EXPERIMENTAL STUDY OF EVAPORATIVE HEAT TRANSFER FOR A NON-AZEOTROPIC REFRIGERANT BLEND AT LOW TEMPERATURE

A Thesis Presented to
The Faculty of the College of Engineering and Technology
Ohio University

In Partial Fulfillment
of the Requirements for the Degree
Master of Science

by
Chuan Weng/
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<td>h</td>
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<td>(-)</td>
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<td>$y_{i,e}$</td>
<td>mole fraction of component i in the bulk vapor</td>
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<td>work done to the first stage of a cascade system</td>
<td>Watts</td>
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<td>$W_B$</td>
<td>work done to the high stage of a cascade system</td>
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<td>a</td>
<td>void fraction</td>
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<td>aₘ</td>
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<td>(-)</td>
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Chapter 1
Introduction

1.1 Objective of This Research Effort

This research is to experimentally examine the heat transfer performance of the non-azeotropic refrigerant blends (NARBs) of R14 (Carbon Tetrafluoride) and R114 (Dichlorotetrafluoroethane) during evaporative flow under various conditions. It is part of a larger project titled Development of Low Temperature Freezer Design Tools, which has been jointly supported by The Edison Seed Development Fund of Ohio's Thomas Edison Program and Forma Scientific, Inc. In this thesis work, heat transfer coefficients of the two phase flow for the NARB and correspondent pressure drop data are to be investigated. The experimental results are compared with those predicted by several empirical heat transfer models, and the most suitable one is recommended. The knowledge will later be applied in a complete simulation of a refrigeration system using the non-azeotropic refrigerant blend of R14 and R114 as its working fluid.

Heat transfer study of NARBs is a fertile research area in heating, refrigeration and air-conditioning industries for the following reasons:

1) NARBs offer unique characteristics as refrigerants. Major improvements in the performance of heat pump
systems such as efficiency, capacity, and capacity modulation can be achieved by utilizing temperature and composition shifts of the working fluid during the phase change of the mixtures.

2) Successful design of components in a refrigeration system using such a refrigerant blend requires thorough understanding of the behavior of the mixture candidates in both the evaporative and condensing flows, which varies from the choices of individual refrigerant components and even the compositions. The applicability of conventional correlations for heat transfer calculations becomes questionable since a new and possibly more limiting phenomenon has been introduced by the existence of the mixture, that of mass diffusion. It has been shown [Radermacher, 1983] that the prediction of the local heat transfer coefficient of a non-azeotropic mixture is a function of the rate at which the more volatile component can diffuse to the vapor bubble region.

3) The environmental issues such as 'ozone layer depletion' and 'green house effect' are becoming so serious that industries are searching for refrigerant alternates that may have a less deteriorating effect to the ozone layer and less of a green house effect. These refrigerants may be
non-azeotropic.

Through experimental observation, this thesis attempts to gain information regarding heat transfer coefficients and pressure drops for evaporative flow of the mixture of R14 and R114 under different conditions.

1.2 Basic Knowledge of Non-Azeotropic Refrigerant Blends

The term, "non-azeotropic refrigerant blend" derives from that of the "azeotropic" refrigerant blend which exhibits a constant boiling temperature under an evaporation process at a constant pressure. For a mixture of two components, according to Raoult's law, the partial pressures of each component are proportional to the mixture concentration. The total pressure equals the sum of the partial pressures. In general, however, the composition of the vapor in equilibrium with the liquid under a certain temperature will be richer in the more volatile component than will the composition in the liquid phase. Figure 1.1 indicates the compositions corresponding to the temperatures.

If there are special cases when the total vapor pressure curve passes through a maximum or minimum, according to the theorem of Gibbs-Konovalov, the equilibrium vapor evolved by this liquid must have the same composition as the liquid; that is, the curves of liquid and vapor composition must touch tangentially at the concentration of maximum or
minimum pressure. This concentration is defined to be the azeotropic composition as can seen in Figure 1.2.

Figure 1.2 Binary solution with a minimum/maximum boiling-point azeotropic.
The azeotropic concentration is slightly temperature dependent. Typical azeotropic refrigerant mixtures are R500(R12/R152a), R501(R22/R12), R502(R22/R115) and others of 500 series. They are minimum boiling point azeotropes (such as Figure 1.2A).

Non-azeotropic blends are therefore defined to be the mixtures that do not form an azeotrope at any given concentration, i.e., mixtures whose total pressure curve does not pass through an extreme value. Figure 1.1 also shows the vapor liquid equilibria of a non-azeotropic refrigerant blend.

At a given pressure, a non-azeotropic mixture will have bubble-point and dew-point curves as shown in principle in the T,x-diagram such as Figure 1.3.

If such a mixture is heated, starting with a liquid under a certain pressure, the concentration is the same as the bulk one, $X_B$. As the temperature rises and reaches the bubble-point curve, the first bubble appears, which is initially rich in the more volatile component. The liquid, thus, becomes a little richer in the less volatile component. The system temperature tends to increase until it reaches a new equilibrium at point C, where the vapor has the concentration of $X_F$ and the liquid, $X_E$. As this process proceeds, the vapor concentration drops relatively because more component B is joining the vapor. Simultaneously, the liquid concentration continues to move
Figure 1.3 NARB operating cycle for evaporation, TX plot.

toward the direction at which the A component is highly concentrated while the system temperature is rising. This process continues until the last drop of liquid disappears, which is most concentrative in the less volatile component at point G.

In this way, the vaporization of a NARB takes place over a range of temperature defined by the intersection of the BD line with the bubble and dew lines. The more volatile component will thus be enriched in the vapor phase, successively increasing the mole fraction for the less volatile component in the liquid phase and increasing the system temperature. In general, as the pressure increases, the bubble and dew lines are shifted upward.
1.3 A Background of Low Temperature Refrigeration

An artificial low temperature environment of -130°C or below is needed in many fields of sciences. There are several ways to achieve the low temperature environment. Three of them are: 1) multi-stage cascade refrigeration systems, 2) liquid nitrogen spray systems, and 3) single-stage mixed refrigerant systems. To give an idea of how these systems work and compare with each other, a review of some aspects of these systems are necessary.

(1) **Multi-stage cascade refrigeration systems**  A cascade cycle is an arrangement of simple vapor-compression cycles in series, such that the condenser of a lower temperature cycle provides the heat input to the evaporator of a higher temperature cycle. Figure 1.4 is a schematic for a cascade refrigeration cycle.

![Figure 1.4 Equipment schematic for a cascade refrigeration cycle.](image-url)
In this figure, a quantity of heat $Q_L$ is transferred from a low-temperature source of temperature $T_L$ to the reversed heat engine. The reversed heat engine operates through a cycle during which net work $W_A$ is added to the engine and a quantity of heat $Q_{EXC}$ is transferred to a higher temperature sink of temperature $T_H$, which is, in this case, the evaporator of the higher temperature cycle. This process can be expressed by the following using the First Law of Thermodynamics:

$$Q_L + W_A - Q_{EXC} = 0 \quad (1-1)$$

In the same way, the higher temperature cycle absorbs the amount of heat dumped by the lower cycle and transfers it to the ambient. Thus the refrigeration process is complete. Equation 1-2 gives the total energy balance for this kind of system:

$$Q_L + W_A + W_B - Q_H = 0 \quad (1-2)$$

The heat, $Q_H$, removed by the system, includes the work done on both cycles. This kind of system achieves a low temperature, splitting the large temperature difference between the condenser and evaporator into two stages. The desired values for the evaporator and the condenser are met by utilizing different refrigerants in each stage. These
refrigerants are selected to have the proper vapor pressure-saturation temperature variations. Although the cycle is reliable, there are practical difficulties in design and service. There are:

A. High compression ratios. A two-stage two-compressor cascade system using R502 and R503 refrigerants with an air-cooled condenser to achieve operating temperatures of -90°C requires operating both compressors at relatively high compression ratios, frequently as high as 14:1 [White, 1984]. Thus, the compressor life is shortened due to an increased operating discharging temperature.

B. Complicated configuration. As can seen from Figure 1.1, the system comprises two totally separate cycles, which increases the complexity of the system. It requires different start-up procedures, different refrigerants and charges. The complicated system can suffer other potential problems such as the compressor overheat, internal leak in the heat interchanger, insufficient insulation on the low stage liquid line which generates excessively flash gas and some others.

(2) Liquid nitrogen cooling system. In the LN_2 vapor system, chamber cooling is achieved by the LN_2 boil-off from a liquid reservoir. Figure 1.5 is a diagram of this
expendable refrigerant system.

When the liquid nitrogen tank is filled and the cold space is loaded, the temperature is selected at the main control. A temperature sensing device anticipates any temperature changes in the cold space, and it allows the controller to open the liquid line solenoid valve to spray the LN\textsubscript{2} to the store space, generating the desired temperature. The physical characteristics of such a design allow a temperature gradient to form throughout the cold chamber, whereby those specimens stored in the lower levels see colder temperatures than those positioned elsewhere. The extent of this gradient could be substantial.

Figure 1.5 A liquid nitrogen cooling system.
However, the system has advantages over others, in that it costs less, easily provides the low temperature environment, and most importantly, the nitrogen gas in the cold chamber takes the place of oxygen, the refrigerated items are preserved in such an inert atmosphere that the negative effect of temperature gradient to the perishable items is therefore compensated.

(3) **Single-stage mixed refrigerant systems** These are the systems that utilize non-azeotropic refrigerant blends as their working fluids. Figure 1.6 is one of the forms of these systems whose performance has shown promise. [White, 1984]

These systems employ non-azeotropic refrigerant mixtures in a single compression circuit. One of the characteristics of these systems is that a portion of the cooling effect is used for some of the condensing of the refrigerant mixtures. As shown in Figure 1.6, a mixture is partially condensed in an air or water-cooled condenser and the vapor and liquid phases are separated in the cooling cycle. The liquid phase, which is rich in higher boiling fractions, is throttled to suction pressure and evaporated to produce cooling. This cooling effect is used in special heat exchangers to condense the separated vapors which contain lower boiling fractions. The resulting condensate is expanded to suction pressure to create very low temperature cooling. These steps are repeated as needed to
achieve the design operating temperature of -135°C. The refrigerant mixture returns to the compressor to repeat the cycle.

Various benefits of refrigerant mixtures have been recognized for nearly a century in the application of refrigeration and heat pump systems, yet their application remains quite limited. However, in recent years, interest in non-azeotropic mixtures has increased dramatically due to the fact that much benefit has been found for improving the performance of refrigeration systems.

Figure 1.6 Schematic diagram of a mixed refrigerant auto-cascade system.
Efforts were undertaken within the last decade to develop analytical models for research and development of systems using such refrigerant blends. Theoretically, the systems enjoy benefits of non-azeotropic refrigerant mixtures from several points:

A. Energy savings: Energy may be saved by improving the coefficient of performance (COP) for the system. One way to accomplish this is to reduce the superheat in the compressor discharge vapor through a properly selected mixture of working fluids. Another is to minimize the temperature difference between the entering temperatures of the outside fluids (heat source and heat sink) of some systems such as heat pumps and the refrigerant temperature during phase change by increasing the heat transfer area and/or the flow rate of the outside fluids. For a single refrigerant cycle system with an isothermal phase change process, these two methods could become impractical because frequently, more non-isothermal heat transfer in the heat sink and source would be desired for the purpose of air heating and conditioning. This non-isothermal heat transfer imposes conflicts to the purpose of improvement of system COP. With a non-azeotropic refrigerant blend, the COP can be increased through a better matching of the
temperature to the heat source and the sink by use of its non-isothermal phase change. To illustrate this effect, Figure 1.7 is a comparison of NARB cycle with the isothermal one in terms of a COP improvement. Under the same conditions with the isothermal cycle, the NARB cycle lowers the temperature lift by 2 degrees by utilizing the temperature shifts during the phase change. The COP is thus increased from 7.13 to 7.29.

![NARB and Isothermal Cycle Comparison](image)

Figure 1.7 NARB cycle compared to isothermal.

B. System optimization: During the evaporation and condensation processes, the composition of the liquid and the vapor changes. By using accumulators, fluid can be stored and released.
Thus, the composition, heat capacity, and temperature shifts of the circulating fluid can be changed, resulting in a capacity variation for a heat pump system and producing both a better match to the heat demand and corresponding savings in auxiliary heat. A best operating state of the system could be obtained without too much difficulty to adapt to the requirements.

C. Low compression ratio: By adding a fluid with a high density to a refrigerant to form a non-azeotropic mixture, the pressures in the condenser and the evaporator can be adjusted to appropriate values so that the configuration can enjoy a compression ratio as low as 6:1 and still reach the low temperature of -135°C using a non-azeotropic refrigerant mixture consisting of complex components [White, 1984]. This allows the compressor to operate in a better condition and its life lasts much longer.

D. Relatively simple configuration: Compared with cascade systems which have successfully achieved low temperatures, a system using a non-azeotropic refrigerant blend could be much less complicated since only one thermodynamic cycle could accomplish the task. This brings significant advantages for system trouble shooting and servicing.
On the other hand, such a system can also have disadvantages of its own. Some of them are so severe that the design of key components for the system calls for unusual efforts.

A. A degradation of the heat transfer effect for heat exchangers: The heat transfer effect for the refrigerant mixture is reduced when it is compared with that of either component. The extent of this effect depends on the specific mixture. Also, predicting of heat transfer becomes extremely uncertain. This generates more difficulties than ever before for the design of heat exchangers.

B. A tougher requirement for compressor: When comparing such a system with other ultra-low temperature systems, the trade-off is a higher compressor displacements versus the compression ratio in order to have a higher refrigerant mass flow rate to partially compensate the lost heat transfer effect.

C. Leakage problem: For a single refrigerant system, a small leakage can be easily fixed without any worry about a decline of the system's performance as far as the required amount of that refrigerant is recharged. Leakage from the mixed-refrigerant system would cause an unknown change of the mixture composition, and a reduced system
capacity. Therefore, the remaining refrigerant would have to be dumped and replaced with new refrigerant of known composition. This process frequently needs special skills and potentially costs more.

Although there are disadvantages, an auto-cascade heat pump system using a non-azeotropic refrigerant blend is a feasible method to achieve low temperatures.

1.4 Literature Review: Heat Transfer Performance of NARB's

Although recently the interest in the application of non-azeotropic refrigerant blends has grown rapidly, the nature of the two phase heat transfer phenomenon of such mixtures under forced convection conditions is virtually unknown. However, the results of a number of tests of heat transfer for various combinations of refrigerants have been reported [Radermacher, 1983]. The tests show that, generally, during evaporation, the heat transfer coefficients of the two-phase flow of non-azeotropic refrigerant blends will be reduced when they are compared with those of either pure component.

Figure 1.8 represents this behavior based on test results of the heat transfer coefficients of the two-phase flow done by the National Bureau of Standards (NBS), in which the pure components are R13B1 and R152a. [Radermacher, 1983] The tests, with phase change involving
refrigerant mixtures of R13B1 and R152a in various concentrations, indicate that the local and average heat transfer coefficients were significantly lower than either pure refrigerant. The tests also revealed that the highest average heat transfer coefficient was for 69% R13B1 (by moles). This average heat transfer coefficient was roughly 90% of the value obtained with pure R152a, and about 70% of that obtained with pure R13B1. The composition of 27% R13B1 (by moles) had the lowest heat transfer coefficients, approximately 35% and 60% below pure R152a and R13B1 respectively.

The reason for this reduction of heat transfer coefficient is most likely due to an increased heat transfer resistance at the interface between the liquid and the vapor phases. During the evaporation process, the more volatile (low boiling-point) component boils preferentially
off the liquid, leaving the remaining liquid more concentrated in the high boiling-point component at the interface of the growing bubble. The local liquid temperature is, thereby, increased. The temperature on the corresponding spot of the wall is thus also increased. The temperature differential between the wall and the bulk fluid is raised. Therefore, this effect produces a reduction in the heat transfer coefficient based on the bulk equilibrium temperature.

Another potential explanation for the observed variation in heat transfer is its dependence on the quantity |y - x|, where y is the vapor composition, and x is the liquid composition. Such a dependence has been shown in pool boiling [Radermacher, 1983]. Figure 1.9 is a group of the two-phase heat transfer coefficient values, hTP, versus |y - x| for R13B1/R152a mixture reported by NBS [Radermacher, 1983]. A strong dependence on the composition difference can be seen. But there remains a question on how consistent this dependence is for different combinations of refrigerants and compositions.

As a result, the unknown nature of heat transfer performance of NARBs could be a barrier for the new design of any heat exchangers that involves the phase change of the blends. This, at the current situation, is better resolved from the actual testing of each combination of components of interest because the analytical model of this
Figure 1.9 Local heat transfer coefficient as a function of liquid-vapor composition difference.

kind has not been proven, and often the results of rough calculation are not satisfactory. As an important outcome of the testing of heat transfer, heat transfer coefficients of similar non-azeotropic refrigerant blend can be inferred.
Chapter 2

Analytical Approach for Evaporative Flow with Binary Mixture

2.1 Heat Transfer in Two-phase Annular Flow for Binary Mixtures

The problems of two-phase, two-component heat transfer have so far been over-shadowed by those of the single-component systems and have therefore received considerably less attention. However, since the introduction of refrigerant blends, the study of heat transfer for refrigerant blends during phase change are becoming more and more important because of the requirements of knowledge for the design of heat exchangers in heat pump systems.

This section attempts to display some of the key factors that affect the evaporative heat transfer effect for a binary blend.

Consider Figure 2.1, a flow model presented by J. G. Collier for evaporative flow of binary mixtures [Collier, 1983], which shows a section of an evaporator heated by a heat source separated from the vaporizing stream by a wall. An annular flow is assumed to be the flow pattern, and the evaporating liquid is displaced to the wall as a thin liquid film. The heat flux through the vapor-liquid interface $q_i$ is less than $q_w$ by the amount of the flux absorbed in heating the liquid mass flow $\Gamma_e \cdot (1-x_e)$ through $dT_e$. 
Figure 2.1 Evaporation of a multicomponent fluid in annular flow.

The amount of heat flux required to heat the liquid mass flow \( \Gamma_e \cdot (1 - x_e) \) is:

\[
q_e = \Gamma_e \cdot (1 - x_e) \cdot \frac{dT_e}{dz} \cdot c_{p1}
\]  

where \( \Gamma_e \) is the flow rate per unit perimeter, \( c_{p1} \) is the liquid heat capacity, \( x_e \) is the fluid quality.

\[
\Gamma_e = \frac{M_e}{A}
\]

\( M_e = \) evaporating mass flow rate

\( A = \) cross section area of evaporator tube
therefore,
\[ q_i = q_w - q_e \]
\[ = q_w - \Gamma_e \cdot (1 - x_e) \cdot \frac{dT_e}{dz} \cdot c_{pl} \]  
(2-2)

where \( q_w \) is the heat flux through the wall.

The interfacial heat flux \( q_i \) is, in turn, made up of two terms, an evaporation flux \( q_v \) at the interface and a sensible heat flux \( q_g \) required to heat the vapor phase through \( dT_e \). The sensible heat flux \( q_g \) from the interface to the vapor core is the sum of the sensible heat flux due to the bulk flow of the vapor from the interface and the heat conducted through the boundary layer:

\[ q_g = (T - T_{e*}) \sum_{i}^{n} \left( n_i \cdot c_{pi} \right) + a_g \cdot \frac{dT}{d\delta} \]  
(2-3)

where \( c_{pi} \) is the molar heat capacity, \( a_g \) is the relative volatility of vapor, and \( T_{e*} \) is dew point of vapor mixture, \( n_i \) is equal to \( n \cdot y_{ie} + J_i \). \( y_{ie} \) is the mole fraction of component \( i \) in the bulk vapor. \( J_i \) is the diffusive molar flux of component \( i \) relative to the mixture total molar flux \( n \).

The thickness of liquid film is represented by \( \delta \). The thickness of liquid film at the interface between the vapor and liquid is \( \delta_0 \) while the thickness on the wall is 0.
Integrating this equation over the liquid film from $\delta = 0$, $T = T_e*$ to $\delta = \delta_0$, $T = T_e$ gives:

\[
q_g = \frac{g}{1 - e^{-a}} \frac{a (T_e* - T_e)}{1 - e^{-a}} \quad (2-4)
\]

where

\[
a = \frac{1}{\Sigma(n_i \cdot \Sigma \cdot \Sigma c_{p_i})} \quad (2-4)
\]

The evaporative heat flux required to form the mixture molar vapor flux $n$ is

\[
q_v = n \cdot h_v \quad (2-5)
\]

where $h_v$ is the mixture molar latent heat of vaporization.

Thus, the $q_i$ is given by

\[
q_i = q_g + q_v
\]

\[
= \frac{a}{1 - e^{-a}} \frac{a (T_e* - T_e)}{1 - e^{-a}} + n \cdot h_v \quad (2-6)
\]

Therefore, $U$, the overall heat transfer coefficient, can be written as the $q_w$ divided by the temperature differential across the wall and the bulk of the evaporating stream.
\[ U = \frac{q_w}{T_w - T_e} \]

\[
\Gamma_e \left(1 - x_e\right) \frac{d T_e}{d z} + a \frac{\alpha_g}{1 - e^{-a}} \left(T_e^* - T_e\right) + n \cdot \nabla \phi
\]

\[
= \frac{\frac{d T_e}{d z} \Gamma_e + \alpha_g \frac{\left(T_e^* - T_e\right)}{1 - e^{-a}} + n \cdot \nabla \phi}{T_w - T_e}
\]

(2-7)

where

\[ n = \sum (n \cdot y_{ie} + J_i) \quad \text{(mixture total molar flux)} \]

\[ y_{ie} = \text{mole fraction of component } i \text{ in the bulk vapor.} \]

\[ J_i = \text{diffusive molar flux.} \]

In general, the solution of such equations is complicated by the fact that the conditions at the interface are not known and must be guessed. An iteration is then necessary to achieve a solution.

Nevertheless, direct and dominant factors that effect the heat transfer can be drawn from this equation.

1. The heat transfer coefficient \( U \) is directly proportional to the mixture mass flow rate \( \Gamma \).

2. The heat transfer is also directly related to the mass diffusion in a positive way. A high effective mass transfer coefficient yields a high diffusive molar flux and the "a" value, and thus the \( U \) value is increased.
3. The equation indicates a heat transfer dependence on the latent heat of the mixture. A high value of latent heat means a more compact molecular structure that allows energy be passed easily, which, in turn, yields a high value of heat transfer effect.

4. A high thermal conductivity of the mixture helps increase the heat transfer coefficient. This is implicitly indicated by $q_e$, expressed by equation 2-1, because $q_e$ can be rewritten by the equation for the conduction heat transfer on the liquid film in the radius direction.

2.2 Empirical Models for the Prediction of Heat Transfer Coefficient and Pressure Drop

Although there is no generally recognized correlation for the prediction of local two-phase heat transfer coefficients of non-azeotropic refrigerant blends, the exploration of the feasibility of applying some of the single component heat transfer correlations to a non-azeotropic refrigerant mixture flow is worthwhile. A comparison between the experimental results for NARBs and those predicted for a single component but using the properties of the NARBs can yield information which may be useful for design.

1. Domanski's Model [Domanski, 1986]

This model was suggested by P.A. Domanski and has the
following form:

$$h_{ev,m} = h \cdot 3.22 \times X_{tt}^{-0.3}$$  \hspace{1cm} (2-8)$$

where $h_{ev,m}$ = evaporative heat transfer coefficient of the refrigerant mixture.

$h$ = forced convection heat transfer coefficient of the liquid film calculated by the equation below for the liquid flow rate equal to two-phase refrigerant flow rate in the tube.

$$Nu = 0.023 \times Re^{0.8} \times Pr^{0.4}$$  \hspace{1cm} (2-9)$$

where $Re = \frac{4 \times m \times D}{\mu \times 3.14 \times D^2}$

$$Pr = \frac{\mu \times C_{PL}}{K_L}$$

$m$, $K_L$, and $C_{PL}$ are liquid mixture dynamic viscosity, thermal conductivity, and heat capacity respectively. $m$ and $D$ are the refrigerant flow rate and the inner diameter of tube. The liquid dynamic viscosity of R14 is predicted by using the method of Morris, and the overall liquid thermal conductivity and liquid viscosity are calculated by using NBS program VISCON (see Appendix H).

$X_{tt}$ is the Lockhart-Martinelli parameter:
\[ X_{tt} = \left[ \frac{1.0 - x}{x} \right]^{0.9} \cdot \left[ \frac{\nu L}{\nu V} \right]^{0.5} \cdot \left[ \frac{\mu L}{\mu V} \right]^{0.1} \] (2-10)

where \( x \) is the fluid quality, \( \nu \) is the fluid specific volume, and \( \mu \) is the fluid viscosity. Parameter \( X_{tt} \) is inversely proportional to flow quality and refers to turbulent vapor turbulent liquid flow. Physically it is equal to the square root of the ratio of the frictional pressure drop of the liquid phase to that of the vapor phase if each of these phases was flowing alone in the tube, i.e.,:

\[ X_{tt} = \left[ \frac{dP}{dL} \right]_L^{0.5} \left[ \frac{dP}{dL} \right]_V \] (2-11)

The correlation of equation 2.8 is based on results of preliminary tests with the uncertainty of heat balances within 10% at the National Bureau of Standards on mixture of R13B1/R152a at a variety of compositions. Comparison shows that the prediction of the above correlation agree within an accuracy of \( \pm 10 \) percent for 50 percent of experimental data, \( \pm 20 \) percent for 89 percent of data, and within \( \pm 31 \) percent for all data. It is also found that this correlation is applicable for annular flow at qualities from 10 percent to 90 percent [Domanski, 1986].

This correlation is:

\[
\frac{h_{TP}}{h_L} = 1.85 \cdot \left[ 10^4 \cdot \frac{Bo}{X_t} + 1.5 \cdot \left( \frac{1}{X_{tt}} \right) \right]^{2/3} \quad (2-12)
\]

where

\[
Bo = \frac{Q}{A} \cdot \frac{G \cdot H_{fg}}{G \cdot H_{fg}}
\]

with

\[
Q = \text{heat input (watts)}.
\]
\[
A = \text{surface area (M}^2\text{)}.
\]
\[
G = \text{mass flux of refrigerants (Kg/s} \cdot \text{M}^2\text{)}.
\]
\[
H_{fg} = \text{enthalpy difference between vapor and liquid phases}.
\]

The boiling number is defined by Schrock and Grossman. When there is a specific mass flow of fluid, the number reflects the influence of heat flux to the degree of boiling.

\[
h_L = 0.023 \cdot \frac{K_L}{D} \cdot Re_L^{0.8} \cdot Pr_L^{0.4} \quad (2-13)
\]

with

\[
Re_L = \frac{G \cdot D}{\mu_L}, \quad Pr_L = \frac{\mu_L \cdot C_{PL}}{K_L}
\]

Again, the liquid viscosity and thermal conductivity are calculated by the NBS program VISCON. The value of heat capacity for the mixture is also calculated by a proper NBS program.
Comparison of experimental local heat transfer coefficients with predicted values for R22 shows that the average negative and positive error are -9.1 percent and 33.5 percent respectively. Average positive error for R13B1 and R152a mixture with 69 percent R13B1 molar composition is 127.6 percent [Radermacher, 1983].

3. Dengler and Addoms Model [Radermacher, 1983]

This model has the following form:

\[
\frac{h_{TP}}{h_L} = 3.0 \cdot \left[ \frac{1}{X_{tt}} \right]^{2/3}
\]  

(2-14)

with \(h_{TP}\) = two-phase heat transfer coefficient.

\(h_L\) = same equation in Schrock and Grossman model.

\(X_{tt}\) = Lockhart-Martinelli parameter.

Experiments showed that this correlation has -21.9 percent of average negative error and 18.7 percent of average positive error for R22. For R13B1 and R152a mixture with 69 percent R13B1 molar composition, average positive and negative errors are 42.2 percent and 22.2 percent respectively [Radermacher, 1983].

4. Happel and Stephan Model [Radermacher, 1983]

The format of the correlation is as follows:

\[
h_{mix} = h_{id} \cdot (1 - K \cdot |y - x|^n)
\]  

(2-15)
where \( h_{\text{mix}} \) = heat transfer coefficient for mixtures.

\( h_{\text{id}} \) = the average heat transfer coefficient based on a mole fraction weighing of the pure refrigerants. It is formulated as the follows:

\[
h_{\text{id}} = X \cdot h_{14} + (1-X) \cdot h_{114}
\]

\( K, n \) = empirically determined constants.

\( y, x \) = vapor and liquid composition respectively.

This form, using a least squares fit to determine the constants (\( k=0.89 \) and \( n=0.5 \)), tended to underpredict the experimental pressure drops for the mixture of R13B1 and R152a [Radermacher, 1983].

5. Pierre's Correlation For Pressure Drop Prediction [Domanski, 1986]

A single-component correlation of Pierre, which was based on experiments with refrigerant 12 and 22, has the following form:

\[
\text{DP} = ( f \left( \frac{L}{D} \right) + \frac{\text{Dx}}{x_m} ) \cdot G^2 \cdot v_m \tag{2-16}
\]

where

\( D \) = inner tube diameter

\( L \) = tube length

\( f \) = friction factor (calculated by equation below)

\( \text{DP} \) = differential pressure
\( x_m = \text{mean quality. (arithmetic average of quality at a calculation segment) } \)

\( D_x = \text{quality change} \)

\( v_m = v_L + x_m \cdot (v_V - v_L) \), \text{mean specific volume} 

The friction factor, \( f \), to be used in the equation above was correlated by Pierre from his experimental data by the following empirical equation valid for \( \text{Re/Kr} > 1 \):

\[
f = 0.0185 \cdot (\text{Kr/Re})^{0.25} \tag{2-17}
\]

where \( \text{Kr} = \frac{J \cdot ifs}{L} \),

\[
\text{Re} = \frac{G \cdot D}{\mu_L},
\]

\( \mu_L = \text{liquid viscosity} \)

\( G = \text{refrigerant mass flux} \)

\( J = \text{mechanical equivalent of heat.} \)

\( ifs = \text{enthalpy difference of liquid and vapor.} \)

This correlation proposed by Pierre contains two terms. The first one is caused by the frictional pressure drop and the second one is due to change of momentum. The friction factor depends on the \( \text{Kr} \) number as much as Reynolds number.

It was found that the friction factor is sensitive to the vapor generation rate at the vapor-liquid interface [Domanski, 1986]. However, an underprediction of 40 percent
of this correlation to the R13B1/R152a mixture suggested a correction factor of 1.4 be used for calculations of pressure drop on mixtures of R13B1/R152a. To be conservative, a correction factor of 2.0 was used when applying this correlation to the design of the test of R14/R114 systems.

These models are programmed into a computer program so that hTP values and pressure drop can be evaluated with other properties of two phase mixture.

2.3 Viscosity and Thermal Conductivity Models

Unfortunately, there are no known published data regarding the absolute viscosity of the refrigerant, R14. Thermal conductivity data for R14 are not complete either. Thus, these parameters needed to be estimated through proposed models.

1. Liquid Viscosity Model

The Morris method [Reid, 1987] is applied to estimate the liquid viscosity of R14. It has the following form,

$$\log \frac{\mu}{c} = J \cdot \left[ \frac{1.0}{Tr} - 1.0 \right]$$

where $c$ is an empirical constant, chosen as 0.148 from Table 9-9 in reference [16], $Tr$ is the reduced temperature
$T/T_c$, $T_c$ is the critical temperature and $J$ is expressed as,

\[
J = \left[ 0.0577 + \sum (b_i \cdot n_i) \right]^{1/2} \tag{2-19}
\]

where $b_i$ = group contribution determined from Table 9-10 in reference [16].

The final format of this estimation of R14 liquid viscosity is written as:

\[
\mu = \{ 0.148 \cdot 10.0^{0.771 \cdot ((227.6/T) - 1.0)} \} \cdot 1.0\text{E}-3
\]

\[
(N/M\cdot S^{-2}) \tag{2-20}
\]

This method is fairly accurate for the estimation of liquid viscosity of n-propylbenzene at 0, 25 and 50°C. The comparison shows a 2.37% average error margin.

2. Vapor Viscosity Model

According to the theory of gas transport properties, the predominant cause of gaseous viscosity is the random, molecular momentum interchange. The Chapman-Enskog treatment of gas viscosity was integrated into the program VISCON because the method considers in detail the interactions between colliding molecules with a potential energy included. The solution for viscosity is written

\[
\mu = \frac{26.69 \cdot (M \cdot T)^{1/2}}{\sigma^2 \cdot \Omega_v} \tag{2-21}
\]
where $\mu = \text{viscosity, } \mu P$

$M = \text{molecular weight, } g/mol$

$T = \text{temperature, } K$

$\sigma = \text{hard-sphere diameter, } A$

$\Omega_v = \text{collision integral.}$

The hard-sphere diameter of R14 is 4.662 A. The collision integral is unity if the molecules do not attract each other. With the intermolecular potential function, the collision integral $\Omega_v$ has been determined by a number of investigators. Neufeld et al. proposed an empirical equation which is written as:

$$\Omega_v = [A \cdot (T^*)^{-B}] + C \cdot [\exp(-D \cdot T^*)] + E \cdot [\exp(-F \cdot T^*)]$$

(2-22)

where $T^* = k \cdot T/\epsilon$, $k$ is the Boltzmann's constant, $\epsilon$ is characteristic of the chemical species of the colliding molecules, $A = 1.16154$, $B = 0.14874$, $C = 0.52487$, $D = 0.77320$, $E = 2.16178$, and $F = 2.43787$ for nonpolar gases. Therefore, it is applicable for R14 vapor. From $0.3 \leq T^* \leq 100$, an average deviation of only 0.064 percent was found. The value of $\epsilon/k$ for R14 vapor is equal to 134.0 [16].

The estimation method is expected to have errors of 0.5 to 1.5 percent for nonpolar compounds and 2 to 4 percent for polar compounds [16].
3. Liquid Thermal Conductivity Model

The R14 liquid thermal conductivity model is chosen from the handbook titled "THERMOPHYSICAL PROPERTIES OF REFRIGERANTS" published by the American Society of Heating, Refrigerating, Air-Conditioning Engineers, Inc. in 1976. It is in the following form,

\[ K \left( \text{Wm}^{-1}\text{K}^{-1} \right) = -0.008 - 0.000792 \cdot t \ (t \text{ in } ^\circ\text{C}) \]  
(2-23)

The values presented should be accurate to be within five percent below 210 K, and the uncertainties increase at higher temperatures to possibly 25 percent at the critical temperature of R14 [21]. Since most of the temperatures in the evaporator are around 210 K, this model is accurate enough to represent the R14 liquid thermal conductivity.

4. Vapor Thermal Conductivity Model

Based on the same handbook mentioned above, the saturated vapor thermal conductivity of R14 is represented by the following equation for the range of -200F to -60F,

\[ K\left(\text{Btu}/(\text{hr})(\text{ft})(\text{F})\right) = 6.054546\cdot10^{-3} + (8.687652\cdot10^{-6} \cdot TF) - (2.667637\cdot10^{-8} \cdot TF^2) + (5.864988\cdot10^{-11} \cdot TF^3) \]

(2-24)

where TF is in degrees Fahrenheit.

The error is within 10 percent at -200F [21].
2.4 Computer Prediction of the Heat Transfer Coefficient and Pressure Drop

The ideal thermodynamic cycle of the system assumes that there is no pressure drop when the mixture flows along the system. However, inside the evaporator, where the refrigeration is accomplished, a certain amount of pressure drop will be generated due to the friction and momentum losses. This result will finally cause a capacity sacrifice to the evaporator. Therefore, the design must consider this effect.

Based on the empirical models above, a program EVAP5 was accomplished. It utilizes NBS programs to provide thermodynamics properties for refrigerant mixtures for the prediction of the heat transfer coefficient. The logic of this program is shown in Figure 2.2.

The program divides the total length of evaporator tubing into large number of segments and treats each segment as a calculation unit in which all parameters are assumed to be constant. The program also allows the user to choose the number of segments when the input file is read. A limit is set at 100 to avoid excessive computation time. Since the concern is about the evaporator capacity in an equilibrium state, enthalpies and inlet pressure are used as the predominant input parameters of this program. This input format also allows the program to apply outputs from a thermodynamic property program, ESTMAT which has the option of inputing measured pressures and temperatures to find the
INPUT INITIAL Cond.

CHOOSE MODEL

CALC. XQ AT EACH SECTION BY CALLING HPIN.

CALC. HEAT TRANSFER COEF. BY CALLING DOM, HS, GROS OR DA.

CALCULATE PRESSURE DROP FOR EACH SECTION BY CALLING EVDP OR SPHDP1.

DP = 2.0 x EVDP
PA(I+1) = PA(I) - DP

J = N+1 ?

Y

PRINT PRESSURE DATA.
PRINT HTCE DATA.
PRINT WALL TEMP. DATA

EXIT

PAOLD = PA(N+1)

Figure 2.2 Logic of program EVAP5.FOR.
correspondent enthalpies. In addition, the program, EVAL5 has the feature of allowing the user to choose various models for prediction of heat transfer coefficients for operating conditions. Therefore, comparison of experimental data to the various heat transfer correlations can be made.

In summary, for different combination of refrigerants, the program output will reveal typical flow parameters and heat transfer coefficients. These values, together with other data for the compressor, condenser, and heat exchanger, provide data for the design purpose. The source code of the Program EVAP5 can be found in Appendix B.
3.1 Motivation

As is discussed above, the heat transfer performance of different kinds of non-azeotropic refrigerant mixtures vary with compositions and combination of refrigerant components. Empirical correlations for the calculation of heat transfer coefficients for pure refrigerants do not agree well with actual results except some of the specific NARBs such as R13b1/R152a [Domanski, 1986].

The motivation, then, is to gain experimental data of heat transfer coefficient and pressure drop for evaporative flow of various mixtures of R14 and R114. Comparison with the various models will thus be possible.

Test runs were performed on an auto-cascade system as shown in Figure 3.1 at Ohio University from December 1988 to July 1989. Two test units for the evaporative heat transfer study are the evaporator, (F), and low temperature double blocks, (G). The evaporative side of the blocks is located at the outlet of the evaporator, and the condensing side of the blocks is on the upstream of the throttling valve. For convenience in understanding the testing system, a brief introduction of the test bed is necessary.
3.2 Description of the Experimental Apparatus Setup

The evaporative flow testing, as mentioned before, is a portion of the viability tests on such a refrigeration system using the non-azeotropic refrigerant mixture R14 and R114.

The complete test system, an auto refrigerating cascade process, combines absorption and vapor compression principles. Figure 3.1, again, is the schematic diagram of the system. In this system, a mixture of R14 and R114 refrigerants is compressed by a single refrigeration compressor. The mixture is then partially condensed in an
air-cooled condenser to a two-phase mixture of the refrigerants. Before it is throttled to the suction pressure to produce cooling, it flows through the high pressure side of a counter flow heat exchanger to continue the condensation process. Actually, this process is dumping the heat to the low pressure side of the heat exchanger which is located at downstream of the evaporator. The refrigerant is in the process of continuing evaporation until it becomes a superheated vapor in the suction line. Figure 3.2, drawn on a 2D plane, attempts to indicate the actual elevations of the test components which might have a bearing on the performance of the system. All components are covered with a heavy layer of Armaflex insulation to have minimum auxiliary heat gain from ambient.

The data acquisition process was automatically conducted by use of an acquisition program written in Quickbasic, a personal computer with a data acquisition board and a data logger with 40 channels. EMF's of Type E and T thermocouple were acquired by an Omega, Model OM-5140, 40 channel, data logger. This data logger was capable of resolving 0.2 °C. It is equipped with a serial communication port which served as the link to the personal computer. An Omega, Model WB-815 data acquisition board inserted inside the personal computer was used for analog to digital conversion of differential pressure transducer signals. The analog-to-digital converters in the WB-815 are 12-bit converters
providing a resolution of 1/4096. The programmable gains could be set to 1, 10, 100, 500.

Therefore, the temperatures and pressures readings could be recorded in data files for subsequent processing.

3.3 The Test Theory and Method for the Evaporator

Figure 3.3 is the diagram of the configuration that allows the local heat transfer coefficient to be determined. Heat input into the evaporator was realized by applying the electrical power to a Teflon-coated Nichrome heater wire wrapped around the evaporator tubing. A layer of epoxy is
applied on top of the wire that secures the wire in place to provide good thermal contact. Along the tube, there are four thermocouple stations. On each station, a 1/16" sheath grounded, type E thermocouple was brazed on a copper column with the thermocouple junction approximately exposed on the center line of the tube so that the bulk fluid temperature could be measured. A type-T thermocouples is attached on the tube wall and another thermocouple is placed one inch from the tube. The difference of readings of the thermocouples on wall and in the insulation was then used to determine heat leakage. Initially, two pressure transducers, one at the inlet and one at the outlet of the evaporator, were used to determine pressure drop. A differential pressure transducer was added later to this device because malfunction of the regular pressure transducers occurred. The evaporator was coiled in a spiral form of approximately three feet in diameter in order to be fit into the test system. Table 3-1 summaries the physical sizes of the evaporator, properties of materials used, and instrumentation for the tests. The electrical heat released by the heater wire constituted the main supply of the energy obtained by the evaporator. This quantity was determined by using a Wattmeter which reads directly the value of the power consumed.
Figure 3.3 Configuration of the experimental evaporator.

