Refrigerant Distribution Model Development

by

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Abstract

The objectives of this research were to evaluate the performance of the evaporator distributors and to construct an analytical model to predict the flow distribution. An experimental apparatus that provided design and off-design conditions for the refrigerant was used to test the distributors. The factors examined included distributor mounting orientation, load conditions, refrigerant degree subcooling, and feeder tube lengths. A refrigerant flow model was constructed based on Baroczy two-phase flow model and fundamental fluid mechanics theory. Algorithms were developed to implement the model. A series of experiments were run to measure the distributors' performance and to validate the model.

The distributors tested were found to give rather uniform distribution with deviation from the mean less than 10%. The model developed successfully predicted the flow distribution when the distributors were subject to a variety of conditions. The model is valid as long as the flow does not choke in the feeder tubes. The feeder tube length was found to be an effective parameter to control the flow distribution. The distributor mounting orientation, load conditions, and refrigerant degree subcooling did not affect the flow distribution for the range of conditions tested.

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To my parents.
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Chapter 1

Introduction

1.1 Problem statement
Carrier Corporation, a leading manufacturer of air conditioners, sponsored my research project for the study of refrigerant flow distribution in air conditioners. The research was motivated when Carrier found that the air that flowed through the evaporator was not uniform in temperature when exiting the evaporator surface. The non-uniformity of temperature indicated that the refrigerants in some of the evaporator circuits did not remove the heat from the air effectively, resulting in a loss of efficiency in air conditioners. The refrigerant flow distribution upstream to the evaporator was suspected to be the cause.

1.2 Background
The most common method of refrigeration in domestic air conditioner systems is a vapor compression cycle (Howell, 1987). As shown in Figure 1.1, such a system consists of four major components: evaporator, compressor, condensor, and throttling valve. The

Figure 1.1: Air conditioner system based on vapor compression cycle schematic.
refrigerant, which is the fluid flowing throughout the system, serves as the heat transfer medium. The refrigerant enters the evaporator circuits at a temperature below room temperature. The heat of the air passing through the evaporator circuits is transferred to the refrigerant. The compressor then compresses the refrigerant to a high temperature and pressure. At the condenser, heat is transferred from the refrigerant to the outdoor air that flows past the condenser circuits. The refrigerant then flows through the throttling valve. The refrigerant undergoes a sharp drop in pressure at the throttling process and its temperature drops back below room temperature. The refrigerant enters the evaporator again and this completes the cycle. Heat generated by humans, lights, and other equipment inside a building is thus continuously transferred to the refrigerant at the evaporator and carried by the refrigerant to the condenser where the heat is transferred outdoors.

As shown in Figure 1.2, the throttling valve consists of some or all of the following devices: thermal expansion valve (TXV), distributor, throttle, and feeder tubes. The TXV

![Thermal expansion valve, distributor, throttle, and feeder tubes.](image)

**Figure 1.2:** Thermal expansion valve, distributor, throttle, and feeder tubes.

modulates the flow rate of the refrigerant. The distributor houses the throttle and distributes the refrigerant into the feeder tubes. Each feeder tube is connected to the corresponding circuit in evaporator.
At the exit of the condensor, the refrigerant is in a subcooled state. After flowing through the throttle at the distributor, the refrigerant flashes and forms a two-phase flow. Because heat is mainly picked up by the refrigerant in liquid form at the evaporator, it is important that the two phases are distributed in the same ratio in each feeder tube that is subsequently connected to the evaporator circuit. If an evaporator circuit contains mostly the vapor refrigerant, only a small amount of heat is transferred from the air to this particular circuit. On the other hand, those circuits that carry more liquid refrigerant will have some of the refrigerant leaving the circuits still in liquid form, without making full use of its heat-absorbing capacity. This results in low efficiency at the evaporator. It was believed that bad refrigerant flow distribution was the cause for the problem in Carrier.

1.3 Objectives

The focus of this research is on how the distributor, throttle, and feeder tubes affect the refrigerant flow distribution. The objectives are to determine the performance of the distributor under real operating environments and to develop an analytical model that can be used to evaluate the performance of a distributor system under a variety of conditions or to give a non-uniform distribution by choice. In addition, this research intends to provide some guidelines to the design engineers what to watch out for in the distributor system design.

Two versions of distributors were provided by Carrier for study. The first type (part number: 314947-315) shown in Figure 1.3 and 1.4, as is called type I in this thesis, is a distributor with nozzle orifice at the inlet. The outlets are five drilled holes that receive capillary tubes (feeder tubes) connected to the evaporator circuits. Figure 1.5 and 1.6 show Type II distributors (part number: 319072-404). It has a movable piston (throttle) and the distributor can function both in refrigeration and heat pump cycle. The refrigerant flows
through the throttle and makes a 90° turn into the plenum before exiting the distributor through the feeder tubes.

1.4 Approaches

To test the distributors under real operating conditions, an experimental apparatus was designed and built. The apparatus provides the design and off-design conditions at the condenser exit and evaporator inlet of an air conditioner. In addition to refrigerant conditions, other variables such as the distributor mounting orientations, load conditions (i.e. refrigerant flow rate), and feeder tube’s length to diameter ratios (L/D) were looked into.

Since the heat is mainly carried away by liquid phase refrigerant, a flow measurement technique to determine the amount of liquid phase distributed to each feeder tube was developed. Sets of experiments were run to evaluate the distributor performance. As specified by the sponsor, R-22 was selected as the working refrigerant for this research.

The analytical model development began with a two-phase flow regime analysis. The model was constructed on an existing two-phase flow model and fundamental fluid mechanics theory. Although the model was developed based on the geometry of Type II distributor, the methodology and concept of the model development was applicable to distributors of other designs.

Chapter 2 gives a discussion of the testing procedures and experimental apparatus used in this research. In Chapter 3, I present the findings of the effects of mounting orientation on the distributor performance. Chapter 4 covers the flow regime map analysis. The development and formulation of the flow distribution model are described in Chapter 5. Chapter 6 discusses the test matrix to study the variables believed to affect the distributor performance and to evaluate the flow distribution model. Chapter 7 gives an in-depth analysis of the results of the experiments and makes comparison between the measured and
model predicted values. Chapter 8 gives conclusions and recommendations resulting from
the research. Appendix A explains the nomenclature in this thesis. The contents of Appen-
dices B through K are referred from the text in the chapters.
Figure 1.3: Type I distributor.

Figure 1.4: A cut-out view of a Type I distributor.
Figure 1.5: Type II distributor.

Figure 1.6: A cut-out view of a Type II distributor.
Chapter 2

Experimental Apparatus and Testing Procedures

Heather Smith, who previously worked on this project, designed and built the apparatus (1993). Since then, I have modified the set-up and added instrumentation. Figure 3.1 shows the apparatus’s schematic. Two pictures of the apparatus are shown in Figure 3.2 and 3.3. There are three major elements in the apparatus.

2.1 Reservoir

The purpose of the reservoir is to store the refrigerant and to provide the refrigerant with conditions that are at the condensor outlet in air conditioners. Once the refrigerant has been charged into the reservoir through valves number (no.) 1, 11, and 13 from a refrigerant supply, the temperature of the refrigerant can be controlled by adjusting the mixing of cold water and steam into the coils inside the reservoir. A thermocouple is mounted in the reservoir to show its temperature.

To create a subcooled state in the reservoir, compressed nitrogen is fed through a pressure regulator into the reservoir to exert extra pressure on the refrigerant. The reservoir remains connected to the nitrogen supply during the experiment. If the reservoir is over-pressurized, the nitrogen can be bled off by opening valve no. 3. In nominal operating conditions, the temperature and pressure of the reservoir is 95°F and 240 psia, respectively.

