve to condenser

Length of line from reversing valve to condenser

Inside diameter of liquid line (from condenser to expansion valve)

Length of liquid line

Inside diameter of line from evaporator to reversing valve

Length of line from evaporator to reversing valve

Inside diameter of line from reversing valve to compressor

Length of line from reversing valve to compressor

Compressor

Compressor performance map (mass flow rate or capacity and power input to compressor as a function of evaporator temperature for different values of condenser temperature)

Refrigerant Flow Control Device

Number of capillary tubes and capillary tube flow factor if the refrigerant flow control device is capillary tube

Rating capacity and rating superheat of thermal expansion valve if the expansion device is thermal expansion valve

Diameter of the orifice if the refrigerant flow control device is short tube orifice

DEPENDENT ON OPERATING CONDITION

Indoor and outdoor air dry-bulb temperatures

Indoor and outdoor air relative humidities

Indoor and outdoor fan airflow rate

Indoor and outdoor fan power input

Discharge line heat loss

Compressor shell heat loss

Superheat or quality at the evaporator exit

Subcooling only if the refrigerant flow control device model is not used

TABLE 2

Comparison of Calculated Value of Discharge Line
Heat Loss with the Correlation (Watts [Btu/hr])

HEAT PUMP NO. 1			HEAT PUMP NO. 2		
OUTDOOR TEMP °C (F)	CALC.	MODEL	OUTDOOR TEMP °C (F)	CALC.	MODEL
-11.7 (12.0)	142 (486)	360 (1230)	-5.1 (22.9)	420 (1433)	431 (1471)
-3 (26.5)	340 (1159)	444 (1515)	-2.9 (26.7)	441 (1505)	429 (1464)
3 (37.5)	354 (1209)	529 (1805)	3.6 (38.4)	448 (1530)	422 (1439)
5.4 (41.7)	494 (1684)	534 (1823)	9 (48.2)	534 (1821)	426 (1452)
10.6 (51.0)	555 (1892)	522 (1780)	13.3 (56.0)	515 (1757)	449 (1531)
17.9 (64.3)	671 (2291)	547 (1867)			
HEAT PUMP NO. 3			HEAT PUMP NO 4		
-8.3 (17)	156 (532)	208 (708)	-8.3 (17.1)	488 (1666)	572 (1952)
1.7 (35)	202 (690)	215 (733)	-6.8 (19.8)	484 (1650)	558 (1903)
8.3 (47)	239 (817)	239 (815)	.7 (30.8)	479 (1635)	509 (1738)
			1.3 (34.3)	461 (1572)	493 (1683)
			3.6 (38.5)	445 (1518)	478 (1630)
			8.3 (47.0)	460 (1570)	446 (1522)

TABLE 3

Comparison of Experimental Data with the Model for Heat Pump No. 1

OUTDOOR AIR TEMPERATURE - °C (F)																		
-11.1 (12)			-3 (26.5)			3 (37.5)			5.4 (41.7)			10.6 (51)			17.9 (64.3)			
EXPERI- MENTAL	MODEL	% DIFFER- ENCE	EXPERI- MENTAL	MODEL	% DIFFER- ENCE	EXPERI- MENTAL	MODEL	% DIFFER- ENCE	EXPERI- MENTAL	MODEL	% DIFFER- ENCE	EXPERI- MENTAL	MODEL	% DIFFER- ENCE	EXPERI- MENTAL	MODEL	% DIFFER- ENCE	
<u>Heating Capacity (Btu/hr) ^a</u>																		
Air Method	23664	--		31469	--		32017	--		36604	--		40800	--		45093	--	
Refrigerant Method	23791	--		31042	--		34868	--		36258	--		38407	--		41417	--	
Average	23727	20901	13.5	31255	27711	12.8	33442	32708	2.2	36431	35038	4.0	39603	38899	1.8	43255	44563	-2.9
<u>Coefficient of Performance</u>																		
Air Method	1.49	--		1.93	--		1.86	--		2.06	--		2.27	--		2.32	--	
Refrigerant Method	1.49	--		1.90	--		2.02	--		2.04	--		2.14	--		2.13	--	
Average	1.49	1.71	-12.9	1.92	1.96	-2.0	1.94	2.09	-7.2	2.05	2.15	-4.7	2.21	2.26	-2.2	2.22	2.34	-5.1
<u>Mass Flow Rate (lb/hr) ^b</u>																		
	256.	256.	0.0	322.	297.	8.4	336.	327.	2.8	351.	345.	1.7	371.	375.	-1.1	395.	427.	-7.5
<u>Refrigerant Temperature (F)</u>																		
Compressor Inlet	5.	-0.3	--	15.7	11.2	40.2	37.2	40.3	-7.7	42.8	45.7	-6.3	53.7	56.6	-5.1	67.2	71.1	-5.5
" Outlet	135.	99.	36.4	178.	163.	9.2	218.	214.	1.9	224.	218.	2.8	228.	228.	0.0	251.	245.	2.4
Condenser Inlet	125.	98.	27.6	160.	140.	14.3	201.	187.	7.5	201.	193.	4.1	205.	205.	0.0	224.	225.	-0.4
" Outlet	77.1	79.1	-2.5	78.	79.5	-1.9	79.1	83.7	-5.5	79.8	80.8	-1.2	77.3	77.8	-0.6	81.8	82.0	-0.2
Evaporator Inlet	14.2	7.7	84.4	27.5	18.4	49.5	34.4	25.9	32.8	37.6	28.7	31.0	42.4	33.4	26.9	48.4	41.4	16.9
" Outlet	7.9	2.6	204.	20.1	13.8	45.7	33.3	33.1	0.6	38.7	38.7	0.0	49.5	50.7	-2.4	63.1	65.9	-4.2
<u>Refrigerant Pressure (psia) ^c</u>																		
Compressor Inlet	43	38.4	12.0	53.5	48.6	10.1	60	57	5.3	63	60.1	4.8	67.5	65.9	2.4	76.	76.6	-0.8
Capillary Tube Inlet	211.	196.	7.7	250.	226.	10.6	265.	256.	3.5	275.	266.	3.4	295.	288.	2.4	325.	337.	-3.6
<u>Air Temperature (F)</u>																		
Evaporator Outlet	9.7	8.5	14.1	21.9	20.6	6.3	30.1	30.2	-0.3	33.5	34.0	-1.5	41.5	41.5	0.0	52.5	54.4	-3.5
Condenser Outlet	90.2	87.5	3.1	96.2	92.5	4.0	100.	99.7	0.3	101.2	98.6	2.6	101.5	98.9	2.6	107.6	105.7	1.8