At a thermocouple station, the heat leakage rate is described as \( q_L \) for an unit length of tubing

\[
q_L = \frac{2.0 \cdot \pi \cdot (T_i - T_w)}{ \ln \left( \frac{r_i}{r_e} \right) \ln \left( \frac{r_e}{r_w} \right) } + \frac{K_i}{K_e} \tag{3-1}
\]

where \( T_i \) is the temperature of insulation at radius of \( r_i \), \( T_w \) is the wall temperature, \( r_i, r_w \) and \( r_e \) are the measured radii of insulation at which the thermocouple was located, tubing wall and epoxy layer, respectively.

The total heat rate per unit length at a the thermocouple station is \( q_L + \frac{Q_H}{L} \).
Table 3.1 Specifications of test apparatus

<table>
<thead>
<tr>
<th>Specification</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Tubing</strong></td>
<td>Nominal OD = 1/2&quot;, Wall = 0.03&quot;, Length = 19.9', Material: Dehydrated Copper.</td>
</tr>
<tr>
<td><strong>Heating Element</strong></td>
<td>Make: Pelican Wire Co. Inc.</td>
</tr>
<tr>
<td><strong>Type of Wire</strong></td>
<td>Nichrome 60/Fused TFE 15 Mils, Wire Size: 19 A.W.G., Length = 46.47', R = 22.8Ω.</td>
</tr>
<tr>
<td><strong>Differential Pressure Transducer</strong></td>
<td>Make: Validyne Corp., Model: DP 15.24, Calibration: P = 7.38E-3·V + 1.31E-2 (P unit: PSI, V unit: mV)</td>
</tr>
<tr>
<td><strong>Pressure Transducer</strong></td>
<td>Make: PSI-TRONIX, Model: PSI-100, Located: Inlet &amp; Outlet of Evaporator, Calibration: Pin = 1.994·V - 0.0378, Pout = 1.995·V - 0.3632 (P unit: PSIA, V unit: mV)</td>
</tr>
<tr>
<td><strong>Thermocouple</strong></td>
<td>Type: T, Size: 26 A.W.G., Quantity: 7, Location: Evenly Placed in Insulation in Axial Direction, Insulation Diameter at which TC locates: 1.5&quot;, 1.437&quot;, 1.5&quot;, 1.437&quot;, 1.445&quot;, 1.461&quot;, 1.531&quot; (Counted downstream), Error Range: ± 1.5°C.</td>
</tr>
<tr>
<td></td>
<td>Type: E, Size: 22 A.W.G., Quantity: 7, Location: Evenly on Copper Wall, Error: ± 1.5°C</td>
</tr>
<tr>
<td></td>
<td>Type: E, Size: 1/16&quot; Sheath Grounded, Quantity: 4, Location: Evenly Placed on the Center Line of Stream, Error Range: ± 1.5°C.</td>
</tr>
<tr>
<td><strong>Insulation</strong></td>
<td>Type: Armaflex, Thickness: Approx. 2.5&quot;, Thermal Conductivity: 0.255 at 75 °F, 0.260 at 90 °F, Unit: BTU/hr·ft² (deg. F/in.)</td>
</tr>
<tr>
<td><strong>Epoxy Layer</strong></td>
<td>Material: Stycast 2057(Emerson &amp; Cuming) Approximately Thickness: 0.03937 (1 mm), Thermal Conductivity: 2.8 BTU·in/hr·ft²·°F</td>
</tr>
</tbody>
</table>
The local heat transfer coefficient can thus be found as

$$h_{TP} = \frac{1}{r \left[ \frac{2 \cdot \pi \cdot (T_w - T_f)}{q_L + (Q_h / L)} - \frac{\ln(r_w/r)}{K_c} \right]}$$

(3-2)

where $r$ is the inner radius of the tubing, $k_c$ is the copper thermal conductivity.

The pressure drop across the evaporator tubing is measured by using a differential pressure transducer and two pressure transducers at both the inlet and outlet of evaporator.

With a reasonable amount of uncertainty, this arrangement allows the temperatures and pressures used as original inputs to find the heat transfer coefficients and other flow parameters.

3.4 The Uncertainty of Measurement for Heat Transfer Coefficient

The accuracy of the measurement of heat transfer coefficients in this experiment, to a large extent, depends upon the reliability of the measurement of the local heat transfer rate and the temperature difference across the tube wall and the fluid.

The uncertainty of heat transfer rate is determined from that of the physical sizes of insulation, and thermal conductivity of material in conjunction with the measurement
error of local temperatures. The overall uncertainty is systematic, and it can then be estimated by considering the uncertainty of each variable in the equation 3-2 using the Pythagorean summation of the discrete uncertainties.

Consider the local heat rate $q_s$ for a tube segment of one meter long. It is expressed as

$$q_s = \frac{2.0 \cdot \pi \cdot (T_i - T_w)}{\ln \left( \frac{r_i}{r_e} \right)} + \frac{Q}{L} + \frac{\ln \left( \frac{r_e}{r_w} \right)}{K_i} + \frac{\ln \left( \frac{r_{e}}{r_w} \right)}{K_e} \quad (3-5)$$

The local heat transfer coefficient is rewritten as

$$h_{FP} = \frac{1}{r \cdot \left[ \frac{2.0 \cdot \pi \cdot (T_w - T_F)}{q_s} - \frac{\ln (r_w/r)}{K_c} \right]} \quad (3-6)$$

To demonstrate how the overall uncertainty of the local heat transfer coefficient, a set of typical, nominal, data at the normal condition for equations above is chosen as follows:

- $T_{INS} = T_i - T_w = 248.85 - 209.65 = 39.2 \, ^\circ C$;
- $T_{WF} = T_w - T_F = 209.85 - 205.45 = 4.4 \, ^\circ C$;
- $r_i = 0.01905 \, m$, $r_e = 0.00735 \, m$, $r = 0.005512 \, m$;
- $r_w = 0.00635 \, m$, $K_i = 0.0367 \, W/m \cdot K$;
- $K_e = 0.0367 \, W/m \cdot K$, $Q = 150 \, Watts$.

The error range of each measurement device is inferred according to experience and references. The temperature difference $T_{INS}$ between the insulation and
tubing wall was determined from the thermocouple readouts at the appropriate locations. Hence, the uncertainty of temperature difference is determined based on the performance of correspondent thermocouples. In the best judgement, a ±1.00 °C error is justified since consideration is given to the calibration of the thermocouples and the performance of the data logger.

The heater power, which was measured by a Wattmeter, is presumed to have an error range of ±5% which, in the case here, is equivalent to ±7.5 Watts. The total length of tube on which the heating element was wrapped is believed to give an error of ±2 centimeter. An uncertainty of 15% is the approximate value assigned to the radius $r_i$ because the insulation material was squeezed by a certain amount after several layers of insulation were overlapped. The thermal conductivity of insulation material is influenced by the application temperature. The normal range of the temperature inside the insulation was between -50°C to -90°C depending on the heat loads of the evaporator. This temperature degraded the performance of the insulation material because heavy ice was formed inside the insulation. Since the thermal conductivity of an Adiprene rubber, a kind of material similar to flexible polyurethane foam rubber, gives a 32.3% error when the applied temperature drops from 300°C to -75°C [Touloukian, 1970], an error of 40% is applied for the estimation of the influence of the thermal
conductivity of Armaflex insulation to the overall uncertainty. The uncertainty of epoxy layer is estimated to have a 10% error on its radius and 10% on its thermal conductivity.

With these values, the local heat rate and heat transfer coefficient can be estimated as described by the followings.

Take the partial derivative of $q_s$ with respective to each variable and substitute the conditions above

\[
\frac{\partial q_s}{\partial T_i} = \frac{2.0\cdot \pi}{\ln \left(\frac{r_i}{r_e}\right) + \ln \left(\frac{r_e}{r_w}\right)} K_i + \frac{2.0\cdot \pi}{\ln \left(\frac{r_i}{r_e}\right) + \ln \left(\frac{r_e}{r_w}\right)} K_e
\]

\[
= \frac{2.0\cdot \pi}{\ln(0.01905/0.00735) + \ln(0.00735/0.00635)} + \frac{2.0\cdot \pi}{0.0367 + 0.4038}
\]

\[
= 0.2388
\]

\[
\frac{\partial q_s}{\partial T_w} = -\frac{2.0\cdot \pi}{\ln \left(\frac{r_i}{r_e}\right) + \ln \left(\frac{r_e}{r_w}\right)} K_i + \frac{2.0\cdot \pi}{\ln \left(\frac{r_i}{r_e}\right) + \ln \left(\frac{r_e}{r_w}\right)} K_e
\]

\[
= -\frac{2.0\cdot \pi}{\ln(0.01905/0.00735) + \ln(0.00735/0.00635)} + \frac{2.0\cdot \pi}{0.0367 + 0.4038}
\]

\[
= -0.2388
\]
\[
\frac{\partial q_s}{\partial Q} = \frac{1}{L} = \frac{1}{6.0} = 0.1667 \quad (3-9)
\]
\[
\frac{\partial q_s}{\partial L} = -\frac{Q}{L^2} = -\frac{150}{6^2} = -4.166 \quad (3-10)
\]
\[
\frac{\partial q_s}{\partial T_{i1}} = -\frac{2\pi T_{i1}}{2} \left[ \ln \left( \frac{r_i}{r_e} \right) + \frac{\ln(r_e)}{K_e} \right]^2 \cdot \frac{1}{K_i \cdot r_i} \quad (3-11)
\]
\[
= -\frac{2\pi \cdot 39.2}{\left[ \ln\left(\frac{0.01905}{0.00735}\right) + \ln\left(\frac{0.00735}{0.00635}\right) \right]^2} \cdot \frac{1}{0.0367 \cdot 0.01905}
\]
\[
= -508.85
\]
\[
\frac{\partial q_s}{\partial K_i} = -\frac{2\pi T_{i1}}{2} \left[ \ln \left( \frac{r_i}{r_e} \right) + \frac{\ln(r_e)}{K_e} \right]^2 \cdot \frac{\ln(r_i)}{K_i^2} \quad (3-12)
\]
\[ \frac{\partial Q_s}{\partial T_e} = - \frac{2 \cdot \pi \cdot T_{\text{INS}}}{\left[ \frac{\ln(r_1/r_e) + \ln(r_e/r_w)}{K_i} + \frac{\ln(r_e/r_w)}{K_e} \right]^2} \cdot \frac{1}{K_e \cdot r_e} \]  

(3-13)

\[ \frac{\partial Q_s}{\partial K_e} = \frac{2 \cdot \pi \cdot T_{\text{INS}}}{\left[ \frac{\ln(r_1/r_e) + \ln(r_e/r_w)}{K_i} + \frac{\ln(r_e/r_w)}{K_e} \right]^2} \cdot \frac{\ln(r_e/r_w)}{K_e^2} \]  

(3-14)
\[
\frac{2 \cdot \pi \cdot 39.2}{\left[ \ln(0.01905/0.00735) + \ln(0.00735/0.00635) \right]^2} = 0.3191
\]

Therefore, the uncertainty in the local heat transfer rate is

\[ \omega_q = \sqrt{\left(0.2388 - 0.75\right)^2 + (0.1667 - 7.5)^2 + (4.1667 - 0.02)^2 +
\left(0.2388 - 0.75\right)^2 + (508.85 - 0.002857)^2 +
(251.55 - 0.01468)^2 + (119.87 - 0.000735)^2 +
(0.3191 - 0.04038)^2} = 4.17 \text{ W/m} \]

The heat transfer rate itself is

\[
qs = \frac{2.0 \cdot \pi \cdot T_{\text{ins}}}{\ln \left(\frac{r_i}{r_e}\right)} + \frac{Q}{L} + \frac{Q}{K_e}
\]

\[ = 34.36 \text{ W/m} \]

Thus, the uncertainty rate is 12.1%.

Take the derivative of \( h_{TP} \) with respective to \( T_w \), \( T_F \) and \( qs \) and find the values for each.

\[
\frac{\partial h_{TP}}{\partial T_w} = -\frac{1}{\left[ \frac{2 \cdot \pi \cdot (T_w - T_F) \cdot r}{qs} - \frac{r \cdot \ln(r_w/r)}{K_c} \right]^2} \cdot \frac{2 \cdot \pi \cdot r}{qs}
\]

\[ (3-16) \]
\[ \frac{\partial h_T}{\partial T_F} = \frac{1}{2\pi \cdot r} \frac{\partial}{\partial T_F} \left[ \frac{2\pi \cdot (T_w - T_F) \cdot r}{qs} - \frac{r \cdot \ln(r_w/r)}{K_c} \right]^{\frac{2}{qs}} \]

\[ = -51.29 \]

\[ \frac{\partial h_T}{\partial q_s} = \frac{1}{2\pi \cdot T_F \cdot r} \frac{\partial}{\partial q_s} \left[ \frac{2\pi \cdot T_w \cdot r}{qs} - \frac{r \cdot \ln(r_w/r)}{K_c} \right]^{\frac{2}{qs^2}} \]

\[ = 51.29 \]
= \frac{1}{2 \cdot \pi \cdot 4.4 \cdot 0.00551 - \ln(0.00635/0.00551) \cdot 0.00551^2 - 380/34.36^2}

= 6.568

As a result, the uncertainty for the local heat transfer coefficient is:

\[ w_h = \left[ (51.29 \cdot 0.75)^2 + (51.29 \cdot 0.75)^2 + (6.568 \cdot 4.17)^2 \right]^{\frac{1}{2}} \]

\[ = 60.91 \text{ W/m}^2\cdot\text{K} \]

but \( h = 225.59 \text{ W/m}^2\cdot\text{K} \), the percentage rate of uncertainty for the heat transfer coefficient is 27.0%.

The accuracy of \( h \)-value measured is thus seen to be influenced by each variable. However, each of them contributes to the overall uncertainty in a different way. The difference between the wall and the fluid presents a high sensitivity to the overall uncertainty. The thermal conductivity of insulation material, although it was believed to have a large uncertainty on heat transfer at low temperatures, contributes less to the overall uncertainty. This, in turn, suggests that more attention on the performance of thermocouples should be paid to the measurement of parameter such as the heat transfer coefficient.
3.5 Test Theory and Method for the Low Temperature Double Blocks

The low temperature double blocks, as described above, function as a tiny counter flow heat exchanger. It is the other unit that allows the measurement of the heat transfer coefficient. Figure 3.4 is the diagram of the double block assembly.

![Diagram of low temperature double blocks](image)

Figure 3.4 Low temperature measurement in double blocks.

One of the blocks is brazed on the condensing side of the interstage heat exchanger while the other is on evaporator line. They were machined to have fine flat surfaces contacting with each other. The surfaces were coated with heat-sink grease and mechanically tightened together to minimize contact resistance. Between the cold
blocks is a thin piece of stainless steel; the material being chosen were for a moderate heat resistance. Two thermocouples were inserted into the stream lines on both sides of the blocks. To measure the temperature across the stainless plate, two thermocouples were also embedded inside the body of the copper at a location near both sides of the plate.

The evaporative heat transfer coefficient can then be measured if the heat transfer rate across the stainless steel plate and the temperature difference between the copper and the refrigerant fluid were known. Since the thermocouples measure the temperatures on each side of the stainless steel, the heat rate across the plate is

\[
Q_{ST} = \frac{A_{ST}}{t_{ST}} \cdot K_{ST} \cdot (T_H - T_L)
\]  

\[ (3-19) \]

where \( t_{ST} \) is the thickness of the plate (M), \( A_{ST} \) is the surface area of the plate (M²), \( K_{ST} \) is the thermal conductivity of the stainless steel (W/m°C), \( T_H \) and \( T_L \) are high and low temperatures on each side of the plate respectively.

The heat transfer coefficient is as

\[
h_{TP} = \frac{Q_{ST}}{3.142 \cdot D_B \cdot L_B \cdot (T_c - T_F)}
\]  

\[ (3-20) \]
where $D_b$ and $L_b$ are inner diameter and length of the copper block, $T_c$ and $T_f$ are the copper and fluid temperatures.
4.1 Post-Processing of Raw Data

The post-processing of raw data includes input of the raw data and calculation of heat transfer coefficients, transport properties, mass flow rates and dimensionless parameters such as Reynolds numbers, Stanton numbers, Prandtl numbers and Nusselt numbers. These data are printed out in a tabular form.

The process is accomplished by a post-processing program written in FORTRAN 77 named EXPCAL.FOR for processing the 0.41 composition case and EXPCAL1.FOR for the 0.50 and 0.60 compositions. Since the calculation requires transport properties of the mixture, NBS programs were called to generate the necessary values for the calculation of dimensionless numbers. The called subroutines are listed in the Table 4-1.

The main program for processing the raw data is organized to comprise several subroutines, with each routine implementing a specific function such as the calculation of heat transfer coefficient, refrigerant mass flow rate, dimensionless numbers, and transport properties. The test system was modified to measure the heat dissipated by the compressor case for test runs with 0.50 and 0.60 compositions. Therefore, two versions of the program are
employed to accommodate the variation of data format. For a complete source code of the programs, please refer to appendix B.

Table 4-1 Summary of Subprograms

<table>
<thead>
<tr>
<th>Program Name</th>
<th>Function</th>
<th>Author</th>
</tr>
</thead>
<tbody>
<tr>
<td>BCONST</td>
<td>CURVE FIT CONTS.</td>
<td>MARK MCLINDEN</td>
</tr>
<tr>
<td>BDESC</td>
<td>INFO. OF COMPONENTS</td>
<td>MARK MCLINDEN</td>
</tr>
<tr>
<td>BUBLT</td>
<td>CAL. $P_{SAT}$ &amp; VOLUMES</td>
<td>MARK MCLINDEN</td>
</tr>
<tr>
<td>BUBLP</td>
<td>CAL. $T_{SAT}$ &amp; VOLUMES</td>
<td>MARK MCLINDEN</td>
</tr>
<tr>
<td>CALNUM</td>
<td>CAL. DIMENSIONLESS NUM.</td>
<td>CHUAN WENG</td>
</tr>
<tr>
<td>ENEBAL</td>
<td>CAL. ENERGY BALANCE</td>
<td>CHUAN WENG</td>
</tr>
<tr>
<td>ENTROP</td>
<td>CAL. MOLAR ENTROPY</td>
<td>MARK MCLINDEN</td>
</tr>
<tr>
<td>ESPAR</td>
<td>CAL. EQUATION OF STATE</td>
<td>MARK MCLINDEN</td>
</tr>
<tr>
<td>EXPCAL</td>
<td>MAIN PROGRAM</td>
<td>CHUAN WENG</td>
</tr>
<tr>
<td>HCVCPS</td>
<td>CAL. ENTHALPY &amp; V</td>
<td>MARK MCLINDEN</td>
</tr>
<tr>
<td>HTEXPS</td>
<td>CAL. EXPERIMENTAL $h_T$</td>
<td>CHUAN WENG</td>
</tr>
<tr>
<td>MASFLOS</td>
<td>CAL. MASS FLOW RATE</td>
<td>CHUAN WENG</td>
</tr>
<tr>
<td>PLIMIT</td>
<td>SET GUESSES OF P</td>
<td>MARK MCLINDEN</td>
</tr>
<tr>
<td>PLIMIT</td>
<td>CAL. $P$ BOUNDS</td>
<td>MARK MCLINDEN</td>
</tr>
<tr>
<td>READIN</td>
<td>READ RAW DATA</td>
<td>CHUAN WENG</td>
</tr>
<tr>
<td>TPIN</td>
<td>CAL. ENTHALPY</td>
<td>RUSS TIPTON</td>
</tr>
<tr>
<td>TRANS</td>
<td>CAL. $\mu$, DENSITY, $K$</td>
<td>CHUAN WENG</td>
</tr>
<tr>
<td>VIT</td>
<td>CAL. $V_L$ &amp; $V_v$</td>
<td>MARK MCLINDEN</td>
</tr>
<tr>
<td>VISCON</td>
<td>CAL. $\mu$ &amp; $K$</td>
<td>CHUAN WENG</td>
</tr>
<tr>
<td>OUPUT</td>
<td>OUTPUT RESULTS</td>
<td>CHUAN WENG</td>
</tr>
</tbody>
</table>

4.2 Key Equations in the Data Processing Program

The following summarizes the fundamental equations that were used in the EXPCAL program to calculate the mixture properties, flow parameters, and dimensionless numbers.
**Heat transfer coefficient:**

\[
hr = \frac{1}{r \cdot \left[ \frac{2 \cdot \pi \cdot (Tw - Tf)}{ql + (QH/L)} - \frac{ln(rw/r)}{Kc} \right]}
\]  

(4-1)

where \( r \) is the inner radius of the tubing, \( Kc \) is the copper thermal conductivity. The local heat transfer rate has two terms, with the first one being the heat leakage from ambient in a segment tube of one meter and the second the heater power. The influence of copper heat resistance was actually neglected due to its small contribution to the total heat resistance.

**Molecular weight:**

\[
WMOL = X \cdot WM1 + (1.0-X) \cdot WM2
\]

(4-2)

where \( X \) is the bulk composition of the more volatile component in the mixture, \( WM1 \) and \( WM2 \) are molecular weights for the more volatile component and less volatile component respectively.

**Mass quality:**

\[
Xmass = \frac{Xmol \cdot WMV}{Xmol \cdot WMV + (1.0-Xmol) \cdot WML}
\]

(4-3)

where \( WML \) and \( WMV \) are liquid and vapor molecular weights respectively.

**Void fraction:**

The void fraction, which represents the relative flow
areas of vapor and liquid phases, is defined by the ratio of vapor phase flow area, $A_g$, to the total flow area of the channel, $A_c$. The velocity of the liquid film and that of vapor are assumed to be equal.

$$\alpha = \frac{A_g}{A_c} = \frac{1.0}{1.0 + (1.0 - \frac{X_{MASS}}{X_{MASS}}) \cdot \frac{VL}{X_{MASS} \cdot VV}}$$  \hspace{1cm} (4-4)

**Volumetric efficiency:**

$$\text{VOLE} = \frac{1.0 - C \cdot \frac{RPAB^{1/\tau}}{r}}{1.0 - C}$$  \hspace{1cm} (4-5)

where $C$ is the ratio of compressor clearance volume to the compressor maximum volume, $RPAB$ is the pressure ratio of the fluid across compressor, $\tau$ is the polytropic exponent which is calculated based upon the measured temperatures and pressures at the suction and discharge lines, presuming that the compression process is polytropic. $\tau$ is expressed as

$$\tau = \frac{1}{\ln(T_D/T_s)} \cdot \frac{1}{1 - \ln(P_D/P_s)}$$  \hspace{1cm} (4-6)

where $T_D$ and $P_D$ are discharge temperature and pressure while $T_s$ and $P_s$ are suction temperature and pressure respectively.
Mass flow rate (venturi meter):

The venturi mass flow meter operates with a pressure drop across the venturi meter as an indication of the flow rate. A Validyne, Model DP15-34, differential pressure transducer which senses the pressure drop is connected to a Validyne, Model DM 56, digital meter. Figure 4.1 shows the cross section of the venturi.

![Figure 4.1 Venturi meter flow system.](image)

Consider the one-dimensional flow system in the figure. The continuity relation for this situation is

\[
m_a = \delta_1 \cdot A_1 \cdot u_1 = \delta_2 \cdot A_2 \cdot u_2 \quad (4-7)
\]

where \( u \) is the velocity, \( \delta \) is the fluid density, \( A_1 \) and \( A_2 \) are flow areas at venturi inlet and venturi throat respectively. Assume that the refrigerant flow is adiabatic and frictionless and the fluid is incompressible, the Bernoulli equation may be written
\[
\frac{p_1}{\delta_1} + \frac{u_1^2}{2g_c} = \frac{p_2}{\delta_2} + \frac{u_2^2}{2g_c}
\]  
(4-8)

where now \( \delta_1 = \delta_2 = \delta \). Solving equation (4-7) for u and substituting it into (4-8) simultaneously gives for the mass flow rate

\[
\dot{m}_m = \left[ \frac{2000.0 \cdot \text{PAB6}}{(1/A_2^2 - 1/A_1^2) \cdot \text{VSUCS}} \right]^{0.5}
\]  
(4-9)

where \text{PAB6} = p_1 - p_2 is the measured differential pressure (kpa). \text{VSUCS} = 1/\delta is the mixture specific volume at the suction line. The venturi was calibrated using air flow prior to being installed onto the system line. The accuracy of this measurement of mass flow rate depends, to a large extent, on the accuracy of the mixture specific volume.

**Mass flow rate (volumetric method):**

\[
\dot{m}_v = \frac{\text{VOLE} \cdot \text{DISRATE}}{\text{VSUCS}}
\]  
(4-10)

Again, \text{VSUCS} is the mixture specific volume at the suction line. \text{DISRATE} is the displacement of the compressor. Both the mixture specific volume and measurements are required in this method.
Mass flow rate (condenser method):

\[ m_c = \frac{Q_{\text{COND}}}{h_{\text{CONI}} - h_{\text{CONO}}} \]  \hspace{1cm} (4-11)

where \( Q_{\text{COND}} \) is the amount of heat brought out by the condenser. It is expressed as

\[ Q_{\text{COND}} = m_A \cdot C_P \cdot (T_{AO} - T_{AI}) \]  \hspace{1cm} (4-12)

where \( m_A \) is the air mass flow rate, \( T_{AO} \) and \( T_{AI} \) are the air temperatures at the outlet and inlet of the condenser cage, and \( C_P \) is the heat capacity of the air.

Mass flow rate (evaporator method):

\[ m = \frac{Q_{\text{EVAP}}}{h_{\text{EVAPO}} - h_{\text{EVAPO}}} \]  \hspace{1cm} (4-13)

where \( Q_{\text{EVAP}} \) is the amount of heat obtained by the evaporator, which is evaluated by adding the heat leakage rate to the heater power.

Both these two methods are similar in the way that both use temperatures and pressures at the evaporator inlet and outlet as their original input to evaluate enthalpies at these locations by substituting the conditions into NBS programs.
Reynolds number of Mixture Flow:

\[ Re = \frac{4.0 \cdot m \cdot D}{\mu \cdot \pi \cdot D^2} \]  \hspace{1cm} (4-14)

where \( \mu \) is the mixture dynamic viscosity, \( D \) is the inner diameter of tube, \( m \) is the refrigerant mass flow rate.

The mass flow rate in this equation utilizes the average of those measured by the venturi meter and calculated by the volumetric method.

Nusselt number of Mixture Flow:

\[ Nu = \frac{h_T \cdot D}{K_m} \]  \hspace{1cm} (4-15)

where \( K_m \) is the thermal conductivity of the mixture.

Mixture Prandtl number:

\[ Pr = \frac{C_p \cdot \mu}{K_m} \]  \hspace{1cm} (4-16)

where \( C_p \) is the heat capacity of the mixture. Similarly, these numbers are obtained based on the measurements of temperatures and pressures, and the calculation by NBS programs.
Chapter 5
Experimental Results

5.1 System Testing Conditions

A number of tests were conducted for the evaporative flow of R14/R114 mixtures of different compositions varying from 0.30 to 0.70 at various refrigerant charges. For each test run, the equilibrium low temperature was assured to be within a range of interest by adjusting the heat load to the evaporator and setting the throttling valve at an appropriate position. A pressure of 3275 Kpa in the compressor discharge line was set to be the maximum pressure for the testing system to withstand.

Table 5-1 summarizes the conditions of the most significant tests. The test runs with 0.41 composition of R14 were conducted with a charge of 2.098 Kg of refrigerants in the system. For the series of tests of 0.50 and 0.60 compositions, the refrigerant charges were 2.268 and 1.814 Kg respectively. A new throttling valve with a larger flow coefficient was installed for the tests involving 0.50 and 0.60 compositions. Therefore, higher flow rates were obtained for these runs.

When the system temperature was within the range of interest (a minimum temperature of -80°C or below), the throttling valve settings were changed to allow the refrigerant flow rate to be altered, yielding different heat
Table 5-1 Evaporator test conditions

<table>
<thead>
<tr>
<th>Molar Composition of R14</th>
<th>Refrigerant Charge(Kg)</th>
<th>Throttling Valve Setting</th>
<th>Evap. Load(W)</th>
<th>Lowest Temp.(k)</th>
<th>Pressure Range(KPA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.41</td>
<td>2.088</td>
<td>Full Open</td>
<td>50.48</td>
<td>200.06</td>
<td>114.02</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1/4 Open</td>
<td>509.88</td>
<td></td>
<td>220.50</td>
</tr>
<tr>
<td>0.50</td>
<td>2.268</td>
<td>Full Open</td>
<td>71.42</td>
<td>179.53</td>
<td>287.40</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1/2 Open</td>
<td>196.66</td>
<td></td>
<td>420.00</td>
</tr>
<tr>
<td>0.60</td>
<td>1.814</td>
<td>Full Open</td>
<td>71.41</td>
<td>172.85</td>
<td>232.48</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1/2 Open</td>
<td>207.66</td>
<td></td>
<td>483.15</td>
</tr>
</tbody>
</table>

5.2 General Phenomenon of Evaporation Process of R14/R114 Mixtures

In general, the heat transfer coefficient and refrigerant pressure drop during the evaporation process are affected by both the testing conditions and the characteristics of the refrigerant mixtures. The parameters chosen for describing the behavior of the R14/R114 mixtures are temperature, quality, void faction, and Prandtl number. These parameters vary along the flow direction in the evaporator tube.

Figure 5.1 displays the typical distribution of the temperatures of the fluid and the tube wall of the evaporator for runs with 0.41 composition. Consistently, a 16 °C increase of the fluid temperature from the inlet to the outlet is observed under several heat loads on the evaporator ranging from 240 W/m² to 1700 W/m² when the valve setting is in full-open position. An examination of the
experimental data of tests with 0.60 composition indicates a lower fluid temperature increase in the evaporator compared to that of tests with 0.41 composition. The temperature increase of the fluid is about 5 to 8 °C when the heat loads are in a range of 340.00 W/m² to 860.00 W/m². With different valve settings, the refrigerant mass flow rate changes and produces corresponding changes in fluid temperature increase. At a specific valve setting, the evaporating temperature is also increasing with the increased overall heat flux to the tube. This is reflected in Figure 5.1.

At a fixed heat load, the evaporating temperature of the mixture decreases when the mass flow rate of the NARB is increased. The experimental data (see Appendix G) for runs with the mixture of 0.60 composition indicates that the mass flow rate of refrigerant increases from 0.0462 Kg/s to 0.06769 Kg/s by changing the valve opening. Within this range of mass flow, the temperature decreases from -90°C to -100°C with no electrical heating applied to the evaporator tube.
Figure 5.1 Fluid & wall temperature along Evaporator tube for R14/R114 mixture of 0.41 molar composition.

On the other hand, the temperature differential between the tube wall and the fluid continuously decreases in the downstream direction of the evaporator, as shown in Figure 5.2. A higher heat flux to the evaporator tube causes a larger temperature differential, but the difference consistently decreases in the downstream direction.

Figure 5.3 shows the fluid pressure drop as a function of the mass flow rate of the refrigerant for runs with mixtures of 0.41 and 0.60 composition. Although scattered, the data shows a pronounced trend which indicates that the pressure drop will increase with the mass flux. Apparently, the pressure drop for runs with both 0.41 and 0.60
Figure 5.2 Temperature differential (K, wall to fluid) versus the tube length for R14/R114 mixture of 0.41 molar composition (q as parameter).

Figure 5.3 Evaporative flow pressure drop vs. mass flux.
compositions increases with refrigerant mass flux, but not at a rate expected from the normal relation where the pressure drop is proportional to the square of the mass flow.

Theoretically, the pressure drop caused by the mass flow rates can only be compared under equivalent testing conditions such as an equal refrigerant charge, temperature, pressure, and heat load. However, these conditions are difficult to achieve. The change of mass flow rate is a consequence of either varying the throttling valve settings or vaporizing any liquid accumulated in the system. In general, the higher the concentration of R114 in the mixture, the higher the shear force at the tube wall because of a higher viscosity of liquid R114 at low temperatures.

The compositions of the liquid and vapor phases in the evaporator for a bulk composition of 0.41 are indicated in Figure 5.4. The compositions are calculated by NBS programs (refer to Appendix H) with the measured temperatures and pressures as input parameters.

Most of the vapor phase consists of the more volatile component, in this case, R14. On the other hand, the liquid phase consists of almost a hundred percent of the less volatile component, i.e., R114. This separation occurs when the mixture is expanded through the throttling valve. As a typical example, the R14/R114 mixture at the state of 2895.8 Kpa and -50°C upstream of the throttling valve is expanded
to the state of 241.3 Kpa and -98°C. This process flashes the R14 liquid into vapor. The R14 absorbs its latent heat of vaporization and chills the mixture to the low temperature. Both at the inlet and outlet of the throttling valve, the R114 is in the liquid phase. Therefore, there are very few R114 molecules mixed with the R14 vapor. This is not surprising because the boiling points of R14 and R114 are so far apart. Particularly, the normal boiling points of R114 and R14 are 38.62 F and -198.3 F respectively. The major differences of these two components are summarized in Table 5-2.

Figure 5.4 Mixture liquid and vapor concentration (on R14) vs. tube length for R14/R114 mixture of 0.41 molar composition.
Table 5-2 Properties of Saturated Liquid and Vapor for R114 and R14

<table>
<thead>
<tr>
<th>Temp F</th>
<th>Pressure Psia</th>
<th>Density $\mu$ lbm/ft$^3$</th>
<th>K Sat.</th>
<th>$C_p$ Sat. Lqd.</th>
<th>$C_p$ Sat. Gas</th>
</tr>
</thead>
<tbody>
<tr>
<td>-100</td>
<td>0.19558</td>
<td>29.523*</td>
<td>98.404</td>
<td>4.40</td>
<td>0.0521</td>
</tr>
<tr>
<td>-40</td>
<td>1.8899</td>
<td>26.073*</td>
<td>97.068</td>
<td>2.65</td>
<td>0.0488</td>
</tr>
<tr>
<td>20</td>
<td>9.6992</td>
<td>10.174*</td>
<td>94.669</td>
<td>1.24</td>
<td>0.0421</td>
</tr>
<tr>
<td>38.62</td>
<td>14.696</td>
<td>0.0</td>
<td>93.564</td>
<td>1.10</td>
<td>0.0404</td>
</tr>
<tr>
<td>50</td>
<td>18.619</td>
<td>3.9226</td>
<td>92.780</td>
<td>1.00</td>
<td>0.0400</td>
</tr>
</tbody>
</table>

Refrigerant 14

<table>
<thead>
<tr>
<th>Temp F</th>
<th>Pressure Psia</th>
<th>Density $\mu$ lbm/ft$^3$</th>
<th>K Sat.</th>
<th>$C_p$ Sat. Lqd.</th>
<th>$C_p$ Sat. Gas</th>
</tr>
</thead>
<tbody>
<tr>
<td>-220</td>
<td>5.7209</td>
<td>18.273*</td>
<td>0.2004</td>
<td>N/A</td>
<td>0.0025</td>
</tr>
<tr>
<td>-198.3</td>
<td>14.696</td>
<td>0.0</td>
<td>0.4832</td>
<td>N/A</td>
<td>0.0026</td>
</tr>
<tr>
<td>-180</td>
<td>28.670</td>
<td>13.974</td>
<td>0.9061</td>
<td>N/A</td>
<td>0.0030</td>
</tr>
<tr>
<td>-160</td>
<td>53.634</td>
<td>38.938</td>
<td>1.6489</td>
<td>N/A</td>
<td>0.0035</td>
</tr>
<tr>
<td>-140</td>
<td>92.024</td>
<td>77.328</td>
<td>2.8000</td>
<td>N/A</td>
<td>0.0040</td>
</tr>
<tr>
<td>-120</td>
<td>147.420</td>
<td>132.730</td>
<td>4.5300</td>
<td>N/A</td>
<td>0.0045</td>
</tr>
<tr>
<td>-100</td>
<td>223.720</td>
<td>209.020</td>
<td>7.1271</td>
<td>N/A</td>
<td>0.0054</td>
</tr>
</tbody>
</table>

Note: * In. Hg Vacuum, $\mu$ (viscosity)--lbm/ft$h$, K (thermal conductivity)--Btu/h·ft·F, $C_p$ (specific heat)--Btu/lbm·F

The consequence of the difference of boiling points is that when the system is in equilibrium, the R14 concentrates highly in the vapor component while the R114 concentrates in the liquid. This situation is shown in Figure 5.5, in which there are few R114 molecules in the vapor phase, and few R14 molecules in the liquid phase.

Being a typical example of the entire evaporation process, Figure 5.6 is a plot of the mixture saturation temperature against the mixture composition under a normal low-side pressure.
Figure 5.5 Mixing effect of R14 and R114.

\[ P_{\text{low}} = 293.03 \text{ KPA} \]

Figure 5.6 A typical evaporation process of R14/R114.
Based on Figure 5.4 which indicates the large difference between the compositions of mixture vapor and liquid phases, the liquid and vapor lines in Figure 5.6 are widely separated. The evaporation process starts at the inlet of the evaporator and continues through out the evaporative side of the heat exchanger to the suction line of the compressor. During the tests, some liquid was collected by a liquid accumulator on the suction line just before the compressor inlet for most of the operating conditions. The results calculated by the NBS programs also indicate the existence of two-phase conditions at the inlet of the compressor for a substantial number of the tests. This indicates that the heat transfer on the evaporative side of the mixture flow needed to be increased to ensure a single phase fluid going into the compressor.

The dependence of the mixture Prandtl number on the mixture temperature for the mixtures of 0.41 and 0.60 compositions is shown in Figure 5.7. The Prandtl number decreases as the fluid temperature increases in the downstream flow direction. This means the diffusion rate of heat from the fluid is in decline and/or the thermal boundary layer in the fluid is becoming thicker compared to that of the hydrodynamic boundary layer. This tends to reduce the local convection coefficient. Figure 5.8 shows the evolution of the Reynolds number for the mixture of 0.41 composition in the flow direction. The Reynolds number
increases along the evaporator as a result of the reduction of the fluid viscosity, which is reduced due to the increasing temperature of fluid. It should be noticed that the Reynolds number increases along the tube at a particular heat flux and the Reynolds number rises to a higher level as the heat fluxes rise. Also, according to Berntsson’s theory, a larger heat load enhances the degree of turbulence in the refrigerant flow [Berntsson, 1985]. These effects tend to raise the local heat transfer coefficient.

Although the local boiling conditions inside the evaporator can not be observed, the primary mechanism of heat transfer for the evaporative flow of R14/R114 mixture
is thought to be the intense agitation at the heat-transfer surface. Created at small cavities in the inner surface of the tube, vapor bubbles grow and break away from the surface, moving into the body of liquid and finally into the vapor core. This process increases the heat-transfer rate by the agitation caused by the bubbles. However, the presence of mixtures changes the surface-tension forces of the liquid and vapor interface, and this effect may be significant. Therefore, bubble-growing can be greatly influenced compared to that of single-component two-phase flow. For most of the cases of evaporative flow of mixtures, the heat transfer rate is thus lowered [Whalley, 1987]. In this issue, there is considerable controversy as to exactly what the heat-transfer mechanism is for a two-phase flow of a mixture. It is even pointed out by Collier [Collier, 1972] that a major difference exists between two-component and single-component two-phase heat transfer, in that "boiling" does not even normally constitute the main mechanism of heat transfer in the former case. Instead, the flow pattern plays an important role in the two-component two-phase heat transfer [Collier, 1972]. How this theory applies to a particular non-azeotropic refrigerant mixture needs more investigation.
In summary, the evaporative flow of R14/R114 mixtures is characterized by the following. First, the 0.60 composition mixture achieves the lowest temperature among the tests. Second, the increase of the fluid temperature along the evaporator tube for runs with mixtures of a certain composition appears to be independent of heat loads (see Figure 5.1). Third, the pressure drops for flows with mixtures of 0.41 and 0.60 compositions increase with increasing mass flux but not at the rate expected from the "flow-squared" law. Of all causes, the mass flow rate is the main one which dominates the trend of the pressure drop. The friction loss of the fluid flow exists mainly at two
places. The first one is inner surface of the wall on which the high-viscous R114 component is adhered. The second is at the interface where the liquid waves of various sizes are developed to contribute more resistance to the flow.