Pressure transducer no. 1 measures the gage pressure of the reservoir. Pressure transducer no. 2 is a differential pressure transducer. It measures the hydrostatic pressure change of the liquid refrigerant in the reservoir. The measurement can be used to calculate the refrigerant mass flow rate when the refrigerant is flowing out of the reservoir. The
Figure 2.1: Schematic of the experimental apparatus.
Figure 2.2: Apparatus picture; The meter ruler placed on top of the main vessel is to show the scale of the apparatus. The vessel with a gage pressure meter on is the reservoir. To the left of the reservoir is the condensing tower. Two compressed nitrogen cylinders can also be seen.
Figure 2.3: Apparatus picture; Two sight windows can be seen on the cover of the main vessel. The key below the windows is used to drain the refrigerant from the measurement glasses after flow distribution measurement. The wire leading out from the cover provides the electricity to the light bulb in the main vessel.
copper tube which comes out from the top of reservoir and, on the other end, connected to pressure transducer no. 2 is wrapped with heating tape. The heating tape is turned on during the experiment so that the refrigerant vapor in this tube segment will not condense and make the pressure transducer no. 2 measurement ambiguous.

In addition, there is a sight glass on the side of the reservoir to indicate the liquid refrigerant level. The average of the mass flow rate can be obtained by timing the liquid level drop at the sight glass.

### 2.2 Main vessel

During experiments, the main vessel is filled with refrigerant vapor or a mixture of nitrogen and refrigerant vapor to simulate the inlet conditions of the evaporators. When valve no. 8 is opened (with valve no. 9 closed), the subcooled refrigerant from the reservoir flows into the main vessel and through the approach pipe and distributor. As shown in Figure 2.4, each feeder tube coming out from the distributor is channeled into a measurement glass, respectively. By comparing the amount of liquid phase refrigerant flown through each feeder tube, how well the distributor system distributes the refrigerant can be determined. This is the technique used in this research to evaluate the performance of the distributors. If the liquid levels in the measurement glasses are uniform, it indicates a good flow distribution because the liquid to vapor ratio in all the feeder tubes are about the same. By turning horizontally the turn-table that supports the measurement glasses, each measurement glass can be observed and the liquid level recorded through the sight windows on the cover of the main vessel. Figure 2.5 and 2.6 show two pictures of the measurement glasses and turn-table.

If valves no. 9 and 10 are open (with valve no. 8 closed), the subcooled refrigerant flows through the distributor and feeder tubes which are outside before entering the main
vessel. The pressure transducer no. 3 measures the pressure drop across the throttle while the no. 4 reads the pressure drop across the feeder tubes (see Figure 2.7). These pressure drop test measurements are used to help evaluate the flow distribution model developed later in this thesis.

![Diagram of a distributor system]

**Figure 2.4:** The amount of liquid refrigerant collected in the glass containers indicate the performance of the distributor.

A condensing tower is connected to the main vessel. Ice water circulating inside the condensing tower condenses the refrigerant vapor in the tower and the liquid refrigerant flows back to the main vessel. However, R-22 vapor requires much lower temperature to be effectively condensed and retrieved, the condensing tower was therefore not used in the experiments in this project. After each run of experiments, a refrigerant recovery and recycle unit was used instead to recover the refrigerant from the main vessel and charge it back
Figure 2.5: Measurement glasses, turn-table and a vertically mounted type II distributor.

Figure 2.6: Measurement glasses, turn-table and a vertically mounted type I distributor. Only 5 of the measurement glasses on this turn-table were used.
Figure 2.7: Pressure transducer no. 3 (center) and no. 4
to the reservoir.

The distributor can be mounted with different orientations in the main vessel to study the effect of orientation on flow distribution. The distributor (Type II) can house various sizes of throttles. The throttle size can be used to control the total refrigerant flow rate through the throttle to provide different load conditions. In addition, the feeder tubes can be cut into different lengths in order to get the desired L/D ratios.

2.3 Data acquisition system and pressure transducers

The pressure transducers no. 1 and 5 are strain-gage pressure transducers each with a range of 300 psig. The no. 2, 3, and 4 are variable-reluctance differential pressure transducers with ranges of 0.5, 200, and 32 psi, respectively. All of the five pressure transducers along with the K-typed thermocouple in the reservoir are connected to a Dash-16 data acquisition board. In addition, a cold junction temperature is measured as a reference to the thermocouple. There are a total of 7 measurements taken and 7 channels used. Throughout all the experiments in this research when the data acquisition system was used, the scanning sampling rate was set to 250Hz. Therefore, the interval between readings was 0.028 seconds for each channel.
Chapter 3

Mounting Orientation Test

Before evaluating the distributors for mounting orientation effects, some preliminary tests with a Type I distributor were performed to determine if the flow measurement technique concept was capable of indicating the distributor’s distribution performance. As described in Appendix B, the apparatus gave consistent and reliable measurements on the flow distribution. The preliminary tests also showed that the type I distributor gave rather uniform distribution with deviation less than 10% from the mean.

To study the effects of mounting orientation, I mounted a type II distributor horizontally (see Figure 3.1). This is the orientation for which I would expect to see the worst maldistribution if it were due to the formation of stratified flow in the distributor. Each of the feeder tubes was 20.5 in. (0.5207 m) long and the R-22 was conditioned to 95°F and 240 psia in the reservoir before the experiments began. Two sets of experiments were run.

**Figure 3.1**: Schematic of experiment setup with distributor mounted horizontally (feeder tubes coming out from the distributor are not shown).
3.1 Test (set 1)

Figure 3.2 shows the relative locations of the distributor holes with respect to ground. For the first two runs, the two approach pipes were arranged as that shown in Figure 3.1.

![Diagram of distributor hole orientation](image)

**Figure 3.2**: Distributor hole orientation (with the direction of gravity indicated) for 1st set of experimental runs.

In the 3rd run, I turned the short approach pipe 90° in the horizontal plane to see if there was bias by the approach pipe arrangement (see Figure 3.3). For the 4th run, as shown in Figure 3.4, the short approach pipe was rotated 90° more so that it was right underneath the long one. As shown in Figure 3.5, the flow distribution in all runs were consistent. The flow distribution did not seem to be biased in any way by the approach pipe arrangement. Feeder tubes “a” and “d” consistently received more liquid. The distribution pattern did not indicate the formation of a stratified flow. If it had developed, the feeder tubes coming out from the holes located lower down would have had passed more liquid.
Figure 3.3: Schematic of experiment setup with distributor mounted horizontally after 90 degree turn (feeder tubes coming out from the distributor are not shown).

Figure 3.4: Schematic of experiment setup with distributor mounted horizontally after 180 degree turn (feeder tubes coming out from the distributor are not shown).
3.2 Test (set 2)

With the same experimental set-up, the same distributor and feeder tubes, but with the distributor rotated (Figure 3.6 shows the new hole locations with respect to ground), I did another 3 runs. The results are shown in Figure 3.7. Feeder tubes “a” and “d” once again received the most liquid. The variation of liquid level for this set of runs was smaller but the flow distribution pattern was essentially the same as that of 1st set of runs.

3.3 Test conclusion

Although the flow deviations among the feeder tubes for the distributor tested were as high as 10% from the mean, the flow distribution was consistent between runs regardless of the distributor hole orientation. The results indicate that the distributor non-uniformity, rather than the mounting orientation, plays a role in flow distribution. A flow regime map analysis, which is discussed in Chapter 4, can explain why a stratified flow was not formed in the distributor for these 2 sets of test. In addition, a flow regime map can predict under
what operating conditions that the mounting orientation can be a significant factor for the flow distribution.

**Figure 3.6:** Distributor hole orientation (with the direction of gravity indicated) for 2nd set of experimental runs.

**Figure 3.7:** Normalized refrigerant distribution with distributor hole orientation shown in Figure 3.6.
Chapter 4

Flow Regime Map Analysis

In addition to predicting the effects of mounting orientation on flow distribution, a flow regime map analysis can help select a suitable two-phase flow model for the basis of the flow distribution model developed later in this thesis. The values of the mass flow rate, quality, and thermal properties at the flow segment of interest need to be predicted in order to construct and interpret the flow regime maps.

4.1 Critical flow

For the nominal operating conditions of air conditioners, when the pressures at the condenser and evaporator are 240 psia and 90 psia, respectively, it is expected that the flow through the throttle will be choked (critical flow). The mass flow rate will reach a maximum value which depends on the fixed inlet conditions. The mass flux is called the critical mass flux and the ratio of the throat pressure to the stagnation pressure is called the critical pressure ratio (Whalley, 1987).