a. Multiply by 2.93×10^{-4} to obtain kWb. Multiply by 1.26×10^{-4} to obtain kg/sec

c. Multiply by 6.89 to obtain KPa

TABLE 4

Comparison of Experimental Data with the Model for Heat Pump No. 3

	OUTDOOR AIR TEMPERATURE - °C (F)								
	-8.3 (17)			1.7 (35)			8.3 (47)		
	EXPERI- MENTAL	MODEL	% DIFFER- ENCE	EXPERI- MENTAL	MODEL	% DIFFER- ENCE	EXPERI- MENTAL	MODEL	% DIFFER- ENCE
Heating Capacity (Air Method) (Btu/hr) ^a .	24100	22475	7.2	33800	32444	4.2	39500	40398	-2.2
Coefficient of Performance	2.0	1.97	1.5	2.42	2.38	1.7	2.58	2.59	-0.4
Refrigerant Mass Flow Rate (Lb/hr) ^c .	266. ^b .	252.	5.6	362. ^b .	341.	6.2	408. ^b .	418.	-2.4
<u>Refrigerant Temperature (F)</u>									
Compressor Inlet	6.5	4.1	58.5	21.7	19.	14.2	38.7	38.2	1.3
" Outlet	159.	157.4	1.0	182.	184.8	-1.5	210.	210.	0.0
Condenser Inlet	148.	143.2	3.4	173.	173.	0.0	201.	200.4	0.3
" Outlet	95.4	97	-1.6	89.6	85.7	4.6	92.1	92.0	0.1
<u>Refrigerant Pressure (psia)^d.</u>									
Compressor Inlet	44.2	42.1	5.0	59.	56.6	4.2	69.5	68.3	1.8
" Outlet	201.	206.4	-2.6	--	261.	--	295.	304.	-3.0
<u>Air Temperature (F)</u>									
Evaporator Outlet	12.	12.2	-1.6	--	27.6	--	38.	38.1	-0.3
Condenser Outlet	89.5	87.9	1.8	97.4	94.7	2.9	102.9	103.2	-0.3

a. Multiply by 2.93×10^{-4} to obtain kW

b. Calculated.

c. Multiply by 1.26×10^{-4} to obtain kg/sec

d. Multiply by 6.89 to obtain KPa

TABLE 5

Comparison of Experimental Data with the Simulation Model for Heat Pump No. 4

OUTDOOR TEMP °C (F)		HEATING CAPACITY (Btu/hr) ^a			COEFFICIENT OF PERFORMANCE		
		AIR METHOD	REFRIGERANT METHOD	AVERAGE	AIR METHOD	REFRIGERANT METHOD	AVERAGE
-8.3 (17.1)	Experimental Model % Difference	18528	18941	18735 18730 0.0	2.09	2.13	2.11 2.05 2.9
-6.8 (19.8)	Experimental Model % Difference	20234	20094	20164 19887 1.4	2.23	2.22	2.22 2.13 4.20
.7 (30.8)	Experimental Model % Difference	24151	24317	24234 25011 -3.1	2.48	2.49	2.48 2.41 2.9
1.3 (34.3)	Experimental Model % Difference	25345	26069	25707 27098 -5.1	2.55	2.62	2.58 2.54 1.6
3.6 (38.5)	Experimental Model % Difference	27114	27750	27432 28943 -5.2	2.64	2.70	2.67 2.61 2.3
8.3 (47)	Experimental Model % Difference	30134	31253	30693 33013 -7.0	2.80	2.90	2.85 2.83 0.7

a. Multiply by 2.93×10^{-4} to obtain kW.

TABLE 6

Comparison of Experimental Results of Heat Pump No. 3 with the
Model for Different Compressor Shell Heat Loss Factor (QCANF)
with Reversing Valve Factor Equal to .25

OUTDOOR AIR TEMPERATURE °C (F)	EXPERIMENTAL		MODEL			
	HEATING CAPACITY kW (Btu/h)	COP	QCANF = 0.25		QCANF = .15	
			CAPACITY kW (Btu/h)	COP	CAPACITY kW (Btu/h)	COP
-8.3 (17)	7.06 (24100)	1.998	6.59 (22475)	1.968	6.75 (23027)	2.021
8.3 (47)	11.58 (39500)	2.576	11.84 (40398)	2.595	12.11 (41316)	2.652

TABLE 7

Comparison of Experimental Results of Heat Pump No. 3 with the
Model for Different Reversing Valve Factor (RVF) with
Compressor Shell Heat Loss Factor Equal to .25