Fourth, the cross section of the flow can be anticipated to consist of a thin layer of highly R114-concentrated liquid adhered on the tube wall and a vapor core which comprises mainly R14-concentrated vapor. Part of resistance for convective heat transfer exists cross the liquid film, where the liquid is highly concentrated on R114 and has a high surface tension and a low thermal conductivity. The overall resistance for the heat transfer includes the resistance between the vapor core of R14 and the R114 liquid film. Diffusion of molecules from the liquid into the vapor phase is believed to be important. Fifth, Bier's theory [Schulz, 1985] may apply to the flow of R14/R114 mixtures. This theory states that there will be a reduction in the heat transfer coefficient when the equilibrium concentration difference between the liquid and the vapor of a mixture increases. Thus, the large difference in compositions of vapor and liquid in R14/R114 mixtures could result in an extremely large reduction of the heat transfer coefficients compared to that of single-component flow. Sixth, increasing the R14 fraction in the mixture lowers the mixture's Prandtl number. To compensate for the lowered heat transfer capability of the mixture, thereafter, the
Reynolds number of the fluid flow must be raised sufficiently.

5.3 Discussion of Experimental Results Regarding Heat Transfer Coefficients

The heat transfer coefficient of evaporative flow of R14/R114 mixtures is influenced by a number of factors such as the mass flow rate of the refrigerant, mixture composition, flow pattern, fluid quality, and heat flux. This section presents the measured heat transfer coefficient as functions of these factors. Also, heat transfer effects of the R14/R114 mixtures will be shown by the more conventional technique of plotting Nusselt numbers as a function of the Reynolds numbers.

Mass Flow Rate, Mixture Composition

When the mass velocity of the refrigerant flow increases, the heat transfer coefficient increases responsively. Figure 5.9 and Figure 5.10 show heat transfer coefficients as a function of the mixture mass flux measured in the evaporator for mixtures of 0.41 and 0.60 compositions respectively. As a comparison, curves in Figure 5.11 and Figure 5.12 are similar plots measured from the cold temperature double blocks. Since the block on the evaporative side is located just downstream of the evaporator outlet and is heavily insulated (see system diagram Figure 3.1 and 3.3), the conditions in the heat
Figure 5.9 Heat transfer coefficient versus mass flux for R14/R114 mixtures measured from evaporator \([X = 0.41 \text{ (molar, on R14)}]\).

Figure 5.10 Heat transfer coefficient versus mass flux for R14/R114 mixtures measured from evaporator \([X = 0.60 \text{ (molar, on R14)}]\).
Heat transfer coefficient versus mass flux for R14/R114 mixture measured from the cold temperature double blocks [X=0.41, (molar)].

Heat transfer coefficient versus mass flux for R14/R114 mixture measured from the cold temperature double blocks [X=0.60, (molar)].
transfer block are the same as those at the outlet of the evaporator. Therefore, a continuity of heat transfer exists on both the evaporator and the cold block.

Figure 5.10 shows that the h-value responds slowly to the change of the mass flux at a low heat flux such as 326.36 W/m². The h-value rises as the heat flux increases. Generally, increasing the mass flux causes a higher magnitude of flow turbulence. Therefore, at the interface of the vapor and the liquid film, greater wave roughness reduces the heat transfer resistance. The flow of a mixture with a lower concentration of R14 should result in a higher heat transfer coefficient because the introduction of more R14 increases the system pressure (see discussion at page 87). The test runs with a mixture of 0.41 composition are indicated in Figure 5.9. At the heat flux of 760 W/m², the h-value varies from 170 to 240 W/K·m² for a change of the mass velocity from 80 to 160 Kg/m²·s. Figure 5.10 shows that a change of h-value from 200 to 225 W/K·m² requires the mass flux to increase from 470 to 820 Kg/m²·s for the same heat flux during the runs with the mixture of 0.60 composition. The lowered heat transfer coefficients for runs with 0.60 composition are in accordance with expectation. The influence of composition on the heat transfer coefficient can be seen once again in Figures 5.11 and Figure 5.12, in which the data were obtained from the low temperature blocks. The comparison of the two plots
indicates that the experimental results agree well with the conclusions drawn previously. For example, an h-value of 600 W/K·m² is approximately correspondent to 325 Kg/s·m² for the flow of the mixture of 0.60 composition, while the same h-value requires only 100 Kg/s·m² for the flow of a mixture of 0.41 composition. Nevertheless, the effect of the mixture composition on the heat transfer remains uncertain because no complete comparison can be made due to the difficulties in controlling the equivalent test conditions such as temperature, heat flux, and mass flow rate for the comparison.

In general, the effect of mixture composition on the heat transfer can be physically deduced according to the theory of Stephan and Koner [Collier, 1972]. The theory argues that the heat transfer coefficient will be reduced due to the increase of the system pressure by introduction of a volatile component to the mixture system. In fact, it was experienced by a number of researchers [Schulz, 1985] that an increased system pressure reduces the local h-value. More details are given below.

The temperature difference(K) between the actual temperature of the mixture and its saturation temperature under a correspondent pressure and with a given heat flux can be expressed as a function of $\Delta T_{sat,e}, \Delta T_{sat,e}$ is the temperature difference(K) between the fluid temperature and saturation temperature for the equivalent pure fluid which
would function as the mixture under the same conditions.

\[(\Delta T_{\text{sat}})_{\text{mixture}} = \Delta T_{\text{sat},e}(1 + \theta) \quad (5-1)\]

where \(\Delta T_{\text{sat},\text{mixture}} = T_w - T_{\text{sat},e}\). While \(T_{\text{sat},e}\) is given by

\[\Delta T_{\text{sat},e} = x_1 \cdot \Delta T_{\text{sat},1} + x_2 \cdot \Delta T_{\text{sat},2} \quad (5-2)\]

where \(x_1\) and \(x_2\) are the mole fractions (in the liquid phase) of components 1 and 2, and \(\Delta T_{\text{sat},1}\) and \(\Delta T_{\text{sat},2}\) are the corresponding temperature differences. The dimensionless parameter \(\theta\) is given by

\[\theta = A_0 \cdot (0.88 + 0.12 \cdot p) \cdot |y - x| \quad (5-3)\]

where \(x\) is the mole fraction of the more volatile component in the liquid phase, and \(y\) is the mole fraction of the more volatile component in the vapor phase. \(p\) is the system pressure (bar), and \(A_0\) is an empirical constant depending on the binary mixture.

If the R14 concentration is increased, the system pressure will rise and \(\theta\) will be increased as well. The R14 concentration in the liquid phase is very small as shown in Figure 5.4. According to equation 5-2, the \(\Delta T_{\text{sat},e}\) could be approximately equal to \(\Delta T_{\text{sat},2}\) which, in this case, is that
of R114. Therefore, $\Delta T_{\text{sat},e}$ will not change while the R14 charge increases. $(\Delta T_{\text{sat}})$mixture will be increasing due to the system pressure, $p$, and parameter $\theta$ with more R14 introduced into the system. Consequently, the heat transfer coefficient will be reduced while the heat flux remains the same. Physically, a decrease in heat transfer coefficient caused by an increase in system pressure is explained by the fact that the increased pressure increases the liquid holdup $(1-a)$ and thus reduces the liquid velocity. This, in turn, lowers the heat transfer performance.

One of the commonly known effects of using a non-azeotropic refrigerant blend is that the heat transfer coefficient of the mixture flow is lower than that for pure fluid flow. The mechanism causing a reduced heat transfer coefficient is believed to be as illustrated in Figure 5.13. The shaded area around the growing vapor bubble represents a layer of liquid around the bubble which has been partially depleted of R14, hence the saturation temperature of the liquid in this layer has been increased. The effective temperature potential available for driving the heat transfer, $T_w - T_{\text{sat}}$, has therefore been reduced and the heat transfer rate falls. The gross effect is that the heat transfer coefficient has been reduced.
Flow Pattern, and Quality

The evaporative flow of the R14/R114 mixture is presumed to be annular in the cross section of the evaporator as shown in Figure 5.14 which indicates the evolution of void fraction along the tube length also. The value of void fraction used in this study is based upon the homogeneous model for two phase flow, which assumes that the vapor and liquid phases traveled at the same velocity (refer to Chapter 4 for calculation of void fraction).

The main features of annular flow are shown in Figure 5.15. In the figure, the liquid is travelling partly as a thin liquid film on the channel walls and partly as drops entrained in the vapor core. As the fluid absorbs a certain amount of heat from the heat source, both the liquid drops
Figure 5.14 Void factor versus tube length for the R14/R114 mixture of 0.41 molar ratio of R14.

Figure 5.15 Main features of annular flow.
and the liquid film will eventually evaporate and become part of the vapor. It is noticed that the void fraction values are fairly close to unity, which symbolizes that the majority of the flow area in the tube is in the vapor phase. The liquid film will thus be very thin. For example, for the value of the void fraction of 0.98, the corresponding thickness of the liquid layer for this evaporator is approximately 5.5E-5 meters. Considering that the heat resistance across this liquid film is caused mainly by R114, one will have a temperature drop of 0.9 °C across the layer when the heat input to the evaporator is 300 Watts. The measured temperature difference between the wall and the fluid is roughly in the range of 4.5 to 5.0 °C with the same conditions as shown in Figure 5.2 previously. Therefore, about 80 percent of heat resistance exists on the interface and the vapor core. The temperature drop across such a thin liquid film is still significant because of its 20 percent contribution to the total temperature drop.

In the annular flow pattern, the heat transfer coefficient of such a two-component flow shows a considerable dependence not only on mass velocity but also on the volumetric ratio($j_g/j_f$) [Collier, 1972].

The volumetric ratio can be determined by the following equation

$$\frac{j_g}{j_f} = \frac{X}{(1 - X)} \cdot \frac{\delta_g}{\delta_f} \quad (5-5)$$
where $j_g$ is the volumetric flux of the vapor core and $j_f$ is the volumetric flux of the liquid film. Physically, volumetric flux is the same with average velocity of vapor or liquid phase. $\rho_g$ and $\rho_f$ are vapor and liquid film densities, respectively. $X$ is the fluid quality. In fact, since the volumetric ratio is the function of fluid quality which relates to the heat transfer coefficient, the dependence of heat transfer coefficient on the volumetric ratio is reflected by the relation between the heat transfer coefficient and the fluid quality. Figure 5.16 shows the fluid quality along the evaporator tube where the fluid quality continuously increases. Therefore, the magnitude of local volumetric flux ratio is consistently raised by the changes of fluid quality as the fluid temperature increases. The local heat transfer is thus related to the volumetric flow through the change of fluid quality.

The heat transfer coefficient increases as the fluid quality increases at a specific heat flux. Figures 5.17 and 5.18 are the local heat transfer coefficients plotted against the mass quality of the fluid flow for runs with mixtures of 0.41 and 0.60, respectively. As more vapor is formed, the average velocity of the fluid increases. Thus, the heat transfer coefficient is expected to be increased.

If the heat flux to the evaporator is sufficient, there should be a dryout region where all the liquid mixture evaporates completely ($X \geq 1$). This could be evidenced by a
sudden increase in the temperature difference between the wall and the fluid, and the heat transfer coefficient would drop to a lower value. Unfortunately, these data could not be obtained in the current test bed. This situation should occur in the evaporative side of the counter flow heat exchanger for a number of test cases.

Figure 5.16 Fluid quality versus tube length for the R14/R114 mixture of 0.41 molar ratio of R14.
Figure 5.17  Heat transfer coefficient versus fluid quality for R14/R114 mixture [X = 0.41, (molar)].

Figure 5.18  Heat transfer coefficient versus fluid quality for R14/R114 mixture [X = 0.60, (molar)].
Heat fluxes

The mass flow rate, mixture composition and heat flux are the main factors affecting heat transfer coefficients that were varied in the experiments. The results of tests runs with mixtures of 0.41 and 0.60 composition with heat flux as a parameter are shown in Figures 5.19 and 5.20.

Although low overall heat transfer coefficient were observed for runs with mixtures of 0.41 and 0.60 compositions, all curves in the figures show a consistent proportional rise with increasing heat flux. Figure 5.19 is a typical example of the tests with the mixture of 0.41 composition. It shows that the h-value increases with heat rate. Raising the heat flux from 241.45 W/m² to 769.11 W/m² gives an increase by the factor of approximately 3, when the heat transfer coefficient within this heat flux range is increased by the factor of 2.4 from approximately 100 W/K·m² to 240 W/K·m². In all cases, as the heat flux rises to its highest magnitude, so does the h-value. The results of test runs with the mixture of the 0.60 composition indicate the similar trend of runs with 0.41 composition. The physical reasons for such an increased heat transfer effect may be a higher liquid surface roughness and a broader distribution of surface cavity diameters at which the boiling bubbles are originally generated.

The local h-value increases along the downstream direction of flow at a certain amount of heat flux. A
typical increase is as much as 200 W/K.m². The consequence of this is that the temperature differential between the tube wall and the fluid in the downstream direction becomes much less as the fluid quality increases.

Figure 5.19 Local heat transfer coefficient versus tube length for R14/R114 mixture, X_{R14} = 0.41 (Molar).
Summary of Heat Transfer Effects

The heat transfer of evaporative flow of R14/R114 mixtures of 0.41 and 0.60 compositions has shown a dependence on many factors such as the mixture composition, the local heat flux, the fluid quality, the flow pattern, and the mass flow rates of refrigerant. These values are more or less related to each other. The relative importance of these individual dependencies is not easy to clarify. Resolution of their dependence requires more planned work on local, and maybe non-intrusive measurements on these particular values. Once these interdependencies are established, improved correlations which include a composition variable may be possible.
Nevertheless, plots of the mixture Nusselt number versus Reynolds number can be organized to exhibit the heat transfer performance of a mixture of R14/R114. Figure 5.21 through Figure 5.24 are arranged with an increasing mass flow rate for test runs with a mixture of 0.41 composition. Figures 5.25 through 5.28 show the test results for runs with 0.60 composition. Although the mass flow for each plot is a constant, it is noted that the increasing Reynolds number of mixtures inside the tube was a result of the increasing of the mixture velocity. The Nusselt number increases together with increasing Reynolds number and heat flux. The test runs with mixtures of 0.60 composition seem to require a higher Reynolds number to achieve the same level of the Nusselt number than those of the 0.41 composition.
Figure 5.21 Nusselt number versus Reynolds number (mass flow = 0.0088 Kg/s, X = 0.41).

Figure 5.22 Nusselt number versus Reynolds number (mass flow = 0.0106 Kg/s, X = 0.41).
Figure 5.23  Nusselt number versus Reynolds number (mass flow = 0.0147 Kg/s, X = 0.41).

Figure 5.24  Nusselt number versus Reynolds number (mass flow = 0.0148 Kg/s, X = 0.41).
Figure 5.25 Nusselt number versus Reynolds number (mass flow = 0.046 Kg/s, X = 0.60).

Figure 5.26 Nusselt number versus Reynolds number (mass flow = 0.051 Kg/s, X = 0.60).
MIXTURE: R14/R114
COMPOSITION: 0.60 (MOLAR)
MASS FLOW: 0.065 KG/S

MIXTURE: R14/R114
COMPOSITION: 0.60 (MOLAR)
MASS FLOW: 0.067 KG/S

Figure 5.27 Nusselt number versus Reynolds number
(mass flow = 0.065 Kg/s, X = 0.60).

Figure 5.28 Nusselt number versus Reynolds number
(mass flow = 0.067 Kg/s, X = 0.60).
5.4 Discussion of Experimental Results Regarding Pressure Drops

The observations of pressure drop across the evaporator are aimed at its relation to the refrigerant mass flux. Changing the throttling valve position or the heat load altered the refrigerant mass flow rates. The corresponding pressure drops were compiled as function of the mass flux. The plot in Figure 5.3 shows the results as a function of mass flux for the test runs with compositions of 0.41 and 0.60. The relationship of the pressure drop to the mass flow can be clearly seen no matter what causes the mass flow to change.

The pressure drop increases with the mass flux for runs with mixtures of both 0.41 and 0.60 compositions. Bulk composition appears to have little effect on the pressure drop. This follows from the observation that an imaginative line, as an extension of the 0.41 composition P-line, would approach the P-line of runs with 0.60 composition. In the test runs with 0.60 composition, the pressure drop is larger than the ones with 0.41 composition by a factor of approximately two. This effect can be attributed to the increased mass flow rates for the test runs with the 0.60 composition mixture, as a result of the installation of the larger throttling valve. Although scattered, the data for the 0.60 composition indicates that the rate of increase of the pressure drop with mass flux is slightly lower than the one of the 0.41 composition mixture.
5.5 Validation of Critical Parameters and Values

This section addresses the questions related to A) how different the actual circulating composition $X_c$ of the fluid is from the bulk value $X_B$ and how $X_c$ is determined, B) how accurate the heat load to the evaporator is from the point of total energy balance of the entire system, and C) how reliable the mass flow rate is.

Circulating Composition

The circulating composition of the mixture is defined as an apparent composition of the fluid as it flows along the system under equilibrium conditions. It is a composition which would explain observed measurements related to energy balance. The bulk composition is determined from the amount of refrigerant of R14 and R114 charged to the system. When the system is operating in a steady state, the compositions are not necessarily equal. This is because there are locations in the evaporator that allows part of the liquid to be trapped. Therefore, a difference between the circulating composition and the bulk composition results.

The circulating composition was calculated based on the energy balance across the throttling valve. As described in Chapter 3, the system was wrapped with effective insulation. Therefore, the heat gain from ambient should be negligible for such a valve with a small surface area. Consequently, the enthalpy downstream from the valve should be equal to
that upstream. Applying measured pressures and temperatures downstream and upstream, the NBS programs were used to calculate the enthalpy values for various, assumed compositions. The "circulating composition" is thus obtained when the difference between the two enthalpies is within an acceptable error value. This circulating composition is assumed to be the actual flow composition and is used for the evaluation of other flow parameters.

Figure 5.29 shows the circulating composition as a function of evaporator heater power. It should be realized that the evaporator heater power is a controllable factor in the system instead of the direct reason to cause the composition shift.

Evidently, the circulating composition sees the greatest difference from the bulk value at a low temperature. The difference is as much as 13 percent. As the heater power was increased, the circulating composition approaches the bulk value. This indicates that the accumulated R114 liquid is thus vaporizing, and becoming part of the circulating fluid. The trapping of R114 liquid may happen in the evaporator unit and possibly also in the evaporative side of the counter flow heat exchanger as well, due to the uphill flow of the fluid. The fact that the 0.41-composition case shows a lower value of circulating composition compared to the bulk one, implies that the system might have suffered from a leakage when being charged or operated. The escaping
part of the mixture may be mostly R14 since it is in gas or saturated vapor state for nearly the entire cycle. If the liquid accumulation is sufficiently severe, the system performance will be greatly degraded.

In determining the circulating composition, the "coupling coefficients" of F0 and F1 used in NBS program BCONST showed inconsistency on their impact to the enthalpy calculation. The resolution of this problem would certainly require large quantity of data points. The trend of variation of influence by the coefficients can thus be studied.
Validation of Energy Balance to the Testing System

The analysis of heat transfer uncertainty in Chapter 4 for the evaporator explains the primary causes that affect the accuracy of the measurement on the local heat flux, and the heat transfer coefficient. From the viewpoint of energy balance of the testing system, it is convenient to find the operational range in which the heat load to the evaporator and other components can be estimated with good accuracy.

The testing system is isolated as a closed system. Energy transfer across the boundary of this system is listed below.

Energy Input to the Testing System $Q_{IN}$:

- Heat Input to the Evaporator $Q_E(W)$ — measured by Wattmeter plus heat leak calculated.
- Electrical Power Input to the Compressor $E_c(W)$ — measured by Wattmeter.
- Heat Leakage to the Heat Exchanger from Ambient $Q_{EXC}(W)$ — unknown.

Energy Output from the Testing System to Ambient $Q_{OUT}$:

- Heat Dissipated by the Condenser $Q_c(W)$ — measured by air-side energy balance.
- Heat Dissipated by the Compressor Case $Q_p(W)$ — measured by air-side energy balance ($X=0.50$, 0.60 only).

The following quantities are known:

- $Q_E$ - Electrical Power + Heat Leak
- $E_c$ - Measured by a Wattsprobe
- $Q_{EXC}$ - Unknown, varied with system temperature
- $Q_p = m_a \cdot C_{pa} \cdot T_{hs}$
- $Q_c = m \cdot C_{pa} \cdot T$
where $m_a$ and $m$ are the air flow across the compressor housing and condenser cage, respectively, $T_{HS}$ and $T$ are the air temperature differences at the inlet and outlet of the housings. $C_{pa}$ is the heat capacity of the air.

The energy input $Q_{IN}$ is designated as positive. The energy output $Q_{OUT}$ is defined to be negative. Since the heat leak to the heat exchanger is unknown, the energy balance of the system $Q_g$ is calculated:

$$Q_g = Q_{IN} - Q_{OUT} = Q_E + Ec + Q_{EXC} - Q_C - Q_P$$

Since $Q_{EXC}$ could not be measured in this test bed, it was assumed zero when calculating $Q_g$. Figure 5.30 is a plot of such "$Q_g$" against the system's low temperature for R14/R114 mixtures of 0.60 composition. It can be seen that in the temperature range of 190 to 200 K, the energy balance is within the range of -40 to 80 Watts, which indicates that the heat leakage to the heat exchanger can be negligible and the energy measurement of each component is within a reasonable error range. As the temperature falls below 183 K, the data tend to be more negative, showing that the heat leak to the heat exchanger is becoming significant and its neglect causes the measurement inaccuracy on the entire system balance. Since the total energy transferred through the system can be as high as 1500 Watts, the range of -40 to 80 Watts for the energy balance is acceptable and is
contributed by a number of measurement errors.

From the above analysis, the heat load of the evaporator is better known at temperature range of 190 to 200 K than other temperature ranges. Therefore, the heat transfer measured at this range is more accurate than those at other ranges.

![Figure 5.30 Analysis of the system energy balance (X_{buls}=0.60).](image)

**Comparison of Flow Rate Based upon Measurement Methods**

As mentioned before, three methods were applied to find the refrigerant mass flow rate. They are the venturi measurement, volumetric method, and energy balance method for the condenser and evaporator. Their accuracies rely on
various measurements and estimations. Discrepancy between them is thus generated.

The mass flow rate measured from the venturi meter is based on the application of Bernoulli equation and the theory of continuity across the venturi meter. The key parameter in the corresponding calculation is the mixture density which would directly influence the correctness of the mass flow rate. However, there is uncertainty on the mixture density because the mixture on the suction line may have included the liquid phase for some of the test conditions. Consequently, the mass flow rate measured by the venturi meter could be much different from the actual value. The volumetric method for estimating the mass flow rate is based on the consideration of the compressor displacement and volumetric efficiency. Its accuracy mainly depends on the mixture density in the suction line and the calculation of the polytropic exponent $\tau$. The method used for the energy balance on both the condenser and the evaporator depends not only on the measurement of the heat transfer, but also on the determination of fluid enthalpy from the NBS programs at the inlets and outlets of the two components. Figure 5.31 shows the typical flow rates obtained by these methods. The two flow rates calculated by the energy balance methods are relatively close to each other, both being below 0.015 Kg/s. The flow rates obtained by the method of the venturi meter present the highest of
the four, which ranges from 0.070 to 0.075 Kg/s and is 8
times greater than the values obtained by the energy balance
method. The volumetric method gives a mass flow rate of
0.028 Kg/s. Such discrepancies are caused by the
measurement errors, off-range data, and most significantly,
uncertainty produced from the calculations of enthalpy and
density by NBS programs.

The method of venturi measurement and volumetric method
involve only the single phase for most of the test cases.
Therefore, the average of the mass flows calculated from
these two methods was used to represent the actual flow rate
for the data presented in this report.

Figure 5.31 Refrigerant mass flow rate versus evaporator
heat load based upon different methods of
measurements (R14/R114 mixture, $X_{\text{BULK}}=0.60$).
5.6 Comparison of Experimental Results with Theoretical Predictions

The measured heat transfer coefficients are compared among four correlation models: Domanski's, Dengler and Addom's, Schrock and Grossman's, and Happel and Stephan's. The comparisons are shown in the following figures.

Figure 5.32 and Figure 5.33 show the experimental heat transfer coefficients against Domanski's correlation for test runs with 0.41 and 0.60 compositions. It seems that at the entire h-value range, this method does not give proper response to the change of the flow parameters. This may result from the inaccuracy of the estimation of the mixture viscosity and other parameters for determining the Xit number. The two groups of data on different sides of the plots are obtained most likely due to the fact that the correlation does not give consistent values for various testing conditions.

The h-values of runs with 0.41 and 0.60 compositions are shown in Figure 5.34 and 5.35 as the comparison with the predictions from the Happel and Stephan correlation. The correlation shows an underestimation of the heat transfer coefficient for runs with mixtures of 0.41 and 0.60 compositions. However, the Happel and Stephan correlation appears to better represent the data trend than the Domanski's correlation. The actual heat transfer coefficient can be achieved by multiplying the correlation results by a factor of 1.75.
The Schrock and Grossman's correlation is compared with the experimental results in Figure 5.36 and Figure 5.37 for 0.41 and 0.60 compositions. This correlation consistently overestimates the heat transfer value by approximately 25 percent.

The Dengler and Addom's model shows its comparison with the experimental results in Figure 5.38 and Figure 5.39. It appears that this method best represents the results of the experimental runs of the 0.41 and 0.60 compositions of all methods.
Figure 5.32 Comparison of Domanski's correlation to runs with 0.41 composition mixture.

Figure 5.33 Comparison of Domanski's correlation to runs with 0.60 composition mixture.
Figure 5.34 Comparison of Happel-Stephan's Correlation to runs with 0.41 composition mixture.

Figure 5.35 Comparison of Happel-Stephan's Correlation to runs with 0.60 composition mixture.
Figure 5.36 Comparison of Schrock-Grossman's correlation to runs with 0.41 composition mixture.

Figure 5.37 Comparison of Schrock-Grossman's correlation to runs with 0.60 composition mixture.
Figure 5.38 Comparison of Dengler-Addom's Correlation to runs with 0.41 composition mixture.

Figure 5.39 Comparison of Dengler-Addom's correlation to runs with 0.60 composition mixture.
The pressure drop across the evaporator is compared with the Pierre's prediction in Figure 5.40 for test runs with mixtures of 0.41 and 0.60 compositions against the mass flow rate. For test runs with the mixture of 0.60 composition, the correlation gives much lower value ranges compared with the experimental data for the same range of the mass flow. However, the predictive correlation has a higher value range than the experimental results for the mixture of 0.41 composition. This indicates that the correlation may have a sensitivity to the mixture composition. This relation is again shown in Figure 5.41, where the pressure drop values are plotted against the Pierre's correlation. It is seen that the data with the 0.41 composition mixture matches the correlation better than the data with runs of 0.60 composition.

The reason may be related to the fact that the Pierre's correlation does not consider the effect of mixtures on transport properties. For more comparative conditions, special equipment may be required to find the relation between the pressure drop and refrigerant mass flow rates.
Figure 5.40 Comparison of experimental pressure drop with the prediction for R14/R114 mixture.

Figure 5.41 Comparison of Pierre's Correlation to runs with 0.41 and 0.60 compositions.
Chapter 6
Conclusions and Recommendations

Although uncertainty still remains in the feasibility of applying an R14/R114 mixture for achieving the target low temperature of -85°C for commercial freezers, this study reveals, to a certain extent, the nature of the heat transfer phenomenon of R14/R114 mixture at typical compositions during the evaporative process of this mixture inside an evaporator. Specifically, the conclusions and recommendations are summarized as follows:

1. Being capable of generating a low temperature of -85°C and lower under an evaporative flow, the combination of R14 and R114 shows an extremely low heat transfer coefficient over the entire process. For instance, heat transfer coefficient in the evaporator is in the range of 275 to 325 W/m²·K under the heat flux of 861 W/m² and the mixture Reynolds number ranging from 30,000 to 35,000 for runs with R14/R114 mixture of 0.60 composition. It is reported that for the mixture 53%/47% (by weight) R12/R114, the heat transfer coefficient in a falling film evaporator is measured to be approximately 500 W/m²·K when the heat flux is 5000 W/m² and the mixture Reynolds number is 23,000 [Berntsson, 1985]. The two heat transfer coefficients would be within the same order of magnitude if the experimental conditions
were the same. According to the experimental results of Berntsson [Berntsson, 1985], the evaporative heat transfer coefficient of pure R114 increases from roughly 1,000 w/m$^2$·K at the conditions of Re=23,000, $q$ (w/m$^2$)=1,000. The result is obviously greater than that of R14/R114 mixtures flow. This indicates that a higher heat resistance was experienced after the introduction of a second substance and this is consistent with the conclusions deduced by others. When designing a commercial evaporator using such a mixture, one needs to use large surface areas with acceptable pressure drops while maintaining adequate flow velocities.

2. The evaporation process of this mixture, under the low-temperature conditions required for the freezer application, occurs over a large range of temperatures. Typically, the temperature shift is more than 100°C for the mixture to reach completely superheated vapor before entering the compressor. This would particularly require an enhanced heat transfer effect for the evaporative side of the counter flow heat exchanger in such a refrigeration system.

3. The local heat transfer correlations for single component flow do not appear very adequate for R14/R114 mixtures. However, the correlation
suggested by Happel-Stephan seems to be the best one to represent the trend of the heat transfer coefficients although it underestimates the value by 10 to 15 percent. The Pierre's correlation for prediction of pressure drop does not show a promising representation for this refrigerant blend. This may be due to the lack of accurate data of R14 viscosity and thermal conductivity. The estimation of thermophysical properties for R14 by the Morris and Chapman-Enskog methods seems to need improvements through experiments.

4. The influence of composition on the heat transfer coefficient could be better seen provided that there were more sets of comparative data. However, a mixture with a high concentration in R14 seems to give a lower heat transfer coefficient due to the low thermal conductivity of R14. An optimal composition may exist that would provide a best heat transfer effect.

5. The dependence of the pressure drop on composition is not apparent. The higher pressure drop for the 0.6 molar ratio can be explained by the fact that a higher mass flow rate was existing. A specific experiment set up should be devised if further research would be necessary on this subject.

6. Designs of commercial evaporators should
particularly avoid any uphill flow for the mixture to run through. An accumulation of liquid in the suction line of the test system was experienced for most of the experimental test cases. This accumulation could be formed even one or two days after the system was in equilibrium state. This could become a potential problem for the application of R14/R114 mixture in terms of a reduction of evaporator capacity after long periods of operation.

7. There remains questions of how good the NBS programs are in predicting the enthalpy and density of mixture. Large discrepancies exist between refrigerant mass flow rate based on energy balance calculation and measurement by venturi meter. This suggests that further experiments are required to verify outputs of NBS subprograms. Typically, the average mixture density could be verified through accurate weighing and chamber with a known volume.

8. To choose a non-azeotropic refrigerant blend for the future, in addition to the basic requirements of refrigerant, one should particularly pay attention to molecular structures of the interested candidate components. Molecular structures without polarity could cause separation of refrigerant components when the mixture evaporates, which pays a price in two major ways: 1. It is extremely difficult to design
and build a feasible heat exchanger. 2. There is almost no hope of improving the system C.O.P. and managing the system capacity because of excessively high concentration of the most volatile component in vapor. To verify these conclusions and satisfy EPA regulations on refrigerants, one should further carry out specific experiments on these subjects. The test bed should incorporate flexibility for any alterations.

Nevertheless, the combination of R14/R114 is still a successful one for an auto cascade system to reach -100°C through a nonazeotropic evaporation process should there be effective improvements on the interstage heat exchanger and other components. This, most importantly, is an significant experience on new design of such systems.
REFERENCES


Appendix A

The Experimental Evaporator Model
Appendix A  Experimental Evaporator Model

As described briefly in chapter 3, the experimental evaporator model is considerably important in the system in terms of the determination of energy balance analysis of the test system, instrumentations and capacity control of the system. This can lead to the evaluation of the performance of the test system.

A.1 Design Objective

The objective of developing an experimental evaporator model for such a refrigeration test system is that the model should allow to investigate heat transfer performance of non-azeotropic mixtures of R14/R114 in forced-convection boiling flow. In the meantime, it accomplishes basic requirements of commercial evaporator inside a freezer.

A.2 Design Requirements

In general, the experimental evaporator model should possess the following characteristics and functions.

1. Electrically-controlled heat input. Unlike any evaporator inside a freezer, the model should have electrically-controlled heat input so that the total amount of heat absorbed by the refrigerant mixture can be monitored and adjusted.

2. A proper volume for refrigerant charge. Internal volume of the evaporator tubing should be as small as possible in order to achieve a low refrigerant charge and high velocity of refrigerant flow.
3. An acceptable pressure drop. The pressure drop across the inlet and outlet of evaporator should compromise the measurement purpose and the requirement of evaporator capacity.

4. A proper size of copper tubing. In addition to the consideration above, the OD/ID of copper tubing should be selected so that the copper tubing will serve as a frame for the heating element to be wound around it.

5. Instruments. Thermocouples sensitive to low temperature should be installed in the fluid stream, along the copper wall and inside insulation materials so that the temperature differentials can be measured and recorded accurately. Due to the possibility of low pressure drop across the evaporator, a differential pressure transducer is desirable to use in such an experimental apparatus.

6. High tensile strength of heating element. Heating element must have relatively high tensile strength to sustain large tensile force while it is wound around the copper tubing.

7. Low temperature coefficients of resistance of heating element. Low temperature coefficients of resistance are important because the resistance of wire needs to be stable for varying temperature zones.

8. Low coefficient of linear expansion. Heating element must have low coefficients of linear expansion so that it would not shrink much after temperature changes.
9. Excellent electrical insulation. Electrical insulation is so important that any broken spots could cause tragedy to the instrumentation and even the evaporator itself.

10. DC signal controls. Control circuit should be designed to use DC signal (0.0-5.0V) to adjust AC voltage across the heating element. The actual AC voltage should be reflected by output of DC signals. This feature allows a computer to record heating power data and send DC control signal to the circuit.

A.3 Determination of tubing size

The EVAP5 program provides the design data for various tubing sizes as shown by Figure A1 and Table A1.

![Figure A1](image_url)  
**Figure A1** Calculated results under the design conditions (Q = 350 W, OD = 1/2 in, X = 0.66).
Table A1. Calculation Results (L=6.0M)

<table>
<thead>
<tr>
<th>Input Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tubing Length: 6.00 meter</td>
</tr>
<tr>
<td>Total Heat Input: 350.0 Watts</td>
</tr>
<tr>
<td>Inlet Pressure: 150.0 KPa</td>
</tr>
<tr>
<td>Refrigerant Mixture: R14 &amp; R114</td>
</tr>
<tr>
<td>Enthalpy Inlet: -21.445 KJ/KG</td>
</tr>
<tr>
<td>OD/ID: 0.0127/0.011 meter</td>
</tr>
<tr>
<td>Mass Ratio of R-14: 0.5</td>
</tr>
<tr>
<td>Iteration: 70</td>
</tr>
<tr>
<td>F0,F1: 0.0,0.0</td>
</tr>
<tr>
<td>Enthalpy Outlet: 15.88 KJ/KG</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Calculated Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model</td>
</tr>
<tr>
<td>h_{e,v,m} at Locations Downstream (W/M²*K)</td>
</tr>
<tr>
<td>measured from inlet in meters</td>
</tr>
<tr>
<td>P(KPa)</td>
</tr>
<tr>
<td>0.0  1.0  2.0  3.0  4.0  5.0  6.0</td>
</tr>
<tr>
<td>Domanski</td>
</tr>
<tr>
<td>160.8 169.1 176.2 182.3 187.4 191.2 193.8</td>
</tr>
<tr>
<td>Happel</td>
</tr>
<tr>
<td>94.8 96.0 97.3 99.3 102.1 105.6 109.47</td>
</tr>
<tr>
<td>Grossman</td>
</tr>
<tr>
<td>92.2 438.0 440.4 444.5 453.8 469.6 496.0</td>
</tr>
<tr>
<td>Dengler</td>
</tr>
<tr>
<td>277.5 300.3 319.5 337.7 358.3 382.0 412.7</td>
</tr>
</tbody>
</table>

Figure A.1 and Table A.1 show that various calculation methods do not agree with each other. The Schrock and Grossman model has the highest values of heat transfer coefficients. Happel and Stephan model, on the other hand, shows the lowest h values. These results would become useful only after the comparison was made with experimental results.

A.4 Sizing of the Heating Element

The selection of heating element for the evaporator is based upon the Ohm's Law.

Under the ideal situation, the system should be able to take in 500 watts heat input from the evaporator. The element will be operating at most cases, on 110 watts. The
resistance is determined as:

\[
R = \frac{V^2}{W} = \frac{110^2}{500} = 24.2 \, \Omega
\]  

(A.1)

From the product manual provided by Pelican Wire Company, 19 gauge size of Nichrome 60 wire with 15 mils Teflon insulation is suitable. It has 0.5208 Ohm per foot. Therefore, the total length of the wire should be:

\[
L_{\text{wire}} = \frac{R}{0.5208} = 46.47 \, \text{Ft}
\]  

(A.2)

Since the tubing outside diameter is \(D(=0.5\text{inch})\), the wire coil pitch \(DL\) is expressed as:

\[
L_{\text{wire}} \approx \frac{L_{\text{tubing}}}{DL} \ast \left[ DL^2 + (\pi \ast D)^2 \right]^{1/2}
\]  

(A.3)

where \(L_{\text{tubing}}\) is the tubing length.

The pitch can thus be solved to approximately equal to 0.734 inch. The total turns are found to be roughly 234.5.

The final assembly of the heating element has 22.8 \(\Omega\) resistance with a actual length of 43.77 feet. The error is 5.8 percent.

A.5 Power Control Circuit

The circuit designed to control the heating element, mainly, consists of a transformer, a triac and an opto isolator as indicated in Figure A2. Once the triac is set at a certain position, DC voltage signals will control the
Figure A2  Schematic of heater control circuit.

on-off of the AC power source to the heating element and the power delivered by the heating element will be a certain function of the DC input signals. However, this circuit requires further calibration because it was not used in the experiment.

On the other hand, the actual AC voltage is reduced to a smaller voltage output. This voltage is converted internally into a DC correspondent signal and can be sensed by the computer.

The significance of this circuit is that if a DC signal representing the actual load is fed into a computer, the computer can be programmed to process this signal so that a desired power can be adjusted on the heating element.
Appendix B

Program Evap5.Foo and Associated Subprograms
$DEBUG

SUBROUTINE DA(AM, AK, CP, DOUT, D, RATE, XTT, HTC)
C THIS ROUTINE COMPUTES HEAT TRANSFER COEFFICIENTS FOR
C EVAPORATIVE FLOW BASED ON DENGLER AND ADDOMS MODEL.
C CHUAN WENG, JULY 2, 1989
C INPUTS:
C AM: MIXTURE LIQUID ABSOLUTE VISCOSITY(KG/M*S)
C AK: MIXTURE LIQUID THERMAL CONDUCTIVITY(W/M*K)
C CP: MIXTURE LIQUID SPECIFIC HEAT AT CONSTANT
C PRESSURE(KJ/KG*C)
C DOUT: OUTER DIAMETER OF TUBING(METER)
C D: INNER DIAMETER OF TUBING(METER)
C RATE: MASS FLOW RATE OF MIXTURE(KG/S)
C XTT: XTT NUMBER
C OUTPUT:
C HTC: HEAT TRANSFER COEFFICIENTS(W/M**2*K)
C BEGIN
PR=AM*CP/AK
RE=RATE*4.0/(AM*3.14159*D)
HLL=0.023*AK*(RE**0.8)*(PR**0.4)/D
HTC=HLL*3.0*( (1.0/XTT)**(2.0/3.0))
RETURN
END
SUBROUTINE DOM(CP,AM,AK,RMASS,D,XTT,HTC)
GIVEN SPECIFIC HEAT, THERMAL CONDUCTIVITY, MASS
FLOW RATE, INNER DIAMETER OF TUBING, XTT NUMBER,
THIS ROUTINE COMPUTES HEAT TRANSFER COEFFICIENTS
FOR EVAPORATIVE FLOW OF MAB BASED ON MONANSKI'S
MODEL.
CHUAN WENG, JUNE 27, 1989
INPUTS:
CP: SPECIFIC HEAT OF REFRIGERANT MIXTURE
LIQUID(BTU/LBM*F)
AM: DYNAMIC VISCOSITY OF REFRIGERANT MIXTURE
LIQUID(LBM/FT*H)
AK: THERMAL CONDUCTIVITY OF REFRIGERANT MIXTURE
LIQUID(BTU/H*F*FT)
RMASS: MASS FLOW RATE(LBM/H)
D: TUBE INNER DIAMETER(FT)
XTT: XTT NUMBER
OUTPUT:
HTC: HEAT TRANSFER COEFFICIENTS FOR EVAPORATIVE
FLOW (BTU/H*FT**2*F)
HTL=SPHTC(CPLE,AML,AKL,RMASS,DE)
HTC=3.23*HTL*XTT**(-0.3)
RETURN
END
PROGRAM EVAL5.FOR

GIVEN DESIGN AND TEST CONDITIONS FOR EVAPORATOR, THIS PROGRAM CALCULATES THE EVAPORATIVE HEAT TRANSFER COEFFICIENT AND PRESSURE DROP BASED UPON SEVERAL CORRELATIONS. IT IS DEVELOPED BY CHUAN WENG FOR THE OHIO UNIVERSITY/FORMA SCIENTIFIC, INC RESEARCH PROJECT ENTITLED DEVELOPMENT OF LOW-TEMPERATURE FREEZER DESIGN TOOLS.