The best model for predicting the critical mass flux is Henry and Fauske’s two-phase critical flow model (Henry, 1971) (see Appendix C). The model can be applied to nozzle, orifice, and short tube choked flow. To get the mass flow rate, I used the conditions at the condenser exit as the stagnation conditions for the Henry and Fauske’s model. Therefore, $T_o$ was 95 °F and $P_o$ was 240 psia. The iteration result showed that the critical mass flux was 46.14 lbm/s/in$^2$, and it corresponded to a mass flow rate of 0.232 lbm/s for the distributor with throttle size of 0.08 inches. The critical pressure ratio was 0.72.

The next step was to iterate the refrigerant quality. Since I was mainly interested in the flow regime after the throttle, I iterated the quality at the throttle discharge. Appendix D
shows the formulation for the quality iteration. The calculation also involves Baroczy’s model for two-phase flow pressure drop across a tube. The details of Baroczy’s model is discussed in Appendix E. For the quality calculation, I knew that the enthalpy did not change during a throttling process. Therefore, I made a guess of the pressure at the throttle discharge, and iterated the value until the guessed pressure drop across the feeder tubes matched the one predicted by Baroczy’s model. For mass flow rate of 0.232 lbm/s, the quality at the throttle discharge was 8.76%. The saturated temperature and pressure were 72 °F and 140.37 psia.

4.2 Taitel-Dukler Flow Regime Maps

I used Marc Hodes’ algorithm, which is based on the Taitel-Dukler model, to generate the flow regime maps (Hodes, 1994; Taitel, 1990). Four flow regimes are defined in the model. They are bubble, annular, intermittent, and stratified.

Figure 4.1 shows the flow regime map for the flow in a horizontally mounted type II distributor for nominal operating conditions. The input diameter is 0.00549m (0.216 in), which is the diameter of the throttle discharge. The 0 degree input for the angle of inclination indicates that it is a horizontal flow. The thermal property inputs correspond to the conditions in the throttle discharge where the predicted saturated temperature and pressure are 72 °F and 140.37 psia. Figure 4.2 shows the flow regime map for a vertically downward flow of the same thermal conditions.

Once the mass flow rate, \( \dot{m} \), and the quality, \( x \), at the throttle discharge were known, I could calculate the gas superficial velocity, \( j_g \), and liquid superficial velocity, \( j_l \), at the throttle discharge. First, the mass flux, \( G \), gas mass flux, \( G_g \), and liquid mass flux, \( G_l \), were calculated by
**Figure 4.1:** Flow regime map for horizontal flow with $P=140.37$ psia, $T=72\, ^\circ \text{F}$, and $D=0.00549$ m (stagnation conditions: $T_o = 95\, ^\circ \text{F}$, $P_o = 240$ psia); The crossed-out area indicates the flow operating region.

**Figure 4.2:** Flow regime map for vertically downward flow with $P=140.37$ psia, $T=72\, ^\circ \text{F}$, and $D=0.00549$ m (stagnation conditions: $T_o = 95\, ^\circ \text{F}$, $P_o = 240$ psia); The crossed-out area indicates the flow operating region.
\[ G = \frac{\dot{m}}{A_T}, \]  
(4.1)

\[ G_g = G \cdot x, \]  
(4.2)

and \[ G_l = G \cdot (1 - x) \]  
(4.3)

where \( A_T \) is the cross-sectional area of the throttle discharge.

Next, the superficial velocities were obtained by

\[ \dot{j}_g = \frac{G_g}{\rho_g}, \]  
(4.4)

and \[ \dot{j}_l = \frac{G_l}{\rho_l}. \]  
(4.5)

where \( \rho \) is the density.

The gas and liquid superficial velocities were 9.49 m/s and 3.36 m/s, respectively. An operating region based on the flow rate and quality range of the experiments described in Chapter 6 is shown in each flow regime map. As shown in Figure 4.1 and 4.2, the velocity values indicate that the flow is in the high velocity intermittent flow regime. This flow regime explains why the stratified flow was not formed in the orientation tests described in Chapter 3. If the flow rate were substantially reduced and both the gas and liquid superficial velocities decreased, it would be possible for the flow regime to fall into the stratified flow which would possibly cause non-uniform distribution if the distributor were oriented horizontally.
To study if the flow regime maps are highly sensitive to the input parameters, I generated flow regime maps with different thermal properties, tube diameters, and angles of inclination. Figure 4.3 shows the flow regime map with different thermal property inputs. Figure 4.4 shows the flow regime map where the diameter is that of the feeder tube. In Figure 4.5, the tube is horizontal with the feeder tube diameter. The maps show that the flow regime boundaries vary negligibly as the input parameters change. Also note that there is no stratified flow regime if the flow is vertical. For the operating conditions considered, Figure 4.2 and 4.4 show that there is no significant difference in the flow regime maps for either the flow is in the throttle discharge or feeder tubes. It can be concluded that the flow regime maps can indicate a flow's regime quite well given the flow's gas and liquid superficial velocities.

**Figure 4.3:** Flow regime map for vertically downward flow with $P=129.94$ psia, $T=67^\circ F$, and $D=0.00549$ m (stagnation conditions: $T_o = 85^\circ F$, $P_o = 240$ psia); The crossed-out area indicates the flow operating region.
**Figure 4.4:** Flow regime map for vertically downward flow with \( P = 140.37 \text{ psia}, T = 72 \, ^\circ\text{F}, \) and \( D = 0.00248 \, \text{m} \) (stagnation conditions: \( T_o = 95 \, ^\circ\text{F}, P_o = 240 \, \text{psia} \)); The crossed-out area indicates the flow operating region.

**Figure 4.5:** Flow regime map for horizontal flow with \( P = 140.37 \, \text{psia}, T = 72 \, ^\circ\text{F}, \) and \( D = 0.00248 \, \text{m} \) (stagnation conditions: \( T_o = 95 \, ^\circ\text{F}, P_o = 240 \, \text{psia} \)); The crossed-out area indicates the flow operating region.
Chapter 5

Model Development

Since the predictions of the flow regimes in the throttle discharge indicate that they are in a high velocity intermittent flow regime, a separated flow model like the Baroczy’s model is therefore appropriate.

5.1 Formulation

Figure 5.1 shows a cross-sectional view of the type II distributor assembly. The total pressure drop from the discharge of the throttle to the discharge of the feeder tubes is the same for each feeder tube path. The flow rates in the feeder tubes depend on the tube

![Cross-sectional view of a distributor (Type II) assembly.](image)

Figure 5.1: Cross-sectional view of a distributor (Type II) assembly.
lengths. If they are all of the same length, it can be expected, from symmetry, the same
flow rate in each tube and thus the same sector angle, $\theta_0'$, in the feeder tube inlet plenum
(The sector angles are shown in Figure 5.2 where a, b, c, d, and e denote each of the five
feeder tubes). A shorter feeder tube generates smaller back pressure at the feeder tube inlet
plenum sector and therefore tends to get higher flow rate as a result of the additional flow
from the adjoining tubes which have higher back pressure.

![Figure 5.2: Feeder tube inlet plenum and the sector division.](image)

The mass flow rate at section 1 of the sector is described as

$$\dot{m} = \rho V_1 R_f h_1,$$  \hspace{1cm} (5.1)

and at section 2, the mass flow rate is

$$\dot{m} = 2\pi R_f \rho V_2.$$  \hspace{1cm} (5.2)

Equating (5.1) and (5.2),
The Bernoulli equation between sections 1 and 2 is

\[ P_1 - P_2 = \frac{1}{2} \rho (V_2^2 - V_1^2) \]  \hspace{1cm} (5.4)

Substitute (5.3) into (5.4),

\[ P_1 - P_2 = \frac{1}{2} \rho \left( \frac{V_1^2 \theta R_f^2}{4\pi^2 R_f^2} - V_1^2 \right) \]  \hspace{1cm} (5.5)

At section 3, the mass flow rate is

\[ \dot{m} = \rho V_3 \pi R_f^2 \]  \hspace{1cm} (5.6)

equating (5.1) and (5.6),

\[ V_3 = \frac{V_1 \theta R_f h}{\pi R_f^2} \]  \hspace{1cm} (5.7)

