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Simulation of a Heat Pump Operating with a Nonazeotropic Mixture

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ABSTRACT

This paper provides an overview of the model developed for simulation of steady-state performance of a heat pump working with a nonazeotropic binary mixture. The modeled heat pump consists of a hermetic, reciprocating compressor, flat-finned tube heat exchangers, a constant flow area expansion device, an accumulator, a four-way valve, and connecting tubing.

The paper discusses basic concepts in formulation of models of major heat pump components and overall program iteration scheme. Verification of the model is presented in the cooling and heating operating modes for a heat pump charged with R13B1/R152a mixture.

INTRODUCTION

Nonazeotropic refrigerant mixtures have gained extensive interest in recent years due to advantages such mixtures may offer over single-component refrigerants when used in refrigeration machines. Certain thermodynamic advantages have been hypothesized (Vakil 1983a, 1983b), in some cases measured in the laboratory (Cooper and Borchard 1979; Didion and Mulroy 1984). To facilitate verification of performance potentials of a vapor compression machine charged with a nonazeotropic refrigerant mixture, a first principle simulation model, HPBI, was developed for simulation of performance of an air-to-air heat pump working with such mixtures. This work evolved from previous modeling projects (Chi 1979; Chi and Didion 1982) and particularly (Domanski and Didion 1983, 1984) from which the model discussed here was derived. This paper provides an overview of the HPBI model and its capabilities. A full description is presented in the comprehensive report on the model (Domanski n.d.).

EVALUATION OF THERMODYNAMIC PROPERTIES

Equation of state, applied to evaluate the thermodynamic state of a nonazeotropic mixture, has to be able to provide predictions for a broad range of composition at which circulating mixtures may assume during heat pump operation. It is also important that the equation of state is accurate in prediction above the critical point of mixture components, since it may happen in practice that the calculated state of a refrigerant mixture, being below mixture critical temperature, is above critical temperature of a more volatile component.

Applied here the equation of state (Morrison 1985), consisting of strong short repulsion and weak long attraction terms, is capable to describe both the liquid and vapor properties of single-component refrigerants as well as mixtures above the critical point. The equation has the following form:

$$\frac{Pv}{RT} = \frac{1 + y + y^2 - y^3}{(1 - y)^3} - \frac{a}{RT(v + b)} \quad ; \quad y = \frac{b}{4v} \tag{1}$$

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For single-component refrigerants, parameters a and b are calculated by second degree polynomials:

$$a = a_0 + a_1T + a_2T^2 \qquad (2)$$

$$b = b_0 + b_1T + b_2T^2 \qquad (3)$$

where a_0, a_1, a_2, b_0, b_1, b_2 are constants for a given fluid.

For a binary mixture, parameters a and b are determined based on mixture component coefficients a and b and using appropriate mixing rules. For these rules and thermodynamic relationships used for evaluation of other thermodynamic properties, the reader may refer to the report on the equation of state (Morrison 1985) or to Domanski and Didion (1985), and Domanski (n.d.).

EVALUATION OF TRANSPORT PROPERTIES

Transport properties of interest are thermal conductivity and absolute viscosity of the liquid and vapor. Algorithms for these properties consist of curve-fitted correlations for prediction of properties of pure components and some kind of mixing rules. Reid et al, (1977) gave a comprehensive review of mixing rules that could be used depending on mixture components' molecular structure and component property data availability. The following four correlations were selected for R13B1-R152a mixture:

liquid	thermal	conductivity	Filoppov equation
vapor	thermal	conductivity	Wassiljewa equation
liquid	absolute	viscosity	Lobe correlation
vapor	absolute	viscosity	Chapman-Enskog kinetic theory based equation

OPERATION OF A HEAT PUMP

The purpose of heat pump operation is to pump heat from a low temperature reservoir to a high temperature reservoir. The major heat pump components from the thermodynamic standpoint to facilitate this operation are a compressor, two heat exchangers, and an expansion device. For practical reasons a heat pump also contains other elements important for both operation and for a modeling effort, like connective tubing, four-way valve, and an accumulator. A schematic of the heat pump configuration with these components is shown in Figure 1.