OUTDOOR AIR TEMPERATURE °C (F)	EXPERIMENTAL			MODEL					
	HEATING CAPACITY kW (Btu/h) COP			RVF = 0		RVF = .25		RVF = .5	
				CAPACITY kW (Btu/h)	COP	CAPACITY kW (Btu/h)	COP	CAPACITY kW (Btu/h)	COP
-8.3 (17)	7.06 (24100)	1.998		6.56 (22391)	1.969	6.59 (22475)	1.968	6.61 (22570)	1.967
8.3 (47)	11.58 (39500)	2.576		11.81 (40299)	2.597	11.84 (40398)	2.595	11.87 (40510)	2.596

Figure 1. Variation of compressor shell heat loss with respect to evaporating temperature

APPENDIX C

Derivation of Heat Pump Input Parameters From Experimental Data

APPENDIX C

Derivation of Heat Pump Input Parameters From Experimental Data

The steady-state computer simulation model requires some input data which is not directly accessible in published manufacturers data. These are:

- o Compressor shell heat loss,
- o Discharge line heat loss,
- o Liquid line heat loss (this was found to have a minor effect on heating capacity and COP),
- o Level of superheat at the evaporator exit,
- o Level of subcooling in the condenser (only necessary if the refrigerant flow control device model is not used).

The values used for these parameters were derived from experimental data. The following tables summarize the procedure by which these parameters were obtained for the four heat pumps studied. The first table for each heat pump (Tables C1-1, C2-1, C3-1 and C4-1) presents the steady state performance data at different outdoor air temperatures (References 1, 2, 3, 4). The values given in subsequent tables are calculated from these figures.

For heat pump No. 3 the mass flow rate was not measured therefore the nature of the applicable data is different than corresponding data for other heat pumps and deserves a brief explanation. The results of mass flow rate calculations are presented in Table C3-2 for this heat pump. Compressor shell heat loss has been calculated on the basis of compressor inlet and outlet conditions and the calculated mass flow rates. Since the compressor inlet conditions were only known at 47F outdoor temperature, the compressor shell heat loss could only be calculated at this temperature. This resulted in a value of .25 for the compressor shell heat loss factor. This value was assumed for all other operating conditions. On the basis of this assumption, compressor inlet conditions were calculated at other operating conditions and

the results are shown in Table C3-3. From compressor inlet conditions, the vaporator outlet conditions were calculated. These results are presented in Table C3-4. The other tables for heat pump No. 3 are the same as corresponding tables for other heat pumps.

Discharge line and liquid line heat losses are modeled according to the equations discussed in Section 2.2.3 of this report.

References

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Table C1-1. Steady-state Experimental Performance Data for
Heap Pump No. 1

	OUTDOOR TEMPERATURE (F)					
	<u>12</u>	<u>26.5</u>	<u>37.5</u>	<u>41.7</u>	<u>51</u>	<u>64.3</u>
Air Temperature (F)						
Evaporator Outlet	9.7	21.9	30.1	33.5	41.5	52.5
Condenser Outlet	90.2	96.2	100.0	101.2	101.5	107.6
Refrigerant Temperature (F)						
Evaporator Inlet	14.7	27.5	34.4	37.6	42.4	48.4
Evaporator Outlet	7.9	20.1	33.3	38.7	49.5	63.1
Compressor Inlet	5.0	15.7	37.2	42.8	53.7	67.2
Compressor Outlet	135	178	218	224	228	251
Condenser Inlet	125	160	201	201	205	224
Condenser Outlet	77.1	78.0	79.1	79.8	77.3	81.8
Refrigerant Pressure (psia)						
Compressor Inlet	43	53.5	60	63	67.5	76
Flow Control Inlet	211	250	265	275	295	325
Refrigerant Mass Flow Rate (lb/hr)						
	256	322	336	351	371	395
Heating Capacity (Btu/hr)						
Refrigerant Method	23791	31042	34868	36258	38407	41417
Air Method	23664	31462	32017	36604	40800	45093
Average	23727	31255	33442	36431	39603	43255
Coefficient of Performance						
Refrigerant Method	1.49	1.90	2.02	2.04	2.14	2.13
Air Method	1.49	1.93	1.86	2.06	2.27	2.32
Average	1.49	1.92	1.94	2.05	2.21	2.22

Table C1-2. Estimate of Compressor Shell Heat Loss From
Experimental Data for Heat Pump No. 1

	OUTDOOR AIR TEMPERATURE (F)					
	<u>12.0</u>	<u>26.5</u>	<u>37.5</u>	<u>41.7</u>	<u>51.0</u>	<u>64.3</u>
Refrigerant Temperature (F)						
Inlet	5.0	15.7	37.2	42.8	53.7	67.2
Outlet	135	178	218	224	229	251
Refrigerant Pressure (psia)						
Inlet	43	53.5	60	63	67.5	76
Outlet*	220	263	283	295	318	345
Refrigerant Enthalpy (Btu/lb)						
Inlet	88.2 ⁺	100.7 ⁺	109.1	109.9	111.5	113.3
Outlet	119.2	126.4	134.2	135.1	135.5	139.3
Refrigerant Mass Flow Rate * (lb/hr)	256	322	336	351	371	395
Compressor Energy (watts)	3520	3670	3960	4090	4170	4660
Shell Heat Loss (Btu/hr)	4074	4247	5078	5110	5324	5630
Shell Heat Loss Factor (%)	33.9	33.9	37.6	36.6	37.4	35.4

* Estimated by ORNL from experimental data.

+ Determined from evaporator cooling capacity.