The entrance loss equation at section 3 is

\[ P_2 - P_3 = K_L \frac{1}{2} \rho |V_3| \cdot V_3 \]  \hspace{1cm} (5.8)

Substituting (5.7) into (5.8),

\[ P_2 - P_3 = K_L \frac{1}{2} \rho \frac{|V_1| \cdot V_1 \theta^2 R_f^2 h^2}{\pi^2 R_f^2} \]  \hspace{1cm} (5.9)
Note that in equation (5.8) and (5.9), I use $|V| \cdot V$ instead of $V^2$ in order to retain the sign which indicates the flow direction and how the pressure changes. Applying Baroczy correlation to the feeder tubes,

$$P_3 - P_4 = f_B \left( \frac{L}{2R_f} \right) \frac{1}{2} \rho |V| \cdot V_3 \quad .$$  

(5.10)

Again, substitute (5.7) into (5.10),

$$P_3 - P_4 = f_B \left( \frac{L}{2R_f} \right) \frac{1}{2} \rho \frac{|V_1| \cdot V_1 \theta^2 R_f^2 h^2 \pi^2 R_f^4}{\pi^2 R_f^4} \quad .$$  

(5.11)

Combining (5.5), (5.9), and (5.11), I get a general equation:

$$P_1 - P_4 = \frac{1}{2} \rho \left( \frac{V^2 \theta^2 R_f^2}{4\pi^2 R_f^2} - V^2 \right) + K_L \frac{1}{2} \rho \frac{|V_1| \cdot V_1 \theta^2 R_f^2 h^2 \pi^2 R_f^4}{\pi^2 R_f^4} + f_B \left( \frac{L}{2R_f} \right) \frac{1}{2} \rho \frac{|V_1| \cdot V_1 \theta^2 R_f^2 h^2 \pi^2 R_f^4}{\pi^2 R_f^4} \quad .$$  

(5.12)

Applying equation (5.12) to all the feeder tubes, for tube “a”,

$$P_1 - P_4 = \frac{1}{2} \rho \left( \frac{V^2 \theta^2 R_f^2}{4\pi^2 R_f^2} - V^2 \right) + K_L \frac{1}{2} \rho \frac{|V_1| \cdot V_1 \theta^2 R_f^2 h^2 \pi^2 R_f^4}{\pi^2 R_f^4} + f_B \left( \frac{L_a}{2R_f} \right) \frac{1}{2} \rho \frac{|V_1| \cdot V_1 \theta^2 R_f^2 h^2 \pi^2 R_f^4}{\pi^2 R_f^4} \quad ,$$  

(5.13)

for tube “b”,

$$P_1 - P_4 = \frac{1}{2} \rho \left( \frac{V^2 \theta^2 R_f^2}{4\pi^2 R_f^2} - V^2 \right) + K_L \frac{1}{2} \rho \frac{|V_1| \cdot V_1 \theta^2 R_f^2 h^2 \pi^2 R_f^4}{\pi^2 R_f^4} + f_B \left( \frac{L_b}{2R_f} \right) \frac{1}{2} \rho \frac{|V_1| \cdot V_1 \theta^2 R_f^2 h^2 \pi^2 R_f^4}{\pi^2 R_f^4} \quad ,$$  

(5.14)

for tube “c”,

for tube “c”,
\[ P_1 - P_4 = \frac{1}{2} \rho \left( \frac{V_1^2 \theta_c^2 R_f^2}{4 \pi^2 R_f^2} - V_1^2 \right) + K_L \frac{1}{2} \rho \frac{|V_1| V_1 \theta_c^2 R_f^2 h^2}{\pi^2 R_f^4} + f_{Be} \left( \frac{L_c}{2 R_f} \right) \frac{1}{2} \rho \frac{|V_1| V_1 \theta_c^2 R_f^2 h^2}{\pi^2 R_f^4} , \] (5.15)

for tube “d”,

\[ P_1 - P_4 = \frac{1}{2} \rho \left( \frac{V_1^2 \theta_c^2 R_f^2}{4 \pi^2 R_f^2} - V_1^2 \right) + K_L \frac{1}{2} \rho \frac{|V_1| V_1 \theta_c^2 R_f^2 h^2}{\pi^2 R_f^4} + f_{Be} \left( \frac{L_d}{2 R_f} \right) \frac{1}{2} \rho \frac{|V_1| V_1 \theta_c^2 R_f^2 h^2}{\pi^2 R_f^4} , \] (5.16)

for tube “e”,

\[ P_1 - P_4 = \frac{1}{2} \rho \left( \frac{V_1^2 \theta_c^2 R_f^2}{4 \pi^2 R_f^2} - V_1^2 \right) + K_L \frac{1}{2} \rho \frac{|V_1| V_1 \theta_c^2 R_f^2 h^2}{\pi^2 R_f^4} + f_{Be} \left( \frac{L_e}{2 R_f} \right) \frac{1}{2} \rho \frac{|V_1| V_1 \theta_c^2 R_f^2 h^2}{\pi^2 R_f^4} . \] (5.17)

The angles of sectors add up to 360 degrees, so

\[ \theta_a + \theta_b + \theta_c + \theta_d + \theta_e = 2\pi . \] (5.18)

There are 6 equations and 6 unknowns in this set of simultaneous equations. Unknowns are

\[ \theta_a, \theta_b, \theta_c, \theta_d, \theta_e, (P_1 - P_4) \ . \]

5.2 Limitations

There are a few assumptions involved in this model. Although Henry and Fauske's model and quality iteration show that the predicted pressure at the throttle discharge is about 140 psia, and the pressure drop across feeder tubes is about 50 psi, I assume that the refrigerant mixture density is constant throughout the throttle discharge and feeder tubes. In addition, to simplify the model, I assume that the Baroczy correlation factor is constant in each feeder tube.
The model does not account for the bend losses which occur in the refrigerant stream when it turns into or out of the feeder tube inlet plenum. In addition, this model does not consider the friction pressure drop in the feeder tube inlet plenum. Figure 5.3 shows an arrangement of a set of feeder tubes where tube “a”, “b”, and “e” are of the same length and are shorter than tube “c”, and “d” which are of identical length. The model predicts that tube “a”, “b”, and “e” would all have the same higher flow rate whereas tube “c” and “d” would both have the same, lower flow rate. It can be argued, however, that tube “a”, “b”, and “e” might not actually have the same flow rate because tube “b” and “e” might get more refrigerant flow than tube “a” due to their closer locations to tube “c” and “d”. If this same set of feeder tubes are arranged differently around the circle of the feeder tube inlet plenum as shown in Figure 5.4, the feeder tube “d” might get the highest flow rate this time whereas the model will predict the flow rates only based on the feeder tube length but not the feeder tube arrangement.

Figure 5.3: A feeder tube inlet plenum with a set of feeder tubes.
**Figure 5.4**: A feeder tube inlet plenum with the same set of feeder tubes as shown in Figure 5.3, but with different arrangement.

It is not clear how important the bend losses and the friction are in the inlet plenum until experimental data are taken. For the friction factor, it might turn out to be negligible because the length/hydraulic diameter in the plenum is small. Section 7.1.1 in Chapter 7 will address this question based on the experimental results.
Chapter 6

Experiments and Results

6.1 Test matrix

A test matrix was designed to evaluate a Type II distributor and to examine the flow distribution model (see Table 6.1). In order to keep the number of experiments manageable, the range for the variables of interest were selected.

**refrigerant**: R-22

**distributor mounting orientation**: vertical direction

Since mounting orientation was not important (as discussed in Chapter 3) for conditions of interest, only vertical orientation was used.

**piston (throttle) size**: 0.052 in. and 0.080 in. (diameter)

These two sizes provided two distinct flow rates through the distributor.

**measurement type**:

Two types of measurement were performed with the apparatus.