DATE: 7-12-89

********************************************************************************
*** SPECIFICATION OF VARIABLES AND SUBPROGRAMES
********************************************************************************

 INPUTS:
 XM: MASS COMPOSITION OF MORE VOLATILE COMPONENT
 P: EVAPORATOR INLET PRESSURE (KPA)
 Q: TOTAL HEAT INPUT TO REFRIGERANT FLUID (WATTS)
 D: INSIDE DIAMETER OF TUBE (METER)
 DOUT: OUTSIDE DIAMETER OF TUBE (METER)
 LENGTH: TOTAL LENGTH OF TUBING (METER)
 N: NUMBER OF SEGMENTS FOR CALCULATION
 H1,H2: ENTHALPIES AT INLET AND OUTLET OF TUBING (KJ/KG)

 OUTPUTS:
 T: TEMPERATURE ON EVERY SECTION (K)
 P: PRESSURE ON EVERY SECTION (KPA)
 HTCEV: HEAT TRANSFER COEFFICIENTS ON EACH SECTION (W/SQ.M*K)
 CP: SPECIFIC HEAT ON EACH SECTION (KJ/KG*K)
 TPW: TUBING WALL TEMPERATURE (K)

 OTHER SUBROUTINES REFERENCED:
 EVDP,PLIMIT,BDESC,TQIN,ESPAR, CRITX,SPHTC,HPIN,BUBLT,HPXSP,TEMPW, HCHCP5,BUBLP,ENTROP,VIT, PXQIN,TQIN,BCONST,SPHDP1,VISC1,DOM, HS,OUTFIL,EVDPM,GROS,DA.

********************************************************************************
** DEFINITION OF VARIABLES
********************************************************************************

REAL F0,F1
 F0,F1: INTERACTION PARAMETERS FOR USE OF CSD EQUATION OF STATE

REAL H1,H2,DH,LENGTH,D,DOUT,RMASS,Q,XQ1,XQ2
 DH: H2-H1
RMASS: REFRIGERANT MASS FLOW RATE (LBm/H)
Q: TOTAL HEAT INPUT (W)
XQ1, XQ2: QUALITY AT THE INLET AND OUTLET (MOLAR QUALITY)

REAL H, PAI, XM
H: ENTHALPY (KJ/KGMOL)
PAI: PRESSURE AT THE INLET OF EVAPORATOR (KPA)

REAL HTCEV(100), PA(100), DIST(100), TK(100), TPW(100)

HTCEV: EVAPORATIVE HEAT TRANSFER COEFFICIENTS
(W/SQ.M*K)
PA: LOCAL PRESSURE OF REFRIGERANT (KPA)
DIST: DISTANCE MEASURED FROM THE TUBING INLET (METER)

TK: REFRIGERANT STREAM TEMPERATURE (K)
TPW: OUTER WALL TEMPERATURE OF TUBING (K)

REAL HTL
HTL: FOR CALC. OF HTCEV

MV, ML, MVM, MLM: VAPOR AND LIQUID MOLAR WEIGHT

REAL UVR14(100), ULR14(100), UVR114(100),

$ ULR14(100), CVR14(100),
$ CLR14(100), CVR114(100), CLR114(100)

DIMENSION COEFF(9,20), CRIT(5,20)

COMMON/ESDATA/ COEFF, CRIT
COMMON/PIPAR/LENGTH, D, DOUT, Q, N
COMMON/HTPAR/HTCEV
COMMON/PAPAR/PA
COMMON/DISPAR/DIST
COMMON/TKPAR/TK
COMMON/TPWPAR/TPW
COMMON/MASDATA/WM1, WM2
COMMON/TRANS1/ UVR14
COMMON/TRANS2/ ULR14
COMMON/TRANS3/ UVR114
COMMON/TRANS4/ ULR114
COMMON/TRANS5/ CVR14
COMMON/TRANS6/ CLR14
COMMON/TRANS7/ CVR114
COMMON/TRANS8/ CLR114

CHARACTER*5 REFRN1
CHARACTER*5 REFRN2
CHARACTER*64 FNAM01
CHARACTER*64 FNAM02
CHARACTER*64 FNAM03
CHARACTER*64 FNAM04
CHARACTER*1 ANS1
CHARACTER*1 ANS2
CHARACTER*1 ANS3
CHARACTER*1 ANS4
CHARACTER*1 ANS5
CHARACTER*5 MODEL

REFRN1,2 ARE REFRIGERANTS TO BE MIXED
C              ANS1..5 ARE CONTROLERS FOR THE OUTPUT FILES
C************************************************************C
C DEFINE I/O FILES
C*******************************************************************************C
C                 WRITE(*,100)
100              FORMAT( ' INPUT FILE NAME=? ' )
                 READ(*,700) FNAMEI
                 OPEN(UNIT=5, FILE=FNAMEI)
                 WRITE(*,105)
105              FORMAT( ' PLEASE CHOOSE ONE OF THE FOLLOWING MODELS.' )
                 WRITE(*,106)
106              FORMAT( ' DOM, HS, GROS, DA, --' )
                 READ(*,700) MODEL
                 WRITE(*,110)
110              FORMAT( ' DO YOU WANT THE OUTPUT FILE OF ALL PARAMETERS?(Y/N)'' )
                 READ(*,700) ANS1
                 IF(ANS1.EQ. 'Y') THEN
                 WRITE(*,120)
120              FORMAT( ' OUTPUT FILE NAME OF TOTAL OUTPUT=?' )
                 READ(*,700) FNAM01
                 WRITE(*,130)
130              FORMAT( ' DO YOU WANT THE LISTING OF TRANSPORT PROPERTIES OF THE REFRIGERANTS?(Y/N)' )
                 READ(*,700) ANS2
                 OPEN(UNIT=6, FILE=FNAM01, STATUS='NEW')
                 ELSE
                 WRITE(*,130)
                 READ(*,700) ANS2
                 IF(ANS2.EQ. 'Y') THEN
                 WRITE(*,140)
140              FORMAT( ' OUTPUT FILE NAME OF THE LISTING=?' )
                 READ(*,700) FNAM01
                 OPEN(UNIT=6, FILE=FNAM01, STATUS='NEW')
                 ENDIF
                 WRITE(*,150)
150              FORMAT( ' PLEASE ANSWER THE NEXT QUESTION, THANK YOU.' )
                 ENDIF
                 WRITE(*,160)
160              FORMAT( ' DO YOU WANT THE DATA FILE OF HEAT TRANSFER COEFFICIENTS?(Y/N)' )
                 READ(*,700) ANS3
                 IF(ANS3.EQ. 'Y') THEN
                 WRITE(*,170)
170              FORMAT( ' OUTPUT FILE NAME OF HTC=?' )
                 READ(*,700) FNAM02
                 OPEN(UNIT=7, FILE=FNAM02, STATUS='NEW')
                 ELSE
WRITE(*,180)
FORMAT( 'PLEASE ANSWER THE NEXT QUESTION, THANK
YOU.' )
ENDIF
WRITE(*,190)
FORMAT( 'DO YOU WANT THE DATA FILE OF FLUID
PRESSURE FOR TELAGRAF PLOTTING?(Y/N)' )
READ(*,700) ANS4
IF(ANSM.EQ.'Y') THEN
WRITE(*,200)
FORMAT( 'OUTPUT FILE NAME OF PRESSURE=?' )
READ(*,700) FNAM03
OPEN(UNIT=8, FILE=FNAM03,STATUS='NEW')
ELSE
WRITE(*,210)
ENDIF
WRITE(*,220)
FORMAT( 'PLEASE ANSWER THE NEXT QUESTION, THANK
YOU.' )
ENDIF
WRITE(*,230)
FORMAT( 'DO YOU WANT THE DATA FILE OF TUBING
WALL TEMP?(Y/N)' )
READ(*,700) ANS5
IF(ANSM.EQ.'Y') THEN
WRITE(*,240)
FORMAT( 'OUTPUT FILE NAME OF WALL TEMP.OF
TUBE=?' )
READ(*,700) FNAM04
OPEN(UNIT=9, FILE=FNAM04,STATUS='NEW')
ELSE
CONTINUE
ENDIF
WRITE(*,250)
FORMAT( 'END OF THE QUESTIONS. PLEASE WAIT...' )
700 FORMAT(A)
BEGIN
READ(5,*) LENGTH,DOUT,D,Q
READ(5,*) PAI,XM,N
READ(5,250) REFRN1,REFRN2
READ(5,*) FO,FI
READ(5,*) H1,H2
250 FORMAT(2A5)
IF(ANSM.EQ.'Y') THEN
WRITE(6,260) LENGTH,DOUT,D,Q
WRITE(6,270) PAI,XM,N
260 FORMAT('LENGTH=',F5.2,'(METER)','2X,
'OD/ID=',2F7.5,'(METER)'
$2X,'TOTAL HEAT=',F6.2,'(WATTS)'
WRITE(6,280) PAI,XM,N
270 FORMAT('PRESSURE=',F6.2,'(KPA)','2X,
'XM=',F4.3,2X,'N=',I3)
WRITE(6,*) 'ENTER REFRIGERANT1 & REFRIGERANT2 ,'
$R14,R114,R13,R12,R142B, etc.'
WRITE(6,280) REFRN1,REFRN2
280 FORMAT('REFRIGERANTS TO BE USED ARE ',2A10)
WRITE(6,*) 'ENTER INTERACTION PARAMETERS F0,F1'
WRITE(6,290) F0,F1
290 FORMAT( 'F0=','F6.5','F1=','F6.5)
WRITE(6,*) 'ENTER ENTHALPY AT 1 AND 2(H1,H2;
$UNIT IN KJ/KG)
WRITE(6,*) 'H1=','H1,'(KJ/KG)','H2=','H2,'(KJ/KG)'
ELSE
WRITE(*,300)
300 FORMAT( ' TOTAL OUTPUT FILE HAS BEEN SKIPPED.')
ENDIF
C
C*************************************************************************
C ASSIGN CODES FOR REFRIGERANTS
C*************************************************************************
C
IF(REFRN1.EQ.'R11 ') THEN
IR1=1
ELSEIF(REFRN1.EQ.'R12 ') THEN
IR1=2
ELSEIF(REFRN1.EQ.'R13 ') THEN
IR1=3
ELSEIF(REFRN1.EQ.'R13B1') THEN
IR1=4
ELSEIF(REFRN1.EQ.'R14 ') THEN
IR1=5
ELSEIF(REFRN1.EQ.'R22 ') THEN
IR1=6
ELSEIF(REFRN1.EQ.'R23 ') THEN
IR1=7
ENDIF
IF(REFRN2.EQ.'R113 ') THEN
IR2=8
ELSEIF(REFRN2.EQ.'R114 ') THEN
IR2=9
ELSEIF(REFRN2.EQ.'R142B') THEN
IR2=10
ELSEIF(REFRN2.EQ.'R152A') THEN
IR2=11
ELSEIF(REFRN2.EQ.'R216A') THEN
IR2=12
ENDIF
C*************************************************************************
C SET INITIAL VALUES
C*************************************************************************
DIST(1)=0.0
PA(I)=PAI
PAOLD=PAI
WM1=CRIT(1,IR1)
WM2=CRIT(1,IR2)
XM=XM/WM1/(XM/WM1+(1.0-XM)/WM2)
WMOL=XM*WM1+(1.0-XM)*WM2
CALL BCONST(IR1,IR2,F0,F1)
CONTINUE
H1=H1*WMOL
H2=H2*WMOL
DH=H2-H1
RMASS=Q*WMOL*0.001*2.0462*3600.0/DH
IF(ANS1.EQ.'Y') THEN
WRITE(6,*) 'RMASS=',RMASS*1.26E-4,'(Kg/S)'
WRITE(6,310)
310 FORMAT( '-----------------------------------------------')
WRITE(6,320)
320 FORMAT( 'DIST PRES TEMP XQ HT.COEF Cp
$ VISCOSITY DENS. Re ENTHAPY')
WRITE(6,330)
330 FORMAT( ' (M) (KPa) (K) (ML) (W/m**2*K) (SI)
$ (Kg/m*s) (Kg/M**3) (KJ/KgMOL)')
WRITE(6,340)
340 FORMAT( '-----------------------------------------------')
ENDIF
DL=LENGTH/N
C********************************************************************C
C CALCULATE ENTHALPY ON EACH SEGMENT
C********************************************************************C
DO 10 J=1,N+1
IF(J.EQ.1) THEN
  HOLD=H1
ELSE
  HOLD=HNEW
ENDIF
HNEW=(DH*(DIST(J)+DL)/LENGTH)+H1
CALL HPIN(HOLD,PA(J),XM,TK(J),AQ,
$ XML,XMV,VLM,VVM,PL,HV)
C********************************************************************C
C IDENTIFY FLOW STATUS
C********************************************************************C
IF(XQ.LE.0.)THEN
C SUBCOOLED LIQUID
CALL VISCON(1,TK(J),XML,AM,IR1,IR2,J)
CALL VISCON(2,TK(J),XML,AK,IR1,IR2,J)
CALL HVCPS(3,TK(J),VLM,XM,DUM,CV,CP,DUM1)
C********************************************************************C
C UNIT CONVERSION (FROM SI TO ENGLISH UNIT)
C********************************************************************C
**HEAT TRANSFER COEFFICIENT AND PRESSURE DROP**

**HEAT TRANSFER COEFFICIENT**

\[
HTCEV(J) = 5.6782 \times \text{SPHTC} \left( CPE, AME, AKE, RMASS, DE \right)
\]

**Pressure Drop**

\[
DP = \text{SPHDP1} \left( RMASS, DLE, DE, VLME, AME \right)
\]

\[
CP = \frac{CP}{WMOL}
\]

\[
UMIX = AM
\]

\[
VSP = \frac{VLM}{WMOL}
\]

\[
\text{DESITY} = 1.0 / VSP
\]

\[
FLM = RMASS \times 1.604 \times 10^{-4} / D^2
\]

\[
\text{REN} = \frac{FLM \times D}{UMIX}
\]

**Write Output**

\[
\text{WRITE}(6,350) \text{ DIST}(J), PA(J), TK(J), XQ, HTCEV(J), CP, UMX, DESITY, REN, HOLD
\]

**UPDATE PRESSURE VALUE**

\[
\text{IF} (J .EQ. (N+1)) \text{ THEN}
\]

\[
\text{GO TO 380}
\]

\[
\text{ELSE}
\]

\[
PA(J+1) = PA(J) - DP \times 6.89476
\]

**ELSEIF**

\[
\text{ELSEIF} (XQ .GT. 0 .AND. XQ .LT. 1.) \text{ THEN}
\]

**CALCULATION OF TRANSPORT PROPERTIES FOR TWO PHASE**

**CALL VISCON**

\[
\text{CALL VISCON}(1, TK(J), XML, AML, IR1, IR2, J)
\]

\[
\text{CALL VISCON}(3, TK(J), XMV, AMV, IR1, IR2, J)
\]

\[
\text{CALL VISCON}(2, TK(J), XML, AKL, IR1, IR2, J)
\]

\[
\text{CALL VISCON}(4, TK(J), XMV, DDD, IR1, IR2, J)
\]

**CALL HCVCPS**

\[
\text{CALL HCVCPS}(3, TK(J), VLM, XML, DUM, CVL, CPL, DUM1)
\]

**CONVERSION OF MOLAR BASE TO MASS BASE**

\[
\text{XMLMS} = \text{XML} \times WM1 / (\text{XML} \times WM1 + ((1.0 - \text{XML}) \times WM2))
\]

\[
\text{XMVM5} = \text{XMV} \times WM1 / (\text{XMV} \times WM1 + ((1.0 - \text{XMV}) \times WM2))
\]

\[
\text{WML} = \text{XML} \times WM1 + ((1.0 - \text{XML}) \times WM2)
\]

\[
\text{WMV} = \text{XMV} \times WM1 + ((1.0 - \text{XMV}) \times WM2)
\]

\[
\text{HL} = \text{HL} / \text{WML}
\]

\[
\text{HV} = \text{HV} / \text{WMV}
\]

\[
\text{MV} = (XQ \times \text{XMV} \times WM1) + (XQ \times (1.0 - \text{XMV}) \times WM2)
\]
\begin{verbatim}
ML=((1.0-XQ)*(XML*WM1)+((1.0-XQ)*(1.0-XML)*WM2)
XQMASS=MV/(MV+ML)
C********************************************************C
C CALCULATION OF MASS FLUX AND REYNOLDS NUMBER
C********************************************************C
RATIO=(1.0-XQMASS)/XQMASS
XTT=(RATIO**0.9)*((VLM/VVM)**0.5)*((AML/AMV)**0.1)
XTT1=(RATIO**(-0.9))*((VLM/VVM)**0.5)*
((AML/AMV)**0.1)
UMIX=XQMASS*AMV+(1.0-XQMASS)*AML
VSP=XQ*VVM+(1.0-XQ)*VLM
VSP=VSP/(XM*WM1+(1.0-XM)*WM2)
DESITY=1.0/VSP
FLM=RMASS*1.604E-4/(D**2)
REN=FLM*D/UMIX
C********************************************************C
C CONVERSION OF VARIABLE TO ENGLISH UNIT
C********************************************************C
AMLE=AML*2419.2
AKLE=AKL*0.5778
CPLE=CPL*0.23884/WML
DE=D*3.2808
C********************************************************C
C CHOOSE OF MODELS FOR HEAT TRANSFER COEFFICIENT
C********************************************************C
IF(MODEL.EQ. 'DaM ') THEN
CALL DOM(CPLE,AMLE,AKLE,RMASS,DE,XTT,HTC)
WRITE(*,*) XTT,HTC
HTCEV(J)=5.6782*HTC
ELSEIF(MODEL.EQ. 'GROS ') THEN
CALL GROS(AML,AKL,CPL,DOUT,D,RMASS*1.26E-4,
XTT1,HL,HV,Q,L,HTC)
WRITE(*,*) XTT1,HTC
HTCEV(J)=HTC
ELSEIF(MODEL.EQ. 'DA ') THEN
CALL DA(AML,AKL,CPL,DOUT,D,RMASS*1.26E-4,XTT,HTC)
WRITE(*,*) XTT,HTC
HTCEV(J)=HTC
ELSEIF(MODEL.EQ. 'HS ') THEN
CALL HS(XM,RMASS*1.26E-4,D,XMLMS,XMVMS,J,HTC)
WRITE(*,*) XLMS,XMVMS,HTC
HTCEV(J)=HTC
ENDIF
CP=((1.0-XQ)*CPL+XQ*CPV)/WMOL
IF(ANS1.EQ. 'Y') THEN
WRITE(6,360) DIST(J),PA(J),TK(J),XQ,HTCEV(J),CP,
UMIX,DESITY,REN,HOLD
360 FORMAT(F5.2,2X,F6.2,2X,F6.2,2X,F4.2,2X,F6.2,2X,
F6.5,2X,F6.5,2X,F5.2,2X,F7.1,2X,F8.1)
ENDIF
C********************************************************C
C CALCULATION OF PRESSURE DROP DUE TO FRICTION
\end{verbatim}
V=(1.0-XQ)*VLML+XQ*VVML
VE=V*16.0179
HOLDE=HOLD*0.4299/WMOL
HNEWE=HNEW*0.4299/WMOL
DLE=DL*3.2808

DP=2.0*EVDP(RMASS,HOLDE,HNEWE,VE,AMLE,DLE,DE)

DP=DP*6.89476

PA(J+1)=PA(J)-DP

XQM=MVM/(MVM+MLM)
CALL EVDPM(RMASS,XQMASS,XQM,V,D,DPP)

PA(J+1)=PA(J+1)-DPP*2.0
ELSEIF(XQ.GE.1.) THEN

C********************************************************C
C CALCULATION OF HEAT TRANSFER COEFFICIENT AND
C DELTA P
C********************************************************C

UMIX=AM
VSP = VVM / WMOL
DESIITY = 1.0 / VSP
CP = CP / WMOL
FLM = RMASS * 1.604E-4 / (D**2)
RENM = FLM * D / UMIX
HTCEV(J) = 5.6782 * SPHTC(CPE, AME, AKE, RMASS, DE)
DP = SPHDP1(RMASS, DLE, DE, VVME, AME)
IF (ANS1.EQ. 'Y') THEN
  WRITE (6, 370) DIST(J), PA(J), TK(J), XQ, HTCEV(J),
  CP, UMIX, DESITY, REN, HOLD
ENDIF
370 FORMAT (F5.2, 2X, F6.2, 2X, F6.2, 2X, F4.2,
         2X, F6.2, 2X, F6.5, 2X, F6.5, 2X,
         F5.2, 2X, F7.1, 2X, F8.1)
ENDIF
C******************************************************************************C
C               UPDATE OF PRESSURE VALUE                                      C
C******************************************************************************C
IF (J.EQ.(N+1)) THEN
  GO TO 380
ELSE
  PA(J+1) = PA(J) - DP * 6.89476
ENDIF
ENDIF
DIST(J+1) = DIST(J) + DL
380 CONTINUE
10 CONTINUE
40 CONTINUE
CALL TEMPW
CALL OUTFIL(ANS2, ANS3, ANS4, ANS5, N)
STOP
END
$DEBUG

FUNCTION EVDP(RMS,H1,H2,VMIX,VISL,AL,D)

C

C*****PURPOSE:
C TO COMPUTE FRICTIONAL EVAPORATION PRESSURE DROP
C FOR FLOW IN A TUBE
C

C*****INPUT DATA:
C
C AL: TUBE LENGTH (FT)
C VISL: LIQUID DYNAMIC VISCOSITY (LBM/H*FT)
C D: TUBE INSIDE DIAMETER (FT)
C RMS: REFRIG. MASS FLOW RATE (LBM/H)
C H1: REFRIG. ENTHALPY AT TUBE INLET (BTU/LBM)
C H2: REFRIG. ENTHALPY AT TUBE OUTLET (BTU/LBM)
C VMIX: REFRIG. AVERAGE SREC. VOLUME IN A TUBE
C (FT**3/LBM)
C

C*****OUTPUT DATA:
C
C EVDP: FRICTIONAL EVAPORATION PRESSURE DROP
C (PSI)
C

BEGIN

AC=1.6654E-11 @ 32.174*144.*3600.*2
G=RMS/(0.78539816*D*D)
RE=G*D/VISL
AKF=778.26*(H2-H1)/AL
RATIO=RE/AKF
IF(RATIO.LT.1.)RATIO=1.
F=0.0185/RATIO**0.25
EVDP=AC*F*AL*G*G*VMIX/D
RETURN
END
SUBROUTINE EVDPM(RMASS, XQMASS, XQM, V, D, DPP)

C
C********PURPOSE:
C TO COMPUTE PRESSURE DROP DUE TO THE MOMENTUM
C CHANGE
C CHUAN WENG , DATE: OCT.23,1988
C********INPUT DATA:
C RMASS: MASS FLOW RATE(Lbm/H)
C XQMASS: MASS QUALITY FOR SECTION INLET
C XQM: MASS QUALITY FOR SECTION OUTLET
C V: MEAN SPECIFIC VOLUME(M**3/Kg)
C D: I.D. OF TUBING(METER)
C********OUTPUT DATA:
C DPP: PRESSURE DROP FOR THE SMALL SECTION(KPa)
C
C BEGIN
REAL RMASS, XQMASS, XQM, V, D, DPP, XMEAN, DX
DX=XQM-XQMASS
XMEAN=(XQM+XQMASS)/2.0
G=(RMASS*0.45359237*4.0)/(3600.0*3.1415926*D*D)
DPP=DX*V*G**2/(XMEAN*1000.0)
RETURN
END
SUBROUTINE GROS(AM, AK, CP, DOUT, D, RATE, XTT, HL, HV, Q, L, HTC)

C THIS ROUTINE COMPUTES HEAT TRANSFER COEFFICIENTS FOR
C EVAPORATIVE FLOW BASED ON SCHROCK AND GROSSMAN
C MODEL.
C CHUAN WENG, JULY 2, 1989
C INPUTS:
C AM: MIXTURE LIQUID ABSOLUTE VISCOSITY(KG/M*S)
C AK: MIXTURE LIQUID THERMAL CONDUCTIVITY(W/M*K)
C CP: MIXTURE LIQUID SPECIFIC HEAT AT CONSTANT
C PRESSURE(KJ/KG*C)
C DOUT: OUTER DIAMETER OF TUBING(METER)
C D: INNER DIAMETER OF TUBING(METER)
C RATE: MASS FLOW RATE OF MIXTURE(KG/S)
C XTT: XTT NUMBER
C HL: LIQUID ENTHALPY(KJ/KG)
C HV: VAPOR ENTHALPY(KJ/KG)
C Q: TOTAL HEAT INPUT TO EVAPORATOR(WATTS)
C L: LENGTH OF TUBING(METER)
C OUTPUT:
C HTC: HEAT TRANSFER COEFFICIENTS(W/M**2*K)

BEGIN
  PR=AM*CP/AK
  RE=RATE*4.0/(AM*3.14159*D)
  HLL=0.023*AK*(RE**0.8)*(PR**0.4)/D
  BO=Q*0.001/(3.14159*DOUT*L*(RATE/
    (3.14159*(D**2)/4.0))*(HV-HL))
  HRAT=1.85*(BO*10000.0+1.5*(1.0/XTT))**(2/3)
  HTC=HLL*HRAT
RETURN
END
SUBROUTINE HS(XM,RMASS,D,XMLMS,XMVMS,J,HTC)
C THIS ROUTINE CALCULATES THE HEAT TRANSFER
C COEFFICIENTS OF NARB IN FORCED-CONVECTIVE
C EVAPORATIVE FLOW BASED ON HAPPEL AND STEPHAN
C MODEL.
C CHUAN WENG, JUNE 23, 1989
C INPUTS:
C XM:  MOLAR COMPOSITION OF MIXTURE
C RMASS: MASS FLOW RATE OF MIXTURE(KG/S)
C D: INNER DIAMETER OF TUBING(METER)
C XMLMS: LIQUID COMPOSITION OF MIXTURE(MASS BASE)
C XMVMS: VAPOR COMPOSITION OF MIXTURE(MASS BASE)
C OUTPUTS:
C HTC:  CALCULATED HEAT TRANSFER
C REAL UVR14(100),UVR114(100),
C CVR14(100),CVR114(100)
C COMMON/TRANS1/UVR14
C COMMON/TRANS3/UVR114
C COMMON/TRANS5/CVR14
C COMMON/TRANS7/CVR114
C COMMON/HSPURE/D1,D2,D3,D4,CPA,CPB
C COMMON/MASDATA/WM1,WM2
C CPAE=CPA*0.23884/WM1
C CPBE=CPB*0.23884/WM2
C UVAE=UVR14(J)*2419.2
C UVBE=UVR114(J)*2419.2
C CVAE=CVR14(J)*0.5778
C CVBE=CVR114(J)*0.5778
C RMASSE=RMASS*2.20462*3600.0
C DE=D*3.2808
C HTCA=SPHTC(CPAE,UVAE,CVAE,RMASSE,DE)
C HTCB=SPHTC(CPBE,UVBE,CVBE,RMASSE,DE)
C HID=5.6782*(XM*HTCA+(1.0-XM)*HTCB)
C DLTCOM=XMVMS-XMLMS
C HTC=HID*(1.0-0.89*(ABS(DLTCOM))**0.5)
RETURN
END
FUNCTION SPHDP1(AM, AL, D, VSP, AMU)

C TO COMPUTE FRICTIONAL PRESSURE DROP FOR SINGLE
C PHASE FLOW IN A TUBE
C WENG, CHUAN  7-10-88

C PARAMETERS
   REAL AL, AM, AMU, D, VSP
   AL: TUBE LENGTH (FT)
   AM: FLUID MASS FLOW RATE (LBm/H)
   AMU: FLUID DYNAMIC VISCOSITY (LBm/H*FT)
   D: TUBE DIAMETER (FT)
   VSP: FLUID SPECIFIC VOLUME (FT**3/LBm)
C
BEGIN

   ACC=3.3309E-11
   G=0.78539816*D*D
   G=AM/G
   RE=G*D/AMU
   F=0.046/RE**0.2
   SPHDP1=ACC*F*VSP*AL*G*G/D
   RETURN
END
FUNCTION SPHTC(CP, AM, AK, RMASS, D)
C
C**********PURPOSE:
C TO CALCULATE SINGLE PHASE HEAT TRANSFER
C COEFFICIENT FOR FLOW INSIDE TUBE
C**********INPUT DATA:
C AM: FLUID DYNAMIC VISCOSITY (LBM/FT*H)
C AK: FLUID THERMAL CONDUCTIVITY (BTU/H*F*FT)
C CP: FLUID SPECIFIC HEAT AT CONST. PRESSURE
C (BTU/LBM*F)
C D: TUBE DIAMETER (FT)
C RMASS: FLUID MASS FLOW RATE (LBM/H)
C
C**********OUTPUT DATA:
C SPHTC: SINGLE PHASE HEAT TRANSFER COEFFICIENT
C (BTU/H*F*FT**2)
C
G=RMASS/(0.7853982*n*D)
RE=D*G/AM
IF(RE.GE.2000) GOTO10
SPHTC=4.36*AK/D
GOTO20
10 PR=(AM*CP/AK)**0.4
RE=RE**0.8
SPHTC=0.023*AK*PR*RE/D
20 RETURN
END
SUBROUTINE OUTFIL(ANS2,ANS3,ANS4,ANS5,N)
C THIS SUBROUTINE GIVES THE DATA FILES FOR TELAGRAF
C PLOTTING BY ANSWERING THE QUESTIONS IN THE MAIN
C PROGRAM.
C CHUAN WENG DATE: SEP. 18, 1988
C PARAMETERS
C ANS2..5 : CONTROLLERS FOR THE OUTPUT FILES(Y/N)
C
INTEGER N
N: NUMBER OF SECTIONS
REAL HTCEV(100),PA(100),DIST(100),TK(100),TPW(100)
REAL UVR14(100),ULR14(100),UVR114(100),
$ ULR114(100),CVR14(100),
$ CLR14(100),CVR114(100),CLR114(100)
COMMON/HTPAR/HTCEV
COMMON/PAPAR/PA
COMMON/DISPAR/DIST
COMMON/TKPAR/TK
COMMON/TPWPAR/TPW
COMMON/TRANS1/UVR14
COMMON/TRANS2/ULR14
COMMON/TRANS3/UVR114
COMMON/TRANS4/ULR114
COMMON/TRANS5/CVR14
COMMON/TRANS6/CLR14
COMMON/TRANS7/CVR114
COMMON/TRANS8/CLR114
CHARACTER*1 ANS1
CHARACTER*1 ANS2
CHARACTER*1 ANS3
CHARACTER*1 ANS4
CHARACTER*1 ANS5

BEGIN
IF(ANS3.EQ. 'y') THEN
WRITE(7,500)
WRITE(7,*) 'HEAT TRANSFER COEFFICIENT INSIDE TUBE:HTCEV'
ELSE
WRITE(*,100)
ENDIF
IF(ANS4.EQ. 'y') THEN
WRITE(8,500)
WRITE(8,*) 'PRESSURE OF MIXTURE INSIDE THE TUBE:PA'
ELSE
WRITE(*,110)
ENDIF
IF(ANS5.EQ. 'y') THEN
WRITE(9,500)
WRITE(9,*) 'WALL TEMP. OF TUBE:TPW'
ELSE
WRITE(*,110)
ENDIF
ELSE
WRITE(*,120)
ENDIF
DO 50 I=1,N+1
IF(ANS3.EQ.'Y') WRITE(7,510) DIST(I),HTCEV(I)
IF(ANS4.EQ.'Y') WRITE(8,510) DIST(I),PA(I)
IF(ANS5.EQ.'Y') WRITE(9,510) DIST(I),TPW(I)
CONTINUE
IF(ANS3.EQ.'Y') WRITE(7,600)
IF(ANS4.EQ.'Y') WRITE(8,600)
IF(ANS5.EQ.'Y') WRITE(9,600)
IF(ANS2.EQ.'Y') THEN
WRITE(6,130)
WRITE(6,140)
130 FORMAT(' THERMOPHYSICAL PROPERTIES OF R $14$')
WRITE(6,*) '------------------------------
140 FORMAT( ' TEMP.(K) MUV(N*S/M**2) $ MUL(N*S/M**2) $ KV($W/M*K$) $ KL(W/M*K')
WRITE(6,*) '----------------------------'
DO 999 I=1,N+1
WRITE(6,*) TK(I),UVR14(I),ULR14(I),
$ CVR14(I),CLR14(I)
WRITE(6,*) '----------------------------'
999 CONTINUE
WRITE(6,*) ' THERMOPHYSICAL PROPERTIES OF R1 $ 14 $'
WRITE(6,*) '----------------------------'
DO 333 I=1,N+1
WRITE(6,*) TK(I),UVR114(I),ULR114(I),
$ CVR114(I),CLR114(I)
WRITE(6,*) '----------------------------'
333 CONTINUE
ELSE
CONTINUE
ENDIF
100 FORMAT(' TELAGRAF DATA FILE FOR H HAS BEEN $ SKIPPED.')
110 FORMAT(' TELAGRAF DATA FILE FOR PRESSURE HAS BEEN
   SKIPPED.')
$ 120 FORMAT(' TELAGRAF DATA FILE FOR WALL TEMP.HAS BEEN
   SKIPPED.')
$ 500 FORMAT(' INPUT DATA.')
  510 FORMAT(2F15.3)
  600 FORMAT(' END OF DATA.')
RETURN
END
SUBROUTINE TEMPW
C      CALCULATE THE WALL TEMPERATURE OF TUBING
C      WENG, CHUAN 6-19-88
C
REAL LENGTH, DOUT, D, Q, COND
C     LENGTH: TOTAL LENGTH OF TUBING (M)
C     DOUT/D: OD/ID OF TUBING (M)
C     Q:      TOTAL HEAT TRANSFERED (W)
C     COND: COPPER CONDUCTIVITY (W/M°C)
REAL TK(100), TPW(100), HTCEV(100)
C     TK:     FLUID TEMPERATURE (K)
C     TPW:    WALL TEMP. OF TUBING (K)
COMMON/TPWPAR/TPW
COMMON/HTPAR/HTCEV
COMMON/PIPAR/LENGTH, DOUT, D, Q, N
C
BEGIN
PI=3.1415926
COND=407.0
DO 10 I=1, N+1
   AA=(Q*(LOG(DOUT/D))/COND)+(2.0*Q/(HTCEV(I)*D))
   TPW(I)=TK(I)+(AA/(2.0*PI*LENGTH))
10  CONTINUE
RETURN
END
SUBROUTINE VISCON(IQ,T,XM,ANSWER,IR1,IR2,K)

C MODIFIED BY CHUAN WENG FROM THE NBS ROUTINE VISCON FOR THE CALCULATION OF VISCOSITY OF R14 UNDER LOW TEMPERATURE DATE: 7-11-88

C**** PURPOSE:
C TO CALCULATE ABS. VISCOSITY OR THERMAL CONDUCTIVITY OF LIQUID OR VAPOR OF A BINARY MIXTURE

C**** INPUT:
C IQ - OUTPUT QUALIFIER
C = 1, IF LIQUID ABS. VISCOSITY REQUIRED
C = 2, IF LIQUID THERMAL CONDUCTIVITY REQUIRED
C = 3, IF VAPOR ABS. VISCOSITY REQUIRED
C = 4, IF VAPOR THERMAL CONDUCTIVITY REQUIRED
C T - TEMPERATURE (K)
C XM - MIXTURE COMPOSITION (MOLAR FRACTION OF MORE VOLATILE COMPONENT)
C IR1 - REFRIGERANT NUMBER OF MORE VOLATILE COMPONENT
C IR2 - REFRIGERANT NUMBER OF LESS VOLATILE COMPONENT

C**** OUTPUT:
C ANSWER - LIQ. ABS. VISCOSITY (IQ=1) (N*S/M**2)
C - LIQ. THERMAL CONDUCTIVITY (IQ=2) (W/(M*K))
C - VAP. ABS. VISCOSITY (IQ=3) (N*S/M**2)
C - VAP. THERMAL CONDUCTIVITY (IQ=4) (W/(M*K))

REAL UVR14(100),ULR14(100),UVR114(100),
$ ULR114(100),CVR114(100),
$ CLR114(100),CVR14(100),CLR14(100)
COMMON/TRANS1/UVR14
COMMON/TRANS2/ULR14
COMMON/TRANS3/UVR114
COMMON/TRANS4/ULR114
COMMON/TRANS5/CVR14
COMMON/TRANS6/CLR14
COMMON/TRANS7/CVR114
LIQUID VISCOSITY

The pure component viscosity equations are from Thermophysical Properties of Refrigerants 1976 ASHRAE and are for saturated liquid. Temperature limits for these equations are shown in the reference. They are not included here because a realistic number (not necessarily very accurate) is needed for a system model. The mixing rule is the method of Grunberg & Nissan p. 474 in the Properties of Gases & Liquids 4th Edition by Reid, Prausnitz, and Poling. The interaction parameter g12 is assumed to be 0 with experimental data available.

IREF = IREF
I = 1
15 IF (IREF .EQ. 3) THEN
   IF (T .LT. 270.) THEN
      AMU(I) = EXP(-4.08106 + 564.472/T) / 1000.
   ELSE
      AMU(I) = (-2.27223 + 0.019373*T - 3.875E-5*T**2) / 1000.
   ENDIF
ELSEIF (IREF .EQ. 5) THEN
   THIS IS A R14 LIQUID
   VIS = 0.7715568*((227.6/T) - 1.0)
   AMU(I) = (0.148*(10.0**VIS))*1.0E-3
   ULR14(K) = AMU(I)
ELSEIF (IREF .EQ. 9) THEN
   IF (T .LT. 360.) THEN
      AMU(I) = EXP(-4.4636 + 1011.47/T) / 1000.
      ULR114(K) = AMU(I)
   ELSE
      AMU(I) = (8.98644 - 0.07248*T + 2.02802E-4*T**2 - 1.92593E-7*T**3) / 1000.
   ENDIF
   ULR114(K) = AMU(I)
ELSE
   WRITE(6,*) ' IREF=', IREF, ' LIQUID VISCOSITY DATA NOT AVAILABLE'
   STOP
ENDIF
IF (I .EQ. 1) THEN
   I = 2
   IREF = IR2
   GO TO 15
ENDIF
IF (T .GE. TC(1)) THEN
C LIQUID THERMAL CONDUCTIVITY
C THE PURE COMPONENT THERMAL CONDUCTIVITY EQUATIONS ARE
C FROM THERMOPHYSICAL PROPERTIES OF REFRIGERANTS 1976
C ASHRAE AND ARE FOR SATURATED LIQUID. TEMPERATURE
C LIMITS FOR THESE EQUATIONS ARE SHOWN IN THE
C REFERENCE. THEY ARE NOT INCLUDED HERE FOR THE SAME
C REASON STATED IN LIQUID VISCOSITY. THE MIXING RULE
C IS THE METHOD OF FILLIPOV P. 562 IN THE PROPERTIES OF
C GASES & LIQUIDS 4TH EDITION BY REID, PRAUSNITZ, AND
C POLING
20 I=1
IREF=IR1
25 IF(IREF.EQ.3)THEN
   AKT(I)=0.0497-0.000522*(T-273.15)
ELSEIF(IREF.EQ.5)THEN
   C THIS IS A R14 LIQUID.
   AKT(I)=-0.0008-0.000792*(T-273.15)
   CLR14(K)=AKT(I)
ELSEIF(IREF.EQ.9)THEN
   AKT(I)=0.071-0.000261*(T-273.15)
   CLR114(K)=AKT(I)
ELSE
   WRITE(6,*),'IREF=',IREF,' LIQUID THERMAL
$ CONDUCTIVITY DATA', ' NOT AVAILABLE'
   STOP
ENDIF
IF(I.EQ.1)THEN
   I=2
   IREF=IR2
   GO TO 25
ENDIF
IF(T.GE.TC(1))THEN
   ANSWER=AKT(2)
ELSE
   W1=1./((1.+(1.-XM)/XM*WM(2)/WM(1)))
   W2=1.-W1
   ANSWER=W1*AKT(1)+W2*AKT(2)-0.72*
$ W1*W2*ABS(AKT(2)-AKT(1))
ENDIF
GO TO 1000
C VAPOR VISCOSITY
C THE PURE COMPONENT VISCOSITY EQUATIONS ARE FROM
C THERMOPHYSICAL PROPERTIES OF REFRIGERANTS 1976 ASHRAE
C AND ARE FOR GAS AT 1 ATM. TEMPERATURE LIMITS FOR
C THESE EQUATIONS ARE SHOWN IN THE REFERENCE. THEY ARE
C NOT INCLUDED HERE FOR THE SAME REASON STATED IN
LIQUID VISCOSITY. THE MIXING RULE IS THE METHOD OF
WILKE P. 407 IN THE PROPERTIES OF GASES & LIQUIDS 4TH
EDITION BY REID, PRAUSNITZ, AND POLING.