Table C1-3. Estimate of Discharge Line Heat Loss from
Experimental Data for Heat Pump No. 1

	OUTDOOR TEMPERATURE (F)					
	<u>12.0</u>	<u>26.5</u>	<u>37.5</u>	<u>41.7</u>	<u>51.0</u>	<u>64.3</u>
Refrigerant Temperature (F)						
Compressor Outlet	135	178	218	224	229	251
Condenser Inlet	125	160	201	201	205	224
Refrigerant Pressure (psia)						
Compressor Outlet	220	263	283	295	318	345
Condenser Inlet*	215	258	278	290	313	340
Refrigerant Enthalpy (Btu/lb)						
Compressor Outlet	119.2	126.4	134.2	135.1	135.5	139.3
Condenser Inlet	117.1	122.8	130.6	130.3	130.4	133.7
Refrigerant Mass Flow Rate (lb/hr)	256	322	336	351	371	395
Discharge Line Loss (Btu/hr)	486	1159	1209	1684	1892	2291

* Condenser inlet pressure is assumed to be 5 psi less than the compressor outlet pressure.

Table C1-4. Estimate of Liquid Line Heat Loss From Experimental Data for Heat Pump No. 1

	OUTDOOR TEMPERATURE (F)					
	<u>12.0</u>	<u>26.5</u>	<u>37.5</u>	<u>42.7</u>	<u>51.0</u>	<u>64.3</u>
Refrigerant Temperature (F)						
Condenser Outlet	77.1	78.0	79.4	79.8	77.7	81.8
Expansion Valve Inlet	67.8	72.5	75.1	75.4	75.9	80.1
Difference	9.3	5.5	4.3	4.4	1.8	1.7
Refrigerant Enthalpy (Btu/lb)						
Condenser Outlet	32.1	32.5	32.9	33.1	32.4	33.7
Expansion Valve Inlet	29.5	30.9	31.6	31.8	31.9	33.1
Difference	2.6	1.6	1.3	1.3	0.5	0.6
Refrigerant Mass Flow Rate (lb/hr)	256	322	336	351	371	395
Liquid Line Loss (Btu/hr)	666	515	437	456	186	237

Table C1-5. Estimates of Evaporator Superheat Level and Suction Line Heat Gain from Experimental Data for Heat Pump No. 1

Compressor Conditions	OUTDOOR TEMPERATURE (F)					
	12.0	26.5	37.5	41.7	51.0	64.3
Inlet Pressure (psia)	43	53.5	60	63	67.5	76
Inlet Temperature (F)	5	15.7	37.2	42.8	53.7	67.2
Inlet Saturation Temp. (F)	5.13	16.0	22.0	24.6	28.3	34.9
Inlet Superheat (F) [quality]	[.82]	[.94]	15.2	18.2	25.4	32.3
Inlet Enthalpy (Btu/lb)	88.2 ⁺	100.7 ⁺	109.1	109.9	111.5	113.3
Evaporator Conditions						
Outlet Pressure*	45.5	58	62	65	69.5	78
Outlet Saturation Temp.	7.9	20.1	23.8	25.3	29.9	36.3
Outlet Temperature	7.9	20.1	33.3	37.7	49.5	63.1
Outlet Superheat (ORNL)[quality]	[.82]	[.94]	11.7	44.7	21.6	28.7
Outlet Superheat	***	***	9.5	12.7	19.6	26.7
Outlet Enthalpy	87.7	99.8	108.3	109.1	110.6	112.5
Refrigerant Mass Flow Rate (lb/hr)	256	322	336	351	371	395
Suction Line Gain (Btu/hr)	***	***	269	281	334	316

* Calculated assuming a 2 psi drop across the reversing valve. Where two phase suction line conditions exist, the reversing valve pressure drop is assumed to be that causing a saturated outlet condition.

+ Determined from evaporator cooling capacity by ORNL.

** The quality cannot be determined using this method to estimate the suction line loss.

Table C1-6. Estimate of Subcool Level in Condenser from Experimental
Data for Heat Pump No. 1

	OUTDOOR TEMPERATURE (F)					
	<u>12.0</u>	<u>26.5</u>	<u>37.5</u>	<u>41.7</u>	<u>51.0</u>	<u>54.3</u>
Refrigerant Pressure (psia)						
Condenser Outlet	215	258	278	290	313	340
Refrigerant Temperature (F)						
Condenser Outlet	77.1	78.0	79.4	79.8	77.7	81.8
Condenser Outlet Saturation	101.5	115.2	121.0	124.3	130.4	137.2
Condenser Outlet Subcooling	24.4	39.2	41.6	44.5	52.7	55.4
Subcooling Calculated by ORNL	26.0	39.0	42.2	44.9	52.8	56.6

Table C2-1. Steady-state Experimental Performance Data for
Heat Pump No. 2

	OUTDOOR TEMPERATURE (F)				
	<u>22.9</u>	<u>26.7</u>	<u>38.4</u>	<u>48.2</u>	<u>56.0</u>
Air Temperature (F)					
Evaporator Outlet	20.7	23.7	33.6	41.6	46.5
Condenser Inlet	70.5	69.9	70.5	70.3	70.0
Condenser Outlet	88.8	89.9	95.3	98.7	100.6
Refrigerant Temperature (F)					
Evaporator Inlet	18.4	21.5	30.9	38.7	43.8
Evaporator Outlet	13.3	16.2	25.0	32.3	37.5
Compressor Inlet	11.2	13.9	22.3	29.5	34.9
Compressor Outlet	170.0	173.1	182.3	193.4	209.3
Condenser Inlet	140.5	144.6	158.5	170.3	186.7
Condenser Outlet	96.2	96.3	96.4	95.8	97.7
Flow Control Inlet	89.7	88.3	90.2	90.3	91.8
Refrigerant Pressure (psia)					
Compressor Inlet	46.7	49.8	58.4	69.4	71.4
Flow Control Inlet	213.9	220.3	254.7	281.7	296.5
Air Flow Rate (CFM)					
Outdoor Fan	3244	3198	3170	3034	3029
Indoor Fan	1125	1090	1080	1138	1156
Refrigerant Mass Flow Rate (lb/hr)	235	255	306	357	382
Power Input (watts)					
Outdoor Fan	442	440	432	425	421
Indoor Fan	628	630	630	589	630
Compressor	2610	2730	3100	3390	3580
Heating Capacity (Btu/hr)					
Refrigerant Method	21431	23258	27815	32472	35656
Air Method	21739	23014	28292	33942	37426
Coefficient of Performance					
Refrigerant Method	1.71	1.79	1.96	2.16	2.26
Air Method	1.73	1.77	1.99	2.26	2.37