1. refrigerant flow distribution

2. pressure drops across throttle and feeder tubes

**feeder tube configuration**:

Table 6.2 shows the four configurations selected. The slight difference between the feeder tube sets for distribution and pressure drop measurement was due to the discrepancy in cutting the tubes at the early preparation stage of the experiments.
<table>
<thead>
<tr>
<th>Test run name</th>
<th>piston size (inch)</th>
<th>Feeder tube configuration</th>
<th>Reservoir temperature (°F)</th>
<th>Reservoir pressure (psia)</th>
<th>Degree subcooling (°F)</th>
<th>Main vessel pressure (psia)</th>
<th>Measurement drop</th>
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<td>1.1 99.8</td>
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<td>13.9 103.8</td>
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<td></td>
<td></td>
</tr>
<tr>
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<td>11.7 105.2</td>
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<tr>
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<td></td>
<td></td>
<td>97.9 244.9</td>
<td>13.1 103.1</td>
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<td>-3.4 102.8</td>
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<td>10.6 99.1</td>
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<td>0.6 90.4</td>
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<td>15.2 96.9</td>
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<td>95.7 225.5</td>
<td>9.2 94.0</td>
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<td>7-4</td>
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<td></td>
<td>97.0 245.3</td>
<td>14.3 91.7</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
(i) original package as sent from Carrier. The purpose of this set of experiments was to see if the effect of tube length on distribution could be predicted for an as-provided distributor.

(ii) three adjacent, same-length, long tubes and two adjacent, same-length, short tubes. The purpose of this set of experiments was to see if the distribution could be predicted for large variations in tube length.

(iii) two same-length, long tubes and three same-length, short tubes, with one short tube located between the long tubes. This set of experiments was to examine if the tube layout had an effect.

(iv) One long tube, one short tube, and three same-length, intermediate length tubes. This set of experiments would test whether the effect of an extreme variation in tube length could be predicted.

The comparison between configuration (ii) and (iii) runs can show the effects of friction pressure drop in the feeder tube inlet plenum.

Table 6.2: Feeder tube configuration

<table>
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<tr>
<th>configuration</th>
<th>distribution measurement</th>
<th>pressure drop measurement</th>
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<tr>
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<td>(i)</td>
<td>(ii)</td>
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<td>&quot;a&quot;</td>
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<td>&quot;d&quot;</td>
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</tr>
<tr>
<td>&quot;e&quot;</td>
<td>23.0</td>
<td>11.0</td>
</tr>
</tbody>
</table>

1. The lengths of feeder tubes indicated in this thesis are the total lengths that include the 3-inch long, same-diameter copper tube connected to the end of feeder tubes that channel the flow into the measurement glass containers.
stagnation (reservoir) conditions:

The stagnation conditions of R-22 were set approximately around the nominal operating conditions with intended range of subcooling to be from 0° to 30° F. The reservoir temperature, pressure, and degree of subcooling are average values over the full period or partial period during which the refrigerant was flowing in the distributor. Each value was obtained based on the data plots (details discussed in Data and results section). Some negative subcooling values in Table 6.1 gives indication about the thermocouple’s measurement accuracy and possible R-22 property variation. For the flow model calculation, zero subcooling values were used instead.

receiving (main vessel) pressure:

The nominal receiving pressure is 90 psia. Similarly, the values of the main vessel pressure on Table 6.1 were average values calculated from the data plots.

6.2 Data and results

During the experiment, the data acquisition system was started one or two seconds before the refrigerant began running through the throttle. Thus, the initial, transient, and steady state conditions could be recorded. Appendix F contains all the data graphs plotted for each experiment.

In the reservoir hydrostatic pressure vs. time plots, the liquid refrigerant hydrostatic pressure in the reservoir drops during the experiment. Although there is small amplitude, high frequency noise in the measurements, each plot, however, is a straight line overall. In appendix G, I discuss the methods of determining the mass flow rate by using either the hydrostatic pressure change or stop watch measurement. The methodology of getting the hydrostatic pressure change rate from the experimental data is shown in Appendix H. In addition, Appendix I presents the calculation of the effective cross-sectional area of the
reservoir, which is required as well to determine the mass flow rate. The measured mass flow rates are listed in Table 6.3.

The reservoir temperature vs. time plots show the temperature variation of the liquid refrigerant in the reservoir. In pressure drops across the throttle and feeder tubes vs. time plots, the sharp rise in the pressure drops can be noted when the valve was opened. In some plots, when the pressure drop across the feeder tubes is over the range (32 psi) of the pressure transducer, a horizontal line can be seen.

The reservoir and main vessel pressure vs. time plots show that the reservoir pressure did not hold at the initial pressure during the experiment even though the reservoir was connected to the compressed nitrogen supply. Therefore, average values were taken for the reservoir pressure, temperature, and main vessel pressure. Also note that the initial non-equilibrium transient state shown in the main vessel pressure measurements. The pressure fluctuated a few seconds after the valve was opened and then steadily increased to achieve equilibrium.

To obtain the average values for the reservoir temperature and pressure, main vessel pressure, pressure drop across the throttle and the feeder tubes, I summed up the values in a range and divided it by the total number of data points taken in this range. For example, in experiment E2-20, I selected the range to be from time = 10 seconds to time = 42 seconds to determine the average values of the variables.
<table>
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<th>experiment</th>
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<th>mass flow rate from reservoir hydrostatic pressure plot (lbm/s)</th>
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Chapter 7

Analysis

7.1 Flow distribution

Algorithms based on the flow distribution model were written (see Appendix J). Computer simulation was run to compare the model and the experimental results. The mass flow rate input for the flow distribution model was based on the Henry-Fauske's model. The useful values generated by the simulation and experiments were tabulated in Appendix K.

7.1.1 Measured results vs. model

Figure 7.1 through 7.12 show the comparison between the model and experimental results on the flow distribution for each of the flow distribution tests. In this section, the limitation of the model and the effects of friction pressure drop in feeder tube inlet plenum are also discussed.

Figure 7.1: Experimental and model flow distribution results for E4-1.
Figure 7.2: Experimental and model flow distribution results for E4-2.

Figure 7.3: Experimental and model flow distribution results for E4-3.
Figure 7.4: Experimental and model flow distribution results for E4-4.

Figure 7.5: Experimental and model flow distribution results for E4-5.
**Figure 7.6:** Experimental and model flow distribution results for E4-6.

**Figure 7.7:** Experimental and model flow distribution results for E5-2.
Figure 7.8: Experimental and model flow distribution results for E5-4.

Figure 7.9: Experimental and model flow distribution results for E6-1.
Figure 7.10: Experimental and model flow distribution results for E6-5.

Figure 7.11: Experimental and model flow distribution results for E7-1.
Figure 7.12: Experimental and model flow distribution results for E7-4.

It can be seen that the model and measured results agree within certain accuracy in predicting the flow distribution. Generally, a shorter feeder tube gets more liquid refrigerant due to lower back pressure at the feeder tube inlet plenum.

In experiments E7-1 and E7-4 (Figure 7.11 and 7.12), the shortest tube (tube “e”) did not get as much refrigerant flow as predicted by the model. It is suspected that the flow in the short tube might have choked at the exit as the sharp pressure drop in the feeder tube could cause a second choked plane to form in the flow. Although there is no existing critical flow model that predicts the choked flow in long tubes (the L/D ratio of the feeder tube is too large to be considered to be the short tube in Henry-Fauske model), I used the Henry-Fauske model anyway as a reference to predict the likelihood of the occurrence of the second choked plane. With the temperature, pressure, and quality at the throttle discharge used as the stagnation conditions to Henry-Fauske’s model, the critical pressure
ratio is about 0.7. The implication is that the freedom to control the flow distribution by adjusting the tube length is limited by the onset of choking in the shortest tube. If we allow a lower pressure drop across the throttle, the large pressure drop across the feeder tubes can create a choked plane which will nullify the prediction of the proposed flow distribution model.

If 0.8 is used as the high value for the critical pressure ratio in the feeder tubes, a pressure of more than 1/0.8=1.25 times the absolute evaporator pressure in the feeder tubes will subject the flow to a second choked plane in the feeder tubes. In other words, to safely use the model to predict the flow distribution, the pressure in the feeder tubes should not be more than 25% higher than that of evaporator.

From Figures 7.7 through 7.10, in which the feeder tubes are in configuration (ii) or (iii), the friction pressure drop in the feeder tube inlet plenum has a negligible impact on flow distribution.