Manufacturers usually establish the amount of refrigerant charge in a heat pump during a prototype cooling test at specified conditions. The criteria for the charge is an appropriate amount of superheat at the compressor inlet. In a heat pump with a capillary tube, vapor superheat at the compressor inlet will vary with changing operating conditions. While during cooling operation, some superheat almost always exists, during heating operation, incomplete evaporation takes place in the evaporator; liquid refrigerant is stored in the accumulator; and high quality vapor enters the compressor. If the heat pump is charged with a nonazeotropic refrigerant mixture, accumulation of liquid in the accumulator has a further implication on operation. Since by definition a nonazeotrope exists in equilibrium phase change with its saturated liquid and vapor having different compositions (see Figure 2), stored liquid has a different composition than the circulating mixture. Furthermore, any increase in liquid accumulation in the accumulator causes a shift in composition of the circulating refrigerant making it richer in a more volatile component. If that more volatile component is the refrigerant of greater capacity (e.g., R13B1 is relative to R152a) then this composition shift will tend to mitigate the net capacity loss due to the decrease in mass flow rate of the circulating mixture. The amount of collected refrigerant in the accumulator and the magnitude of the composition shift increases with decrease of outdoor temperature during heat pump heating operation. This reduction in net compressor capacity loss is provided at operating conditions at which the supplemental (usually strip resistant) heater is used to match the heating load. Thus, the potential savings offered by mixtures is enhanced when applied to today's residential heat pump systems.

DESCRIPTION OF SUBCOMPONENT MODELS

The HPBI simulation has a modular structure. It consists of independent models of heat pump components, which are supplemented by a mass inventory and are tied together by several iteration schemes that tend to converge toward a unique solution. All the submodels are explained in detail in the comprehensive report on the model (Domanski n.d.). Models of major components (compressor, constant flow area expansion device, evaporator, and condenser) and mass inventory are briefly described below.

The major assumption in compressor simulation is that the highly dynamic process in the compressor results in a steady refrigerant vapor flow and steady thermodynamic states at four internal locations identified in Figure 1. Simulation of the compressor is performed by evaluating refrigerant pressure and enthalpy changes at flow between assigned locations. The process inside the compressor cylinder is modeled with the assumption of constant suction and discharge pressures and assuming both compression and reexpansion processes to be polytropic with the same polytropic index, n, following the equation:

$$P \cdot v^n = \text{const} \tag{4}$$

The refrigerant enthalpy increase during polytropic compression, $i_6 - i_5$ (see Figure 3), is evaluated by the equation derived from the expressions for isentropic and polytropic work of compression at the same compression ratio:

$$i_6 - i_5 = (i_{6s} - i_5) \frac{1}{\eta_p} \frac{\left(\frac{P_6}{P_5}\right)^{\frac{n-1}{n}} - 1}{\left(\frac{P_6}{P_5}\right)^{\frac{\gamma-1}{\gamma}} - 1} \tag{5}$$

where i_{6s} denotes the enthalpy obtained during the isentropic compression.

The isentropic index, γ, and the polytropic index, n, are related by the polytropic efficiency of the compressor, η_p:

$$\eta_p = \frac{\frac{\gamma - 1}{\gamma}}{\frac{n - 1}{n}} \tag{6}$$

Refrigerant pressure drop within the compressor during flow between the key locations is due to dynamic or frictional effects and is evaluated by expressions based on the following relationships, respectively:

$$\Delta P \propto \frac{m^2}{\rho} \tag{7}$$

$$\Delta P \propto \mu^{0.2} \frac{m^{0.8}}{\rho} \tag{8}$$

where
- m = the mass flow rate
- μ = the absolute viscosity
- ρ = the density

Refrigerant enthalpy change, Δi, is governed by forced convection and is evaluated based on the forced convection heat transfer equation:

$$Q = m \cdot \Delta i = h \cdot \Delta T \tag{9}$$

where h, the forced convection heat transfer coefficient, is calculated with the aid of the nondimensional heat transfer parametric expression in terms of Nusselt, Reynolds, and Prandtl numbers:

$$Nu \propto Re^{0.8}\ Pr^{0.333} \tag{10}$$

Five heat transfer parameters, four pressure drop parameters and polytropic efficiency have to be submitted to the program based on one compressor (heat pump) test. The program also requires standard electric motor characteristics: rpm and efficiency versus load.

The refrigerant pressure drop through the capillary tube or orifice is due to sudden contraction of flow area at the entrance and to flow through the expansion device itself. Pressure drop due to sudden contractions, ΔP, which consists of acceleration loss and entrance friction loss, is evaluated by the equation:

$$\Delta P = (1 + K)\ \rho\ \frac{V^2}{2} \tag{11}$$

where K = the contraction coefficient

ρ = the density (based on smaller cross section)

V = the velocity (based on smaller cross section)

For evaluation of pressure drop due to flow inside the tube itself, the tube is assumed to have a constant inner diameter, and to be straight and horizontal, and the flow is assumed to be one-dimensional, homogeneous, and adiabatic.