Table C2-2. Estimate of Compressor Shell Heat Loss From
Experimental Data for Heat Pump No. 2

	OUTDOOR AIR TEMPERATURE (F)				
	<u>22.9</u>	<u>26.7</u>	<u>38.4</u>	<u>48.2</u>	<u>56.0</u>
Refrigerant Temperature (F)					
Inlet	11.2	13.9	22.3	29.5	34.9
Outlet	170.0	173.1	182.3	193.4	209.3
Refrigerant Pressure (psia)					
Inlet	46.7	49.8	58.4	69.4	71.4
Outlet*	224	230	265	292	307
Refrigerant Enthalpy (Btu/lb)					
Inlet	105.7	105.9	106.7	107.0	108.0
Outlet	126.4	126.7	127.2	128.4	131.2
Refrigerant Mass Flow Rate (lb/hr)	235	255	306	357	382
Compressor Energy (watts)	2610	2730	3100	3390	3580
Shell Heat Loss (Btu/hr)	4041	4011	4304	3927	3353
Shell Heat Loss Factor (%)	45.6	43.0	40.7	34.0	27.5

*Compressor outlet pressure is assumed to be 10 psi above the flow control device pressure.

Table C2-3. Estimate of Discharge Line Heat Loss from
Experimental Data for Heat Pump No. 2

Refrigerant Temperature (F)	OUTDOOR TEMPERATURE (F)				
	<u>22.9</u>	<u>26.7</u>	<u>38.4</u>	<u>48.2</u>	<u>56.0</u>
Compressor Outlet	170.0	173.1	182.3	193.4	209.3
Condenser Inlet	140.5	144.6	158.5	170.3	186.7
Refrigerant Pressure (psia)					
Compressor Outlet*	224	230	265	292	307
Condenser Inlet ⁺	220	226	261	298	303
Refrigerant Enthalpy (Btu/lb)					
Compressor Outlet	126.4	126.7	127.2	128.4	131.2
Condenser Inlet	120.3	121.0	122.3	123.7	126.6
Refrigerant Mass Flow Rate (lb/hr)	235	255	306	357	382
Discharge Line Loss (Btu/hr)	1433	1505	1530	1821	1757

* Compressor outlet pressure is assumed to be 10 psi greater than the flow control device inlet pressure.

+ Condenser inlet pressure is assumed to be 6 psi greater than flow control device inlet pressure.

Table C2-4. Estimate of Liquid Line Heat Loss from
Experimental Data for Heat Pump No. 2

	OUTDOOR TEMPERATURE (F)				
	<u>22.9</u>	<u>26.7</u>	<u>38.4</u>	<u>48.2</u>	<u>56.0</u>
Refrigerant Temperature (F)					
Condenser Outlet	96.2	96.3	96.4	95.8	97.7
Expansion Valve Inlet	89.7	88.3	90.2	90.3	91.8
Difference	6.5	8.0	6.2	5.5	5.9
Refrigerant Enthalpy (Btu/lb)					
Condenser Outlet	38.1	38.1	38.1	37.9	38.6
Expansion Valve Inlet	36.1	35.6	36.2	36.2	36.7
Difference	2.0	2.5	1.9	1.7	1.9
Refrigerant Mass Flow Rate (lb/hr)	235	255	306	357	382
Liquid Line Loss (Btu/hr)	470	637	581	607	726

Table C2-5. Estimates of Evaporator Superheat Level and Suction Line Heat Gain from Experimental Data for Heat Pump No. 2

	OUTDOOR TEMPERATURE (F)				
	<u>22.9</u>	<u>26.7</u>	<u>38.4</u>	<u>48.2</u>	<u>56.0</u>
Compressor Conditions					
Inlet Pressure (psia)	46.7	49.8	58.4	69.4	71.4
Inlet Temperature (F)	11.2	13.9	22.3	29.5	34.9
Saturation Temp. (F)	9.2	12.4	20.6	29.8	31.4
Inlet Superheat (F)	2.0	1.5	1.7	~ 0	3.5
Inlet Enthalpy (Btu/lb)	105.7	105.9	106.7	107.0	108.0
Evaporator Conditions					
Outlet Pressure* (psia)	48.7	51.8	60.4	71.4	73.4
Outlet Saturation Temp. (F)	11.3	14.4	22.4	30.7	32.9
Outlet Temperature (F)	13.3	16.1	25.0	32.3	37.5
Outlet Superheat (ORNL) (F)	2.0	1.5	1.7	0.0	3.5
Outlet Superheat (F)	2.0	1.7	2.6	1.6	4.6
Outlet Enthalpy (Btu/lb)	106.0	106.1	107.1	107.6	108.3
Refrigerant Mass Flow Rate (lb/hr)	235	255	306	357	382
Suction Line Gain ⁺ (Btu/hr)	-70	-51	-122	-214	-115

* Calculated assuming a 2 psi drop across the reversing valve. Where 2 phase suction line conditions exist, the reversing valve pressure drop is assumed to be that causing a saturated outlet condition.