7.1.2 Subcooling effect on flow distribution

Figure 7.13 and 7.14 show that the flow distributions are almost the same for different

![Figure 7.13: Experimental flow distribution results with the same throttle size (D_t = 0.080 in) but different stagnation conditions.](image-url)
Figure 7.14: Experimental flow distribution results with the same throttle size ($D_t = 0.052$ in) but different stagnation conditions.

degree of refrigerant subcooling. The flow distribution model also predicts the same result as shown in Figure 7.15 and 7.16. For the range of stagnation conditions run in the experiments, the degree subcooling did not have observable effects on the flow distribution.

Figure 7.15: Model flow distribution results with the same throttle size ($D_t = 0.080$ in) but different stagnation conditions.
Figure 7.16: Model flow distribution results with the same throttle size ($D_t = 0.052$ in) but different stagnation conditions.

7.1.3 Mass flow rate on flow distribution

In Figure 7.17, the flow distribution does not vary much for the two runs which have almost the same stagnation conditions but different flow rates (using different throttle sizes). The flow distribution model also predict the same trend as shown in Figure 7.18.

In Figure 7.19 through 7.21, although the flow distribution vary between runs with different stagnation conditions and flow rates, it can be seen that the flow distribution is essentially dictated by the feeder tube lengths. The corresponding flow distribution model results are shown in Figure 7.22 through 7.24.
Figure 7.17: The experimental flow distribution results for two runs with similar stagnation conditions but different mass flow rates.

Figure 7.18: The model flow distribution results for two runs with similar stagnation conditions but different mass flow rates.
**Figure 7.19:** The experimental flow distribution results for E5-2 and E5-4.

**Figure 7.20:** The experimental flow distribution results for E6-1 and E6-5.
E7-1 (Po=216.0 psia, To=93.5 F, subcooled=8.3 F, mass flow rate=0.26 lbm/s)
E7-4 (Po=245.3 psia, To=96.9 F, subcooled=14.3 F, mass flow rate=0.0911 lbm/s)

E5-2 (Po=241.5 psia, To=97.2 F, subcooled=12.8 F, mass flow rate=0.0821 lbm/s)
E5-4 (Po=215.9 psia, To=98.8 F, subcooled=2.9 F, mass flow rate=0.1409 lbm/s)

**Figure 7.21:** The experimental flow distribution results for E7-1 and E7-4.

**Figure 7.22:** The model flow distribution results for E5-2 and E5-4.
Figure 7.23: The model flow distribution results for E6-1 and E6-5.

Figure 7.24: The model flow distribution results for E7-1 and E7-4.
7.2 Mass flow rate

The mass flow rates in the experiments were determined by two methods: stop watch measurement and reservoir hydrostatic pressure change measurement. Figure 7.25 shows the measurement results from both methods.

![Figure 7.25: Mass flow rates determined by stop watch measurement vs. mass flow rates from reservoir hydrostatic pressure change measurement.](image)

There are small discrepancies between these two measurements. I believe the stop watch measurement is more reliable because the reading at the pressure differential transducer can be affected easily by the formation of condensate at one end of the transducer, especially for the small range of pressure differential it measured. Therefore, the stop watch measured mass flow rate is used to compare to that predicted by Henry-Fauske's model, as shown in Figure 7.26.
It can be noted that Henry-Fauske’s model predicts the mass flow rate quite well for small flow rates. However, it tends to overestimate the flow rate for higher mass flow rates.

### 7.3 Pressure drops across the throttle and feeder tubes

The flow distribution model predicted the pressure drops across the throttle within certain accuracy as shown in Figure 7.27.

Figure 7.28 shows the pressure drops across the feeder tubes. The discrepancies are likely due to the poor measurement taken by one of the pressure transducers used. As it can be seen from the pressure drop across feeder tubes vs. time plots in Appendix F, or...
Figure 7.27: Model predicted and measured pressure drop across the throttle.

Figure 7.28: Model predicted and measured pressure drop across the feeder tubes.
table K.7 in Appendix K, the zero offsets of the transducer varied between -7.7 psi and -0.2 psi. The diaphragm in the differential pressure transducer could have been sticky, causing the zero offset fluctuation.
Chapter 8

Conclusions

The major findings of this research are as follows:

(1) The flow regime, in both the distributor throttle discharge and feeder tubes for all conditions tested, was in the high velocity, intermittent flow regime, as shown in Figures 4.1 through 4.5. The maps also show the velocity level where stratification becomes important.

(2) The approach pipe orientation upstream to the distributor made virtually no difference in the flow distribution for the type II distributor (see Figure 3.5).

(3) Individual distributor tube differences, probably burrs on the inlet, gave rise to variations (as high as 10% from the mean) in the mass flow rates into the feeder tubes (see Figures 3.5 and 3.7 for type II distributor, and Figures B.2, B.4, B.5, B.6, and B.7 for type I distributor).

(4) When the feeder tube pressure drop is greater than 25% of the evaporator inlet pressure, a feeder tube may choke and the use of feeder tube length to control distribution becomes impossible (see Figures 7.11 and 7.12). Note how, as tube “e”, the short tube, gets shorter in proceeding from Figure 7.10 to 7.11 and 7.12, the deviation from the Baroczy theory becomes more severe.

(5) The flow distribution model (based on the Baroczy pressure drop model), used as indicated, does a fine job of predicting the distribution (see Figures 7.1 through 7.10). However, the model has not been proven to give good absolute pressure drop predictions due to
poor pressure drop measurements from one of the differential pressure transducers used (see Figures 7.27 and 7.28).

(6) The Henry-Fauske critical flow model gave good predictions for low flows but poor ones for large flows (see Figure 7.26). This appears to be due to flashing upstream of the distributor which greatly reduces the choked flow values.
References


Appendix A

Nomenclature

\( A \) : area; cross-sectional area
\( a_0 \) : y-axis intercept
\( a_1 \) : slope
\( D \) : diameter
\( e \) : error
\( f \) : friction factor
\( G \) : mass flux
\( h \) : height; feeder tube inlet plenum height; enthalpy
\( j \) : superficial velocity
\( K_L \) : entrance loss coefficient
\( L \) : length
\( \dot{m} \) : mass flow rate
\( N \) : experimental parameter
\( P \) : pressure
\( R \) : radius
\( Re \) : Reynolds number
\( s \) : entropy
\( T \) : temperature
\( V \) : velocity
\( \nu \) : specific volume
\( x \) : quality

\( \eta \) : critical pressure ratio
\( \theta \) : sector angle
\( \mu \) : viscosity
\( \rho \) : density
\( \Omega \) : Baroczy correction factor

\( \bar{2} \) : Baroczy multiplier
subscripts:

1: denotes section 1; area of section 1 = $2\pi R_f h$
2: denotes section 2; area of section 2 = $2\pi R_f h$
3: denotes section 3; area of section 3 = $\pi R_f^2$
4: denotes section 4; area of section 4 = $\pi R_f^2$

a: denotes feeder tube "a"
B: Baroczy model
b: denotes feeder tube "b"
c: denotes feeder tube "c"; critical flow
d: denotes feeder tube "d"
E: equilibrium (corresponding to local static pressure)
e: denotes feeder tube "e"
f: feeder tube
g: gas; vapor
(guess) a guessed value
l: liquid
m: model
o: stagnation conditions
T: throttle discharge
r: throat
v: main vessel
Appendix B

Apparatus Evaluation

Apparatus Consistency Test

To ascertain the experimental apparatus can effectively measure the performance of the distributors, I first ran a set of redundancy experiments to evaluate the apparatus and to gain a preliminary insight of the flow distribution in the distributor assembly. I used a Type I distributor to do the test and it was mounted in the main vessel in an orientation as that shown in Figure 2.4 in Chapter 2. Each feeder tube was labeled and their arrangement with respect to the approach pipe is shown in Figure B.1. The feeder tubes were all 26.5 inches (0.6731 m) long and 0.0975 inches (2.4765e-3 m) in inner diameter (ID).

![Diagram of distributor assembly]

**Figure B.1:** Top view of the distributor assembly

To begin with the experiments, R-22 was conditioned to 95°F and 240 psia in the reservoir while the main vessel was filled with R-22 vapor at 90 psia. Once the valve was opened, the subcooled R-22 entered the approach pipe and then flowed horizontally before
making a 90° turn and entered vertically into the distributor. The refrigerant then flowed spirally and smoothly into the glass containers. No splash, nor leakage of the liquid from the containers were observed. Once the refrigerant liquid entered the glass containers, evaporation occurred immediately. However, the evaporation rate was not fast enough to affect the liquid level readings (the liquid level dropped about an negligible amount of 1/32 inch after all the measurements were made). Therefore, the order of recording the liquid level was not critical.