In most general cases, refrigerant flow in a capillary tube consists of a liquid flow part and two-phase flow part. Single-phase flow and two-phase flow are considered by the model separately, applying the equation of motion to the respective portions of the tube:

$$\left(\frac{A}{m}\right)^2 \int_{P_{in}}^{P_{out}} \rho dP + \frac{2}{D}\int_{L_{in}}^{L_{out}} f dL + \ln \frac{\rho_{in}}{\rho_{out}} = 0 \tag{12}$$

where A = the flow cross-section area

D = the inside diameter of a tube

f = the Fanning friction factor

L = the length of a tube

P = the pressure

For single-phase (liquid) flow, the third term of this equation equals zero and the equation reduces to the Fanning pressure drop formula. For two-phase flow, the momentum equation is solved in its full form.

Simulation of a capillary tube and orifice is performed in a highly iterative scheme since the number of factors affecting the final solution for mass flow rate are functions of mass flow itself. These factors are: pressure drop at the entrance, portion of the tube with a single-phase and two-phase flow, critical pressure, and flow friction factors. It should be noted that this capillary tube and orifice model is based on fundamental fluid mechanics without sufficient experimental data for its precise verification. The two-phase portion of the flow, most critical for a capillary tube, is modeled utilizing the Fanno flow theory just as if it was a single-component flow.

Condenser and evaporator models are based on a tube-by-tube simulation method. This method depends on imaginary isolation of each tube with associated fins from the coil assembly and evaluation of performance of each coil tube independently. Summation of all tube performances result in total coil capacity. This simulation approach adds considerably to the number of iterations and, thus, overall run time; however, it requires no empirical data as to the heat exchangers' performance, only its geometry. The coil simulation consists of two parts: logical part and performance evaluation part.

The logical part traces refrigerant flow through the coil, taking into account merging and splitting points of circuit, and selects tubes for calculation in appropriate order. The logical part of the program also traces airflow and air properties when passing through the coil. Each coil tube is associated with a unique set of refrigerant and air inlet parameters resultant from heat transfer and pressure drop at upstream tubes.

Heat transfer to/from an individual tube is calculated with the aid of a cross-flow heat exchanger theory. Derived equations allow for detailed consideration of the tube in which change from one flow mode into another occurs; lengths of the tube with single- and two-phase flows are evaluated and heat transfer and pressure drop for each part of the tube are evaluated separately by appropriate relationships. Overall heat transfer coefficient for a tube is calculated using the following equation:

$$U = \left[\frac{A_o}{h_i A_{p,i}} + \frac{A_o x_p}{A_{p,m} k_p} + \frac{\delta}{k_w} + \frac{1}{h_o \left(1 - \frac{A_f}{A_o}(1 - \Phi)\right)} \right]^{-1} \quad (13)$$

<u>where</u> $$h_o = h_c \left[1 + \frac{i_{fg,w}(w_a - w_w)}{C_{pa}(T_a - T_w)} \right] \quad (14)$$

A_f = the surface area of fins associated with a tube

$A_{p,i}$ = the inside surface area of a tube

$A_{p,m}$ = the mean surface area

A_o = the total outside surface area of a finned tube

C_{pa} = the specific heat at constant pressure of air

h_c = the air-side forced-convection heat transfer coefficient for a finned dry tube

h_i = the convection heat transfer coefficient inside a tube

h_o = the air-side forced-convection heat transfer coefficient for a finned dry tube

$i_{fg,w}$ = the latent heat of evaporation of water

k_p = the thermal conductivity of air

k_w = the thermal conductivity of the tube material

T_a = the temperature of air

T_w = the temperature of water or frost

w_a = the humidity ratio of air

w_w = the humidity ratio of saturated air at T_w temperature

x_p = the thickness of a tube wall

δ = the thickness of a liquid (frost) layer on a fin

Φ = the efficiency of a fin

Air-side forced convection heat transfer coefficient for a finned flat tube, h_c, is calculated by the following correlation (Briggs and Young 1962):

$$h_c = 0.134 \frac{k_a}{D_o} Re^{0.681} Pr^{0.333} \left(\frac{z}{s}\right)^{0.2} \left(\frac{z}{t}\right)^{0.1134} \tag{15}$$

<u>where</u> D_o = the outside diameter of a tube

k_a = the thermal conductivity of air

s = the height of a fin

t = the thickness of a fin

z = the distance between adjacent fins

This correlation, sensitive on finned-tube geometry, allows for detailed analysis of air-side heat transfer in case of air being cooled down with occurring condensation and/or frosting.