⁺ This number should be positive. It appears there are some experimental errors in the measured parameters.

Table C2-6. Estimate of Subcool Level in Condenser from Experimental Data for Heat Pump No. 2

	OUTDOOR TEMPERATURE (F)				
	<u>22.9</u>	<u>26.7</u>	<u>38.4</u>	<u>48.2</u>	<u>56.0</u>
Refrigerant Pressure (psia)					
Condenser Outlet*	220	226	261	288	303
Refrigerant Temperature (F)					
Condenser Outlet	96.2	96.3	96.4	95.8	97.7
Condenser Outlet Saturation	103.2	105.2	116.0	123.7	127.8
Condenser Outlet Subcooling	7.0	8.9	19.6	27.9	30.1

* Condenser outlet pressure is assumed to be 6 psi greater than the flow control device inlet pressure.

Table C3-1. Steady-state Experimental Performance Data
For Heat Pump No. 3

	OUTDOOR TEMPERATURE (F)		
	17	35	47
Air Temperature (F)			
Evaporator Outlet	12	--	38
Condenser Inlet	70	70	70
Condenser Outlet	89.5	97.4	102.9
Refrigerant Temperature (F)			
Compressor Inlet	6.5	21.7	38.7
Compressor Outlet	158.6	181.8	210.0
Condenser Inlet	148.2	173.0	200.6
Condenser Outlet	95.4	89.6	92.1
Flow Control Inlet	87.5	84.9	88.3
Refrigerant Pressure (psia)			
Compressor Inlet	44.2	59.0	69.6
Compressor Outlet	201	273	300
Air Flow Rate (CFM)			
Outdoor Fan	2133	2200	2311
Indoor Fan	1145	1199	1109
Power Input (watts)			
Indoor Fan	430	420	430
Compressor	2700	3290	3710
Heating Capacity (Btu/hr)			
Air Method	24100	33800	39500

Table C3-2. Estimate of Mass Flow Rate From Experimental Data for Heat Pump No. 3

	OUTDOOR TEMPERATURE (F)		
	17	35	47
Capacity (Btu/hr)	24100	33800	39500
Indoor Fan Energy (Btu/hr)	1467	1433	1433
Condenser Inlet Pressure* (psia)	198	270	297
Refrigerant Temperatures (F)			
Condenser Inlet	148.2	173.0	200.6
Condenser Outlet	95.4	89.6	92.1
Refrigerant Enthalpy (Btu/lb)			
Condenser Inlet	122.8	125.1	130.0
Condenser Outlet	37.8	36.0	36.8
Refrigerant Mass Flow Rate ⁺ (lb/hr)	266.0	362.8	408.4

* Condenser inlet pressure is assumed to be 3 psi less than the compressor outlet pressure.

+ mass flow rate = $\frac{\text{capacity} - \text{indoor fan energy}}{\text{condenser enthalpy drop}}$

Table C3-3. Estimate of Compressor Inlet Conditions From
Experimental Data for Heat Pump No. 3

	OUTDOOR TEMPERATURE (F)		
	17	35	47
Compressor Inlet Pressure (psia)	44.2	59.0	69.5
Compressor Energy (watts)	2700	3290	3710
Compressor Heat Loss (Btu/hr)	2303	2806	3142*
Compressor Outlet Pressure (psia)	201	273	300
Compressor Outlet Temperature (F)	158.6	181.8	210
Compressor Outlet Enthalpy (Btu/hr)	124.8	126.9	132.1 ⁺
Compressor Inlet Enthalpy	98.8	103.7	108.8 ⁺
Refrigerant Mass Flow Rate (lb/hr)	266.0	362.8	408.4
Refrigerant Inlet Superheat (F) [quality]	[93.2]	[96.9]	8.8

*At 47F, compressor shell heat loss was calculated from experimental data for inlet and outlet conditions using:

Heat Loss = Compressor energy - mass flow rate x enthalpy
change across compressor

+This was calculated from experimentally determined temperature and pressure.

Table C3-4. Estimate of Evaporator Outlet Conditions From Experimental Data for Heat Pump No. 3

	OUTDOOR TEMPERATURE (F)		
	<u>17</u>	<u>35</u>	<u>47</u>
Compressor Inlet Pressure (psia)	44.2	59.0	69.5
Compressor Inlet Enthalpy (Btu/lb)	98.8	103.7	108.8
Suction Line Heat Gain* (Btu/lb)	126.4	163.0	204.0
Refrigerant Mass Flow Rate (lb/hr)	266.0	362.8	408.4
Evaporator Outlet Enthalpy (Btu/lb)	93.3	103.2	108.3
Evaporator Outlet Pressure ⁺ (psia)	46.2	61.0	71.5
Evaporator Outlet Superheat (F) [quality]	[92.5]	[96.5]	5.0

* Suction line heat gain is assumed to be 25% of discharge line loss.

+ Determined assuming a suction line pressure drop (including reversing valve) of 2 psi.

Table C3-5. Estimate of Refrigerant Line Heat Loss From
Experimental Data for Heat Pump No. 3

	OUTDOOR TEMPERATURE (F)		
	<u>17</u>	<u>35</u>	<u>47</u>
Refrigerant Temperature (F)			
Compressor Outlet	158.6	181.8	210.1
Compressor Inlet	148.2	173.0	200.6
Refrigerant Pressure (psia)			
Compressor Outlet	201	273	300
Condenser Inlet	198	270	297
Refrigerant Enthalpy (Btu/lb)			
Compressor Outlet	124.8	127.0	132.0
Condenser Inlet	122.9	125.1	130.0
Refrigerant Mass Flow Rate (lb/hr)	266.0	362.8	408.4
Discharge Line Loss* (Btu/hr)	532	690	817

* The section of discharge line located in the outdoor unit was insulated according to the manufacturer causing the line loss to be about half that predicted by the model. Therefore, in this case, the derived values were used in the simulation model.