I=1
IREF=IR1
IF(IREF.EQ.3)THEN
AMU(I)=(SQRT(T)/(0.41108+$
 328.51/T-27963./T**2))*1.E-6
ELSEIF(IREF.EQ.5)THEN
THIS IS A R14 VAPOR.
USLK=134.0
SIGMA=4.662
TSTAR=T/USLK
A=1.16145
B=0.14874
C=0.52487
D=0.77320
E=2.16178
F=2.43787
OMEGA=(A/(TSTAR**B))+(C/EXP(D*TSTAR))$
  +(E/EXP(F*TSTAR))
AMU(I)=1.0E-7*(26.69*(WM(1)*T)**0.5)/(OMEGA*SIGMA**2)
UVR14(K)=AMU(I)
ELSEIF(IREF.EQ.9)THEN
AMU(I)=(SQRT(T)/(0.51357+434.124/$
  T-41996./T**2))*1.E-6
UVR114(K)=AMU(I)
ELSE
WRITE(6,*) ' IREF=',IREF,' NO VAPOR VISCOSITY DATA
$ AVAILABLE'
STOP
ENDIF
IF(I.EQ.1)THEN
I=2
IREF=IR2
GO TO 35
ENDIF
PHI12=(1.+(AMU(1)/AMU(2))**0.5*$(
(WM(2)/WM(1))**0.25)**2/
&
((8.*(1.+WM(1)/WM(2)))**0.5)
PHI21=PHI12*AMU(2)/AMU(1)*WM(1)/WM(2)
ANSWER=XM*AMU(1)/(XM+(1.-XM)*PHI12)+(1.-XM)*AMU(2)/$
&
((1.-XM)+XM*PHI21)
GO TO 1000

VAPOR THERMAL CONDUCTIVITY
THE PURE COMPONENT THERMAL CONDUCTIVITY
EQUATIONS ARE FROM THERMOPHYSICAL PROPERTIES OF
REFRIGERANTS 1976 ASHRAE AND ARE FOR GAS AT 1 ATM.
TEMPERATURE LIMITS FOR THESE EQUATIONS ARE SHOWN IN
THE REFERENCE. THEY ARE NOT INCLUDED HERE FOR THE
SAME REASON STATED IN LIQUID VISCOSITY. THE MIXING
RULE IS THE METHOD OF MASON AND SAXENA P. 531 IN
PROPERTIES OF GASES & LIQUIDS 4TH EDITION BY REID,
PRAUSNITZ, AND POLING. EPSILON=1.0 AS SUGGESTED BY
THE AUTHORS.

40 I=1
IREF=IR1
45 IF(IREF.EQ.3)THEN
  AKT(I)=SQRT(T*1.8)/(529.2954+4.0556715E5/(T*1.8)+
  1.84449E8/(T*1.8)**2)
ELSEIF(IREF.EQ.5)THEN
  THIS IS A R14 SATURATED VAPOR.
  TF=32.0+((T-273.15)*9.0/5.0)
  AKT(I)=6.054546E-3+(8.687652E-6*TF)+
  (5.864988E-11*TF*TF*TF)+
  AKT(I)=AKT(I)*1.7307
  CVR14(K)=AKT(I)
ELSEIF(IREF.EQ.9)THEN
  AKT(I)=SQRT(T)/{-2716.32+
  3.30226E6/T-9.71911E8/T**2+
  1.1228E11/T**3)
  CVR114(K)=AKT(I)
ELSE
  WRITE(6,*)' IREF=',IREF,' NO VAPOR THERMAL
  CONDUCTIVITY', ' DATA AVAILABLE'
  STOP
ENDIF
IF(I.EQ.1)THEN
  I=2
  IREF=IR2
  GO TO 45
ENDIF
PHI12=(1.+(AKT(1)/AKT(2)**
  0.5*(WM(2)/WM(1))**0.25)**2/
  (8.*(1.+WM(1)/WM(2))**0.5)
PHI21=PHI12*AKT(2)/AKT(1)*WM(1)/WM(2)
ANSWER=XM*AKT(1)/(XM*(1.-XM)*PHI12)+(1.-XM)*AKT(2)/
  (1.-XM)*XM*PHI21
GO TO 1000
1000 RETURN
END
Appendix C

List of Data-Processing Program (V.1)
SUBROUTINE CALNUM
C THIS ROUTINE CALCULATES THE
C NUSSELT, PRANDTL, STANTON AND REYNOLDS NUMBERS AT
C THE MEASURED LOCATIONS.
C CHUAN WENG, 8-23-89
REAL RE(20), NU(20), ST(20), PR(20)
REAL U(20), V(20), CP(20), TMC(20), H(20)
COMMON/RENUM/RE
COMMON/NUNUM/NU
COMMON/STNUM/ST
COMMON/PRNUM/PR
COMMON/MISC1/X, F0, F1, D
COMMON/MASF/AMDOTA
COMMON/TRAN1/U
COMMON/TRAN2/V
COMMON/TRAN3/TMC
COMMON/HTCOEF/H
COMMON/TRAN4/CP
BEGIN
DO 10 I = 11, 8, -1
RE(I) = 4.0 * AMDOTA * D / (U(I) * 3.14159 * D**2)
NU(I) = H(I) * D / TMC(I)
PR(I) = 1000.0 * CP(I) * U(I) / TMC(I)
ST(I) = NU(I) / (RE(I) * PR(I))
CONTINUE
10 WRITE(*,*) 'Re, Nu, .. NUMBER CALCULATION COMPLETED.'
RETURN
END
SUBROUTINE ENEBAL
C THIS ROUTINE DOES THE ENERGY BALANCE CALCULATION
C BASED ON MEASURED PARAMETERS.
C CHUAN WENG, 8-21-89
C
REAL ENERIN,ENEROUT,ENCHAG
C ENERIN: TOTAL ENERGY INTO THE SYSTEM.(W)
C ENERGY: TOTAL ENERGY OUT OF THE SYSTEM.(W)
C ENCHAG: ENERIN-ENERGY.(W)
COMMON/MISC2/PDIS,PSUCS,TDIS,TSUCS,VOLEF,WPUMP
COMMON/RESLT1/TINCR,SHDLU,DTAVG,HAVG,HFLX
COMMON/RESLT8/AMEVP,AMCON,AMHAVG,QCOND
COMMON/ENB/ENERIN,ENEROUT,ENCHAG
C BEGIN
ENERIN=WPUMP+SHDLU
ENEROUT=QCOND
ENCHAG=ENERIN-ENEROUT
WRITE(*,*) 'ENERGY BALANCE ANALYSIS COMPLETED.'
RETURN
END
THIS PROGRAM CALCULATES HEAT TRANSFER DATA
AND GENERATES TABLES FROM EXPERIMENTAL RAW DATA
CHUAN WENG, 8-12-89
REAL QUILIF
INTEGER INDEX
COMMON/DEX/INDEX
COMMON/JUG/QUILIF
CHARACTER*64 FNAMEI
CHARACTER*64 FNAMEO

C BEGIN

WRITE(*,10)
10 FORMAT( ' INPUT FILE NAME.')
READ(*,20) FNAMEI
WRITE(*,30)
30 FORMAT( ' OUTPUT FILE NAME.')
READ(*,20) FNAMEO
20 FORMAT(A)
OPEN(UNIT=5, FILE=FNAMEI)
OPEN(UNIT=6, FILE=FNAMEO,STATUS='NEW')
WRITE(*,*) ' INPUT THE STARTING TEST ID NUMBER.'

READ(*,*), INDEX
40 CONTINUE
CALL READIN
CALL HTEXPS
CALL TRANS
CALL MASFLOS
CALL CALNUM
CALL ENEBAL
CALL OUTPUT
INDEX=INDEX+1
IF(QUILIF.EQ.3.0) THEN
GOTO 50
ELSE
GO TO 40
ENDIF
50 CONTINUE
STOP
END
SUBROUTINE HTEXPS
C THIS ROUTINE IS USED TO TABULATE THE HEAT TRANSFER
C COEFFICIENTS FROM MEASUREMENTS OF TEMPERATURES
C INSIDE INSULATION, ON THE SURFACE OF COPPER TUBING
C AND FLUID STREAM.
C CHUAN WENG 5-30-89
REAL T(20),H(20),DT(20),DINSUL(20)
REAL KINSUL,KEPOXY
COMMON/TEMP/T
COMMON/MISC/Q
COMMON/HTCOEF/H
COMMON/TEMPDT/DT
COMMON/RESLT1/TINCR,SHDLU,DTAVG,HAVG,HFLX
DATA DEPOXY,DOUTC,DINC,KINSUL,
$    KEPOXY,PI/0.0147,0.0127,
$    0.01107,0.0368,0.404,3.1415926/
DATA DINSUL(11),DINSUL(10),DINSUL(9),
$    DINSUL(8)/0.0381,
$    0.0381,0.0365,0.0389/
SDT=0.0
HSUM=0.0
SHDLU=0.0
DO 10 I=12,15
    J=J+2
   IF(I.EQ.12) J=1
    QDLU=2.0*PI*(T(I)-T(J))
    QDLL1=(LOG(DINSUL(I-4)/DEPOXY))/KINSUL
    QDLL2=(LOG(DEPOXY/DOUTC))/KEPOXY
    QDLL=QDLL1+QDLL2
    QDL=QDLU/QDLL
    QHEATER=Q/6.0
    HDLU=QHEATER+QDL
    HDLL=(T(J)-T(I-4))*0.03477
    N=I-4
    DT(N)=T(J)-T(N)
    SDT=SDT+DT(N)
    H(N)=HDLU/HDLL
    HSUM=H(N)+HSUM
    SHDLU=SHDLU+(HDLU*1.5)
10 CONTINUE
TINCR=T(8)-T(11)
DTAVG=SDT/4.0
HAVG=HSUM/4.0
HFLX=SHDLU/(3.14159*0.0127*6.0)
WRITE(*,*) 'HEAT TRANSFER CALCULATION COMPLETED.'
RETURN
END
SUBROUTINE MASFLOS
C THIS PROGRAM DOES THE CONVERSION OF FAKE MASS
C FLOW RATE OUTPUT FROM DATA REQUITION PROGRAM TO
C ACTURAL MASS FLOW.
C CHUAN WENG, JUNE 26, 1989
REAL MASFLX,MASFLZ
REAL T(50),P(20),ENTH(20)
REAL AMDOTA,VRATIO,DISRATE,A1,A2,GAMA,GAM
C DENNS: SUCTION FLUID DENSITY(KG/M**3)
C DENSd: DISCHARGE FLUID DENSITY(KG/M**3)
C VRATIO: 1/18
C DISRATE: VOLUME FLOW RATE (KG**3/S)
C A1,A2,GAMA,GAM: CONSTANTS
COMMON/TEMP/T
COMMON/PSE/P
COMMON/ENTHAL/ENTH
COMMON/MASF/AMDOTA,MASFLX,MASFLZ
COMMON/MISC/Q,DP,POUT,DISRATE,
$ AMDOTM,VRATIO,A1,A2,GAM
COMMON/MISC1/X,F0,F1,D
COMMON/MISC2/PDIS,PSUCS,TDIS,TSUCS,VOLEF
COMMON/VS/VSUCS,WMOL
COMMON/RESLT/TINCR,SHDLU,DTAVG,HAVG
COMMON/RESLT/AMDOV,AMDOM,VOLE,SHDLU,AMHVG,PSUCS,TDIS,TSUCS,VOLEF
COMMON/RESLt/AMDOV,AMDOM,VOLE,SHDLU,AMHVG,PSUCS,TDIS,TSUCS,VOLEF
COMMON/RESLT8/AMEVP,AMCON,AMHVG,QCOND
BEGIN
RPAB=((1.0-(VOLE*VRATIO))/VRATIO)**GAM
PAB6=((1.0/A2**2)-(1.0/A1**2))*$ (AMDOTM**2)*0.157/2000.0
$ RP=PSUCS/PDIS
RG=LOG(RP)/LOG(RP)
GAMA=1.0/(1.0+RG)
GAMAR=1.0/GAMA
VOLE=(1.0-VRATIO*RPAB**GAMAR)/(1.0-VRATIO)
AMDOV=VOLE*DISRATE/VSUCS
AMDOM=(2000.0*PAB6/((1.0/A2**2)-(1.0/A1**2))**VSUCS))**0.5
$ AMDOTA=(AMDOV+AMDOM)/2.0
MASFLX=AMDOTA*4.0/(3.14159*D**2)
CALCULATE MASS FLOW RATE BASED ON ENTHALPY
C DIFFERENCE
CALL TPIN(T(16),PDIS,X,HCONO,
$ Z1,Z2,Z3,Z4,Z5,Z6,Z7)
CALL TPIN(TDIS,PDIS,X,HCONI,Z1,
$ Z2,Z3,Z4,Z5,Z6,Z7)
HCONI=HCONI/WMOL
HCONO=HCONO/WMOL
QCOND=1.1774*15.85*8.107E-3*
$ 1.004E+3*(T(18)-T(17))
AMCON=QCOND*0.001/(HCONI-HCONO)
AMEVP=SHDLU*0.001/(ENTH(8)-ENTH(11))
AMHAVG = (AMEVP + AMCON) / 2.0
MASFLZ = AMHAVG * 4.0 / (3.14159 * D**2)
WRITE(*,*) ' MASS FLOW RATE CALCULATION COMPLETED.'
RETURN
END
SUBROUTINE OUTPUT

THIS PROGRAM TABULATES REDUTED RESULTS FROM RAW DATA.

CHUAN WENG, 8-12-89

REAL U(20),QL(20),CP(20),TMC(20),DT(20),H(20),V(20)

REAL T(50),P(50),ENTH(20),ALFA(20),XML(20),XMV(20)

REAL RE(20),NU(20),ST(20),PR(20)

REAL MASFLX,MASFLZ,ENEIN,ENEOUT,ENCHAG

INTEGER INDEX

COMMON/DEX/INDEX

COMMON/TEMP/T

COMMON/PSE/P

COMMON/ENTHAL/ENTH

COMMON/ALFAB/ALFA

COMMON/HTCOEF/H

COMMON/TEMPDT/DT

COMMON/MISC1/X,F0,F1,D

COMMON/RESLT1/TINCR,SHDLU,DTAVG,HAVG,HFLX

COMMON/TRAN1/U

COMMON/TRAN2/V

COMMON/TRAN3/TMC

COMMON/TRAN4/CP

COMMON/RESLT2/QL

COMMON/XMLB/XML

COMMON/XMLB/XML

COMMON/RESLT7/AMDOV,AMDOM,VOLE,GAMA

COMMON/RESLT8/AMEVP,AMCON,AMHAVG,QCOND

COMMON/MAFL/MASFLX,MASFLZ

COMMON/ENC/ENERIN,ENEROUT,ENCHAG

COMMON/RENUM/RE

COMMON/NUM/NU

COMMON/STNUM/ST

COMMON/PNUM/PR

BEGIN

WRITE(*,5) INDEX
WRITE(6,5) INDEX

5 FORMAT( ' THE TEST ID NUMBER IS',I3,'.' )
WRITE(*,10)
WRITE(6,10)

10 FORMAT(1X,'------------------------------------
------------------------------------
------------------------------------')

WRITE(*,20)
WRITE(6,20)

20 FORMAT(2X,'L',5X,'T',6X,'P',5X,'H',6X,'h',5X,'DT',

5X,'U',5X,'V',5X,'K',6X,

3X,'ALFA',3X,'XMV',

3X,'XML',5X,'Re',6X,'Nu',5X,'Pr',6X,'St')
WRITE(*,30)
WRITE(6,30)
FORMAT(2X, 'M', 5X, 'K', 5X, 'KPA',
$ 2X, 'KJ/KG', 1X, 'W/M**2*K',
$ 1X, 'K', 2X, 'KG/M*S', 1X, 'M**3/KG',
$ 1X, 'W/M*K', 1X, 'KJ/KG*K')
WRITE(*, 10)
WRITE(6, 10)
DO 40 I=11, 8, -1
IF(I.EQ.11) THEN
  DEX=0.0
ELSEIF(I.EQ.10) THEN
  DEX=2.0
ELSEIF(I.EQ.9) THEN
  DEX=4.0
ELSE
  DEX=6.0
ENDIF
WRITE(*, 50) DEX, T(I), P(I), ENTH(I), H(I), DT(I), U(I),
$ V(I), TMC(I), CP(I), ALFA(I), QL(I), XMV(I), XML(I),
$ RE(I), NU(I), PR(I), ST(I)
WRITE(6, 50) DEX, T(I), P(I), ENTH(I), H(I), DT(I), U(I),
$ V(I), TMC(I), CP(I), ALFA(I), QL(I), XMV(I), XML(I),
$ RE(I), NU(I), PR(I), ST(I)
CONTINUE
WRITE(*, 10)
WRITE(6, 10)
FORMAT(1X, F3.1, 1X, F6.2, 1X, F6.2, 1X,
$ F5.2, 1X, F7.2, 1X, F4.2, 1X,
$ F6.5, 1X, F5.4, 1X, F5.4, 1X, F6.4,
$ 1X, F5.4, 1X, F6.2, 1X, F6.3, 1X, F7.6)
WRITE(*, 60) AMDOV, AMDOM, AMDOTA, MASFLX, X
WRITE(*, 70) TINCR, SHDLU, DTAVG, HAVG
WRITE(*, 80) AMEVP, AMCON, AMHAVG, MASFLZ
WRITE(*, 90) VOLE, GAMA, ENERIN, ENEROUT, ENCHAG
WRITE(*, 100) HFLX
WRITE(6, 60) AMDOV, AMDOM, AMDOTA, MASFLX, X
WRITE(6, 70) TINCR, SHDLU, DTAVG, HAVG
WRITE(6, 80) AMEVP, AMCON, AMHAVG, MASFLZ
WRITE(6, 90) VOLE, GAMA, ENERIN, ENEROUT, ENCHAG
WRITE(6, 100) HFLX
FORMAT(1X, 'AMDOTV=', F6.5,
$ '(KG/S)', ', ', 'AMDOTM=', F6.5,
$ '(KG/S)', ', ', 'AMDOTA=', F6.5,
$ '(KG/S)', ', ', 'MASFLX=', F7.2,
$ '(KG/S**M**2)', ', ', 'X=', F4.3, '(MOLAR)')
70 FORMAT(1X, 'TINCR=', F5.2, ' (K)', ', ', 'HEAT
$ RATE=', F6.2, ' (W)', ', ', 'DT(W TO
$ F)=', F4.2, ' (K)', ', ', 'H.T
$ COEFF. (AVG.)=', F6.2, ' (W/M**2*K')
80 FORMAT(1X, 'AMEVP=', F6.5, '(KG/S)', ', ',
$ 'AMCON=', F6.5, '(KG/S)', ', ',
$ 'AMHAVG=', F6.5, '(KG/S)', ', ')
$ 'MASFLZ=', F6.2, '(KG/S*MM**2)')
FORMAT(1X, 'VOLEF=', F4.3, ', ', 'GAMA=',
$ F5.3, ', ', 'ENERGY(IN)=',
$ F7.2, '(W)', ', ', 'ENERGY(OUT)=', F7.2,
$ ', '(W)', ', ', 'ENERGY_BALANCE=', F7.2, '(W)')
100 FORMAT(1X, 'HEAT FLUX=', F7.2, '(W/MM**2)')
RETURN
END
SUBROUTINE READIN
C THIS ROUTINE READS EXPERIMENTAL RESULTS SUCH AS
C TEMPERATURES, PRESSURES AND MASS FLOW RATES DE-
C TERMINED BY DATA ACQUISITION PROGRAM TO GENERATE
C NECESSARY INFORMATION FOR PLOTTHS.
C CHUAN WENG, 8-23-89
REAL T(50),P(20)
C T: MEASURED STREAM TEMPERATURES INSIDE
C EVAPORATOR.(K)
C P: STREAM PRESSURES.(KPA)
REAL Q,DP,AMDOTM,VOLEF,POUT
C Q: HEATER POWER.(WATTS)
C DP: PRESSURE DROP ACROSS EVAPORATOR.(KPA)
C AMDOTM: MASS FLOW RATE DETERMINED BY
C METER.(KG**3/S)
C VOLEF: VOLUMETRIC EFFICIENCY.
C POUT: PRESSURE AT OULET OF EVAPORATOR.(KPA)
REAL A1,A2,GAM,DISRATE,VRATIO
C A1,A2,GAM,DISRATE,VRATIO: CONSTANT USED IN
C ACQUISITION PROGRAM.
COMMON/JUG/QUILIF
COMMON/TEMP/T
COMMON/PSE/P
COMMON/MISC/Q,DP,POUT,DISRATE,
AMDOTM,VRATIO,A1,A2,GAM
COMMON/MISC1/X,F0,F1,D
COMMON/MISC2/PDIS,PSUCS,TDIS,TSUCS,VOLEF,VPUMP
BEGIN
READ(5,*) QUILIF
READ(5,*) Q
DO 30 J=1,18
READ(5,*) T(J)
CONTINUE
30 START READING DATA FOR MASS FLOW CALCULATION.
READ(5,*) PDIS,PSUCS,TDIS,TSUCS,VPUMP
READ(5,*) X,AMDOTM,VOLEF,DP,POUT
F0=0.0
F1=0.0
D=0.011
A1=1.5608E-04
A2=2.7983E-05
GAM=1.11
DISRATE=0.00277
VRATIO=1.0/18.0
WRITE(*,*) ' INPUT DATA COMPLETED.'
RETURN
END
$DEBUG

SUBROUTINE TRANS

THIS ROUTINE CALCULATES TRANSPORT PROPERTIES IN
ORDER TO FIND EXPERIMENTAL RE AND NU NUMBERS.

CHUAN WENG 8-23-89

REAL ENTH(20), ALFA(20)
REAL X, F0, F1, DP, POUT
REAL VSUCS
REAL UVR14(100), ULR14(100),
   UVR14(100), ULR14(100),
   CVR14(100), CLR14(100), CVR14(100), CLR(100)
REAL U(20), CP(20), TMC(20),
   T(20), P(20), QL(20), V(20)

DIMENSION COEFF(9,20), CRIT(5,20)

COMMON/TEMP/T
COMMON/PSE/P
COMMON/MISC/Q, DP, POUT
COMMON/MISC1/X, F0, F1, D
COMMON/MISC2/PSUCS, TDIS, TSUCS, VOLEF
COMMON/TRAN1/U
COMMON/TRAN2/V
COMMON/TRAN3/TMC
COMMON/TRAN4/CP
COMMON/VS/VSUCS, WMOL
COMMON/RESULT2/QL
COMMON/ENTHAL/ENTH
COMMON/ALFAB/ALFA
COMMON/XMLB/XML
COMMON/XMVB/XMV
COMMON/ESDATA/ COEFF, CRIT

BEGIN
DPICR = DP/3.0
P(11) = DP + POUT
P(10) = P(11) - DPICR
P(9) = P(10) - DPICR
P(8) = POUT
CALL BCONST(5, 9, F0, F1)
WM1 = CRIT(1, 5)
WM2 = CRIT(1, 9)
WMOL = XM*WM1 + ((1.0 - XM)*WM2)

DO 10 I = 8, 11
CALL TPIN(T(I), P(I), X, ENTH(I), QL(I), XL(I), XV(I),
   VLM, VVM, D3, D4)

ENTH(I) = ENTH(I)/WMOL
VML = WM1*XLM(I) + (1.0 - XLM(I))*WM2
WMV = WM1*XVM(I) + (1.0 - XVM(I))*WM2
XML(I) = XL(I)*WM1/WMOL
XVM(I) = XM(I)*WM1/WMV
QL(I) = QL(I)*WMV/(QL(I)*WMV + (1.0 - QL(I))*WML)
XM = X*WM1/WMOL
VV = VVM/WMV
VL = VLM/WML
V(I) = VV*QL(I) + (1.0 - QL(I))*VL
ALFA(I) = 1.0/((1.0 + ((1 - QL(I)))*VL/(QL(I)*VV)))
CALL VISCON(1, T(I), X, UL, 5, 9, I)
CALL VISCON(2, T(I), X, TMCL, 5, 9, I)
CALL VISCON(3, T(I), X, UV, 5, 9, I)
CALL VISCON(4, T(I), X, TMCV, 5, 9, I)
U(I) = UV*QL(I) + (1.0 - QL(I))*UL
TMC(I) = TMCV*QL(I) + (1.0 - QL(I))*TMCL
CALL HVCPS(5, T(I), VVM, XL(I), Z, CVV, CPV, VS)
CALL HVCPS(5, T(I), VLM, XL(I), Z, CVL, CPL, VS)
CPV = CPV/WMV
CPL = CPL/WML
CP(I) = CPV*QL(I) + (1.0 - QL(I))*CPL
CONTINUE
CALL TPIN(TSUCS, PSUCS, X, DO, QLSUC, XSL, XSV, $ VLM, VVM, D3, D4)
WML = WM1*XSL + (1.0 - XSL)*WM2
WMV = WM1*XSV + (1.0 - XSV)*WM2
VLS = VLM/WML
VVS = VVM/WMV
IF(QLSUC.GE.1.0) THEN
VSUCS = VVS
ELSEIF(QLSUC.GE.0.0.OR.QLSUC.LT.1.0) THEN
QLSUC = QLSUC*WMV/(QLSUC*WMV + (1.0 - QLSUC)*WML)
VSUCS = VVS*QLSUC/(1.0 - QLSUC)*VLS
ELSE
WRITE(*,*) ' THERE IS SOMETHING WRONG WITH THE $ SUCTION LINE PARAMETER.'
ENDIF
WRITE(*,*) ' TRANSPORT PROPERTY ROUTINE $ COMPLETED.'
RETURN
Appendix D

List of Data-Processing Program(V.2)
SUBROUTINE CALNUM1
C THIS ROUTINE CALCULATES THE
C NUSSELT, PRANDTL, STANTON
C AND REYNOLDS NUMBERS AT THE MEASURED LOCATIONS.
C CHUAN WENG, 11-12-89
REAL RE(20), NU(20), ST(20), PR(20)
REAL U(20), V(20), CP(20), TMC(20), H(20)
COMMON/RENUM/RE
COMMON/NUNUM/NU
COMMON/STNUM/ST
COMMON/PRNUM/PR
COMMON/MISC1/X, F0, F1, D
COMMON/MASF/AMDOTA
COMMON/TRAN1/U
COMMON/TRAN2/V
COMMON/TRAN3/TMC
COMMON/HTCOEF/H
COMMON/TRAN4/CP
C BEGIN
DO 10 I=11, 8, -1
RE(I) = 4.0 * AMDOTA * D / (U(I) * 3.14159 * 3**2)
NU(I) = H(I) * D / TMC(I)
PR(I) = 1000.0 * CP(I) * U(I) / TMC(I)
ST(I) = NU(I) / (RE(I) * PR(I))
10 CONTINUE
WRITE(*,*) 'Re, Nu, .. NUMBER CALCULATION COMPLETED.'
RETURN
END
SUBROUTINE ENEBAL1
C THIS ROUTINE DOES THE ENERGY BALANCE CALCULATION
C BASED ON MEASURED PARAMETERS.
C CHUAN WENG, 11-21-89
C
REAL T(40)
REAL ENERIN,ENEROUT,ENCHAG
C ENERIN: TOTAL ENERGY INTO THE SYSTEM.(W)
C ENERGY: TOTAL ENERGY OUT OF THE SYSTEM.(W)
C ENCHAG: ENERIN-ENERGY.(W)
COMMON/TEMP/T
COMMON/MISC/DUM1,DUM2,DUM3,WPUMP
COMMON/RESLT1/TINCR,SHDLU,DTAVG,HAVG,HFLX
COMMON/RESLT8/AMEVP,AMCON,AMHAVG,QCOND
COMMON/ENB/ENERIN,ENEROUT,ENCHAG
C BEGIN
WRITE(*,*) 'THE ENERGY BALANCE ANALYSIS IS IN
$ PROGRESS.'
HRCOMP=(T(38)-T(39))*9.397*
$ 2.1556E-3*1.1774*1.0057E+3
ENERIN=WPUMP+SHDLU
ENEROUT=QCOND+HRCOMP
ENCHAG=ENERIN-ENEROUT
WRITE(*,*) 'THE ENERGY BALANCE ANALYSIS IS
$ COMPLETED.'
RETURN
END
C THIS PROGRAM CALCULATES HEAT TRANSFER DATA
C AND GENERATES TABLES FROM EXPERIMENTAL RAW DATA
C CHUAN WENG, 8-12-89
REAL QUILIF
INTEGER INDEX
COMMON/DEX/INDEX
COMMON/JUG/QUILIF
CHARACTER*64 FNAMEI
CHARACTER*64 FNAMEO
C BEGIN
        WRITE(*,10)
  10 FORMAT( ' INPUT FILE NAME.' )
        READ(*,20) FNAMEI
        WRITE(*,30)
  30 FORMAT( ' OUTPUT FILE NAME.' )
        READ(*,20) FNAMEO
        FORMAT(A)
        OPEN(UNIT=5, FILE=FNAMEI)
        OPEN(UNIT=6, FILE=FNAMEO,STATUS='NEW')
        WRITE(*,*), ' INPUT THE STARTING TEST ID NUMBER.'
        READ(*,*), INDEX
        CONTINUE
        CALL READIN1
        CALL HTEXPS1
        CALL TRANSP
        CALL MASFLOS1
        CALL CALNUM1
        CALL ENEBAL1
        CALL OUTPUT1
        INDEX=INDEX+1
        IF(QUILIF.EQ.3.0) THEN
          GOTO 50
        ELSE
          GO TO 40
        ENDIF
  50 CONTINUE
        STOP
        END
SUBROUTINE HTEXPS1
C THIS ROUTINE IS USED TO TABULATE THE HEAT TRANSFER
C COEFFICIENTS FROM MEASUREMENTS OF TEMPERATURES
C INSIDE INSULATION, ON THE SURFACE OF COPPER TUBING
C AND FLUID STREAM.
C CHUAN WENG 11-16-89
REAL T(40),H(20),DT(20),DINSUL(20)
REAL KINSUL,KEPOXY
COMMON/TEMP/T
COMMON/MISC/Q
COMMON/HTCOEF/H
COMMON/TEMPDT/DT
COMMON/RESULT/TINCR,SHDLU,DTAVG,HAVG,HFLX
DATA DEPOXY,DOUTC,DINC,KINSUL,
  KEPOXY,PI/0.0147,0.0127,
  0.01107,0.0368,0.404,3.1415926/
DATA DINSUL(11),DINSUL(10),
  0.0381,0.0381,0.0365,0.0389/
SDT=0.0
SHDLU=0.0
DO 10 I=12,15
  J=I+2
  IF(I.EQ.12) J=1
  QDLU=2.0*PI*(T(I)-T(J))
  QDLL1=(LOG(DINSUL(I-4)/DEPOXY))/KINSUL
  QDLL2=(LOG(DEPOXY/DOUTC))/KEPOXY
  QDLL=QDLL1+QDLL2
  QDL=QDLU/QDLL
  QHEATER=Q/6.0
  HDLU=QHEATER+QDL
  HDLL=(T(J)-T(I-4))*0.03477
  N=I-4
  DT(N)=T(J)-T(N)
  SDT=SDT+DT(N)
  H(N)=HDLU/HDLL
  HSUM=H(N)+HSUM
  SHDLU=SHDLU+(HDLU*1.5)
CONTINUE
TINCR=T(8)-T(11)
DTAVG=SDT/4.0
HAVG=HSUM/4.0
HFLX=SHDLU/(3.14159*0.0127*6.0)
WRITE(*,*)'HEAT TRANSFER CALCULATION COMPLETED.'
RETURN
END
SUBROUTINE MASFLOSl
C THIS ROUTINE CALCULATES THE MASS FLOW RATES
C BASED UPON THE VOLUMETRIC FLOW OF COMPRESSOR AND
C THE VENTURI METER.
C CHUAN WENG, 11-15-1989
REAL MASFLX,MASFLZ
REAL T(50),P(20),ENTH(20)
REAL AMDOTA,VRATIO,DISRATE,A1,A2,GAMA,GAM
C DENST: SUCTION FLUID DENSITY(KG/M**3)
C DENS: DISCHARGE FLUID DENSITY(KG/M**3)
C VRATIO: 1/18
C DISRATE: VOLUME FLOW RATE (KG**3/S)
C A1,A2,GAMA: CONSTANTS
COMMON/TEMP/T
COMMON/PSE/P
COMMON/ENTHAL/ENTH
COMMON/MAST/AMDOTA,MASFLX,MASFLZ
COMMON/MISC/Q,DISRATE,VRATIO,WPUMP
COMMON/MISC1/X,F0,F1,D,A1,A2
COMMON/MISC4/PDIS,PSUCS,TDIS,TSUCS
COMMON/VS/VSUCS,WMOL
COMMON/RESLT1/TINC,SHDLU,DTAVG,HAVG
COMMON/RESLT7/AMDOV,AMDOM,VOLE,GAMA
COMMON/RESLT8/AMEVP,AMCON,AMHAVG,QCOND
BEGIN
WRITE(*,*) 'THE MASS FLOW RATE CALCULATION IS IN PROGRESS.'
RT=TDIS/TSUCS
RP=PSUCS/PDIS
RG=LOG(RT)/LOG(RP)
GAMA=1.0/(1.0+RG)
GAMAR=1.0/GAMA
VOLE=(1.0-VRATIO*RP**(-GAMAR))/(1.0-VRATIO)
AMDOV=VOLE*DISRATE/VSUCS
AMDOM=(2000.0*PSUCS/(((1.0/A2**2)-(1.0/A1**2)))*VSUCS)**0.5
AMDOTA=(AMDOV+AMDOM)/2.0
MASFLX=(AMDOTA+AMDOM)/2.0
C CALCULATE MASS FLOW RATE BASED ON ENTHALPY DIFFERENCE
CALL TPIN(T(16),PDIS,X,HCONO,
 Z1,Z2,Z3,Z4,Z5,Z6,Z7)
CALL TPIN(TDIS,PDIS,X,HCONI,
 Z1,Z2,Z3,Z4,Z5,Z6,Z7)
HCONI=HCONI/WMOL
HCONO=HCONO/WMOL
QCOND=1.1774*15.85*8.107E-3*
$ 1.004E+3*(T(40)-T(39))
AMCON=QCOND*0.001/(HCONI-HCONO)
AMEVP=SHDLU*0.001/(ENTH(8)-ENTH(11))
AMHAVG=(AMEVP+AMCON)/2.0
MASFLZ=AMHAVG*4.0/(3.14159*D**2)
WRITE(*,*) ' THE MASS FLOW RATE CALCULATION IS COMPLETED.'
RETURN
END
SUBROUTINE OUTPUT1
C THIS PROGRAM TABULATES REDUCED RESULTS FROM RAW
C DATA.
C CHUAN WENG, 11-12-89
REAL U(20),QL(20),CP(20),TMC(20),
DT(20),H(20),V(20)
REAL T(50),P(50),ENTH(20),ALFA(20),XML(20),XMV(20)
REAL RE(20),NU(20),ST(20),PR(20)
REAL MASFLX,MASFLZ,ENEIN,ENEOUT,ENCHAG
INTEGER INDEX
COMMON/DEX/INDEX
COMMON/TEMP/T
COMMON/PSE/P
COMMON/ENTHAL/ENTH
COMMON/ALFAB/ALFA
COMMON/HTCOEF/H
COMMON/TEMPDT/DT
COMMON/PRESDEF/DP
COMMON/MISC1/X,F0,F1,D
COMMON/RESLT1/TINCR,SHDLU,DTAVG,HAVG,HFLX
COMMON/TRAN1/U
COMMON/TRAN2/V
COMMON/TRAN3/TMC
COMMON/TRAN4/CP
COMMON/RESLT2/QL
COMMON/XMLB/XML
COMMON/XMVB/XMV
COMMON/RESLT7/AMDOV,AMDOM,VOLE,GAMA
COMMON/RESLT8/AMEVP,AMCON,AMHAVG,QCOND
COMMON/MA SF/AMDOTA,MASFLX,MASFLZ
COMMON/ENB/ENERIN,ENEROUT,ENCHAG
COMMON/RENUM/RE
COMMON/NUNUM/NU
COMMON/STNUM/ST
COMMON/PRNUM/PR
BEGIN
WRITE(*,5) INDEX
WRITE(6,5)
FORMAT( 'THE TEST ID NUMBER IS',13,'.'
WRITE(*,10)
WRITE(6,10)
FORMA T(1X,'---------------------------'
$---------------------------'
$---------------------------'
WRITE(*,20)
WRITE(6,20)
FORMA T(2X,'L',5X,'T',6X,'P',6X,'H',7X,'h',4X,'DT',
$ 5X,'U',6X,'V',5X,'K',6X,'CP',
$ 4X,'ALFA',4X,'XQ',3X,'XMV',
$ 3X,'XML',4X,'Re',8X,'Nu',5X,'Pr',5X,'St'
 WRITE(*,30)
WRITE(6,30)
FORMAT(2X,'M',5X,'K',5X,'KPA',
$ 3X,'KJ/KG',1X,'W/M**2*K',
$ 1X,'K',3X,'KG/M*S',1X,'M**3/KG',
$ 1X,'W/M*K',1X,'KJ/KG*K')
WRITE(*,10)
WRITE(6,10)
DO 40 I=11,8,-1
IF(I.EQ.11) THEN
  DEX=0.0
ELSEIF(I.EQ.10) THEN
  DEX=2.0
ELSEIF(I.EQ.9) THEN
  DEX=4.0
ELSE
  DEX=6.0
ENDIF
WRITE(*,50) DEX,T(I),P(I),ENTH(I),H(I),DT(I),U(I),
$ V(I),TMC(I),CP(I),ALFA(I),QL(I),XMV(I),XML(I),
$ RE(I),NU(I),PR(I),ST(I)
WRITE(6,50) DEX,T(I),P(I),ENTH(I),H(I),DT(I),U(I),
$ V(I),TMC(I),CP(I),ALFA(I),QL(I),XMV(I),XML(I),
$ RE(I),NU(I),PR(I),ST(I)
40 CONTINUE
WRITE(*,10)
WRITE(6,10)
FORMAT(1X,F3.1,1X,F6.2,1X,F6.2,1X,
$ F6.2,1X,F7.2,1X,F4.2,1X,
$ F6.5,1X,F6.4,1X,F5.4,1X,
$ F6.4,1X,F6.4,1X,F6.4,1X,F5.4,1X,
$ F5.4,1X,F8.2,1X,F7.2,1X,F6.3,1X,F7.6)
WRITE(*,60) AMDOV,AMDOM,AMDOTA,MASFLX,X
WRITE(*,70) TINCR,SHDLU,DTAVG,HAVG
WRITE(*,80) AMEVP,AMCON,AMHAVG,MASFLZ
WRITE(*,90) VOLE,GAMA,ENERIN,ENEROUT,ENCHAG
WRITE(*,100) HFLX,DP
WRITE(6,60) AMDOV,AMDOM,AMDOTA,MASFLX,X
WRITE(6,70) TINCR,SHDLU,DTAVG,HAVG
WRITE(6,80) AMEVP,AMCON,AMHAVG,MASFLZ
WRITE(6,90) VOLE,GAMA,ENERIN,ENEROUT,ENCHAG
WRITE(6,100) HFLX,DP
60 FORMAT(1X,'AMDOTV=',F6.5,'
$ (KG/S)',',',',', 'AMDOTM=',F6.5,
$ (KG/S)',',',',', 'AMDOTA=',F6.5,
$ ' (KG/S)',',',',', 'MASFLX=',F7.2,
$ ' (KG/S**M**2)',',',',', 'X=',F4.3,'(MOLAR)')
70 FORMAT(1X,'TINCR=',F5.2,'(K)',',',',', 'HEAT
$ RATE=' ,F6.2,'(W)',
$ ',',',', 'DT(W TO F)=',F4.2,'(K)',',',',', 'H.T
$ COEFF.(AVG.)=',F6.2,
$ ' (W/M**2*K)' )
80 FORMAT(1X,'AMEVP=',F6.5,'(KG/S)',',',',',
$ (KG/S)',',',',', 'X=',F4.3,'(MOLAR)')
'AMCON=', F6.5, '(KG/S)', '

'AMHAVG=', F6.5, '(KG/S)', '

'MASFLZ=', F6.2, '(KG/S*M**2)'

FORMAT(1X, 'VOLEF=', F7.3, ',

'GAMA=', F5.3, ', 'ENERGY(IN)='

,F7.2, '(W)', ', 'ENERGY(OUT)=' F7.2, ,

'(W)', ', 'ENERGY BALANCE=' F7.2, '(W)'

HEAT FLUX=' F7.2, '(W/M**2)'

PRESSURE DROP=' F6.2, '(KPA)'

RETURN

END
SUBROUTINE READIN1
C THIS ROUTINE READS IN EXPERIMENTAL VALUES
C SUCH AS TEMPERATURES, VENTURI METER READINGS, POWER
C CONSUMED BY THE COMPRESSOR AND OTHER READINGS FOR
C MASS FLOW RATE CALCULATION, HEAT TRANSFER
C COEFFICIENT CALCULATION AND OTHER PARAMETERS.
C CHUAN WENG, 11-12-89
REAL T(50)
C T: MEASURED STREAM TEMPERATURES INSIDE
C EVAPORATOR.(K)
REAL Q,F0,F1,D,A1,A2,DISRATE,VRATIO
C Q: RESISTANCE HEATER POWER.(WATTS)
C F0,F1: CONSTANTS FOR NBS ROUTINES.
C D: TUBING ID.(METER)
C A1,A2: CONSTANTS FOR VENTURI METER.
C DISRATE: COMPRESSOR PISTON DISPLACEMENT.
C VRATIO: CLEAR VOLUME RATIO.
COMMON/JUG/QUALIF
COMMON/TEMP/T
COMMON/MISC/Q,DISRATE,VRATIO,WPUMP
COMMON/MISC1/X,F0,F1,D,A1,A2
COMMON/MISC3/PAD,PBD,PAS,PBS,PAEI,PBEI,PAEO,PBEO
COMMON/PTVOLT/VOLTEI,VOLTEO,VOLTS,VOLTD
C BEGIN
READ(5,*), QUALIF
READ(5,*), Q
DO 30 J=1,40
READ(5,*), J,T(J)
30 CONTINUE
T(11)=T(37)
READ(5,*), WPUMP,X
READ(5,*), VOLTEI,VOLTEO,VOLTS,VOLTD
C START READING CONSTANTS FOR PRESSURE CALIBRATION.
PAD=-0.2469
PBD=2990.48
PAS=-0.5523
PBS=2994.13
PAEI=-0.0378
PBEI=2991.56
PAEO=-0.3632
PBEO=2992.73
F0=0.0
F1=0.0
D=0.011
A1=1.5608E-4
A2=2.7983E-5
DISRATE=0.00277
VRATIO=1.0/18.0
WRITE(*,*) ' INPUT DATA COMPLETED.'
RETURN
END
SUBROUTINE TRANSP

THIS ROUTINE CALCULATES TRANSPORT PROPERTIES IN ORDER TO FIND EXPERIMENTAL RE AND NU NUMBERS.