Inside tube forced convective heat transfer coefficients are calculated by the following correlations: single-phase - McAdams (1933), two-phase with condensation - Traviss et al. (1971), two-phase with evaporation - correlation in the form of Dengler and Addoms (1956) based on analysis of data of R13B1/R152a mixture evaporative flow (Radermacher et al. 1982).

Inside tube pressure drop is calculated by the following correlations: single-phase - Fanning equation, two-phase with condensation - Lockhart and Martinelli (1949), two-phase with evaporation - Pierre (1964) corrected for pressure drop of R13B1/R152a mixture observed during tests (Mulroy 1984).

Thermodynamic equilibrium inside the tube is assumed for calculating refrigerant two-phase pressure drop and heat transfer coefficient. Thermodynamic and transport properties of the vapor and liquid in the tube are calculated for average temperature and pressure taken between the tube's inlet and outlet. Compositions of liquid and vapor phase being in equilibrium are obtained with the aid of the equation of state. As indicated, only evaporative heat transfer and pressure drop data of nonazeotropic mixture were available for this modeling effort. Condensation case is handled using straightforward single-component fluid correlations supplied with the mixture properties.

MASS INVENTORY

Since a heat pump is a hermetric system, the amount of refrigerant contained in it is the same all the time although refrigerant may be differently distributed among system components at different operating conditions. This heat pump simulation performs a mass inventory of the mixture and mass inventory of one of the components in the course of iteration for the final solution. Mass inventory depends on determination of internal volume of the particular system components and evaluation of the average refrigerant density occupying particular internal volume. Thus the internal volume must be determined and supplied to the program with other heat pump input data.

The average density calculation method depends upon the refrigerant flow pattern. For single-phase flow or low quality two-phase flow (slug flow with slip ratio close to 1) densities of liquid and vapor are obtained from the equation of state and are weighted with quality where applicable. Average density for annular flow regime is evaluated in terms of densities of saturated liquid and vapor, and the void fraction, α, as correlated based on Lockhart-Martinelli experiment (Wallis 1969):

$$\alpha = (1 + X_{tt}^{0.8})^{-0.378} \tag{16}$$

where X_{tt} is the Lockhart-Martinelli parameter.

In case of incomplete evaporation and liquid refrigerant storage in the accumulator, the amount of stored liquid is determined by calculating the required static pressure difference at the oil return hole to obtain the known vapor quality at the compressor inlet.

Accuracy of mass inventory depends on accurate determination of average density and internal volume occupied by the refrigerant. In order to minimize error of computation, mass inventory results are used on a relative basis, i.e., calculated charge is compared, not to the actual charge, but to the mass of refrigerant calculated for operating condition at which superheat of compressor inlet is known as a design parameter.

PROGRAM LOGIC

During steady-state operation of a heat pump at a given operating condition, steady refrigerant parameters are established throughout the system. These parameters are unique to that operating condition and are established due to balances taking place in the system. This model recognizes 13 key system locations as identified in Figure 1. These locations are also shown on the thermodynamic cycle illustrated in the pressure-enthalpy diagram in Figure 3. The program iterates refrigerant parameters at these locations utilizing the following four balances: pressure balance, enthalpy balance, total refrigerant mixture mass conservation, and mass conservation of one of mixture components.

The pressure balance implies that the increase of refrigerant pressure across the compressor is equal to the summation of the refrigerant pressure drops throughout the rest of the components. Furthermore, the pressure drop through each component has to be adequate so the mass flow rate through them is the same. The enthalpy balance (derived from energy balance and the fact that mass flow rate through each of the components is the same) implies that the net refrigerant enthalpy change in the system has to be equal to zero, as is true with all properties in a cycle.

Mass inventory of refrigerant mixture and mass inventory of one of the mixture components is a straight application of mass conservative law to a hermetic system. It should be noted that by inventoring both the total mixture mass and the mixture component mass, a verification of the composition of the circulating mixture is obtained.