Three identical runs were done. As shown in Figure B.2, glass “d” consistently had the lowest liquid level, and either glass “a” or “e” received the most liquid. The plot for each run shows a consistent trend in the distribution of the liquid. It is noted that the feeder tubes at the far side of the approach pipe tend to get higher liquid level. The variation of liquid level in each individual glass for different runs was within about 6%.

![Figure B.2](image-url)

**Figure B.2:** Normalized refrigerant liquid level in each glass container with feeder tube “a” pointing straight away from the approach pipe.
**Bias test**

I turned the distributor 180° horizontally and had feeder tubes “a”, “b”, and “e” on the near side of the approach pipe (see Figure B.3). The results show that feeder tube “a” and “e” did not receive the most liquid with this orientation and feeder tube “c”, which was on the far side, had the highest liquid level (see Figure B.4). I repeated the experiments with different arrangements by having different feeder tube pointing straight away from the approach pipe. Figure B.5, B.6, and B.7 show the results respectively. I observed that all the glass containers on the far side (with the exception of “d”) consistently received more liquid than those on near side. Glass “d” received less liquid no matter how the distributor was positioned with respect to the approach pipe.

**Figure B.3:** Top view of distributor assembly after 180 degree turn.
Figure B.4: Normalized refrigerant liquid level in each glass container with feeder tube “a” pointing towards the approach pipe.

Figure B.5: Normalized refrigerant liquid level in each glass container with feeder tube “b” pointing straight away from the approach pipe.
Figure B.6: Normalized refrigerant liquid level in each glass container with feeder tube “d” pointing straight away from the approach pipe.

Figure B.7: Normalized refrigerant liquid level in each glass container with feeder tube “e” pointing straight away from the approach pipe.
**Apparatus Evaluation Conclusion**

The redundancy experiments verified that the test apparatus used is reliable and is capable of indicating the flow distribution of a distributor assembly. Care has to be taken when evaluating the data because the apparatus introduces bias in the measurement with the far-side glasses generally receiving more liquid. The bias is likely due to the non-fully-developed flow as a result of the 90° bend of the pipe right before entering the distributor. However, the bias was only found in type I distributor and the later experiments on type II distributor mounted in the same orientation did not show such bias.

The redundancy experiments also showed that one of the holes in the distributor tested allowed less liquid flow. The disparity might be caused by the distributor fabrication process or defects that were associated with this particular distributor used. However, by looking at all the plots, it can be noted that none of the liquid level variations was larger than 10% from the mean. This small degree of variation indicates that the distributor used in fact did distribute the flow quite uniformly.
Appendix C

Henry and Fauske’s Two-Phase Critical Flow Model (Henry, 1971)

The Henry-Fauske model is developed for the two-phase critical flow of one-component mixtures through convergent nozzles. The model is extended to the orifices and short tubes. It accounts for the nonequilibrium nature of the two-phase critical flow.

The model can predict the critical mass flux based on the stagnation conditions. For subcooled stagnation conditions where quality, \( x_o \), is zero, the square of the critical mass flux is

\[
G_c^2 = \frac{1}{(v_{gE} - v_{lE})} \left| N \int ds \frac{dS}{dP} \right|,
\]

where

\[
N = \frac{x_{Et}}{0.14},
\]

and

\[
x_{Et} = \frac{s_o - s_{lE}}{s_{gE} - s_{lE}}.
\]

The predicted critical pressure ratio, \( \eta_m \), is

\[
\eta_m = 1 - \frac{v_{lE} \cdot G_c^2}{2 \cdot P_o}.
\]
The knowns are the stagnation conditions: $v_o$, $s_o$, and $P_o$. Since equation (C.1) is evaluated at the throat conditions, first the pressure at the throat, $P_{t(guess)}$ has to be guessed. Based on this value, the remaining terms can be evaluated and $\frac{ds_{tg}}{dP}$ can be obtained by interpolation.

Next, equation (C.4) can be evaluated. By comparing $\eta_m$ to $\eta_{(guess)}$, where

$$\eta_{(guess)} = \frac{P_{t(guess)}}{P_o},$$

(C.5)

if the two values are equal, then $G_c$ is the critical mass flux. Otherwise, keep guessing $P_{t(guess)}$ and iterate until $\eta_m$ and $\eta_{(guess)}$ are equal.
Appendix D

Throttle Discharge Quality Iteration

Similar to the iteration in the Henry-Fauske model, an initial guess for the pressure at the throttle discharge, \( P_{T(\text{guess})} \) is required. Since the enthalpy at the discharge remains constant as long as the exit kinetic energy can be neglected (as it can here), we get

\[
x_f = \frac{h_e - h_i}{h_{tg}} \bigg|_{P_{T(\text{guess})}}.
\]  

(D.1)

Next, I use Baroczy correlation (see Appendix E) to formulate the pressure drop across the feeder tubes, \( \Delta P_f \),

\[
\Delta P_f = f_b \left( \frac{L_f}{2R_f} \right)^{\frac{1}{2}} \rho v_f^2 = f \cdot \varphi^2 \cdot \Omega \cdot \left( \frac{L_f}{2R_f} \right)^{\frac{1}{2}} \cdot C_f^2 \cdot v_f
\]

(D.2)

Note that I only use one equation to account for the pressure drop across the feeder tubes even though the feeder tubes can be of different lengths and each can have different mass flux. In order to simplify this preliminary calculation, I believe taking the average value for the feeder tube lengths and mass flux is sufficient. The final model developed in this thesis takes into account the variation in the feeder tube lengths and mass flux. In addition, the model considers the Bernoulli pressure drop in the feeder tube inlet plenum and entrance loss.

In equation (D.2), the feeder tube length is determined by

\[
L_f = \frac{L_u + L_b + L_c + L_d + L_e}{5}.
\]

(D.3)
The feeder tube mass flux is obtained by

\[ G_f = \frac{\dot{m}_f}{A_f} \quad \text{(D.4)} \]

where

\[ \dot{m}_f = \frac{\dot{m}_T}{5} \quad \text{(D.5)} \]

To determine \( f \), I need to calculate the Reynolds number, \( \text{Re} \), in the feeder tubes.

\[ \text{Re}_f = \frac{G_f \cdot (2R_f)}{\mu} \quad \text{(D.6)} \]

where \( \mu \) is the viscosity of the refrigerant evaluated at \( P_{T(\text{guess})} \). With Reynolds number, we can look up Moody Chart to get the friction factor. The two-phase multiplier, \( \Theta_{f_0} \), and correction factor, \( \Omega \), can be obtained from Baroczy correlation charts.

Another assumption is made in equation (D.2). I assume the thermal property changes are small from throttle discharge to the end of feeder tube. That is the reason the viscosity in equation (D.6) is evaluated at throttle discharge conditions. For the same reason, the specific volume of the refrigerant in the feeder tube is obtained by

\[ v_f = v_T = v_l + x \cdot (v_g - v_l) \bigg|_{P_{T(\text{guess})}} \quad \text{(D.7)} \]

The iteration continues until

\[ \Delta P_f = P_{T(\text{guess})} - P_v \quad \text{(D.8)} \]

where \( P_v \) is the pressure in the main vessel (which simulates the conditions of the evaporator inlet).
Appendix E

Baroczy Correlation Model (Wallis, 1969)

As shown in equation (E.1), the Baroczy model uses a correlation factor -- the Baroczy correlation factor, $f_B$, to replace the friction factor in a single-phase pressure drop equation.

\[ \Delta P = f_B \left( \frac{L}{D} \right) \frac{1}{2} \rho V^2 \]  

(E.1)

\[ \rho \] is the density of the two-phase mixture and is obtained by taking the inverse of the mixture’s specific volume. The specific volume is expressed as,

\[ v = v_i + x \cdot (v_g - v_i) \]  

(E.2)

\[ V \] is the velocity of the two-phase mixture. It can be determined by

\[ V = \frac{m}{\rho \cdot A} \]  

(E.3)

The Baroczy correlation factor, $f_B$, is a product of single-phase Moody chart friction factor, $f$, two-phase multiplier, $\Omega^2$, and correction factor, $\Omega$, i.e.

\[ f_B = f \cdot \Omega^2 \cdot \Omega \]  

(E.4)

Figure E.1 shows the chart for getting the two-phase multiplier values for $G = 10E6 \text{ lb}/(\text{hr})/(\text{ft}^2)$. The two-phase multiplier is expressed as a function of property index

\[ \frac{\mu_f}{\mu_g}^{0.2} \cdot \frac{\rho_f}{\rho_g} \] (which is denoted as \( (\mu_f/\mu_g)^{0.2}/(\rho_f/\rho_g) \) in Wallis). When $G =$
10E6 lb/(hr)/(ft²), the correction factor is not necessary and thus is equal to 1. Figure E.2 shows the chart for correction factor values for other mass flux values.