CHUAN WENG 11-12-89

REAL ENTH(20), ALFA(20)
REAL X, F0, F1, DPICR
REAL VSUCS
REAL UVR14(100), ULR14(100),
    CVR14(100), CLR14(100), CVR114(100), CLR(100)
REAL U(20), CP(20), TMC(20),
    T(40), P(20), QL(20), V(20)
REAL UVR1(100), ULR1(100),
    CVR1(100), CLR1(100), CVR11(100), CLR11(100)
REAL U(20), CP(20), TMC(20),
    T(40), P(20), QL(20), V(20)

DIMENSION COEFF(9, 20), CRIT(5, 20)
COMMON/PTVOLT/VOLTEI, VOLTEO, VOLTS, VOLTD
COMMON/TEMP/T
COMMON/PSE/P
COMMON/PRESDEF/DP
COMMON/MISC/Q, DISRATE, VRATIO, WPUMP
COMMON/MISC1/X, F0, F1, D, A1, A2
COMMON/MISC3/PAD, PBD, PAS, PBS, PAEI, PBEI, PAOE, PBEO
COMMON/MISC4/PDIS, PSUCS, TDIS, TSUCS
COMMON/TRAN1/U
COMMON/TRAN2/V
COMMON/TRAN3/TMC
COMMON/TRAN4/CP
COMMON/VS/VSUCS, WMOL
COMMON/ESLT2/QL
COMMON/ENTHAL/ENTH
COMMON/ALFAB/ALFA
COMMON/XMLB/XML
COMMON/XMV/B/XMV
COMMON/ESDATA/Coeff, CRIT

BEGIN

WRITE(*,*) ' TRANSPORT PROPERTIES OF MIXTURE IN CALCULATION.'

P(11) = 6.895 * ((VOLTEI * PBEI / 1500.0) + PAEI)
P(8)  = 6.895 * ((VOLTEO * PBEO / 1500.0) + PAEO)
PSUCS = 6.895 * ((VOLTS * PBS / 1500.0) + PAS)
PDIS = 6.895 * ((VOLTD * PBD / 1500.0) + PAD)
DP = P(11) - P(8)
DPICR = DP / 3.0
P(10) = P(11) - DPICR
P(9)  = P(10) - DPICR
TSUCS = T(21)
TDIS = T(33)
CALL BCONST(5, 9, F0, F1)
WM1 = CRIT(1, 5)
WM2 = CRIT(1, 9)
WMOL = X * WM1 + ((1.0 - X) * WM2)

CALL TO...
DO 10 I=8,11
CALL TPIN(T(I),P(I),X,ENTH(I),QL(I),XL(I),XV(I),
         VLM,VVM,D3,D4)
ENTH(I)=ENTH(I)/WMOL
WML=WM1*XL(I)+(1.0-XL(I))*WM2
WMV=WM1*XV(I)+(1.0-XV(I))*WM2
XML(I)=XL(I)*WM1/WML
XMV(I)=XV(I)*WM1/WMV
QL(I)=QL(I)*WMV/(QL(I)*WMV+(1.0-QL(I))*WML)
XM=X*WM1/WMOL
VV=VVM/WMV
VL=VLM/WML
V(I)=VV*QL(I)+(1.0-QL(I))*VL
ALFA(I)=1.0/(1.0+((1-QL(I))*VL/(QL(I)*VV)))
CALL VISCON(1,T(I),X,UL,5,9,I)
CALL VISCON(2,T(I),X,TMCL,5,9,I)
CALL VISCON(3,T(I),X,UV,5,9,I)
CALL VISCON(4,T(I),X,TMCV,5,9,I)
U(I)=UV*QL(I)+(1.0-QL(I))*UL
TMC(I)=TMCV*QL(I)+(1.0-QL(I))*TMCL
CALL HCVCP5(5,T(I),VVM,XV(I),Z,CVV,CPL,VS)
CALL HCVCP5(5,T(I),VLM,XL(I),Z,CVL,CPL,VS)
CPV=CPV/WMV
CPL=CPL/WML
CP(I)=CPV*QL(I)+(1.0-QL(I))*CPL
CONTINUE
WRITE(*,*) 'TSUCS=',TSUCS,'PSUCS=',PSUCS,'X=',X
CALL TPIN(TSUCS,PSUCS,X,DO,QLSUC,XSL,XSV,
         VLM,VVM,D3,D4)
WML=WM1*XSL+(1.0-XSL)*WM2
WMV=WM1*XSV+(1.0-XSV)*WM2
VLS=VLM/WML
VVS=VVM/WMV
IF(QLSUC.GE.1.0) THEN
  VSUCS=VVS
ELSEIF(QLSUC.GE.0.0.AND.QLSUC.LT.1.0) THEN
  VSUCS=VVS
ELSE
  WRITE(*,*) 'THERE IS SOMETHING WRONG WITH THE
               SUCTION LINE PARAMETER.'
ENDIF
WRITE(*,*) 'TRANSPORT PROPERTY ROUTINE COMPLETED.'
RETURN
Appendix E

List of Input and Output Data of EXPCAL for Test Runs with R14/R114 mixture of 0.41 Molar Composition

The following is the input data for the program EXPCAL which calculates the experimental results. The input data includes measured temperatures, power input into the evaporator, etc.

Line 1: QUILIF
QUILIF = indicator, used to control calculation

Line 2: Q, T1, T2, T3, T4, T5, T6
Q = power input to the evaporator (Watts)
T1..T7 = temperatures measured by TCs placed evenly on the tubing wall in the counterflow direction (K)

Line 3: T7, T8, T9, T10, T11, T12, T13
T8..T11 = temperatures measured by TCs placed evenly on the center of the tube, T11 is the inlet temperature, T8 is the outlet temperature (K)

Line 4: T14, T15, T16, T17, T18, PDIS, PSUCS
T12..T15 = temperatures measured by TCs placed evenly inside insulation, they are correspondent to positions of T8 to T11.
T16 = temperature at the outlet of the condenser
T17, T18 = temperatures at the air inlet and outlet of the condenser cage (K)

Line 5: TDIS, TSUCS, WPUMP, X, AMDOTM, VOLEF, DP
TDIS = discharge temperature (K)
TSUCS = suction temperature (K)
WPUMP = compressor power (Watts)
X = fluid circulating composition (-)
AMDOTM = refrigerant mass flow rate (by meter, determined by data acquisition program) (Kg/s)
VOLEF = volumetric efficiency determined by data acquisition program (-)

DP = pressure drop measured by the differential pressure transducer (Kpa)

Line 6: POUT
POUT = pressure at outlet of evaporator (Kpa)

Note on the input data:

The refrigerant mass flow rate determined by the data
acquisition program and the volumetric efficiency were incorrect. They were corrected by the program EXPCAL.

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Specification of Variables in the Data Output Lists

L—thermocouple stations at downstream direction
T—measured temperatures by each thermocouple
P—pressure reading at each thermocouple station
H—mixture enthalpy
h—measured heat transfer coefficient
DT—temperature difference between wall and fluid
U—mixture viscosity
V—mixture specific volume
K—mixture thermal conductivity
Cp—mixture heat capacity
ALFA—flow void fraction
XQ—fluid quality (mass based)
XML—liquid composition
XMV—vapor composition
Re—flow Reynolds number
Nu—flow Nusselt number
Pr—mixture Prandtl number
St—flow Stanton number
AMDOTV—mass flow measured by volumetric method
AMDOTM—mass flow measured by venturi meter method
AMDOT—average of AMDOTV and AMDOTM
MASFLX—refrigerant mass flux based on AMDOT
X—mixture molar composition
TINC;R—temperature increase over the evaporator tube
HEAT RATE—total heat load to the evaporator
DT(W TO F)—average DT on measurement stations
H.T.COEFF.(AVG.)—average heat transfer coefficient over the evaporator
AMEVP—mass flow rate based on the energy balance on the evaporator
AMCON—mass flow rate based on the energy balance on the condenser
AMHAVG—average of AMCON and AMEVP
MASFLZ—mass flux based on AMHAVG
VOLEF—volumetric efficiency
GAMA—polytropic exponent
ENERGY(IN)—measured energy input to the test system
ENERGY(OUT)—measured energy output from the system
ENERGY BALANCE—difference between ENERGY(IN) and ENERGY(OUT)
HEAT FLUX—heat load per unit surface area
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AMDOP = 0.0125 (KG/S), AMDON = 0.00705 (KG/S), AMHAD = 0.0170 (KG/S), WASSPL= 122.16 (KG/S**2)
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THE TEST ID NUMBER IS 51.

AMDOTV: 0.0191 (KG/S), AMDOTW: 0.01488 (KG/S), AMDOTA = 0.0148 (KG/S), Z = 0.380 (W/MAL)
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4.0 248.77 199.45 57.89 547.85 6.10 .00045 .0358 .0539 .6680 .9880 .3387 .7235 .0159 3898.89 110.31 7.212 .002946
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ANDOTM=.0173(KG/S), ANDOTM=.0123(KG/S), ANDOM=.0159(KG/S), MASPL= 159.77(KG/S**2), z=-400(MOLAR)
TIME=19.00(K), HEAT RATE=64.38(W), DT(W TO P)=5.22(K), H.T COEFF. (AVG.)=444.31(W/M**2*K)
AMVP=.0157(KG/S), ANDOM=.00657(KG/S), AMHAF=.0112(KG/S), MASPL=117.02(KG/S**2)
VOLE=.557, GAM=1.156, ENERGY (IN)=1595.39(W), ENERGY (OUT)=1108.84(W), ENERGY BALANCE= 486.54(W)
HEAT FLUX=194.04(W/M**2)
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4.0 243.36 193.17 54.55 492.44 4.00 .00049 .0363 .0543 .8900 .9880 .3297 .7424 .0161 4510.46 98.61 7.472 .002295
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AMVP=.01654(KG/S), ANDOM=.00827(KG/S), AMHAF=.01065(KG/S), MASPL=109.39(KG/S**2)
VOLE=.676, GAM=1.175, ENERGY (IN)=1521.18(W), ENERGY (OUT)=1002.89(W), ENERGY BALANCE= 488.29(W)
HEAT FLUX=177.61(W/M**2)
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2.0 236.35 287.22 29.17 321.64 5.20 .00061 .0240 .0601 .8825 .9771 .2788 .8901 .0329 5316.53 58.65 8.757 .001264
4.0 242.05 284.36 46.86 429.87 3.90 .00055 .0220 .0578 .8803 .9792 .2964 .6485 .0287 6058.33 81.73 7.996 .001693
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ANDOTM=.03548(KG/S), ANDOTM=.01932(KG/S), ANDOM=.02740(KG/S), MASPL= 288.33(KG/S**2), z=-420(MOLAR)
TIME=17.20(K), HEAT RATE=349.51(W), DT(W TO P)=4.63(K), H.T COEFF. (AVG.)=368.81(W/M**2*K)
AMVP=.01621(KG/S), ANDOM=.00649(KG/S), AMHAF=.01135(KG/S), MASPL=119.43(KG/S**2)
VOLE=.580, GAM=1.202, ENERGY (IN)=1409.51(W), ENERGY (OUT)=1048.09(W), ENERGY BALANCE= 361.42(W)
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198

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AMDOTV=.01495(KG/S), AMDOTK=.00127(KG/S), AMDOTA=.00811(KG/S) MASPLX= 33.24(KG/MM**2), T=390(MOLAR)
TINCR=157.0(K), HEAT RATE=273.90(W), DT(W TO F)=4.02(K), H.T COEFF. (AVG.)=.318.08(W/MM**2)
AMEPK=.01099(KG/S), AMCON=.00540(KG/S), AMHAVG=.30925(KG/MM**2)
VOLPK=.44, GAMA=1.155, ENERGY(IN)=149.90(W), ENERGY(OUT)= 881.00(W), ENERGy BALANCE= 368.89(W)

THE TEST ID NUMBER IS 58.

| L | T | P | h | DT | U | V | X | K | CP | ALPA | XQ | XWF | IML | Re | Nu | Pr | St |
|---|---|---|---|----|---|---|---|---|----|------|----|----|----|----|----|----|----|----|
| W | K | KPA | KG/MM**2 | K | KG/MM**2 | KG/MM**2 | K | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 |

| TINCR=13.60(K), HEAT RATE=184.27(W), DT(W TO F)=3.37(K), H.T COEFF. (AVG.)=264.44(W/MM**2)
AMEPK=.01127(KG/S), AMCON=.00531(KG/S), AMHAVG=.00824(KG/S), MASFLX= 86.12(KG/MM**2)
VOLPK=.50, GAMA=1.163, ENERGY(IN)=1124.27(W), ENERGY(OUT)= 650.62(W), ENERGy BALANCE= 273.65(W)

THE TEST ID NUMBER IS 59.

| L | T | P | h | DT | U | V | X | K | CP | ALPA | XQ | XWF | IML | Re | Nu | Pr | St |
|---|---|---|---|----|---|---|---|---|----|------|----|----|----|----|----|----|----|----|
| W | K | KPA | KG/MM**2 | K | KG/MM**2 | KG/MM**2 | K | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 | KG/MM**2 |

| TINCR=10.50(K), HEAT RATE= 5.95(W), DT(W TO F)=2.70(K), H.T COEFF. (AVG.)= 99.92(W/MM**2)
AMEPK=.00472(KG/S), AMCON=.00025(KG/S), AMHAVG=.00099(KG/S), MASFLX= 41.34(KG/MM**2)
VOLPK=.204, GAMA=1.136, ENERGY(IN)= 804.61(W), ENERGY(OUT)= 486.07(W), ENERGy BALANCE= 318.56(W)

THE TEST ID NUMBER IS 60.

| L | T | P | h | DT | U | V | X | K | CP | ALPA | XQ | XWF | IML | Re | Nu | Pr | St |
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- Test ID number: 61.

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**THE TEST ID NUMBER IS 65.**

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**HEAT FLUX** = 1708.71 (W/M**2**) **THE TEST ID NUMBER IS 66.**

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**AKDOTV**: 0.0102 (KG/S), **AMDODW**: 0.0078 (KG/S), **AMDODA**: 0.00001 (KG/S), **MASPLI**: 94.82 (KG/S/M**2**), **X**: 0.410 (W/MOLAR)

**VAC**: 23.30 (E), **HEAT RATE**: 405.51 (W), **UT (W TO P)**: 5.30 (K), **H.T COEFF** (AVG): 414.00 (W/M**2**K)

**AMOS**: 0.0404 (KG/S), **AMOSW**: 0.0041 (KG/S), **AMOAVG**: 0.00648 (KG/S), **MASPL2**: 66.20 (KG/S/M**2**)

**VAC**: 0.0199, **CAMA**: 1.122, **ENERGY IN**: 1235.51 (W), **ENERGY OUT**: 668.35 (W), **ENERGY BALANCE**: 567.16 (W)

**HEAT FLUX**: 1693.92 (W/M**2**)
Appendix F

List of Input and Output Data of EXPCAL1 for Test Runs with R14/R114 Mixture of 0.50 Molar Composition

The following is the input data for the program EXPCAL1 which, unlike the EXPCAL, uses input data of different format as shown below. The program calculates the experimental results based on the measured temperatures and pressures at various locations. The input data includes measured temperatures, power input into the evaporator, etc.

**Line 1:**  QUILIF, Q

- **QUILIF:** indicator, used to control the calculation of results (-)
- **Q:** heat input to the evaporator (Watts)

**Line 2:**  J1, T1, J2, T2, J3, T3, J4

- J1..J4 = channel number on data logger (-)
- T1..T3 = wall temperatures (see specification for runs of 0.41 molar composition case) (K)

**Line 3:**  T4, J5, T5, J6, T6, J7, T7

- J5..J7 = channel number on data logger (-)
- T4..T7 = temperatures on the wall (same as T1..T3) (K)

**Line 4:**  J8, T8, J9, T9, J10, T10, J11

- J8..J11 = channel number on data logger (-)
- T8..T10 = fluid temperatures (K)

**Line 5:**  T11, J12, T12, J13, T13, J14, T14

- T11 = fluid temperature at the inlet of evaporator (K)
- J12 = index (-)
- T12..T14 = temperatures in the insulation (K)

**Line 6:**  J15, T15, J16, T16, J17, T17, J18

- J15..J18 = channel numbers on data logger (-)
- T15 = temperature of insulation corresponding to T11 (K)
- T16 = temperature of fluid in the cold block at upstream of the expansion valve (K)
- T17 = temperature of fluid in the cold block at downstream of evaporator (K)

**Line 7:**  T18, J19, T19, J20, T20, J21, T21

- J19..J21 = channel numbers on the data logger
- T18 = temperature of the stainless plate on the evaporating side of the cold blocks (K)
- T19 = temperature of the copper block on the
evaporating side (K) 
T20 = temperature of the stainless plate on the condensing side of the cold blocks (K) 
T21 = temperature of the copper block on the condensing side of the cold blocks (K) 

Line 8: 
J22, T22, J23, T23, J24, T24, J25 
J22..J25 = channel numbers on the data logger (-) 
T22 = fluid temperature at the evaporative side of hot blocks (K) 
T23 = fluid temperature at the condensing side of hot blocks (K) 
T24 = temperature of the stainless plate on the evaporating side of the hot blocks (K) 

Line 9: 
T25, J26, T26, J27, T27, J28, T28 
J26..J28 = channel numbers on the data logger (-) 
T25 = temperature of the hot block on the evaporating side (K) 
T26 = temperature of stainless plate on the condensing side of hot blocks (K) 
T27 = temperature of copper block of hot blocks (K) 
T28 = temperature at surface of heat exchanger at inlet of evaporating side (K) 

Line 10: 
J29, T29, J30, T30, J31, T31, J32 
J29..J32 = channel numbers on the data logger (-) 
T29 = temperature at surface of heat exchanger at outlet of condensing side (K) 
T30 = temperature at surface of heat exchanger at outlet of evaporating side (K) 
T31 = temperature at surface of heat exchanger at inlet of condensing side (K) 

Line 11: 
T32, J33, T33, J34, T34, J35, T35 
J33..J35 = channel numbers on data logger (-) 
T32 = surface temperature on the expansion valve (K) 
T33 = discharge temperature (K) 
T34..T35 = temperatures of heat exchanger surface (K) 

Line 12: 
J36, T36, J37, T37, J38, T38, J39 
J36..J39 = channel numbers on data logger 
T36, T37 = heat exchanger surface temperatures (K) 
T38 = air outlet temperature of compressor cage (K) 

Line 13: 
T39, J40, T40, J41, WPUMP, J42, X 
T39 = temperature of air inlet of condenser (K) 
J40 = channel 40 on data logger (-) 
T40 = temperature of air outlet of condenser (K) 
J41 = index (-) 
WPUMP = compressor power (Watts) 
J42 = index (-) 
X = fluid circulating composition (-) 

Line 14: 
J43, VOLTEI, J44, VOLTEO, J45, VOLTS, J46
\( J_{43} \ldots J_{46} = \text{indexes(-)} \)

\( \text{VOLTEI} = \text{voltage output measured on the pressure transducer at the inlet of evaporator (mV)} \)

\( \text{VOLTEO} = \text{voltage output measured on the pressure transducer at the outlet evaporator (mV)} \)

**Line 15:**

\( \text{VOLTD} = \text{voltage output measured on the pressure transducer at the discharge line of system (mV)} \)

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For the specification of calculated results, please refer to that for runs with 0.41 composition at page 194.
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HEAT FLUX= 815.32(W/M**2), PRESSURE DROP= 15.05(KPA)
Appendix G

List of Input and Output of EXPCAL1 for Test Runs with R14/R114 Mixture of 0.6 Molar Composition

The followings are the input data for test runs with 0.6 composition and the calculated results. For specifications of input data, please refer to those in Appendix F at page 203. The specifications of results can be found at page 194.

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**ANEVP:** 0.0034 (KG/S), **AMCON:** 0.00347 (KG/S), **ANMARCH:** 0.00347 (KG/S), **X:** 601.8 (KG/S**2**/K)

**VOFLP:** 1.457 (KG/S), **MENETY (IN):** 1981.94 (W), **MENETY (OUT):** 1984.69 (W), **MENETY BALANCE:** -10.65 (W)

**HEAT FLUX:** 501.45 (W/K**2**), **PRESSURE DROP:** 23.83 (KPA)

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**TTONS:** 11.50 (K), **HEAT RATE:** 162.6 (W), **DT (W TO F):** 3.10 (K), **H.T COEF.:** 254.5 (W/K**2**/K)

**ANEVP:** 0.014 (KG/S), **AMCON:** 0.0065 (KG/S), **ANMARCH:** 101.16 (KG/S**2**/K)

**VOFLP:** 0.803 (KG/S), **MENETY (IN):** 1980.04 (W), **MENETY (OUT):** 1984.27 (W), **MENETY BALANCE:** -3.23 (W)

**HEAT FLUX:** 673.33 (W/K**2**), **PRESSURE DROP:** 18.80 (KPA)

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**HEAT FLUX:** 256.13 (W), **PRESSURE DROP:** 24.11 (KPA)

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**TINC:** 14.20 (K), **HEAT RATE:** 205.65 (W), **DT (W TO F):** 1.38 (K), **H.T COEFF. (AVG.):** 235.46 (W/K**2*K)

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**VOLSP:** 77.1, **GAMA:** 1.394, **ENERGY IN:** 1976.33 (W), **ENERGY OUT:** 1965.88 (W), **ENERGY BALANCE:** 10.45 (W)

**HEAT FLUX:** 256.13 (W), **PRESSURE DROP:** 24.11 (KPA)

**THE TEST ID NUMBER IS 62.**

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**AMDOTV:** 0.0232 (KG/S), **AMDOTM:** 0.00875 (KG/S), **AMDOTA:** 0.0083 (KG/S), **MASFLX:** 534.34 (KG/S*K**2), **x:** 0.380 (MOLAR)

**TINC:** 14.20 (K), **HEAT RATE:** 205.65 (W), **DT (W TO F):** 1.38 (K), **H.T COEFF. (AVG.):** 235.46 (W/K**2*K)

**AMV:** 0.0306 (KG/S), **AMCON:** 0.0056 (KG/S), **MASFLX:** 100.60 (KG/S*K**2)

**VOLSP:** 77.1, **GAMA:** 1.394, **ENERGY IN:** 1976.33 (W), **ENERGY OUT:** 1965.88 (W), **ENERGY BALANCE:** 10.45 (W)

**HEAT FLUX:** 256.13 (W), **PRESSURE DROP:** 24.11 (KPA)

**THE TEST ID NUMBER IS 62.**
### Table 1: Thermodynamic Properties

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### Table 2: Thermodynamic Properties

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### Notes
- The test ID number is 63.
- The test ID number is 64.
**TINCE= 8.90(K), HEAT RATE=161.55(W), DT(W TO F)=2.37(K), H.T COEFF.(AVG.)=263.47(W/M^2*K)**

**RAfE=161.53(W), DT(W TO F):2.37(K)**

**AMEV=31990(KG/S), AMCON=0.00753(KG/S), AMH AVG=0.01331(KG/S), MASPL=109.04(KG/S**2)**

**VOL=F=596, GAMMA=1.445, ENERGY(IN)=2107.51(W), ENERGY(OUT)=2419.17(W), ENERGY BALANCE= -88.36(W)**

**HEAT FLUX= 674.75(W/M^2), PRESSURE DROP= -50.72(KPA)**

THE TEST ID NUMBER IS 68.

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**ANDOTV=0.04492(KG/S), ANDOTM=1.1047(KG/S), ANDOTA=.07781(KG/S), MASPLX= 818.80(KG/S**2), k=.590(MOLAL)**

**TINCE=10.50(K), HEAT RATE=206.11(W), DT(W TO F)=3.23(K), H.T COEFF.(AVG.)=319.53(W/M**2)**

**AMEV=0.1653(KG/S), AMCON=0.00805(KG/S), AMH AVG=0.01222(KG/S), MASPL=123.66(KG/S**2)**

**VOL=F=579, GAMMA=1.479, ENERGY(IN)=2106.11(W), ENERGY(OUT)=2574.27(W), ENERGY BALANCE= 151.14(W)**

**HEAT FLUX= 660.39(W/M**2), PRESSURE DROP= 32.33(KPA)**
Appendix H

Subprograms for the Calculation of Refrigerant Mixture Properties (Provided by the National Bureau of Standards)
SUBROUTINE BCONST (IR1,IR2,F0,F1)

C
C DEVELOPED BY MARK MCLINDEN AND GRAHAM MORRISON AT THE
C NATIONAL BUREAU OF STANDARDS UNDER FUNDING FROM THE
C ELECTRIC POWER RESEARCH INSTITUTE AND NBS.
C
C THIS ROUTINE ACCESSES THE CURVE FIT COEFFICIENTS TO THE
C EQUATION OF STATE PARAMETERS (STORED IN BLOCK DATA
C BDESC) FOR THE REFРИGERANT PAIR OF INTEREST. THE
C REFERENCE STATES FOR ENTHALPY AND ENTROPY ARE ALSO
C COMPUTED. THIS SUBROUTINE MUST BE CALLED BEFORE ANY
C OTHER PROPERTY ROUTINES ARE REFERENCED AND ALSO IF
C THE MIXTURE OR THE VALUES OF THE INTERACTION
C COEFFICIENTS F0, F1 ARE CHANGED.
C
C INPUTS:
C IR1, IR2 - CODE NUMBERS FOR PURE COMPONENTS
C F0, F1 - COEFFICIENTS TO THE CURVE FIT FOR THE
C INTERACTION PARAMETER: F = F0 + F1*T
C
C OUTPUTS (VIA COMMON BLOCKS):
C A - ARRAY OF A COEFFICIENTS FOR THE TWO PURE
C COMPONENTS
C B - ARRAY OF B COEFFICIENTS FOR THE PURE COMPONENTS
C CP - ARRAY OF PURE COMPONENT CP0 COEFFICIENTS
C HR - TWO ELEMENT ARRAY OF PURE COMPONENT REFERENCE
C ENTHALPIES; THESE ARE EQUAL TO THE SATURATED
C LIQUID ENTHALPY AT THE REFERENCE TEMPERATURE MINUS
C THE PERFECT GAS ENTHALPY AT THE REFERENCE
C TEMPERATURE
C SR - REFERENCE ENTROPIES; EQUAL TO THE DIFFERENCE
C BETWEEN THE SATURATED LIQUID AND PERFECT GAS
C ENTROPIES AT THE REFERENCE TEMPERATURE
C TC - PURE COMPONENT CRITICAL TEMPERATURES
C TREF - REFERENCE TEMPERATURES AT WHICH HR AND SR ARE
C COMPUTED
C WM - PURE COMPONENT MOLECULAR WEIGHS
C
C OTHER SUBROUTINES REFERENCED:
C BUBLT - COMPUTE SATURATED LIQUID AND VAPOR
C CONDITIONS
C HCVCP5 - COMPUTE ENTHALPY AT REFERENCE STATE
C ENTROP - COMPUTE REFERENCE ENTROPY
C
C IMPLICIT REAL (A-H,O-Z)
C DIMENSION COEFF(9,20),CRIT(5,20),
C 1 WM(2),TC(2),A(3,2),B(3,2),
C 1 CP(3,2),HR(2),SR(2),VR(2),
C 1 TREF(2),HZERO(20),SZERO(20)
C INTEGER IR(2)
C CHARACTER*6 HREF(20)
COMMON /ESDATA/ COEFF, CRIT
COMMON /HREF1/ HREF
COMMON /RDATA1/ A, B, FF0, FF1
COMMON /RDATA2/ WM, TC
COMMON /CPDATA/ CP
COMMON /REF/ TREF, HR, SR, VR
COMMON /HSZERO/ HZERO, SZERO
IR(1) = IR1
IR(2) = IR2
IR1 = ABS(IR1)
IR2 = ABS(IR2)
FF0 = F0
FF1 = F1
DO 100 KR = 1, 2
IF (IR(KR).GT.0) THEN
  IF IR IS NEGATIVE READ IN COEFFICIENTS FROM DATA FILE
  ELSE IF (IR(1).EQ.ABS(IR(2)) .AND. KR.EQ.2) THEN
    HREF(IR1) = HREF(IR2)
    DO 66 KC = 1, 5
    CRIT(KC, IR2) = CRIT(KC, IR1)
    HZERO(IR2) = HZERO(IR1)
    SZERO(IR2) = SZERO(IR1)
    DO 68 KC = 1, 9
    COEFF(KC, IR2) = COEFF(KC, IR1)
  ELSE
    IRK = ABS(IR(KR))
    READ (*,1000) HREF(IRK)
    READ (*,*) (CRIT(KC, IRK), KC=1, 5),
    HZERO(IRK), SZERO(IRK),
    (COEFF(KC, IRK), KC=1, 9)
  END IF
  IR(KR) = ABS(IR(KR))
  WM(KR) = CRIT(1, IR(KR))
  TREF(KR) = CRIT(2, IR(KR))
  TC(KR) = CRIT(3, IR(KR))
  HR(KR) = 0.0
  SR(KR) = 0.0
  DO 100 KC = 1, 3
  A(KC, KR) = COEFF(KC, IR(KR))
  B(KC, KR) = COEFF(KC + 3, IR(KR))
100 CP(KC, KR) = COEFF(KC + 6, IR(KR))
  CALL ESPAR (-2, TREF(1), 0.0, AB, BB)
CALL BUBBLE POINT ROUTINE TO CALCULATE SATURATED LIQUID
AND VAPOR VOLUMES AND THEN CALL ENTHALPY AND ENTROPY
ROUTINE TO DETERMINE REFERENCE VALUES. THE HZERO AND
SZERO ALLOW AN ARBITRARY VALUE TO BE ASSIGNED TO THE
SATURATED LIQUID H OR S AT THE REFERENCE TEMPERATURE.
CALL BUBLT (TREF(1), 1.0, XV, P, VR(1), VV, .TRUE., .FALSE.)
CALL BUBLT (TREF(2), 0.0, XV, P, VR(2), VV, .TRUE., .FALSE.)
DO 160 I = 1, 2
XL=FLOAT(2-I)
CALL HVCVCP (I,TREF(I),VR(I),XL,HR(I),CV,XCP,VSND)
HR(I)=HR(I)-HZERO(IR(I))
SR(I)=ENTROP(TREF(I),VR(I),XL)-SZERO(IR(I))
160 CONTINUE
RETURN
100 FORMAT (A6)
END

BLOCK DATA BDESC

THIS ROUTINE INITIALIZES THE COMMON BLOCKS CONTAINING
INFORMATION ABOUT THE PURE COMPONENTS. IT IS NOT
REFERENCED DIRECTLY BY ANY OTHER SUBROUTINE BUT MUST BE
INCLUDED IN THE EXECUTABLE ELEMENT. DATA ARRAYS ARE
DIMENSIONED TO ACCOMODATE ADDITIONAL PURE COMPONENTS.

EXPLANATION OF CONSTANTS:
COEFF(I,J) - FOR REFRIGERANT J, COEFFICIENTS OF A,
B, CP0
CURVE FITS:
A = A0 * EXP(A1*T + A2*T*T) (KJ M**3/KMOL**2)
CP0 = C0 + C1*T + C2*T*T (KJ/KMOL K)
(STORED IN ORDER A0,A1,A2,B0,B1,B2,C0,C1,C2)
CRIT(I,J) - FOLLOWING INFORMATION FOR REFRIGERANT J:
I = 1 - MOLECULAR WEIGHT
2 - REFERENCE TEMPERATURE FOR ENTHALPY AND
ENTROPY
(K)
3 - CRITICAL TEMPERATURE (K)
4 - CRITICAL PRESSURE (KPA)
5 - CRITICAL VOLUME (M**3/KMOL)
HREF(J) - REFRIGERANT NAME (ASHRAE DESIGNATION)
HZERO(J) - VALUE OF SATURATED LIQUID ENTHALPY OF
REFRIGERANT
J AT ITS REFERENCE TEMPERATURE (KJ/KMOL)
SZERO(J) - VALUE OF SATURATED LIQUID ENTROPY AT
REFERENCE
TEMPERATURE (KJ/KMOL K)
R - GAS CONSTANT (KJ/KMOL K)
TOLR - RELATIVE CONVERGENCE TOLERANCE FOR ITERATION
LOOPS
SHOULD BE AT LEAST 10 TIMES LARGER THAN MACHINE
PRECISION
ITMAX - MAXIMUM ITERATION COUNT FOR ITERATIVE LOOPS
LUP - LOGICAL UNIT TO WHICH ANY WARNING MESSAGES ARE
WRITTEN
A, B COEFFICIENTS EVALUATED FROM ASHRAE (1981)
SATURATION
DATA UNLESS INDICATED.
IMPLICIT REAL (A-H,O-Z)
DIMENSION COEFF(9,20),CRIT(5,20),HZERO(20),SZERO(20)
CHARACTER*6 HREF(20)
COMMON /ESDATA/ COEFF,CRIT
COMMON /HREF1/ HREF
COMMON /HSZERO/ HZERO,SZERO
COMMON /RDATA4/ R
COMMON /TOL/ TOLR,ITMAX,LUP
COMMON /TOLSH/ TOLH,TOLS
DATA R /8.314/
DATA TOLR /1.0E-7/
DATA TOLH,TOLS /0.01,0.001/
DATA ITMAX,LUP /20,6/

DATA FOR R11, R12, R13, R13B1, R14, R22, R23, R113, R114, R142B, R152A FOLLOW.