The input to the program consists of operating condition, heat pump data, refrigerant data, and composition of charged refrigerant. Additionally, four estimates are required to start calculations, namely: refrigerant dew-point temperature at discharge from compressor cylinder, refrigerant dew-point temperature and superheat (or quality) at compressor inlet, and composition of circulating refrigerant. The iteration scheme contains four nested loops based on the four discussed balances to iterate the initial estimates on a one-by-one basis. The final results of calculation include refrigerant state at all 13 key heat pump locations, refrigerant mass flow rate, system capacity, electrical energy input, and coefficient of performance.

The logic of the program is explained in Figure 4. To iterate solution in the cooling mode, the program required approximately 15 minutes of CPU time on a computer*. A comparable run with a pure component refrigerant (e.g., R22) would required about 5 minutes of CPU time. Neither of these runs included an accounting of a composition shift, and so the increased run time is due primarily to the complexities of the mixture's equation of state and properties determination. Heating mode calculations including iteration of shift in composition of circulating mixture (i.e., the outermost loop in Figure 4) required up to 40 CPU minutes.

MODEL VERIFICATION AND HEAT PUMP PERFORMANCE PREDICTIONS

The model was verified by comparing model predictions with laboratory test results obtained on a split residential heat pump placed in environmental chambers. The heat pump was charged with R13B1/R152a mixture of composition equal 0.65 (weight fraction of R13B1). Laboratory tests and simulation runs were performed at two of the DOE standard cooling and heating rating points. Test and simulation results along with operating conditions are presented in Table 1 and Table 2. Table 2 contains less information than Table 1, since not all measurements were taken during the laboratory tests in the heating mode due to incomplete mixture condensation in the condenser and incomplete evaporation in the evaporator.

Simulation in the cooling mode at 95F outdoor temperature was performed with imposed vapor superheat at the evaporator outlet (9.7F), which was the same as during a laboratory test. This amount of superheat was established during the test by charging the system with appropriate amount of refrigerant. Applying this principal to the model, the amount of refrigerant calculated by the model at these test conditions at the same superheat was used as refrigerant charge for simulation purposes to iterate refrigerant vapor superheat/quality at the compressor inlet at other operating conditions.

*Sperry-Univac 1100/82.

During operation in the cooling mode at 82F outdoor temperature, a superheat existed at the evaporator outlet, no liquid refrigerant collection in the accumulator took place, and the composition of the circulating mixture was equal to that of the charged mixture.

In the heating mode operation, accumulation of liquid refrigerant in the accumulator occured and shift in composition of the circulating mixture took place. Since the amount of stored liquid refrigerant and composition shift have to be evaluated during simulation, the heating mode results are more important for validating the program.

Obtained agreement between simulation and test results is much better than was expected considering the number of simplifications taken in formulating the model. Either the simplifications are insignificant in their effect on performance or there exist some cancelling effects. What can be concluded is that the model is capable of predicting overall heat pump performance within an approximate 6% band.

CONCLUSIONS

The model HPBI described here is capable of simulating the performance of a heat pump equipped with a capillary tube or orifice and charged with a binary non-azeotropic mixture. The model provides performance data at imposed operating conditions in the cooling or heating mode. There is no restriction on refrigerant state anywhere in the system; refrigerant state at system key locations provided with the final solution are iterated entirely utilizing balances contained within the model logic. The model is first principle based; it can be applied to most commercially available residential heat pumps. Applicability of the model is not limited by the size of the modeled hardware, but rather by design, which has to be compatible with the formulated model. The model has been verified at both cooling and heating operation for a heat pump charged with R13B1/R152a nonazeotropic mixture.

The model in its present form can be useful in a number of applications, eliminating the need for expensive laboratory tests. It can be efficiently used to evaluate potentials of different mixtures working in a heat pump system. It can also guide an engineer during the early stages of design by providing data of a system with an individual component being altered to see the effect on performance. Heat pump performance information can also be obtained with the aid of the model for a wide range of circulating compositions and operating conditions.

The model, having modular structure, is also suitable for modifications to examine different hardware configurations and different refrigerant passes. Finally, the model, consisting of 68 functions and subroutines, is a series of nonazeotropic mixture property routines, fluid mechanics and heat transfer routines, and major heat pump component algorithms, which can be used with slight modification independently for specific analyses of interest.

