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.[1] 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 reciprocating compressors, which differ from those levels for which the performance curves were generated.[2]

In order to simulate other heat pump components, several component models were investigated for their applicability and availability.[3-6] 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):

$$QDISL = 10\ (T_R - T_o)$$

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, and the level of refrigerant subcooling at the condenser exit was used as input.

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.[3]

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).[13] 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.[14] 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°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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TABLE 1

Heat Pump Simulation Model Input Data

<u>INDEPENDENT OF OPERATING CONDITION</u>

<u>Condenser and Evaporator</u>

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

<u>Reversing Valve and Interconnecting Tubes</u>

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

<u>Compressor</u>

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)

<u>Refrigerant Flow Control Device</u>

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

<u>DEPENDENT ON OPERATING CONDITION</u>

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)		
	EXPERIMENTAL	MODEL	% DIFFERENCE	EXPERIMENTAL	MODEL	% DIFFERENCE	EXPERIMENTAL	MODEL	% DIFFERENCE	EXPERIMENTAL	MODEL	% DIFFERENCE	EXPERIMENTAL	MODEL	% DIFFERENCE	EXPERIMENTAL	MODEL	% DIFFERENCE
Heating Capacity (Btu/hr) [a.]																		
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
Coefficient of Performance																		
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
Mass Flow Rate (Lb/hr) [b.]	256.	256.	0.0	322.	297.	8.4	336.	327.	2.8	351.	345.	1.7	371.	375.	-1.1	395.	427.	-7.5
Refrigerant Temperature (F)																		
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
Refrigerant Pressure (psia) [c.]																		
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
Air Temperature (F)																		
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 kW

b. 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
Refrigerant Temperature (F)									
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
Refrigerant Pressure (psia)[d]									
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
Air Temperature (F)									
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	18528	18941	18735	2.09	2.13	2.11
	Model			18730			2.05
	% Difference			0.0			2.9
-6.8 (19.8)	Experimental	20234	20094	20164	2.23	2.22	2.22
	Model			19887			2.13
	% Difference			1.4			4.20
.7 (30.8)	Experimental	24151	24317	24234	2.48	2.49	2.48
	Model			25011			2.41
	% Difference			-3.1			2.9
1.3 (34.3)	Experimental	25345	26069	25707	2.55	2.62	2.58
	Model			27098			2.54
	% Difference			-5.1			1.6
3.6 (38.5)	Experimental	27114	27750	27432	2.64	2.70	2.67
	Model			28943			2.61
	% Difference			-5.2			2.3
8.3 (47)	Experimental	30134	31253	30693	2.80	2.90	2.85
	Model			33013			2.83
	% Difference			-7.0			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			
			QCANF = 0.25		QCANF = .15	
	HEATING CAPACITY kW (Btu/h)	COP	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					
			RVF = 0		RVF = .25		RVF = .5	
	HEATING CAPACITY kW (Btu/h)	COP	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

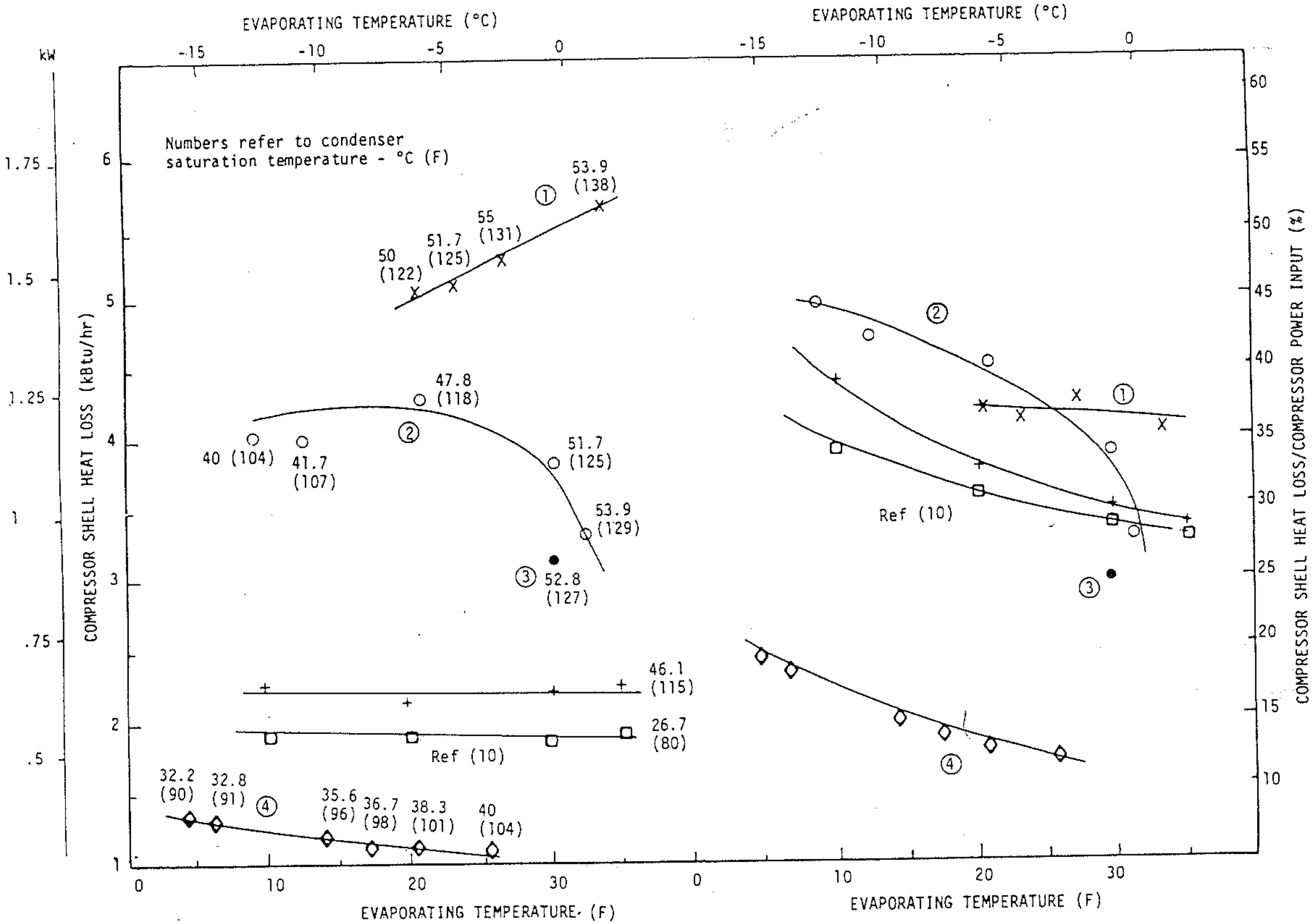

Figure 1. *Variation of compressor shell heat loss with respect to evaporating temperature*