R11, TRICHLOROFUOROMETHANE

DATA HREF(1) /'R11'/
DATA CRIT(1,1),CRIT(2,1),
1 CRIT(3,1),CRIT(4,1),CRIT(5,1)
1 /137.37,233.15,471.2,4467.,0.247/
DATA HZERO(1),SZERO(1) /0.0,0.0/
DATA COEFF(1,1),COEFF(2,1),COEFF(3,1),
1 COEFF(4,1),COEFF(5,1),
1 COEFF(6,1),COEFF(7,1),
1 COEFF(8,1),COEFF(9,1) /4971.54,
1 -2.24669E-3,-5.11943E-7,0.176659,-1.74531E-4,-3.49717E-8,
1 22.0418,0.260895,-2.45319E-4/

R12, DICHLORODIFLUOROMETHANE

DATA HREF(2) /'R12'/
DATA CRIT(1,2),CRIT(2,2),CRIT(3,2),
1 CRIT(4,2),CRIT(5,2)
1 /120.91,233.15,384.95,4180.,0.217/
DATA HZERO(2),SZERO(2) /0.0,0.0/
DATA COEFF(1,2),COEFF(2,2),COEFF(3,2),
1 COEFF(4,2),COEFF(5,2),
1 COEFF(6,2),COEFF(7,2),COEFF(8,2),
1 COEFF(9,2) /3524.12,
1 -2.77230E-3,-6.73180E-7,0.153755,
1 -1.84195E-4,-5.03644E-8,
1 17.5387,0.248546,-2.16271E-4/

R13, CHLOROTRIFLUOROMETHANE

DATA HREF(3) /'R13'/
DATA CRIT(1,3), CRIT(2,3), CRIT(3,3),
1 CRIT(4,3), CRIT(5,3)
1 /104.46, 233.15, 302.0, 3921.0, 0.181/
DATA HZERO(3), SZERO(3) /0.0, 0.0/
DATA COEFF(1,3), COEFF(2,3), COEFF(3,3),
1 COEFF(4,3), COEFF(5,3),
1 COEFF(6,3), COEFF(7,3), COEFF(8,3),
1 COEFF(9,3) /2298.13,
1 -3.41872E-3, -1.52430E-6, 0.128141,
1 -1.84474E-4, -1.07951E-7,
1 13.9300, 0.232181, -1.82929E-4/

C R13B1, BROMOTRIFLUOROMETHANE
C
DATA HREF(4) /'R13B1'/
DATA CRIT(1,4), CRIT(2,4), CRIT(3,4),
1 CRIT(4,4), CRIT(5,4)
1 /148.91, 233.15, 340.2, 4017.0, 0.200/
DATA HZERO(4), SZERO(4) /0.0, 0.0/
DATA COEFF(1,4), COEFF(2,4), COEFF(3,4),
1 COEFF(4,4), COEFF(5,4),
1 COEFF(6,4), COEFF(7,4), COEFF(8,4), COEFF(9,4)
1 /2728.10, -2.79791E-3, -1.50848E-6, 0.139949,
1 -1.82428E-4, -7.75898E-8,
1 19.9537, 0.216394, -1.70241E-4/

C R14, TETRAFLUOROMETHANE
C
DATA HREF(5) /'R14'/
DATA CRIT(1,5), CRIT(2,5), CRIT(3,5),
1 CRIT(4,5), CRIT(5,5)
1 /88.00, 200.00, 227.5, 3795.0, 0.141/
DATA HZERO(5), SZERO(5) /0.0, 0.0/
DATA COEFF(1,5), COEFF(2,5), COEFF(3,5),
1 COEFF(4,5), COEFF(5,5),
1 COEFF(6,5), COEFF(7,5), COEFF(8,5), COEFF(9,5)
1 /1393.60, -4.81985E-3, -1.89167E-6,
1 0.100601, -1.94974E-4, -1.35408E-7,
1 11.0629, 0.209740, -1.40992E-4/

C R22, CHLORODIFLUOROMETHANE
C
COEFFICIENTS EVALUATED FROM LITERATURE DATA
C
DATA HREF(6) /'R22'/
DATA CRIT(1,6), CRIT(2,6), CRIT(3,6),
1 CRIT(4,6), CRIT(5,6)
1 /86.47, 233.15, 369.3, 5054.0, 0.169/
DATA HZERO(6), SZERO(6) /0.0, 0.0/
DATA COEFF(1,6), COEFF(2,6), COEFF(3,6),
1 COEFF(4,6), COEFF(5,6),
1 COEFF(6,6), COEFF(7,6), COEFF(8,6), COEFF(9,6)
1 /2450.27, -2.30087E-3, -1.78103E-6,
R23, TRIFLUOROMETHANE

DATA HREF(7) /'R23'/
DATA CRIT(1,7),CRIT(2,7),CRIT(3,7),
1 CRIT(4,7),CRIT(5,7)
1 /70.01,233.15,299.1,4900.0,0.133/
DATA HZERO(7),SZERO(7) /0.0,0.0/
DATA COEFF(1,7),COEFF(2,7),COEFF(3,7),
1 COEFF(4,7),COEFF(5,7),
1 COEFF(6,7),COEFF(7,7),COEFF(8,7),
1 COEFF(9,7) /2025.93,
1 -4.68206E-3,9.95524E-7,0.103137,
1 -2.29653E-4,1.55760E-7,20.4760,0.106183,-1.21892E-5/
C

R113, 1,1,2-TRICHLOROTRIFLUOROETHANE
C COEFFICIENTS EVALUATED FROM LITERATURE DATA

DATA HREF(8) /'R113'/
DATA CRIT(1,8),CRIT(2,8),CRIT(3,8),
1 CRIT(4,8),CRIT(5,8)
1 /187.38,233.15,487.5,3456.0,0.329/
DATA HZERO(8),SZERO(8) /0.0,0.0/
DATA COEFF(1,8),COEFF(2,8),COEFF(3,8),
1 COEFF(4,8),COEFF(5,8),
1 COEFF(6,8),COEFF(7,8),COEFF(8,8),
1 COEFF(9,8) /7332.59,
1 -2.20396E-3,-7.26656E-7,0.230713,
1 -1.87956E-4,-1.06114E-7,76.2637,0.119641,7.18786E-5/
C

R114, 1,2-DICHLOROTETRAFLUOROETHANE

DATA HREF(9) /'R114'/
DATA CRIT(1,9),CRIT(2,9),CRIT(3,9),
1 CRIT(4,9),CRIT(5,9)
1 /170.92,233.15,419.03,3304.0,0.307/
DATA HZERO(9),SZERO(9) /0.0,0.0/
DATA COEFF(1,9),COEFF(2,9),COEFF(3,9),
1 COEFF(4,9),COEFF(5,9),
1 COEFF(6,9),COEFF(7,9),COEFF(8,9),
1 COEFF(9,9) /9771.35,
1 -5.85557E-3,3.99413E-6,0.306318,
1 -7.96444E-4,7.81059E-7,20.7005,0.464035,-4.17589E-4/
C

R142B, 1-CHLORO-1,1-DIFLUOROETHANE

DATA HREF(10) /'R142B'/
DATA CRIT(1,10), CRIT(2,10), CRIT(3,10), CRIT(4,10), CRIT(5,10)
  1 /100.49, 233.15, 410.3, 4120.0, 0.231/
DATA HZERO(10), SZERO(10) /0.0, 0.0/
DATA COEFF(1,10), COEFF(2,10), COEFF(3,10), COEFF(4,10), COEFF(5,10),
  1 COEFF(6,10), COEFF(7,10), COEFF(8,10), COEFF(9,10) /2990.0,
  1 -5.40563E-4, -4.12642E-6, 0.146006,
  1 -8.92503E-5, -1.80562E-7, 23.7611,
  1 0.231706, -1.06534E-4/

C
R152A, 1,1-DIFLUOROETHANE

DATA HREF(11) /'R152A'/
DATA CRIT(1,11), CRIT(2,11), CRIT(3,11), CRIT(4,11), CRIT(5,11)
  1 /66.05, 233.15, 386.7, 4492.0, 0.181/
DATA HZERO(11), SZERO(11) /0.0, 0.0/
DATA COEFF(1,11), COEFF(2,11), COEFF(3,11), COEFF(4,11), COEFF(5,11),
  1 COEFF(6,11), COEFF(7,11), COEFF(8,11), COEFF(9,11) /2254.37,
  1 -5.87778E-4, -4.37432E-6, 0.116521,
  1 -9.04883E-5, -1.14563E-7, 22.2804,
  1 0.154009, -3.06670E-6/

C
R216A, 1,3-DICHLOROHEXAFLUOROPROPANE

DATA HREF(12) /'R216A'/
DATA CRIT(1,12), CRIT(2,12), CRIT(3,12), CRIT(4,12), CRIT(5,12)
  1 /20.93, 233.15, 453.14, 2754.1, 0.3847/
DATA HZERO(12), SZERO(12) /0.0, 0.0/
DATA COEFF(1,12), COEFF(2,12), COEFF(3,12), COEFF(4,12), COEFF(5,12),
  1 COEFF(6,12), COEFF(7,12), COEFF(8,12), COEFF(9,12) /8431.44,
  1 -2.45916E-3, -9.91754E-7, 0.265720,
  1 -2.20418E-4, -1.68111E-7, 8.79769,
  1 0.654246, -5.39923E-4/

C
R216B, 1,2-DICHLOROHEXAFLUOROPROPANE

DATA HREF(13) /'R216B'/
DATA CRIT(1,13), CRIT(2,13), CRIT(3,13), CRIT(4,13), CRIT(5,13)
  1 /220.93, 233.15, 453.14, 2754.1, 0.3847/
DATA HZERO(13), SZERO(13) /0.0, 0.0/
DATA COEFF(1,13), COEFF(2,13), COEFF(3,13), COEFF(4,13), COEFF(5,13),
  1 COEFF(6,13), COEFF(7,13), COEFF(8,13),
R143A, 1,1,1-TRIFLUOROETHANE
COEFFICIENTS FIT TO LITERATURE DATA

DATA HREF(14) /'R143A'/
DATA CRIT(1,14),CRIT(2,14),CRIT(3,14),
   CRIT(4,14),CRIT(5,14)
   /84.04,233.15,346.3,3758.0,0.194/
DATA HZERO(14),SZERO(14) /0.0,0.0/
DATA COEFF(1,14),COEFF(2,14),COEFF(3,14),
   COEFF(4,14),COEFF(5,14),
   COEFF(6,14),COEFF(7,14),COEFF(8,14),
   COEFF(9,14) /3092.03,
   -3.42907E-3,1.49460E-7,0.137902,
   -1.96301E-4,1.78383E-8,
   14.0656,0.254765,-1.29865E-4/

END
SUBROUTINE BCBLP (P,XL,XV,T,VL,VV,LBUB,LCRIT)

C DEVELOPED BY MARK MCLINDEN AND GRAHAM MORRISON AT THE
C NATIONAL BUREAU OF STANDARDS UNDER FUNDING FROM THE
C ELECTRIC POWER RESEARCH INSTITUTE AND NBS.
C
C GIVEN PRESSURE AND COMPOSITION OF ONE PHASE THIS
C ROUTINE CALCULATES THE SATURATION TEMPERATURE, THE
C COMPOSITION OF THE OTHER PHASE AND THE LIQUID AND VAPOR
C MOLAR VOLUMES.
C
INPUTS:
C   P - SATURATION PRESSURE (KPA)
C   ONLY ONE OF: XL - LIQUID COMPOSITION (MOLE
C                  FRACTION)
C   OR: XV - VAPOR COMPOSITION (MOLE FRACTION)
C   LBUB - LOGICAL VARIABLE
C         IF LBUB=TRUE. LIQUID COMPOSITION IS GIVEN
C         (COMPUTE BUBBLE POINT)
C         IF LBUB=FALSE. VAPOR COMPOSITION IS GIVEN
C         (COMPUTE DEW POINT)
C
OUTPUTS:
C   XL OR XV - COMPOSITION OF CALCULATED PHASE
C   T - SATURATION TEMPERATURE (K)
C   VL - LIQUID VOLUME (M**3/KMOL)
C   VV - VAPOR VOLUME (M**3/KMOL)
C   LCRIT - ERROR FLAG; SET TO .TRUE. WHEN THE INPUT
C            PRESSURE EXCEEDS THE CRITICAL PRESSURE.
C
OTHER SUBROUTINES REFERENCED:
C   VIT - ITERATION FOR MOLAR VOLUME
C   ESPAR - COMPUTATION OF EQUATION OF STATE PARAMETERS
C
GENERAL NOMENCLATURE FOR FIRST LETTER OF VARIABLE NAMES
C   A,B - EQUATION OF STATE PARAMETERS
C   F - MIXING PARAMETER
C   T - TEMPERATURE
C   P - PRESSURE
C   V - MOLAR VOLUME
C   X - COMPOSITION
C   G - GIBBS FREE ENERGY
C   U - CHEMICAL POTENTIAL
C   Y - COMBINATION OF VARIABLES USED IN EQUATION OF
C        STATE
C   TOL - CONVERGENCE TOLERANCE FOR ITERATION LOOPS
C   I,J - INDEX VARIABLES FOR ITERATION AND DO LOOPS
C   L - LOGICAL VARIABLES SUCH AS NON-CONVERGENCE FLAGS
C
GENERAL NOMENCLATURE FOR SECOND OR THIRD LETTER OF
C VARIABLES
A, B - COMPONENTS OF MIXTURE; COMPOSITION IS MOLE FRACTION A
L - LIQUID PHASE
V - VAPOR PHASE
1 - PARENT PHASE (PHASE WITH SPECIFIED COMPOSITION)
2 - INCIPIENT PHASE
(FOR EXAMPLE UA1 REFERS TO CHEMICAL POTENTIAL OF COMPONENT A IN PHASE 1)

IMPLICIT REAL (A-H,O-Z)
LOGICAL LBUB, LV1CON, LV2CON, LXCON, LXPOS, LXNEG,
1 LTPOS, LTNEG, LPPCON, LCRIT
COMMON /ESPAR1/ AA, AB, BA, BB, F, C1, D1, C2, D2
COMMON /RDATA4/ R
COMMON /TOL/ TOLR, ITMAX, LUP
DIMENSION TT(3), FT(2), XX2(3), FX2(2)
SAVE TNEG, TPOS
DATA TC /340./
DATA TNEG, TPOS /9999., -999./

STATEMENT FUNCTIONS FOR GIBBS FREE ENERGY AND CHEMICAL POTENTIAL NOTE THAT SINCE ONLY DIFFERENCES OF G AND U ARE USED IN THE PROGRAM ANY TERMS WHICH WOULD CANCEL ARE OMITTED. BOTH G AND U ARE DIVIDED BY R*T TO OBTAIN DIMENSIONLESS QUANTITIES.

G(T,V,A,B)=-LOG(V)+0.25*
1 B*((8.0*V-2.25*B)*V+0.1875*B*B)/(V-O.25*B)**2/(V-O.25*B)+
1 (A/B*LOG(V/(V+B))-A/(V+B))/(R*T)
U(T,X,V,A,B,AA,AB,BI,F,Y)=(Y*
1 (4.0-3.0*Y)+BI*(4.0-2.0*Y)*Y/
1 (1.0-Y)**2+(BI*A*)
1 LOG(1.0+B/V)/B-BI*A/(V+B)
1 +2.0*(XX*AA+(1.0-F)*(1.0-X)*
1 SQRT(XX*AB))*LOG(V/(V+B)))/
1 /(R*T*B)-LOG(V)

COMPUTE PURE COMPONENT E.S. COEFFICIENTS, THE MIXING PARAMETER, C AND THE E.O.S. COEFFICIENTS FOR PHASE 1

IF (LBUB) THEN
  XL=XL
  XV=XL
ELSE
  XL=XV
END IF

FIND CRITICAL POINT OF THE PURE (OR PSEUDO-PURE) MATERIAL AND BASE INITIAL GUESS FOR TEMPERATURE ON A
SIMPLE, EMPIRICAL RELATIONSHIP BETWEEN REDUCED PRESSURE AND REDUCED TEMPERATURE THAT IS REASONABLY ACCURATE FOR REFRIGERANTS.

LCRIT=.FALSE.
CALL CRITX (X1,TC,PC,VC)
TT(1)=TC/(1.0-0.153*LOG(P/PC))
CALL ESPAR (0,TT(1),X1,A1,B1)
VL=0.8*B1
V2=R*TT(1)/P

ENTER ITERATION FOR PURE COMPONENT. THIS ITERATION VARIES TEMPERATURE UNTIL THE GIBBS FREE ENERGY OF BOTH PHASES ARE EQUAL.
A COMBINATION OF SECANT AND REGULI-FALSI METHODS IS USED FOR THE ITERATION.

IF (X1*XBL .LT. TOLR) THEN
  IF (P.GT.PC) THEN
    LCRIT=.TRUE.
    WRITE (LUP,1010)
    RETURN
  END IF
  J=1
  LTPOS=.FALSE.
  LTNEG=.FALSE.
  LPPCON=.FALSE.
  DO 400 IT=1,ITMAX
  CALL ESPAR (0,TT(J),X1,A1,B1)
  LVICON=.FALSE.
  LV2CON= .FALSE.
  CALL VI (TT(J),P,A1,B1,VL,.TRUE.,LVICON)
  CALL VI (TT(J),P,A1,B1,VV,.FALSE.,LV2CON)
  GL=G(TT(J),VL,A1,B1)
  GV=G(TT(J),VV,A1,B1)
  FT(J)=GL-GV
  WRITE (*,*) IT,TT(J),GL,GV,FT(J)
  IF (ABS(FT(J)).LT.100.*TOLR) GOTO 440
  IF (FT(J).LT.0.0) THEN
    LTNEG=.TRUE.
    FTNEG=FT(J)
    TNEG=TT(J)
  ELSE
    LTPOS=.TRUE.
    FTPOS=FT(J)
    TPOS=TT(J)
  END IF
  IF (IT.LE.1) THEN
    C=EXP(GL-GV)
    P2=P*C
    TT(2)=1.0/(1.0/TT(1)+0.0004*(LOG(P2)-LOG(P)))
    J=2
  ELSE
    C=EXP(GL-GV)
    P2=P*C
    TT(2)=1.0/(1.0/TT(1)+0.0004*(LOG(P2)-LOG(P)))
    J=2
  END IF

WRITE (LUP,1010)
ELSE
   DGDT=(FT(2)-FT(1))/(TT(2)-TT(1))
   IF (DGDT.EQ.0.0) GOTO 440
C. NEXT GUESS FOR TEMPERATURE GIVEN BY SECANT METHOD
   TT(3)=TT(2)-FT(2)/DGDT
C. IF NEXT GUESS FOR TEMPERATURE IS FURTHER FROM SOLUTION
C. THAN PREVIOUS BEST GUESS, USE REGULI-FALSI METHOD FOR
C. NEXT GUESS
   IF (LTNEG.AND.LTPOS) THEN
      IF (TT(3).LT.MIN(TNEG,TPOS) .OR.
           TT(3).GT.MAX(TNEG,TPOS))
         TT(3)=TPOS-FTPOS*(TPOS-TNEG)/(FTPOS-FTNEG)
   END IF
   TT(1)=TT(2)
   FT(1)=FT(2)
   TT(2)=TT(3)
END IF
400 CONTINUE
C. IF ITERATION HAS NOT CONVERGED, SET ERROR FLAG.
   LPPCON=.TRUE.
C. END OF PURE COMPONENT ITERATION
C. 440 IF (LV1CON) WRITE (LUP,1050)
   IF (LV2CON) WRITE (LUP,1055)
   IF (LPPCON) WRITE (LUP,1020)
   T=TT(J)
   RETURN
END IF
C. ENTER ITERATION FOR MIXTURE
C. THE MIXTURE ITERATION CONSISTS OF TWO CONCENTRIC
C. ITERATION LOOPS WHICH VARY THE TEMPERATURE OF THE
C. MIXTURE AND THE COMPOSITION OF THE COMPUTED PHASE TO
C. GIVE EQUAL CHEMICAL POTENTIALS FOR EACH OF THE
C. COMPONENTS BETWEEN THE TWO PHASES.
C. THE INITIAL GUESS FOR THE TEMPERATURE
C. IS GIVEN BY THE CALCULATION
C. ABOVE; THE INITIAL GUESS FOR COMPOSITION
C. IS THAT X2=X1.
C.
   J=1
   X2CONV=X1
   IF (LBUB) THEN
      V1=VL
      V2=VV
   ELSE
      V1=VV
      V2=VL
   END IF
   LTNEG=.FALSE.
LTPOS=.FALSE.
DO 800 ITT=1,ITMAX
XX2(1)=X2CONV
LXCON=.FALSE.
LV1CON=.FALSE.
CALL ESPAR (0,TT(J),X1,A1,B1)
CALL VIT (TT(J),P,A1,B1,V1,LBUB,LV1CON)
C IF VOLUME ITERATION HAS NOT CONVERGED, TRY A NEW
C TEMPERATURE AND RETURN TO THE BEGINNING OF THE
C ITERATION
C IF (LV1CON .OR. LXCON) THEN
   TT(J)=0.95*TT(J)
   GOTO 800
END IF
C COMPUTE CHEMICAL POTENTIALS FOR PHASE 1
Y=0.25*B1/V1
UA1=U(TT(J),X1,V1,A1,B1,AA,AB,BA,F,Y)
UB1=U(TT(J),XB1,V1,A1,B1,AB,AA,BB,F,Y)
C ENTER INNER ITERATION LOOP (FOR COMPOSITION OF PHASE 2)
C JJ=1
LXNEG=.FALSE.
LXPOS=.FALSE.
DO 600 IT=1,ITMAX
LV2CON=.FALSE.
XB2=1.0-XX2(JJ)
C COMPUTE EQUATION OF STATE COEFFICIENTS FOR PHASE 2
CALL ESPAR (0,TT(J),XX2(JJ),A2,B2)
CALL VIT (TT(J),P,A2,B2,V2,.NOT.LBUB,LV2CON)
C IF VOLUME ITERATION HAS NOT CONVERGED, TRY A NEW
C TEMPERATURE AND RETURN TO THE START OF THE TEMPERATURE
C ITERATION.
C IF (LV2CON) THEN
   TT(J)=0.95*TT(J)
   GOTO 800
END IF
C COMPUTE CHEMICAL POTENTIALS OF PHASE 2
Y=0.25*B2/V2
UA2=U(TT(J),XX2(JJ),V2,A2,B2,AA,AB,BA,F,Y)
UB2=U(TT(J),XB2,V2,A2,B2,AB,AA,BB,F,Y)
C CALCULATE THE COMPOSITION OF PHASE 2 FROM THE
C COMPOSITION OF PHASE 1 AND THE CHEMICAL POTENTIALS.
C THE INNER ITERATION LOOP HAS CONVERGED WHEN THE
C CALCULATED COMPOSITION EQUALS
C (WITHIN A CONVERGENCE TOLERANCE) THE GUESSED VALUE OF
C X2.
c
za=x1*exp(ua1-ua2)
z0=xb1*exp(ub1-ub2)
c=za+z0
x2calc=za/c
fx2(jj)=x2calc-xx2(jj)
if (abs(fx2(jj)).lt.tolr) then
    x2conv=xx2(jj)
goto 640
end if
c update positive or negative bounds for use with
c reguli-falsi
if (fx2(jj).lt.0.0) then
    lxneg=.true.
    fxneg=fx2(jj)
xneg=xx2(jj)
else
    lxpos=.true.
    fxpos=fx2(jj)
xpos=xx2(jj)
end if
c update the guess for x2. the composition computed
c above is used for the second guess. a combination of
c secant and reguli-falsi methods is used for third and
c subsequent guesses.
if (jj.le.1) then
    jj=2
    xx2(2)=x2calc
else
    if (fx2(1).eq.fx2(2)) then
        x2conv=xx2(jj)
goto 640
    end if
    xx2(3)=xx2(2)-fx2(2)*(xx2(2)-xx2(1))/(fx2(2)-fx2(1))
    if (lxpos .and. lxneg) then
        if (xx2(3).lt.min(xpos,xneg) .or.
            1 xx2(3).gt.max(xpos,xneg)) then
            xx2(3)=xpos-fxpos*(xpos-xneg)/(fxpos-fxneg)
        end if
    end if
    xx2(1)=xx2(2)
    xx2(2)=xx2(3)
    fx2(1)=fx2(2)
end if
xx2(jj)=min(1.0,max(0.0,xx2(jj)))
600 continue
c if inner iteration loop has not converged, set error
flag
lxcon=.true.
END OF ITERATION LOOP FOR PHASE 2 COMPOSITION

640  \[ FT(J) = 1.0 - C \]

WRITE (*,*) ITT,J,TT(J),FT(J)

OUTER (TEMPERATURE) ITERATION HAS CONVERGED WHEN \( C = 1.000 \) (I.E. WHEN THE CHEMICAL POTENTIALS OF EACH COMPONENT ARE THE SAME IN BOTH PHASES).

IF (ABS(FT(J)).LT.100.*TOLR) GOTO 840

PROVIDED THAT THE X2 ITERATION HAS CONVERGED FOR THE CURRENT GUESS OF TEMPERATURE, UPDATE THE POSITIVE AND NEGATIVE BOUNDS FOR USE WITH THE REGULI-FALSI METHOD.

IF (.NOT.LXCON) THEN
  IF (FT(J).LT.0.0) THEN
    LTNEG=.TRUE.
    FTNEG=FT(J)
    TNEG=TT(J)
  ELSE
    LTPOS=.TRUE.
    FTPOS=FT(J)
    TPOS=TT(J)
  END IF
END IF

COMPUTE NEW GUESS FOR SATURATION TEMPERATURE.

FOR THE SECOND ITERATION, COMPUTE AN APPROXIMATE SATURATION PRESSURE CORRESPONDING TO THE CURRENT GUESS OF TEMPERATURE AND ADJUST THE GUESS FOR TEMPERATURE ACCORDING TO THE DEVIATION BETWEEN THE CALCULATED AND INPUT PRESSURES.

IF (ITT.LE.2) THEN
  IF (LBUB) THEN
    \[ P2 = P\times C \]
  ELSE
    \[ P2 = P/C \]
  END IF
  \[ TT(3) = 1.0/(1.0/TT(J) + 0.0004\times (\log(P2) - \log(P))) \]
  IF (J.LE.1) THEN
    J=2
  ELSE
    \[ TT(1) = TT(2) \]
    \[ FT(1) = FT(2) \]
  END IF
  \[ TT(2) = TT(3) \]
ELSE

FOR THIRD AND SUBSEQUENT ITERATIONS, USE SECANT/REGULI-FALSI

\[ TT(3) = TT(2) - FT(2)\times (TT(2) - TT(1))/(FT(2) - FT(1)) \]
IF ((TT(3).GT.MAX(TNEG,TPOS) .OR. 
1    TT(3).LT.MIN(TNEG,TPOS)) .AND. LTNEG .AND. LTPOS) 
1    TT(3)=TPOS-FTPOS*(TPOS-TNEG)/(FTPOS-FTNEG) 
    TT(1)=TT(2) 
    TT(2)=TT(3) 
    FT(1)=FT(2)
END IF
C
800 CONTINUE
  WRITE (LUP,1040)
840 T=TT(J)
C
ASSIGN RESULTS FOR PHASES 1 AND 2 TO LIQUID AND VAPOR
C PHASES DEPENDING ON WHETHER THE DEW OR BUBBLE POINT WAS
C CALCULATED.
C
IF (LBUB) THEN
  XV=XX2(JJ)
  VL=V1
  VV=V2
ELSE
  XL=XX2(JJ)
  VL=V2
  VV=V1
END IF
C
PRINT WARNING MESSAGES FOR ANY CASES OF NON-CONVERGENCE
C OCCURRING ON FINAL CALL TO EACH ITERATION AND RETURN.
C
IF (ABS(1.0-VL/VV).LT.TOLR) THEN
  LCRIT=.TRUE.
  WRITE (LUP,1010)
END IF
IF (LV1CON) WRITE (LUP,1050)
IF (LV2CON) WRITE (LUP,1055)
IF (LXCON) WRITE (LUP,1060)
RETURN
1010 FORMAT (IX,'CRITICAL POINT EXCEEDED IN BUBLP')
1020 FORMAT (IX,'PURE MATERIAL TEMPERATURE ITERATION IN
1    BUBLP', ' DID NOT CONVERGE')
1040 FORMAT (IX,'MIXTURE TEMPERATURE ITERATION IN BUBLP
1    DID NOT', ' CONVERGE')
1050 FORMAT (IX,'ITERATION IN BUBLP FOR PARENT PHASE
1    VOLUME DID', ' NOT CONVERGE')
1055 FORMAT (IX,'ITERATION IN BUBLP FOR INCIPIENT PHASE
1    VOLUME DID', ' NOT CONVERGE')
1060 FORMAT (IX,'COMPOSITION ITERATION IN BUBLP DID NOT
1    CONVERGE')
END
SUBROUTINE BUBLT (T, XL, XV, P, VL, VV, LBUB, LCRIT)

DEVELOPED BY MARK McLINDEN AND GRAHAM MORRISON AT THE
NATIONAL BUREAU OF STANDARDS UNDER FUNDING FROM THE
ELECTRIC POWER RESEARCH INSTITUTE AND NBS.

GIVEN TEMPERATURE AND COMPOSITION OF ONE PHASE THIS
ROUTINE CALCULATES THE SATURATION PRESSURE, THE
COMPOSITION OF THE OTHER PHASE AND THE LIQUID AND VAPOR
MOLAR VOLUMES.

INPUTS:
T - TEMPERATURE (K)
ONLY ONE OF: XL - LIQUID COMPOSITION (MOLE
FRACTION)
OR: XV - VAPOR COMPOSITION (MOLE FRACTION)
LBUB - LOGICAL VARIABLE
IF LBUB=.TRUE. LIQUID COMPOSITION IS GIVEN
(COMPUTE BUBBLE POINT)
IF LBUB=.FALSE. VAPOR COMPOSITION IS GIVEN
(COMPUTE DEW POINT)

OUTPUTS:
XL OR XV - COMPOSITION OF CALCULATED PHASE
P - SATURATION PRESSURE (KPA)
VL - LIQUID MOLAR VOLUME (M**3/KMOL)
VV - VAPOR MOLAR VOLUME (M**3/KMOL)
LCRIT - ERROR FLAG; IF LCRIT=.TRUE. THE INPUT
TEMPERATURE
EXCEEDS THE CRITICAL TEMPERATURE OF THE PURE
COMPONENT
OR THE PSEUDO-PURE COMPONENT CORRESPONDING TO THE
MIXTURE COMPOSITION AND NO CALCULATIONS ARE DONE.

OTHER SUBROUTINES REFERENCED:
VIT - ITERATION FOR MOLAR VOLUME
PLIMIT - DETERMINES INITIAL BOUNDS ON PRESSURE AND
VOLUME
ESPAR - COMPUTATION OF EQUATION OF STATE PARAMETERS

GENERAL NOMENCLATURE FOR FIRST LETTER OF VARIABLE NAMES
A,B - EQUATION OF STATE PARAMETERS
F - MIXING PARAMETER
T - TEMPERATURE
P - PRESSURE
V - MOLAR VOLUME
X - COMPOSITION
G - GIBBS FREE ENERGY
U - CHEMICAL POTENTIAL
Y - COMBINATION OF VARIABLES USED IN EQUATION OF
STATE
TOL - CONVERGENCE TOLERANCE FOR ITERATION LOOPS
I,J - INDEX VARIABLES FOR ITERATION AND DO LOOPS
L - LOGICAL VARIABLES SUCH AS NON-CONVERGENCE FLAGS

GENERAL NOMENCLATURE FOR SECOND OR THIRD LETTER OF VARIABLES
A,B - COMPONENTS OF MIXTURE; COMPOSITION IS MOLE FRACTION A
L - LIQUID PHASE
V - VAPOR PHASE
1 - PARENT PHASE (PHASE WITH SPECIFIED COMPOSITION)
2 - INCIPIENT PHASE
(FOR EXAMPLE UA1 REFERS TO CHEMICAL POTENTIAL OF COMPONENT A IN PHASE 1)

IMPLICIT REAL (A-H,O-Z)
LOGICAL LBUB,LCRIT,LV1CON,LV2CON,LXCON,LXPOS,LXNEG,
LPPOS,LPNEG,LPPCON
COMMON /ESPAR1/ AA,AB,BA,BB,F,C1,D1,C2,D2
COMMON /RDATA4/ R
COMMON /TOL/ TOLR,ITMAX,LUP
DIMENSION PP(3),FP(2),XX2(3),FX2(2),PL(3)

STATEMENT FUNCTIONS FOR GIBBS FREE ENERGY AND CHEMICAL POTENTIAL
NOTE THAT SINCE ONLY DIFFERENCES OF G AND U ARE USED IN THE PROGRAM ANY TERMS WHICH WOULD CANCEL ARE OMITTED. THE EXPRESSION FOR U IS DIVIDED BY RT TO OBTAIN A DIMENSIONLESS QUANTITY.

G(T,V,A,B)=R*T*(-LOG(V)+0.25*B*((8.0*V-2.25*B)/(V-O.25*B)**2/V))+A/B*LOG(V/(V+B))-A/(V+B)
\*2*(X*AA+(1.0-F)*(1.0-X)*SQRT(AA*AB))
\*LOG(V/(V+B))/(R*T*B)-LOG(V)

LCRIT=.FALSE.

COMPUTE PURE COMPONENT E.S. COEFFICIENTS, THE MIXING PARAMETER, AND THE E.S. COEFFICIENTS FOR PHASE 1

IF (LBUB) THEN
X1=XL
XV=XL
ELSE
X1=XV
XL=XV
END IF
XB1 = 1.0 - X1
CALL ESPAR (O, T, X1, A1, B1)

C DETERMINE IF INPUT TEMPERATURE EXCEEDS CRITICAL POINT;
C IF SO, SET ERROR FLAG AND RETURN
C
TC = A1 / (B1 * 4.398909 * R)
IF (T > 0.99 * TC) THEN
  LCRIT = .TRUE.
  WRITE (LUP, 1010)
  RETURN
END IF

C ENTER ITERATION FOR PSEUDO-PURE COMPONENT. THIS
C ITERATION YIELDS THE FINAL RESULT FOR A PURE COMPONENT
C AND PROVIDES A STARTING GUESS FOR THE PRESSURE OF A
C MIXTURE CALL SUBROUTINE TO DETERMINE THE UPPER AND
C LOWER BOUNDS ON PRESSURE FOR WHICH THERE ARE BOTH
C LIQUID AND VAPOR SOLUTIONS OF THE EQUATION OF STATE
C
CALL PLIMIT (T, A1, B1, VLOW, VUP, PLOW, PUP)

C SET INITIAL GUESSES FOR PRESSURE NEAR THE UPPER AND
C LOWER BOUNDS. IF THE LOWER BOUND FOR PRESSURE IS
C NEGATIVE RESET IT TO A SMALL POSITIVE VALUE.
C
IF (PLOW .LE. 0.0) THEN
  VLOW = 0.8 * B1
  PC = 0.1049995 * R * TC / B1
  PLOW = 1.0E-12 * PC
  PP(2) = PLOW
ELSE
  PP(2) = PLOW + 0.0001 * (PUP - PLOW)
END IF
PP(1) = PUP - 0.0001 * (PUP - PLOW)
PL(2) = LOG(PP(2))
PL(1) = LOG(PP(1))
VL = 0.9 * VLOW
VV = 1.1 * VUP
J = 1
LPPOS = .FALSE.
LPNEG = .FALSE.
LPPCON = .FALSE.

C STARTING WITH INITIAL VALUES OF PRESSURE CLOSE TO THE
C UPPER AND LOWER BOUNDS (FOUND BY SUBROUTINE PLIMIT)
C ITERATE ON LOG (P) UNTIL THE GIBBS FREE ENERGY OF BOTH
C PHASES ARE EQUAL. A COMBINATION OF SECANT AND
C REGULI-FALSI METHODS IS USED FOR THE ITERATION.
C
DO 400 IT = 1, ITMAX
  LV1CON = .FALSE.
LV2CON=.FALSE.
CALL VIT(T,PP(J),A1,B1,VL,.TRUE.,LV1CON)
CALL VIT(T,PP(J),A1,B1,VV,.FALSE.,LV2CON)
GL=G(T,VL,A1,B1)
GV=G(T,VV,A1,B1)
FP(J)=GL-GV
IF (FP(J).LT.0.0) THEN
  LPNEG=.TRUE.
  FPNEG=FP(J)
  PNEG=PL(J)
ELSE
  LPPOS=.TRUE.
  FPPOS=FP(J)
  PPOS=PL(J)
END IF
IF (IT.LE.1) THEN
  J=2
ELSE
  DGDPL=(FP(2)-FP(1))/(PL(2)-PL(1))
  IF (DGDPL.EQ.0.0) GOTO 440
  IF (ABS(FP(J)/(PL(J)*DGDPL)).LT.TOLR) GOTO 440
  C NEXT GUESS FOR LOG (P) GIVEN BY SECANT METHOD
  PL(3)=PL(2)-FP(2)/DGDPL
  C IF NEXT GUESS FOR LOG (P) IS FURTHER FROM SOLUTION THAN
  C PREVIOUS BEST GUESS, USE REGULI-FALSI METHOD FOR NEXT
  C GUESS
  IF ((PL(3).GT.MAX(PNEG,PPOS) .OR.
    OR. PL(3).LT.MIN(PNEG,PPOS)) .AND. LPNEG .AND. LPPOS) THEN
    PL(3)=PPOS-FPPOS*(PPOS-PNEG)/(FPPOS-FPNEG)
    PL(1)=PL(2)
    PL(2)=PL(3)
    FP(1)=FP(2)
    PP(2)=EXP(PL(2))
  END IF
  400 CONTINUE
C IF ITERATION HAS NOT CONVERGED, SET ERROR FLAG.
LPPCON=.TRUE.
C END OF PSEUDO-PURE COMPONENT ITERATION
C FOR A PURE COMPONENT THE ABOVE ITERATION GIVES THE
C FINAL RESULT
C 440 IF (X1*XBl.LE.TOLR) THEN
    IF (LV1CON) WRITE (LUP,1050)
    IF (LV2CON) WRITE (LUP,1055)
    IF (LPPCON) WRITE (LUP,1020)
    P=PP(J)
    RETURN
END IF
C C ENTER ITERATION FOR MIXTURE
THE MIXTURE ITERATION CONSISTS OF TWO CONCENTRIC ITERATION LOOPS WHICH VARY THE SATURATION PRESSURE OF THE MIXTURE AND THE COMPOSITION OF THE COMPUTED PHASE TO GIVE EQUAL CHEMICAL POTENTIALS FOR EACH OF THE COMPONENTS BETWEEN THE TWO PHASES. THE INITIAL GUESS FOR THE PRESSURE IS GIVEN BY THE PSEUDO-PURE ITERATION ABOVE; THE INITIAL GUESS FOR COMPOSITION IS THAT \( x_2 = x_1 \).

ASSIGN INITIAL VALUES OF LIQUID AND VAPOR VOLUMES FROM ABOVE ITERATION TO PHASE 1 AND 2 VOLUMES.

IF (LBUB) THEN
  \( V_1 = V_L \)
  \( V_2 = V_V \)
ELSE
  \( V_1 = V_V \)
  \( V_2 = V_L \)
END IF

\( PP(1) = PP(J) \)

BEGIN ITERATION FOR SATURATION PRESSURE OF MIXTURE

\( J = 1 \)
\( x_{2\text{CONV}} = x_1 \)
\( LPNEG = .FALSE. \)
\( LPPOS = .FALSE. \)

DO 800 ITP = 1, ITMAX
  \( X_{2(1)} = X_{2\text{CONV}} \)
  \( LXCON = .FALSE. \)
  \( LV1CON = .FALSE. \)
  CALL VIT \((T, PP(J), A1, B1, V1, LBUB, LV1CON)\)
END DO

IF VOLUME ITERATION HAS NOT CONVERGED, TRY A NEW PRESSURE AND RETURN TO THE BEGINNING OF THE ITERATION

IF (LV1CON .OR. LXCON) THEN
  \( PP(2) = 0.5 \ast (PP(1) + PP(2)) \)
GOTO 800
END IF

COMPUTE CHEMICAL POTENTIALS FOR PHASE 1

\( Y = 0.25 \ast B_1/V_1 \)
\( UA1 = U(T, X1, V1, A1, B1, AA, AB, BA, F, Y) \)
\( UB1 = U(T, XB1, V1, A1, B1, AB, AA, BB, F, Y) \)

ENTER INNER ITERATION LOOP (FOR COMPOSITION OF PHASE 2)

\( JJ = 1 \)
\( LXNEG = .FALSE. \)
\( LXPOS = .FALSE. \)

DO 600 IT = 1, ITMAX
  \( LV2CON = .FALSE. \)
END DO
XB2=1.0-XX2(JJ)
C COMPUTE EQUATION OF STATE COEFFICIENTS FOR PHASE 2
CALL ESPAR (O,T,XX2(JJ),A2,B2)
CALL VIr (T,PP(J),A2,B2,V2,.NOT.LBUB,LV2CON)
C
C IF VOLUME ITERATION HAS NOT CONVERGED, TRY A NEW
C PRESSURE AND RETURN TO THE START OF THE PRESSURE
C ITERATION.
C
IF (LV2CON) THEN
PP(2)=0.5*(PP(1)+PP(2))
GOTO 800
END IF
C COMPUTE CHEMICAL POTENTIALS OF PHASE 2
Y=0.25*B2/V2
UA2=U(T,XX2(JJ),V2,A2,B2,AA,AB,BA,F,Y)
UB2=U(T,XX2(JJ),A2,B2,AB,AA,BB,F,Y)
C
C CALCULATE THE COMPOSITION OF PHASE 2 FROM THE
C COMPOSITION OF PHASE 1 AND THE CHEMICAL POTENTIALS.
C THE INNER ITERATION LOOP HAS CONVERGED WHEN THE
C CALCULATED COMPOSITION EQUALS (WITHIN A CONVERGENCE
C TOLERANCE) THE GUESSED VALUE OF X2.
C
ZA=X1*EXP(UA1-UA2)
ZB=XB1*EXP(UB1-UB2)
C=ZA+ZB
X2CALC=ZA/C
FX2(JJ)=-X2CALC+XX2(JJ)
IF (ABS(FX2(JJ)).LT.TOLR) THEN
X2CONV=XX2(JJ)
GOTO 640
END IF
C UPDATE POSITIVE OR NEGATIVE BOUNDS FOR USE WITH
C REGULI-FALSI
IF (FX2(JJ).LT.0.0) THEN
LXNEG=.TRUE.
FXNEG=FX2(JJ)
XNEG=XX2(JJ)
ELSE
LXPOS=.TRUE.
FXPOS=FX2(JJ)
XPOS=XX2(JJ)
END IF
C
C UPDATE THE GUESS FOR X2. THE COMPOSITION COMPUTED
C ABOVE IS USED FOR THE SECOND GUESS. A COMBINATION OF
C SECANT AND REGULI-FALSI METHODS IS USED FOR THIRD AND
C SUBSEQUENT GUESSES.
C
IF (IT.LE.1) THEN
J=2
XX2(2) = X2CALC
ELSE
  IF (FX2(1).EQ.FX2(2)) THEN
    X2CONV = XX2(JJ)
    GOTO 640
  END IF

  XX2(3) = XX2(2) - FX2(2) * (XX2(2) - FX2(1))
  IF (LXPOS .AND. LXNEG) THEN
    IF (XX2(3).LT.MIN(XPOS,XNEG) .OR.
        XX2(3).GT.MAX(XPOS,XNEG)) THEN
      XX2(3) = XPOS - FXPOS * (XPOS - XNEG)/(FXPOS - FXNEG)
    END IF
  END IF

  XX2(1) = XX2(2)
  XX2(2) = XX2(3)
  FX2(1) = FX2(2)
END IF

XX2(JJ) = MIN(1.0, MAX(0.0, XX2(JJ)))
600 CONTINUE
C IF INNER ITERATION LOOP HAS NOT CONVERGED, SET ERROR
C FLAG
LXCON = .TRUE.
C END OF ITERATION LOOP FOR PHASE 2 COMPOSITION
C
640 FP(J) = 1.0 - C
C OUTER (PRESSURE) ITERATION HAS CONVERGED WHEN C = 1.000
C (I.E. WHEN THE CHEMICAL POTENTIALS OF EACH COMPONENT
C ARE THE SAME IN BOTH PHASES).
C
  IF (ABS(FP(1)).LT.100.*TOLR) GOTO 840
C PROVIDED THAT THE X2 ITERATION HAS CONVERGED FOR THE
C CURRENT GUESS OF PRESSURE, UPDATE THE POSITIVE AND
C NEGATIVE BOUNDS FOR USE WITH THE REGULI-FALSI METHOD.
C
  IF (.NOT.LXCON) THEN
    IF (FP(J).LT.0.0) THEN
      LPNEG = .TRUE.
      FPNEG = FP(J)
      PNEG = PP(J)
    ELSE
      LPPOS = .TRUE.
      FPPOS = FP(J)
      PPOS = PP(J)
    END IF
  END IF
END IF
C
C COMPUTE NEW GUESS FOR SATURATION PRESSURE.
IF (ITP.LE.2 .OR. FP(1).EQ.FP(2)) THEN
  PP(1)=PP(J)
  FP(1)=FP(J)
  IF (LBUB) THEN
    PP(2)=PP(J)*C
  ELSE
    PP(2)=PP(J)/C
  END IF
  J=2
ELSE
  PP(3)=PP(2)-FP(2)*(PP(2)-PP(1))/(FP(2)-FP(1))
  IF ((PP(3).GT.MAX(PNEG,PPOS) .OR.
      PP(3).LT.MIN(PNEG,PPOS)) .AND. LPNEG .AND. LPPOS) THEN
    PP(3)=PPOS-FPPOS*(PPOS-PNEG)/(FPPOS-FPNEG)
  END IF
END IF
800 CONTINUE
  WRITE (LUP,1040)
840 P=PP(J)
C ASSIGN RESULTS FOR PHASES 1 AND 2 TO LIQUID AND VAPOR PHASES DEPENDING ON WHETHER THE DEW OR BUBBLE POINT WAS CALCULATED.
C IF (LBUB) THEN
  XV=XX2(JJ)
  VL=V1
  VV=V2
ELSE
  XL=XX2(JJ)
  VL=V2
  VV=V1
END IF
C PRINT WARNING MESSAGES FOR ANY CASES OF NON-CONVERGENCE OCCURING ON FINAL CALL TO EACH ITERATION AND RETURN.
C IF (LV1CON) WRITE (LUP,1050)
IF (LV2CON) WRITE (LUP,1055)
IF (LXCON) WRITE (LUP,1060)
RETURN
1010 FORMAT (1X,'CRITICAL POINT OF PURE OR PSEUDO-PURE MATERIAL', ' EXCEEDED IN BUBLT')
1020 FORMAT (1X,'PURE MATERIAL PRESSURE ITERATION IN BUBLT', ' DID NOT CONVERGE')
1040 FORMAT (1X,'MIXTURE PRESSURE ITERATION IN BUBLT DID NOT', ' CONVERGE')
1050 FORMAT (1X,'VOLUME ITERATION FOR PARENT PHASE DID', ' NOT CONVERGE')
1055 FORMAT (1X,'VOLUME ITERATION FOR INCIPIENT PHASE
1  DID', ' NOT CONVERGE')
1060 FORMAT (1X,'COMPOSITION ITERATION IN BUBLT DID NOT
1  CONVERGE')
END
SUBROUTINE CRITX (X, TTC, PC, VC)
C
DEVELOPED BY MARK McCLINDEN AND GRAHAM MORRISON AT THE
NATIONAL BUREAU OF STANDARDS UNDER FUNDING FROM THE
ELECTRIC POWER RESEARCH INSTITUTE AND NBS.
C
THIS ROUTINE FINDS THE CRITICAL POINT PREDICTED BY THE
EQUATION OF STATE. FOR A MIXTURE THE CRITICAL POINT IS
THAT OF A PSEUDO-PURE COMPONENT HAVING THE SAME 'A' AND
'B' PARAMETERS AS THE MIXTURE; IN GENERAL, SUCH A
PSEUDO-PURE CRITICAL POINT WILL BE BELOW THE ACTUAL
MIXTURE CRITICAL POINT.
C
INPUTS:
X - MIXTURE COMPOSITION (MOL FRAC)
TTC - INITIAL GUESS FOR THE CRITICAL TEMPERATURE (K)

OUTPUTS:
TTC - CRITICAL TEMPERATURE (K)
PC - CRITICAL PRESSURE (KPA)
VC - CRITICAL VOLUME (M**3/KMOL)

OTHER SUBROUTINE REFERENCED:
ESPAR - CALCULATION OF EQUATION OF STATE PARAMETERS

COMMON /RDATA4/ R
DIMENSION TC(3), FTC(2)
IF (TTC.LE.0.0) TTC=300.
TC(1)=TTC
J=1
DO 200 IT=1,20
CALL ESPAR (0,TC(J),X,AC,BC)
IF (BC.LE.0.0) THEN
   TC(J)=0.5*TC(J)
   GOTO 200
END IF
TCC=0.2273291*AC/(R*BC)
FTC(J)=TCC-TC(J)
IF (ABS(FTC(J)).LT.0.01) GOTO 240
IF (J.LE.1) THEN
   J=2
   TC(2)=MIN(TCC,1.5*TC(1))
ELSE
   TC(3)=TC(2)-FTC(2)*(TC(2)-TC(1))/(FTC(2)-FTC(1))
   TC(1)=TC(2)
   TC(2)=TC(3)
   FTC(1)=FTC(2)
END IF
200 CONTINUE
WRITE (6,1000)
240 TTC=TC(J)
PC=0.02386944*AC/BC**2
VC=3.006818*BC
RETURN
1000 FORMAT (1X,'*** CRITX DID NOT CONVERGE')
END
FUNCTION ENTROP (T,V,X)

DEVELOPED BY MARK MCCLINDEN AND GRAHAM MORRISON AT THE
NATIONAL BUREAU OF STANDARDS UNDER FUNDING FROM THE
ELECTRIC POWER RESEARCH INSTITUTE AND NBS.