NOMENCLATURE

A = the flow cross-section area

A_f = the surface area of the fins associated with a tube

$A_{p,i}$ = the inside surface area of a finned tube

$A_{p,m}$ = the mean surface area of a finned tube

A_o = the total outside area of a finned tube

a = the equation of state parameter associated with intermolecular attraction

a_o, a_1, a_2 = the single-component fluid constants for the evaluation of parameter a

b = the equation of state parameter associated with the hard core of a molecule

b_o, b_1, b_2 = the single-component fluid constants for the evaluation of parameter b

C_{pa} = the specific heat at constant pressure of air

D = the inside diameter of a tube

D_o = the outside diameter of a tube

DB = the dry bulb

f = the Fanning friction factor

h = the forced-convection heat transfer coefficient calculated by Equation 10

h_c = the air-side forced-convection heat transfer coefficient for a dry finned tube calculated by Equation 15

h_i = the convective heat transfer coefficient inside a tube

h_o = the air-side forced convection heat transfer coefficient for a wet finned tube calculated by Equation 14

i = the enthalpy

$i_{fg,w}$ = the latent heat of evaporation of water

i_5 = the enthalpy of the refrigerant in a cylinder during a suction stroke

i_6 = the enthalpy of the refrigerant in a cylinder during a discharge stroke

K = the contraction coefficient

k_a = the thermal conductivity of air

k_p = the thermal conductivity of the tube material

k_w = the thermal conductivity of water (frost) deposited on the outer finned tube surface

L = the length of a tube

m = the mass flow rate of refrigerant

Nu = the Nusselt number

n = the polytropic index

P = the pressure

P_5 = the suction pressure in a cylinder

P_6 = the discharge pressure in a cylinder

Pr = the Prandtl number

Q = the heat transfer rate

R = the gas constant

Re = the Reynolds number

s = the height of a fin

T = the temperature

T_a = the temperature of air

T_w = the temperature of water or frost

t = the thickness of a fin

U = the overall heat transfer coefficient

V = the velocity

v, v_L, v_V = the specific volume, v_L and v_V refer to a liquid and a vapor, respectively

WB = the wet bulb

w_a = the humidity ratio of air

w_w = the humidity ratio of saturated air at T_w temperature

X_{tt} = $\left(\frac{1 - x}{x}\right)^{0.9} \left(\frac{v_L}{v_V}\right)^{0.5} \left(\frac{\mu_L}{\mu_V}\right)^{0.1}$ = the Lockhart-Martinelli parameter

x_p = the thickness of a tube wall

x = the quality

y = a function defined in Equation 1

z = the distance between adjacent fins

α = the void fraction

γ = the isentropic index

δ = the thickness of a liquid (frost) layer on a fin

η_p = the polytropic efficiency

μ, μ_L, μ_V = absolute viscosity, μ_L and μ_V refer to a liquid and vapor, respectively

ρ = the density

Φ = the efficiency of a fin

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TABLE 1

Laboratory Test and Computer Simulation Results in the Cooling Mode

Outdoor Temperature	Data Source	Circulating Mixture Composition	Capacity	Energy Consumption Rate	EER	Refrigerant Mass Flow Rate	Evaporator Outlet Dew-Point Temperature	Superheat Leaving Evaporator	Compressor Discharge		Condenser Outlet	
									Pressure	Temperature	Pressure	Temperature
F		Weight Fraction of R13B1	Btu/h	Watt	Btu/(h · Watt)	lb/h	F	F	psia	F	psia	F
95F	Simulation	.65	28050	4359	6.43	562	50.5	9.7	304	212	291	101.6
	Test	.65	27880	4322	6.45	568	51.8	9.7	297	208	292	111.0
82F	Simulation	.65	30000	4047	7.41	557	48.6	13.2	263	195	249	88.0
	Test	.65	29670	4048	7.33	552	47.8	14.7	252	196	247	97.2

Indoor Air Conditions - 80F DB/67F WB

TABLE 2

Laboratory Test and Computer Simulation Results in the Heating Mode

Outdoor Temperature	Data Source	Circulating Mixture Composition	Capacity	Energy Consumption	COP	Evaporator Outlet Temperature	Compressor Discharge		Condenser Outlet	
							Pressure	Temperature	Pressure	Temperature
		Weight Fraction of R13B1	Btu/h	Watt	--	F	psia	F	psia	F
47F DB 43F WB	Simulation	.667	35180	3673	2.81	37.1	230	159	220	84.9
	Test	.667	33800	3776	2.62	40.1	231	176	231	93.3
17F DB 15F WB	Simulation	.743	21910	3206	2.00	11.4	211	150	203	80.2
	Test	.734	21303	3298	1.89	14.1	209	177	195	82.1