Table C3-6. Estimate of Liquid Line Heat Loss From Experimental Data for Heat Pump No. 3

	OUTDOOR TEMPERATURE (F)		
	<u>17</u>	<u>35</u>	<u>47</u>
Refrigerant Temperature (F)			
Condenser Outlet	95.4	89.6	92.1
Expansion Valve Inlet	87.5	84.9	88.3
Difference	7.9	4.7	3.8
Refrigerant Enthalpy (Btu/lb)			
Condenser Outlet	37.8	36.0	36.8
Expansion Valve Inlet	35.4	34.6	35.6
Difference	2.4	1.4	1.2
Refrigerant Mass Flow Rate (lb/hr)	266.0	362.8	408.4
Liquid Line Loss (Btu/hr)	638	508	490

Table C3-7. Estimate of Subcool Level in Condenser From
Experimental Data for Heat Pump No. 3

	OUTDOOR TEMPERATURE (F)		
	<u>17</u>	<u>35</u>	<u>47</u>
Refrigerant Pressure (psia)			
Condenser Outlet*	198	270	297
Refrigerant Temperature (F)			
Condenser Outlet	95.4	89.6	92.1
Condenser Outlet Saturation	95.5	118.7	126.2
Condenser Outlet Subcooling	0.1	29.1	34.1

* Assuming condenser pressure drop is negligible.

Table C4-1. Steady-state Experimental Performance Data for
Heat Pump No. 4

	OUTDOOR TEMPERATURE (F)					
	<u>17.1</u>	<u>19.8</u>	<u>30.5</u>	<u>34.3</u>	<u>38.5</u>	<u>47.0</u>
Air Temperature (F)						
Evaporator Outlet	14.7	16.7	25.3	28.2	31.4	37.8
Condenser Inlet	70.3	69.5	69.9	69.7	70.9	70.0
Condenser Outlet	84.5	85.1	88.6	89.4	91.9	93.8
Refrigerant Temperature (F)						
Evaporator Inlet	8.8	11.0	18.9	21.7	25.2	30.0
Evaporator Outlet	10.2	12.5	19.8	22.5	23.7	32.9
Compressor Inlet	19.6	21.3	27.3	30.1	32.5	38.7
Compressor Outlet	212.4	210.1	204.6	202.6	201.5	199.2
Condenser Inlet	167.3	168.2	171.3	171.9	173.1	175.1
Condenser Outlet	84.2	84.2	87.4	87.9	90.7	93.9
Flow Control Inlet	61.3	63.5	71.3	73.1	76.6	79.5
Refrigerant Pressure (psia)						
Compressor Inlet	42.7	44.5	51.4	54.2	57.9	63.8
Flow Control Inlet	175.3	176.7	188.5	192.2	200.6	208.6
Air Flow Rate						
Outdoor Fan (CFM)	2795	2720	2789	2795	2792	2516
Indoor Fan (lbm/hr)	5437	5404	5381	5361	5380	5276
Refrigerant Mass Flow Rate (lb/hr)	191.5	203.7	251.4	271.0	291.9	334.1
Power Input (watts)						
Outdoor Fan	269	268	262	259	260	255
Compressor	1996	2053	2258	2322	2416	2564
Heating Capacity (Btu/hr)						
Refrigerant Method	18941	20094	24317	26069	27750	31253
Air Method	18528	20234	24151	25345	27114	30134
Coefficient of Performance						
Refrigerant Method	2.13	2.22	2.49	2.62	2.70	2.90
Air Method	2.09	2.23	2.48	2.55	2.64	2.80

Table C4-2. Estimate of Compressor Shell Heat Loss From
Experimental Data for Heat Pump No. 4

	OUTDOOR AIR TEMPERATURE (F)					
Refrigerant Temperature (F)	<u>17.1</u>	<u>19.8</u>	<u>30.8</u>	<u>34.3</u>	<u>38.5</u>	<u>47.0</u>
Inlet	19.6	21.3	27.3	30.1	32.5	38.7
Outlet	212.4	210.1	204.6	202.6	201.5	199.2
Refrigerant Pressure (psia)						
Inlet	42.7	44.5	51.4	54.2	57.9	63.8
Outlet	183.6	185.8	199.4	204.0	213.1	221.9
Refrigerant Enthalpy (Btu/lb)						
Inlet	107.2	107.4	108.0	108.3	108.4	109.1
Outlet	135.9	135.4	134.0	133.4	132.9	132.2
Refrigerant Mass Flow Rate (lb/hr)	191.5	203.7	251.4	271.0	291.9	334.1
Compressor Energy (watts)	1996	2053	2258	2322	2416	2564
Shell Heat Loss (Btu/hr)	1314	1301	1168	1121	1092	1064
Shell Heat Loss Factor (%)	19.0	18.6	15.2	14.1	13.2	12.2

Table C4-3. Estimate of Discharge Line Heat Loss from
Experimental Data for Heat Pump No. 4

	OUTDOOR TEMPERATURE (F)					
Refrigerant Temperature (F)	<u>17.1</u>	<u>19.8</u>	<u>30.8</u>	<u>34.3</u>	<u>38.5</u>	<u>47.0</u>
Compressor Outlet	212.4	210.1	204.6	202.6	201.5	199.2
Condenser Inlet	167.3	168.2	171.3	171.9	173.7	175.1
Refrigerant Pressure (psia)						
Compressor Outlet	183.6	185.8	199.4	204.0	213.1	221.9
Condenser Inlet*	179.6	181.8	195.4	200.0	209.1	217.9
Refrigerant Enthalpy (Btu/lb)						
Compressor Outlet	135.9	135.5	134.1	133.5	133.0	132.3
Condenser Inlet	127.2	127.4	127.6	127.7	127.8	127.6
Refrigerant Mass Flow Rate (lb/hr)	191.5	203.7	251.4	271.0	291.9	334.1
Discharge Line Loss (Btu/hr)	1666	1650	1635	1572	1518	1570

* Condenser inlet pressure is assumed to be 4 psi less than the compressor outlet pressure.