COMPUTE MOLAR ENTROPY OF A SINGLE PHASE TWO-COMPONENT
MIXTURE AS A FUNCTION OF TEMPERATURE, MOLAR VOLUME, AND
COMPOSITION

INPUTS:
T - TEMPERATURE (K)
V - MOLAR VOLUME (M**3/KMOL)
X - COMPOSITION (MOLE FRACTION)

OUTPUT:
S - MOLAR ENTROPY (KJ/KMOL K)

OTHER SUBROUTINES REFERENCED BY ENTROP:
ESPAR - COMPUTATION OF EQUATION OF STATE PARAMETERS

IMPLICIT REAL (A-H,O-Z)
COMMON /ESPAR1/ AA,AB,BA,BB,F,C1,D1,C2,D2
COMMON /REF/ TREFA,TREFB,HRA,HRB,SRA,SRB,VRA,VRB
COMMON /CPDATA/ CPA0,CPA1,CPA2,CPB0,CPB1,CPB2
COMMON /HSPURE/ HPA,HPB,SPA,SPB,CPOA,CPOB
COMMON /RDATA4/ R
CALL ESPAR (1,T,X,A,B)
B4=0.25*B
S=X*(SPA-SRA+R*LOG(V/VRA))+
1 *(1.0-X)*(SPB-SRB+R*LOG(V/VRB))
1 +(C1*B-A*D1)/B**2*LOG((V+B)/V)+A*D1/(V+B)
1 -R*B4/(V-B4)**2*(4.0*V-3*B4)
1 -R*T*D1*0.5*V/(V-B4)**3*(2.0*V-B4)
IF (X.GT.0.0 .AND. X.LT.1.0) THEN
  S=S-R*(X*LOG(X)+(1.0-X)*LOG(1.0-X))
END IF
ENTROP=S
RETURN
END
SUBROUTINE ESPAR (IQ,T,X,A,B)

DEVELOPED BY MARK MCLINDEN AND GRAHAM MORRISON AT THE
NATIONAL BUREAU OF STANDARDS UNDER FUNDING FROM THE
ELECTRIC POWER RESEARCH INSTITUTE AND NBS.

THIS ROUTINE CALCULATES THE EQUATION OF STATE
PARAMETERS AND THEIR TEMPERATURE DERIVATIVES AS A
FUNCTION OF TEMPERATURE AND COMPOSITION AS NEEDED BY
THE OTHER PROPERTY ROUTINES. BASED ON THE VALUE OF THE
INPUT QUALIFIER THE NECESSARY PARAMETERS ARE CALCULATED
EXCEPT THAT IF THE TEMPERATURE AND COMPOSITION ARE
UNCHANGED FROM THE LAST CALL THE PREVIOUS VALUES ARE
USED. THE TEMPERATURE DEPENDENCE OF THE A, B, AND CPO
PARAMETERS ARE CONTAINED ENTIRELY WITHIN ESPAR AND THE
STATEMENT FUNCTIONS (IN BUBLT AND BUBLP) FOR GIBBS
FREE ENERGY AND CHEMICAL POTENTIAL; ALTERNATE
EXPRESSIONS FOR 'A' AND 'B' REQUIRE CHANGING ONLY THESE
ROUTINES.

INPUTS:
IQ - INPUT QUALIFIER
= 0 COMPUTE ONLY A AND B
>= 1 ALSO COMPUTE TEMPERATURE DERIVATIVES OF A
AND B
>= 2 ALSO COMPUTE SECOND DERIVATIVE OF A AND B
AND IDEAL GAS HEAT CAPACITY
= 1, 2 OR 3 ALSO COMPUTE CONSTANTS FOR PURE
COMPONENT ENTHALPY AND ENTROPY
T - TEMPERATURE (K)
X - COMPOSITION (MOLE FRACTION COMPONENT A)

OUTPUTS:
A - 'A' PARAMETER FOR MIXTURE AT T, X
B - 'B' PARAMETER FOR MIXTURE AT T, X

OUTPUTS (VIA COMMON BLOCKS):
AA - 'A' PARAMETER FOR PURE COMPONENT A
AB - 'A' PARAMETER FOR PURE COMPONENT B
BA - 'B' PARAMETER FOR PURE COMPONENT A
BB - 'B' PARAMETER FOR PURE COMPONENT B
F - MIXTURE INTERACTION PARAMETER
DADT - TEMPERATURE DERIVATIVE OF A
DBDT - TEMPERATURE DERIVATIVE OF B
D2ADT2 - SECOND DERIVATIVE OF A WITH RESPECT TO
TEMPERATURE
D2BDT2 - SECOND DERIVATIVE OF B WITH RESPECT TO
TEMPERATURE
HPA - INTEGRAL OF CPO WITH RESPECT TO TEMP FOR PURE
A
HPB - INTEGRAL OF CPO WITH RESPECT TO TEMP FOR PURE
SPAT - INTEGRAL OF \((CPO - R)/T\) WITH RESPECT TO TEMP FOR PURE A

SPB - INTEGRAL OF \((CPO - R)/T\) WITH RESPECT TO TEMP FOR PURE B

CPOA - PERFECT GAS HEAT CAPACITY FOR COMPONENT A (KJ/KMOL K)

CPOB - PERFECT GAS HEAT CAPACITY FOR COMPONENT B (KJ/KMOL K)

IMPLICIT REAL (A-H,O-Z)

COMMON /ESPAR1/ AA, AB, BA, BB, F, DADT, DDBT, D2ADT2, D2BDT2
COMMON /RDATA1/ AA0, AA1, AA2, AB0, AB1, AB2, BA0, BA1, BA2,
   1 BB0, BB1, BB2, FO, FI
COMMON /CPDATA/ CPA0, CPA1, CPA2, CPB0, CPB1, CPB2
COMMON /HSPURE/ HPA, HPB, SPA, SPB, CPOA, CPOB
COMMON /REF/ TREFA, TREFB, HRA, HRB, SRA, SRB, VRA, VRB
COMMON /RDATA4/ R
SAVE TLAST0, TLAST1, TLAST2, TLAST3, XLAST0,
1 XLAST1, XLAST2, ALAST,
1 BLAST, SQAB, XB, XB2, XBX, DAA,
1 DAB, DBA, DBB, DSQAB, F0LAST, F1LAST
DATA TLAST0, TLAST1, TLAST2, TLAST3, XLAST0, XLAST1, XLAST2
1 /*-999.*/
IF (IQ.LT.0) THEN
   IQ=ABS(IQ)
END IF
A=ALAST
B=BLAST
IF (T.NE.TLAST0) GOTO 100
IF (FO.NE.F0LAST .OR. F1.NE.F1LAST) GOTO 110
IF (X.NE.XLAST0) GOTO 120
IF (IQ.LE.0) RETURN
IF (T.NE.TLAST1 .OR. X.NE.XLAST1) GOTO 200
IF (T.NE.TLAST3) GOTO 230
IF (IQ.LE.1) RETURN
IF (T.NE.TLAST2 .OR. X.NE.XLAST2) GOTO 300
RETURN
100 AA=AAO*EXP((AA1+AA2*T)*T)
   AB=AB0*EXP((AB1+AB2*T)*T)
   BA=BA0+(BA1+BA2*T)*T
   BB=BB0+(BB1+BB2*T)*T
   TLAST0=T
110 F=F0+F1*T
   F0LAST=F0
   F1LAST=F1
   F=F0+F1*T
120 X2=X*X
   XB=1.0-X
   XB2=XB*XB
\[ \text{XB} = \text{X} \times \text{XB} \]
\[ \text{SQAB} = \sqrt{\text{AA} \times \text{AB}} \]
\[ \text{A} = \text{X}^2 \times \text{AA} + 2.0 \times \text{XB} \times (1.0 - \text{F}) \times \text{SQAB} \times \text{X} \times \text{BB} \]
\[ \text{B} = \text{X} \times \text{BA} + \text{XB} \times \text{BB} \]
\[ \text{ALAST} = \text{A} \]
\[ \text{BLAST} = \text{B} \]
\[ \text{XLASTO} = \text{X} \]
\[ \text{IF} (\text{IQ} \leq 0) \text{ RETURN} \]
\[ \text{DAA} = \text{AA} \times ((\text{AA} + 2.0 \times \text{AA})^2 + 2.0 \times \text{AA}) \]
\[ \text{DAB} = \text{AB} \times ((\text{AB} + 2.0 \times \text{AB})^2 + 2.0 \times \text{AB}) \]
\[ \text{DBA} = \text{BA} \times ((\text{BA} + 2.0 \times \text{BA})^2 + 2.0 \times \text{BA}) \]
\[ \text{DBB} = \text{BB} \times ((\text{BB} + 2.0 \times \text{BB})^2 + 2.0 \times \text{BB}) \]
\[ \text{DSQAB} = 0.5 \times ((\text{AA} \times \text{DAB} + \text{AB} \times \text{DAA}) / \text{SQAB}) \]
\[ \text{DADT} = \text{X}^2 \times \text{DAA} + \text{XB} \times \text{DAB} + 2.0 \times \text{XB} \times ((1.0 - \text{F}) \times \text{DSQAB} - \text{F} \times \text{SQAB}) \]
\[ \text{DBDT} = \text{X} \times \text{DBA} + \text{XB} \times \text{DBB} \]
\[ \text{TLAST1} = \text{T} \]
\[ \text{XLAST1} = \text{X} \]
\[ \text{IF} (\text{IQ} \leq 3) \text{ THEN} \]
\[ \text{HPA} = (\text{CPAO} + (0.5 \times \text{CPA} + \text{CPA} \times 3.0 \times \text{T}) \times \text{T}) \times \text{T} \]
\[ \text{HPB} = (\text{CPB} + (0.5 \times \text{CPB} + \text{CPB} \times 3.0 \times \text{T}) \times \text{T}) \times \text{T} \]
\[ \text{SPA} = (\text{CPA} - \text{R}) \times \text{LOG}(\text{T} / \text{TREF}) + \text{CPA} \times (\text{T} - \text{TREF}) + 0.5 \times \text{CPA}^2 \]
\[ \text{T} \times (\text{T} - \text{TREF})^2 \]
\[ \text{SPB} = (\text{CPB} - \text{R}) \times \text{LOG}(\text{T} / \text{TREF}) + \text{CPB} \times (\text{T} - \text{TREF}) + 0.5 \times \text{CPB}^2 \]
\[ \text{T} \times (\text{T} - \text{TREF})^2 \]
\[ \text{TLAST2} = \text{T} \]
\[ \text{XLAST2} = \text{X} \]
\[ \text{RETURN} \]
\[ \text{END} \]
SUBROUTINE HCVCP (IQ,T,V,X,H,CV,CP,VS)
C
DEVELOPED BY MARK MCLINDEN AND GRAHAM MORRISON AT THE
C NATIONAL BUREAU OF STANDARDS UNDER FUNDING FROM THE
C ELECTRIC POWER RESEARCH INSTITUTE AND NBS.
C
GIVEN TEMPERATURE, MOLAR VOLUME AND COMPOSITION COMPUTE
C ENTHALPY AND/OR HEAT CAPACITY AT CONSTANT VOLUME AND/OR
C PRESSURE AND/OR VELOCITY OF SOUND AS SPECIFIED BY
C OUTPUT QUALIFIER IQ. THIS ROUTINE APPLIES ONLY TO
C SINGLE PHASE SYSTEMS.
C
INPUTS:
C IQ - OUTPUT QUALIFIER
  = 1 COMPUTE ENTHALPY ONLY
  = 2 ENTHALPY AND CONSTANT VOLUME HEAT CAPACITY
  = 3 ENTHALPY AND HEAT CAPACITY AT CONSTANT VOLUME
     AND PRESSURE
  = 4 COMPUTE HEAT CAPACITY AT CONSTANT VOLUME ONLY
  = 5 HEAT CAPACITY AT CONSTANT VOLUME AND AT
     CONSTANT PRESSURE
  = 3,5 ALSO COMPUTE VELOCITY OF SOUND
C T - TEMPERATURE (K)
C V - MOLAR VOLUME (M**3/KMOL)
C X - COMPOSITION (MOLE FRACTION)
C
OUTPUTS:
C H - MOLAR ENTHALPY (KJ/KMOL)
C CV - HEAT CAPACITY AT CONSTANT VOLUME (KJ/KMOL K)
C CP - HEAT CAPACITY AT CONSTANT PRESSURE (KJ/KMOL K)
C VS - VELOCITY OF SOUND (M/S)
C
OTHER SUBROUTINES REFERENCED BY HCVCP:
C ESPAR - COMPUTATION OF EQUATION OF STATE PARAMETERS
C
IMPLICIT REAL (A-H,O-Z)
COMMON /ESPAR1/ AA,AB,BA,BB,F,C1,D1,C2,D2
COMMON /REF/ TREFA,TREFB,HPA,HRB,SRA,SRB,HPB,HRB
COMMON /HSPURE/ HPA,HRB,SRA,SRB,SPA,SPB,CP0A,CP0B
COMMON /RDATA2/ WM(2),TC(2)
COMMON /RDATA4/ R
CALL ESPAR (IQ,T,X,A,B)
B4=0.25*B
VB=V+B
VRL=LOG(V/VB)
\B4=V-B4
VB43=VB4**3
RT=R*T
IF (IQ.LE.3) THEN
C COMPUTE MOLAR ENTHALPY AS A FUNCTION OF T, V, X
C H=X*(HPA-HRA)+(1.0-X)*(HPB-HRB)
```
1   +1(A+(A*D1/B-C1)*T)*VBL+A*(D1*T-B)/VB)/B
1   +2.0*RT*V*(2.0*V*V4)*(B4-0.25*D1*T)/VB43
END IF
   IF (IQ.GE.2) THEN
   COMPUTE CONSTANT VOLUME MOLAR HEAT CAPACITY
   D12=D1*D1
       CV=X*(CP0A-R)+(1.0-X)*(CP0B-R)
       +R*V*((0.375*D12*T/VB4+0.5*D2*T+D1)*(B4-2.0*V)
       +0.125*D12*T)/VB43
       +(1.0/VB+VBL/B)*(A*D2*B+2.0*(C1*D1*B-A*D12))/B
       -C2*VBL-A*D12/VB**2)/B)*T
   IF (IQ.EQ.3 .OR. IQ.EQ.5) THEN
   COMPUTE MOLAR HEAT CAPACITY AT CONSTANT PRESSURE USING
   CV
       Y=B4/V
       DPDT=2.0*R/VB4*(-1.0+(-0.25*T*D1
       +(V*V*(1.0+0.75*T*D1/VB4))/V
       DPDV=-RT*(B4/VB43*(8.0+1.5*B/VB4)*1.0/V**2)
       +A*(2.0*V+B)/(V*VB)**2
       CP=CV-DPDT*DPDT*T/DPDV
   COMPUTE VELOCITY OF SOUND USING C'S AND VOLUME
   CV
       WMOL=X*WM(1)+(1.0-X)*WM(2)
       VS=V*SQRT(-1000.0*CP*DPDV/(WMOL*CV))
END IF
END IF
RETURN
END
```
SUBROUTINE HPIN (H,P,X,T,XQ,XL,XV,VL,VL,HL,HV)

DEVELOPED BY MARK MCLINDEY AND GRAHAM MORRISON AT THE
NATIONAL BUREAU OF STANDARDS UNDER FUNDING FROM THE
ELECTRIC POWER RESEARCH INSTITUTE AND NBS.

THIS ROUTINE CALCULATES THE TEMPERATURE AND QUALITY OF
A SINGLE OR TWO-PHASE MIXTURE GIVEN THE PRESSURE AND
THE OVERALL (BULK) ENTHALPY AND COMPOSITION

INPUTS:
H - ENTHALPY (KJ/KMOL) OF BULK MIXTURE
P - PRESSURE (KPA)
X - BULK COMPOSITION (MOLE FRACTION)

OUTPUTS:
T - EQUILIBRIUM TEMPERATURE OF SYSTEM (K)
XQ - EQUILIBRIUM QUALITY (MOLAR BASIS), NEGATIVE
VALUES INDICATE SUBCOOLED LIQUID. QUALITIES >
1 INDICATE SUPERHEATED VAPOR.
NOTE: THE QUALITIES COMPUTED FOR SINGLE PHASE
CONDITIONS IN THE ROUTINES HPIN, HTIN, SPIN, AND
VTIN WILL NOT BE THE SAME.
XL - COMPOSITION OF LIQUID PHASE (MOLE FRACTION)
XV - COMPOSITION OF VAPOR PHASE (MOLE FRACTION)
VL - MOLAR VOLUME OF LIQUID PHASE (M**3/KMOL)
VV - MOLAR VOLUME OF VAPOR PHASE (M**3/KMOL)
HL - MOLAR ENTHALPY OF LIQUID PHASE (KJ/KMOL)
HV - MOLAR ENTHALPY OF VAPOR PHASE (KJ/KMOL)
NOTE: IF ONLY SUBCOOLED LIQUID IS PRESENT, VV AND
HV ARE COMPUTED AT THE BUBBLE POINT
TEMPERATURE AND VAPOR COMPOSITION IN
EQUILIBRIUM WITH SATURATED LIQUID AT THE
GIVEN P, XL. IF ONLY SUPERHEATED VAPOR IS
PRESENT, VL AND HL ARE COMPUTED AT THE
DEWPOINT TEMPERATURE AND LIQUID COMPOSITION
IN EQUILIBRIUM WITH SATURATED VAPOR
AT THE GIVEN P, XV.

IMPLICIT REAL (A-H,O-Z)
LOGICAL LCRIT
COMMON /TOL/ TOLR,ITMAX,LUP
COMMON /TOLSH/ TOLH,TOLS
C INITIAL GUESSES FOR TBUB, TDEW
TBUB=T
TDEW=T
CALL BUBLP (P,X,XVBUB,TBUB,VLBUB,VVBUB,.TRUE.,LCRIT)
CALL BUBLP (P,XLDEW,X,TDEW,VLDEW,VVDEW,.FALSE.,LCRIT)
CALL HCVCP (1, TBUB, VLBUB, X, HL, CV, CP, VSND)
CALL HVCPS (1, TDEW, VVDEW, X, HV, CV, CP, VSND)
IF (H.LE.HL) THEN
XL=X
ELSE IF (H.GE.HV) THEN
  XL=XLDEW
  XV=X
  VL=VLDW
  CALL HPXSP (H,P,X,TDEW,HV,VVDEW,T,VV,.FALSE.)
ELSE IF (ABS(XVBUB-XLDEW).LT.TOLR) THEN
  XL=X
  XV=X
  VL=VVBUB
  VV=VVDEW
  T=TBUB
ELSE
  XL1=X
  FX1=H-HL
  IF (ABS(FX1).LT.TOLH) THEN
    T=TBUB
    XV=XVBUB
    VL=VVBUB
    VV=VVBUB
    XL2=XL1
    GOTO 240
  END IF
  XLPOS=XL1
  FXPOS=FX1
  XL2=XLDEW
  FX2=H-HV
  IF (ABS(FX2).LT.TOLH) THEN
    T=TDEW
    XV=X
    VL=VLDW
    VV=VVDEW
    GOTO 240
  END IF
  XINEG=XL2
  FXNEG=FX2
DO 200 IT=1,ITMAX
  IF (ABS(FX2).LT.TOLH .OR. ABS(FX2-FX1).LT.0.02*TOLH) GOTO 240
  XL3=XL2-FX2*(XL2-XL1)/(FX2-FX1)
  IF (XL3.LT.MIN(XLNEG,XLPOS) .OR. .GT.MAX(XLNEG,XLPOS))
    XL3=XLPOS-FXPOS*(XLPOS-XLNEG)/(FXPOS-FXNEG)
  XL1=XL2
  XL2=XL3
  FX1=FX2
CALL RUBL (P,XL2,XV,T,VL,VV,.TRUE..LCRIT)
CALL HCVCPS (1,T,VL,XL2,HL,CV,CP,VSND)
CALL HCVCPS (1,T,VV,XV,HV,CV,CP,VSND)
XQ=(X-XL2)/(XV-XL2)
FX2=H-XQ*HV-(1.0-XQ)*HL
IF ( FX2.LT.0.0 ) THEN
   XNNEG=XL2
   FXNEG=FX2
ELSE
   XPOS=XL2
   FXPOS=FX2
END IF
200 CONTINUE
WRITE (LUP,1000)
240 CONTINUE
XL=XL2
END IF
XQ=(H-HL)/(HV-HL)
RETURN
1000 FORMAT (1X,'ROUTINE HPIN DID NOT CONVERGE')
END
SUBROUTINE HPXSP (H,P,X,TSAT,HSAT,VSAT,T2,V,LLIQ)
C
SUBROUTINE TO ITERATE FOR THE TEMPERATURE OF A SINGLE
PHASE BINARY MIXTURE GIVEN ENTHALPY, PRESSURE, AND
COMPOSITION. CALLED ONLY BY HPIN.
C
IMPLICIT REAL (A-H,O-Z)
LOGICAL LLIQ,LVCON
COMMON /TOL/ TOLR,ITMAX,LUP
COMMON /TOLSH/ TOLH,TOLS
T1=TSAT
FT1=H-HSAT
CALL HVCVPS (5,T1,VSAT,X,HSAT,CV,CP,VSND)
T2=TSAT+FT1/CP
V=VSAT
DO 200 IT=1,ITMAX
CALL ESPAR (2,T2,X,A,B)
CALL VIT (T2,P,A,B,V,LLIQ,LVCON)
CALL HVCVPS (1,T2,V,X,HSP,CV,CP,VSND)
FT2=H-HSP
IF (ABS(FT2).LT.TOLH .OR. ABS(FT2-FT1).LT.0.02*TOLH)
RETURN
T3=T2-FT2*(T2-T1)/(FT2-FT1)
T1=T2
T2=T3
200 FT1=FT2
WRITE (LUP,1000)
RETURN
1000 FORMAT (1X,'SINGLE PHASE ITERATION IN HPIN DID NOT
1 CONVERGE')
END
SUBROUTINE PLIMIT (T, A, B, VL, VU, PLOW, PUP)

GIVEN TEMPERATURE AND EQUATION OF STATE PARAMETERS,
THIS ROUTINE CALCULATES THE UPPER AND LOWER BOUNDS ON
PRESSURE FOR WHICH THERE ARE BOTH LIQUID AND VAPOR
SOLUTIONS TO THE EQUATION OF STATE. IT CARRIES OUT TWO
BISECTION METHOD ITERATIONS TO FIND THE POINTS WHERE
THE DERIVATIVE OF PRESSURE W.R.T. VOLUME IS ZERO.

INPUTS:
T - TEMPERATURE (K)
A, B - EQUATION OF STATE PARAMETERS AT TEMPERATURE T

OUTPUTS:
PLOW - LOWER BOUND ON PRESSURE (PLOW CAN BE
NEGATIVE, THE CALLING PROGRAM MUST CHECK AND CORRECT FOR
NEGATIVE PRESSURES)
PUP - UPPER BOUND ON PRESSURE (KPA)
VL - MOLAR VOLUME AT PLOW (M**3/KMOL)
VU - MOLAR VOLUME AT PUP (M**3/KMOL)

OTHER SUBROUTINES REFERENCED:
NONE

IMPLICIT REAL (A-H,O-Z)
COMMON /RDAT A4/ R
COMMON /TOL/ TOLR, ITMAX, LUP

STATEMENT FUNCTIONS FOR THE EVALUATION OF PRESSURE AS A
FUNCTION OF V AND THE DERIVATIVE OF PRESSURE W.R.T
VOLUME AS A FUNCTION OF V

\[ P(\text{RT}, V, Y, A, B) = \frac{\text{RT} \times (1.0 + (1.0 + (1.0 - Y) \times Y) \times Y)}{(1.0 - Y)^3} - \frac{A}{(V + B)} \]
\[ \frac{\text{D}P(\text{RT}, V, A, B, B4, B42)}{\text{D}V} = \frac{-\text{RT} \times (B4^2 \times B4^2 + (-4.0 \times B4^2 \times B4 + (4.0 \times B4^2 \times V + V) \times V) \times V)}{(V - B4)} \]

B4 = 0.25*B
B42 = B4*B4
RT = R*T

STARTING AT A VOLUME OF 12.0*B4 (WHICH HAS A POSITIVE
SLOPE FOR ALL 'REASONABLE' VALUES OF A, B, T) REDUCE
THE VOLUME UNTIL A NEGATIVE SLOPE OF P W.R.T. V IS
FOUND AND THEN BEGIN BISECTION METHOD TO FIND LOWER
BOUND ON VOLUME AND PRESSURE.

VC = 12.0272727*B4
V = V_old
DO 100 IT = 1, ITMAX
DPDV = DP(RT, V, A, B, B4, B42)
IF (DPDV LE 0.0) GOTO 116
VPOS = V
V = 0.5 * (V + B4)
100 CONTINUE

VNEG = V
DO 116 IT = 1, 20
VL = 0.5 * (VNEG + VPOS)
DPDV = DP(RT, VL, A, B, B4, B42)
IF (DPDV LT 0.0) THEN
    VNEG = VL
ELSE
    VPOS = VL
END IF
116 CONTINUE

Y = B4 / VL
PLOW = P(RT, VL, Y, A, B)

C STARTING AT V = 2 * A / RT INCREASE V UNTIL A NEGATIVE
C SLOPE IS FOUND; USE WITH V = 12.0 * B TO BEGIN BISECTION
C ITERATION FOR UPPER BOUND ON PRESSURE
C
VPOS = V_old
V = 2.0 * A / RT
DO 160 IT = 1, ITMAX
DPDV = DP(RT, V, A, B, B4, B42)
IF (DPDV LE 0.0) GOTO 164
VPOS = V
V = 2.0 * V
160 CONTINUE

VNEG = V_old
DO 164 IT = 1, 20
VU = 0.5 * (VNEG + VPOS)
DPDV = DP(RT, VU, A, B, B4, B42)
IF (DPDV LT 0.0) THEN
    VNEG = VU
ELSE
    VPOS = VU
END IF
164 CONTINUE

Y = B4 / VU
PUP = P(RT, VU, Y, A, B)
RETURN
END
FUNCTION PVT (T,V,X)
C
C DEVELOPED BY MARK MCLINDEN AND GRAHAM MORRISON AT THE
C NATIONAL BUREAU OF STANDARDS UNDER FUNDING FROM THE
C ELECTRIC POWER RESEARCH INSTITUTE AND NBS.
C
C THIS ROUTINE CONTAINS THE BASIC FORM OF THE CARNAHAN-
C STARLING-DESANTIS EQUATION OF STATE AND COMPUTES
C PRESSURE AS A FUNCTION OF TEMPERATURE, VOLUME, AND
C COMPOSITION.
C
C INPUTS:
C    T - TEMPERATURE (K)
C    V - MOLAR VOLUME (L/MOL)
C    X - COMPOSITION (MOLE FRACTION)
C
C OUTPUT (VIA FUNCTION VALUE):
C    PVT - PRESSURE (KPA)
C
C OTHER ROUTINE REFERENCED:
C    ESPAR - CALCULATION OF EQUATION OF STATE
C    COEFFICIENTS
C
COMMON /RDATA4/ R
CALL ESPAR (0,T,X,A,B)
Y=0.25*B/V
PVT=(R*T*(1.0+(1.0+(1.0-Y)*Y)*Y)/
     (1.0-Y)**3-A/(V+B))/V
RETURN
END
SUBROUTINE SPXSP (S, X, TSAT, SSAT, VSAT, T2, V, LLIQ)

SUBROUTINE TO ITERATE FOR THE TEMPERATURE OF A SINGLE
PHASE BINARY MIXTURE GIVEN ENTROPY, PRESSURE, AND
COMPOSITION.

IMPLICIT REAL (A-H,O-Z)
LOGICAL LLIQ, LVCON
COMMON /TOL/ TOLR, ITMAX, LUP
COMMON /TOLSH/ TOLH, TOLS
T1 = TSAT
FT1 = S - SSAT
CALL HCVCS (4, T1, VSAT, X, SSAT, CV, CP, VSND)
T2 = TSAT + FT1/CV
V = VSAT
DO 200 IT = 1, ITMAX
CALL ESPAR (2, T2, X, A, B)
CALL VIT(T2, P, A, B, V, LLIQ, LVCON)
SSP = ENTROP(T2, V, X)
FT2 = S - SSP
IF (ABS(FT2).LT.TOLS .OR. ABS(FT2 - FT1).LT.0.02*TOLS)
RETURN
T3 = T2 - FT2*(T2 - T1)/(FT2 - FT1)
T1 = T2
T2 = T3
200 FT1 = FT2
WRITE (LUP, 1000)
RETURN
1000 FORMAT (1X,'SINGLE PHASE ITERATION IN SPIN DID NOT
CONVERGE')
END
SUBROUTINE TQIN (T,XQ,X,P,H,S,XL,XV,VL,VV)

C DEVELOPED BY MARK MCCLINDEM AND GRAHAM MORRISON AT THE
C NATIONAL BUREAU OF STANDARDS UNDER FUNDING FROM THE
C ELECTRIC POWER RESEARCH INSTITUTE AND NBS.
C
C THIS ROUTINE CALCULATES THE PRESSURE, ENTHALPY, AND
C ENTROPY OF A TWO-PHASE MIXTURE GIVEN THE TEMPERATURE,
C QUALITY AND COMPOSITION.
C
INPUTS:
T - TEMPERATURE (K)
XQ - QUALITY (MOLES VAPOR/TOTAL MOLES)
X - BULK COMPOSITION (MOLE FRACTION)

OUTPUTS:
P - SATURATION PRESSURE OF SYSTEM
H - MOLAR ENTHALPY (J/MOL)
S - MOLAR ENTROPY (J/MOL K)
XL - COMPOSITION OF LIQUID PHASE (MOLE FRACTION)
XV - COMPOSITION OF VAPOR PHASE (MOLE FRACTION)
VL - MOLAR VOLUME OF LIQUID PHASE (L/MOL)
VV - MOLAR VOLUME OF VAPOR PHASE (L/MOL)

OTHER ROUTINES REFERENCED:
BUBLT - SATURATION PRESSURE
HCVCPS - ENTHALPY
ENTROP - ENTROPY

IMPLICIT REAL (A-H,O-Z)
COMMON /TOL/ TOLR,ITMAX,LUP
LOGICAL LCRIT,LITER1
DIMENSION XLQ(3),FXQ(2)
J=1
LITER1=.FALSE.
XV=X
DO 200 IT=1,ITMAX
CALL BUBLT (T,XLQ(J),XV,P,VL,VV,LITER1,LCRIT)
FXQ(J)=XQ-(X-XLQ(J))/(XV-XLQ(J))
IF (ABS(FXQ(J)) .LT. 1.0E-05) GO TO 240
IF (FXQ(J) .LT. 0.0) THEN
XLNEG=XLQ(J)
FXNEG=FXQ(J)
ELSE
XLPOS=XLQ(J)
FXPOS=FXQ(J)
END IF
IF (IT.EQ.1) THEN
IF (FXQ(J) .LT. 0.0) THEN
XLPOS=X
FXPOS=0.0
ELSE

240 CONTINUE
XLNEG = X
FXNEG = X
END IF
J = 2
LITER1 = .TRUE.
XLQ(2) = X + XQ*(XLQ(1) - X)
ELSE
XLQ(3) = XLQ(2) - FXQ(2)*(XLQ(2) -
1 XLQ(1)) / (FXQ(2) - FXQ(1))
XLQ(1) = XLQ(2)
XLQ(2) = XLQ(3)
FXQ(1) = FXQ(2)
END IF
IF (XLQ(2) .LT. MIN(XLNEG, XLPOS) .OR.
1 XLQ(2) .GT. MAX(XLNEG, XLPOS))
1 XLQ(2) = XLPOS - FXPOS*(XLPOS - XLNEG) / (FXPOS - FXNEG)
200 CONTINUE
WRITE (LUP, 1020)
240 XL = XLQ(J)
IF (LCRIT) THEN
WRITE (LUP, 1000)
RETURN
END IF
242 CALL HCVCPSP (1, T, VL, XL, HL, CV, CP, VSND)
CALL HCVCPSP (1, T, VV, XV, HV, CV, CP, VSND)
H = XQ*HV + (1.0 - XQ)*HL
SL = ENTROP(T, VL, XL)
SV = ENTROP(T, VV, XV)
S = XQ*SV + (1.0 - XQ)*SL
RETURN
1000 FORMAT (1X, '*** CRITICAL POINT EXCEEDED IN TQIN ***')
1020 FORMAT (1X, '*** ROUTINE TQIN DID NOT CONVERGE ***')
END
SUBROUTINE VIT (T,P,A,B,VS,LLIQI,LVCON)

C DEVELOPED BY MARK MCLINDEN AND GRAHAM MORRISON AT THE  
C NATIONAL BUREAU OF STANDARDS UNDER FUNDING FROM THE  
C ELECTRIC POWER RESEARCH INSTITUTE AND NBS.

C GIVEN TEMPERATURE, PRESSURE, AND EQUATION OF STATE  
C PARAMETERS, THIS ROUTINE CALCULATES THE LIQUID OR VAPOR  
C MOLAR VOLUME THAT SATISFIES THE EQUATION OF STATE.

C INPUTS:
C T - TEMPERATURE (K)  
P - PRESSURE (KPA)
C A,B - EQUATION OF STATE PARAMETERS AT TEMPERATURE T
C VS - INITIAL GUESS FOR VOLUME. IN ABSENCE OF BETTER
C GUESSES SUGGESTED VALUES ARE:
C LIQUID: VS=0.8*B
C VAPOR: VS=R*T/P
C LLIQI - LOGICAL VARIABLE
C IF LLIQI = .TRUE. COMPUTE LIQUID VOLUME
C IF LLIQI = .FALSE. COMPUTE VAPOR VOLUME
C NOTE: IF EITHER THE TEMPERATURE OR THE PRESSURE IS
C ABOVE THE CRITICAL VALUE, ONLY ONE SOLUTION EXISTS
C AND THE VALUE OF LLIQI HAS NO EFFECT.

C OUTPUTS:
C VS - MOLAR VOLUME (M**3/KG MOL)
C LVCON - ERROR FLAG; IF LVCON = .TRUE. THE ITERATION
C HAS NOT CONVERGED

C OTHER SUBROUTINES REFERENCED:
C NONE

C (FOR EXPLANATION OF NOMENCLATURE SEE BUBLT)

C NOTE: THIS ROUTINE IS WRITTEN IN DOUBLE PRECISION
C EXCEPT THAT THE ARGUMENTS ARE SINGLE PRECISION

IMPLICIT DOUBLE PRECISION (A-H,O-Z)
LOGICAL LLIQ,LVCON,LLIQI
REAL T,P,A,B,R,VS,TOLR,TC,PC
COMMON /RDATA4/ R
COMMON /TOL/ TOLR,ITMAX,LUP
LVCON=.FALSE.
LLIQ=LLIQI
V=VS
VL=LOG(V)
PL=LOG(P)
RT=R*T
B4=0.25*B
B4L=LOG(B4)
IF (VL.LT.B4L) VL=B4L+0.5
TC = A/(B*4.398909*R)
PC = 0.02386944*A/B**2
VCL = LOG(12.0272727*B4)
IF (P.GT.PC) THEN
  LIQ = .TRUE.
ELSE IF (T.GT.TC) THEN
  LIQ = .FALSE.
END IF

ENTER NEWTONS METHOD ITERATION FOR VOLUME. FOR LIQUIDS
(OR FLUIDS ABOVE THE CRITICAL PRESSURE) THE ITERATION
IS CARRIED OUT IN TRANSFORMED COORDINATES OF LOG (V).
FOR VAPOR (OR FLUIDS AT SUPERCRITICAL TEMPERATURES BUT
PRESSURES BELOW THE CRITICAL VALUE) THE ITERATION IS IN
TERMS OF LOG (V) AND LOG (P). THE ITERATION HAS
CONVERGED WHEN THE PRESSURE CALCULATED FROM THE
EQUATION OF STATE AGREES WITH THE INPUT PRESSURE.

DO 100 IT=1,ITMAX
  IF (VL.GT.VCL .EQV. LIQ) VL=VCL
  VLS=VL
  Y=B4/V
  VB=V+B

CALCULATE PRESSURE AS A FUNCTION OF VOLUME AND THE

P2=(RT*(1.0+(1.0+(1.0-Y)*Y)*Y)/(1.0-Y)**3-A/VB)/V
DPDLV=RT/V*(-1.0+(-4.0+(-4.0+(4.0-Y)*Y)*Y)/(1.0-Y)**4
1  +A*(2.0*Y+B)/(V*VB*VB)
IF (LIQ) THEN
  IF (DPDLV.GE.0.0) THEN
    VL=0.5*(B4L+VLS)
  ELSE
    FVDP=(P2-P)/DPDLV
    IF (ABS(FVDP/P).LT.0.001*TOLR) THEN
      VS=EXP(VL)
      RETURN
    ELSE
      VL=VL-FVDP
      IF (VL.LE.B4L) VL=0.5*(B4L+VLS)
    END IF
  END IF
ELSE
  IF (DPDLV.GE.0.0 .OR. P2.LE.0.0) THEN
    VL=VL+0.5
  ELSE
    FVDP1=(LOG(P2)-PL)*P2/DPDLV
    IF (ABS(FVDP1).LT.0.001*TOLR) THEN
      VS=EXP(VL)
      RETURN
  END IF
END IF
END IF
VL=VL-FVDPL
IF (ABS(VL-VLS).GT.1.5) VL=VLS+SIGN(1.0D0,VL-VLS)
IF (VL.LT.VCL) VL=0.5*(VLS+VCL)
END IF
END IF
V=EXP(VL)
100 CONTINUE
LVCON=.TRUE.
VS=V
RETURN
END