Indoor Air Temperature 70F

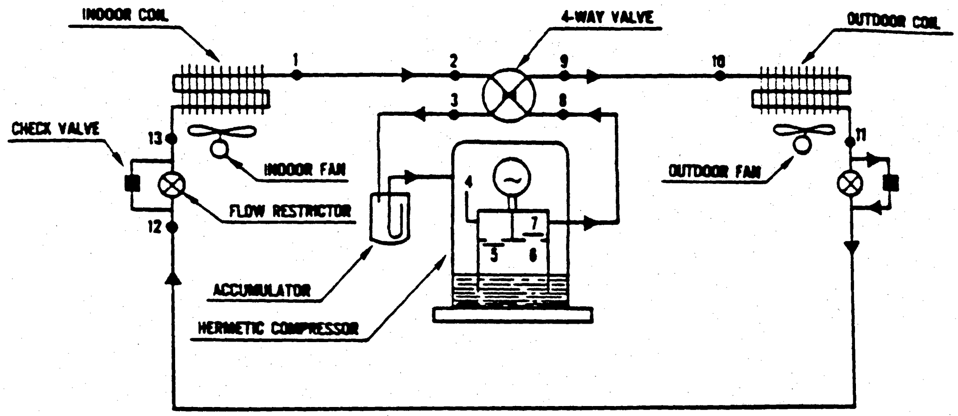
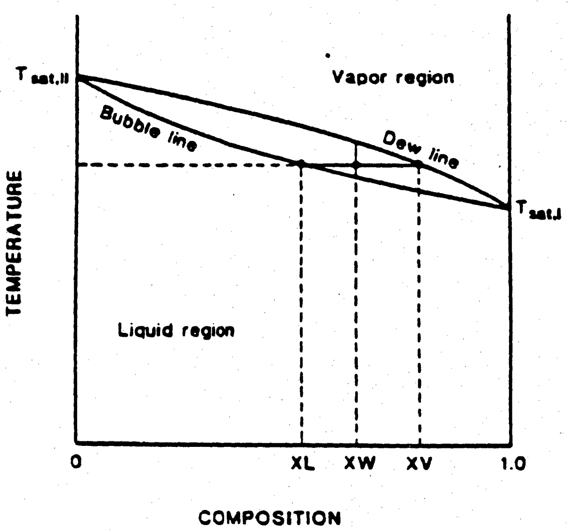


Figure 1. *Schematic of a heat pump*

Note: *Refrigerant flow direction is marked for cooling operating*
Numbers five and six situated in the compressor correspond to refrigerant state before and after the compression process



$T_{sat,I}$ - saturation temperature of more volatile component
$T_{sat,II}$ - saturation temperature of less volatile component
XL - composition of saturated liquid (fraction of more volatile component)
XW - mixture composition (fraction of more volatile component)
XV - composition of saturated vapor (fraction of less volatile component)