Table C4-4. Estimate of Reversing Valve Heat Loss From
Experimental Data for Heat Pump No. 4

	OUTDOOR TEMPERATURE (F)					
Refrigerant Temperature (F)	<u>17.1</u>	<u>19.8</u>	<u>30.8</u>	<u>34.3</u>	<u>38.5</u>	<u>47.0</u>
Compressor Outlet	212.4	210.1	204.6	202.6	201.5	199.2
Reversing Valve Outlet	190.8	189.6	187.1	186.3	186.4	185.7
Refrigerant Pressure (psia)						
Compressor Outlet	212.4	210.1	204.6	202.6	201.5	199.2
Reversing Valve Outlet*	190.8	189.6	187.1	186.3	186.4	185.7
Refrigerant Enthalpy (Btu/lb)						
Compressor Outlet	135.9	135.4	134.0	133.4	132.9	132.2
Reversing Valve Outlet	131.9	131.5	130.6	130.3	130.0	129.6
Refrigerant Mass Flow Rate (lb/hr)	191.5	203.7	251.4	271.0	291.9	334.1
Reversing Valve Loss (Btu/hr)	776	794	855	840	847	869
Discharge Line Loss (Btu/hr)	1666	1630	1609	1545	1489	1537
Ratio of reversing valve loss to discharge line loss	.46	.49	.53	.54	.57	.57

* Reversing valve outlet pressure is assumed to be 4 psi less than the compressor outlet pressure.

Table C4-5. Estimate of Liquid Line Heat Loss from Experimental Data for Heat Pump No. 4

	OUTDOOR TEMPERATURE (F)					
	<u>17.1</u>	<u>19.8</u>	<u>30.8</u>	<u>34.3</u>	<u>38.5</u>	<u>47.0</u>
Refrigerant Temperature (F)						
Condenser Outlet	84.2	84.2	87.4	87.9	90.7	93.9
Expansion Valve Inlet	61.3	63.5	71.3	73.1	76.6	79.5
Difference	22.9	20.7	16.1	14.8	14.1	14.4
Refrigerant Enthalpy (Btu/lb)						
Condenser Outlet	34.4	34.4	35.3	35.5	36.3	37.4
Expansion Valve Inlet	27.5	28.2	30.5	31.0	32.1	33.0
Difference	6.9	6.2	4.8	4.5	4.2	4.4
Refrigerant Mass Flow Rate (lb/hr)	191.5	203.7	251.4	271.0	291.9	334.1
Liquid Line Loss (Btu/hr)	1321	1263	1207	1219	1226	1470

Table C4-6. Estimates of Evaporator Superheat Level and Suction Line Heat Gain from Experimental Data for Heat Pump No. 4

	OUTDOOR TEMPERATURE (F)					
	<u>17.1</u>	<u>19.8</u>	<u>30.8</u>	<u>34.3</u>	<u>38.3</u>	<u>47.0</u>
Compressor Conditions						
Inlet Pressure (psia)	42.7	44.5	51.4	54.2	57.9	63.8
Inlet Temperature (F)	19.6	21.3	27.3	30.1	32.5	38.7
Inlet Saturation Temp. (F)	4.8	6.8	14.0	16.7	20.1	25.3
Inlet Superheat (F)	14.8	14.5	13.3	13.4	12.4	13.4
Inlet Enthalpy (Btu/lb)	107.2	107.4	108.0	108.3	108.4	109.1
Evaporator Conditions						
Outlet Pressure# (psia)	44.7	46.5	53.5	56.2	59.9	65.8
Outlet Saturation Temp. (F)	7.0	9.0	16.1	18.6	22.0	27.1
Outlet Temperature (F)	10.2	12.5	19.8	22.5	23.7	32.9
Outlet Superheat (F)	3.2	3.5	3.7	2.9	1.7	5.8
Outlet Enthalpy (Btu/lb)	105.6	106.0	106.6	106.9	106.8	108.0
Refrigerant Mass Flow Rate (lb/hr)	191.5	203.7	251.4	271.0	291.9	334.1
Suction Line Gain (Btu/hr)	306	285	351	379	467	368
Ratio of Suction Line Gain to Discharge Line Loss	.184	.175	.218	.245	.314	.240

* Calculated assuming a 2 psi drop across the reversing valve. Where 2-phase suction line conditions exist, the reversing valve pressure drop is assumed to be that causing a saturated outlet condition.

Table C4-7. Estimate of Subcool Level in Condenser from Experimental Data for Heat Pump No. 4

	OUTDOOR TEMPERATURE (F)					
	<u>17.1</u>	<u>19.8</u>	<u>30.8</u>	<u>34.3</u>	<u>38.3</u>	<u>47.0</u>
Refrigerant Pressure (psia)						
Condenser Outlet	182.3	183.7	195.5	199.2	207.6	215.6
Refrigerant Temperature (F)						
Condenser Outlet	84.2	84.2	87.4	87.9	90.7	93.9
Condenser Outlet Saturation	89.6	90.2	94.6	96.0	99.0	101.6
Condenser Outlet Subcooling	5.4	6.0	7.2	8.1	8.3	7.7