Figure 2. *Temperature-composition diagram for a nonazeotropic binary mixture*

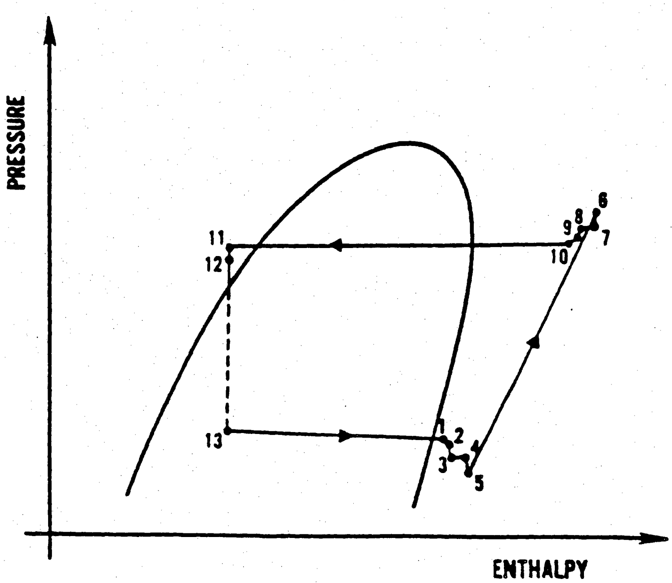
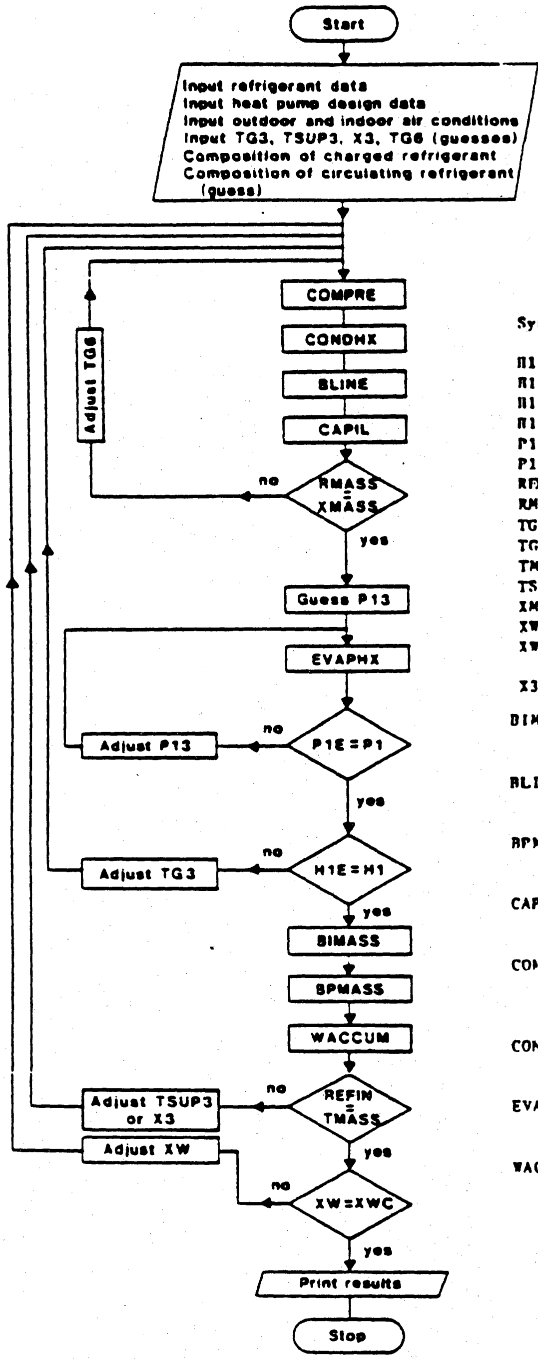


Figure 3. *Thermodynamic cycle realized by a heat pump (for number location refer to Figure 1)*



Symbols:

H1 - refrig. enthalpy at evaporator outlet calculated by COMPRE
H1E - refrig. enthalpy at evaporator outlet calculated by EVAPHX
H12 - refrig. enthalpy at point 12
H13 - refrig. enthalpy at point 13
P1 - refrig. pressure at evaporator outlet calculated by COMPRE
P1E - refrig. pressure at evaporator outlet calculated by EVAPHX
REFIN - mass of refrig. charged into a machine
RMASS - refrig. mass flow rate through a compressor
TG3 - refrig. dew point temp. at point 3
TG6 - refrig. dew point temp. at point 6
TMASS - total mass of refrig. calculated by inventory program
TSUP3 - refrig. superheat at point 3
XMASS - refrig. mass flow rate throuth an expansion device
XW - circulating refrigerant mixture composition used for the calculations
XWC - circulating refrigerant mixture composition resulting from the component refrigerant mass inventory
X3 - refrig. quality at point 3

BIMASS - calculates mass of refrigerant in a coil
Input: refrig. state at each coil tube end
Output: mass of refrig. in a coil

BLINE - calculates pressure drop in a liquid line
Input: RMASS and refrig. state at point 11
Output: refrig. state at point 12

BPMASS - Calculates mass of refrig. in a tube
Input: refrig. state at tube ends
Output: mass of refrig. in a tube

CAPIL - calculates performance of an expansion device
Input: refrig state at point 12 and pressure at point 13
Output: XMASS

COMPRE - calculates performance of a compressor with a 4-way valve and tubing connecting compresor with both coils
Input: refrig. state at point 3 and pressure at point 6
Output: RMASS and refrig. state at points 1 through 10

CONDHX - calculates performance of a condenser
Input: RMASS and refrig. state at point 10
Output: refrig. state at point 11

EVAPHX - calculates performance of an evaporator
Input: RMASS and refrig. state at point 13
Output: refrig. state at point 1

WACCUM - calculates mass of refrigerant in an accumulator
Input: refrig. state in an accumulator
Output: mass of refrig. in an accumulator and mean refrig. composition

Figure 4. Overall logic of the program HPBI