**Figure E.1**: Baroczy's correlation of two-phase multiplier for $G = 10^6 \text{ lb/hr-ft}^2$ (Wallis, 1969).
Figure E.2: Correction factor versus property index for Baroczy correlation (Wallis, 1969).
Appendix F

Experimental Result Plots

Experiment E2-20

Figure F.1: Reservoir hydrostatic pressure vs. time for E2-20.

Figure F.2: Reservoir temperature vs. time for E2-20.
Figure F.3: Pressure drops across the throttle and feeder tubes vs. time for E2-20.

Figure F.4: Reservoir and main vessel pressure vs. time for E2-20.
Experiment E3-1

Figure F.5: Reservoir hydrostatic pressure vs. time for E3-1.

Figure F.6: Reservoir temperature vs. time for E3-1.
Figure F.7: Pressure drops across the throttle and feeder tubes vs. time for E3-1.

Figure F.8: Reservoir and main vessel pressure vs. time for E3-1.
Experiment E3-2

Figure F.9: Reservoir hydrostatic pressure vs. time for E3-2.

Figure F.10: Reservoir temperature vs. time for E3-2.
Figure F.11: Pressure drops across the throttle and feeder tubes vs. time for E3-2.

Figure F.12: Reservoir and main vessel pressure vs. time for E3-2.
Experiment E3-5

**Figure F.13:** Reservoir hydrostatic pressure vs. time for E3-5.

**Figure F.14:** Reservoir temperature vs. time for E3-5.
Figure F.15: Pressure drops across the throttle and feeder tubes vs. time for E3-5.

Figure F.16: Reservoir and main vessel pressure vs. time for E3-5.
Experiment E3-6

Figure F.17: Reservoir hydrostatic pressure vs. time for E3-6.

Figure F.18: Reservoir temperature vs. time for E3-6.
**Figure F.19:** Pressure drops across the throttle and feeder tubes vs. time for E3-6.

**Figure F.20:** Reservoir and main vessel pressure vs. time for E3-6.
Experiment E3-7

**Figure F.21:** Reservoir hydrostatic pressure vs. time for E3-7.

**Figure F.22:** Reservoir temperature vs. time for E3-7.
Figure F.23: Pressure drops across the throttle and feeder tubes vs. time for E3-7.

Figure F.24: Reservoir and main vessel pressure vs. time for E3-7.
Experiment E4-1

**Figure F.25**: Reservoir hydrostatic pressure vs. time for E4-1.

**Figure F.26**: Reservoir temperature vs. time for E4-1.
Figure F.27: Reservoir and main vessel pressure vs. time for E4-1.

Figure F.28: Flow distribution for E4-1.
Experiment E4-2

**Figure F.29:** Reservoir hydrostatic pressure vs. time for E4-2.

**Figure F.30:** Reservoir temperature vs. time for E4-2.
Figure F.31: Reservoir and main vessel pressure vs. time for E4-2.

Figure F.32: Flow distribution for E4-2.
Experiment E4-3

Figure F.33: Reservoir hydrostatic pressure vs. time for E4-3.

Figure F.34: Reservoir temperature vs. time for E4-3.
Figure F.35: Reservoir and main vessel pressure vs. time for E4-3.

Figure F.36: Flow distribution for E4-3.
Experiment E4-4

Figure F.37: Reservoir hydrostatic pressure vs. time for E4-4.

Figure F.38: Reservoir temperature vs. time for E4-4.
Figure F.39: Reservoir and main vessel pressure vs. time for E4-4.

Figure F.40: Flow distribution for E4-4.
Experiment E4-5

Figure F.41: Reservoir hydrostatic pressure vs. time for E4-5.

Figure F.42: Reservoir temperature vs. time for E4-5.
**Figure F.43:** Reservoir and main vessel pressure vs. time for E4-5.

**Figure F.44:** Flow distribution for E4-5.
Experiment E4-6

Figure F.45: Reservoir hydrostatic pressure vs. time for E4-6.

Figure F.46: Reservoir temperature vs. time for E4-6.
Figure F.47: Reservoir and main vessel pressure vs. time for E4-6.

Figure F.48: Flow distribution for E4-6.
Experiment E5-1

Figure F.49: Reservoir hydrostatic pressure vs. time for E5-1.

Figure F.50: Reservoir temperature vs. time for E5-1.
Figure F.51: Pressure drops across the throttle and feeder tubes vs. time for E5-1.

Figure F.52: Reservoir and main vessel pressure vs. time for E5-1.
Experiment E5-2

Figure F.53: Reservoir hydrostatic pressure vs. time for E5-2.

Figure F.54: Reservoir temperature vs. time for E5-2.
Figure F.55: Reservoir and main vessel pressure vs. time for E5-2.

Figure F.56: Flow distribution for E5-2.
Experiment E5-3

**Figure F.57:** Reservoir hydrostatic pressure vs. time for E5-3.

**Figure F.58:** Reservoir temperature vs. time for E5-3.
Figure F.59: Pressure drops across the throttle and feeder tubes vs. time for E5-3.

Figure F.60: Reservoir and main vessel pressure vs. time for E5-3.
Experiment E5-4

Figure F.61: Reservoir hydrostatic pressure vs. time for E5-4.

Figure F.62: Reservoir temperature vs. time for E5-4.
Figure F.63: Reservoir and main vessel pressure vs. time for E5-4.

Figure F.64: Flow distribution for E5-4.
Experiment E6-1

Figure F.65: Reservoir hydrostatic pressure vs. time for E6-1.

Figure F.66: Reservoir temperature vs. time for E6-1.
**Figure F.67:** Reservoir and main vessel pressure vs. time for E6-1.

**Figure F.68:** Flow distribution for E6-1.
Experiment E6-3

Figure F.69: Reservoir hydrostatic pressure vs. time for E6-3.

Figure F.70: Reservoir temperature vs. time for E6-3.
Figure F.71: Pressure drops across the throttle and feeder tubes vs. time for E6-3.

Figure F.72: Reservoir and main vessel pressure vs. time for E6-3.
Experiment E6-4

Figure F.73: Reservoir hydrostatic pressure vs. time for E6-4.

Figure F.74: Reservoir temperature vs. time for E6-4.
**Figure F.75**: Pressure drops across the throttle and feeder tubes vs. time for E6-4.

**Figure F.76**: Reservoir and main vessel pressure vs. time for E6-4.
Experiment E6-5

Figure F.77: Reservoir hydrostatic pressure vs. time for E6-5.

Figure F.78: Reservoir temperature vs. time for E6-5.
**Figure F.79:** Reservoir and main vessel pressure vs. time for E6-5.

**Figure F.80:** Flow distribution for E6-5.
Experiment E7-1

Figure F.81: Reservoir hydrostatic pressure vs. time for E7-1.

Figure F.82: Reservoir temperature vs. time for E7-1.
**Figure F.83:** Reservoir and main vessel pressure vs. time for E7-1.

**Figure F.84:** Flow distribution for E7-1.
Experiment E7-2

Figure F.85: Reservoir hydrostatic pressure vs. time for E7-2.

Figure F.86: Reservoir temperature vs. time for E7-2.
Figure F.87: Pressure drops across the throttle and feeder tubes vs. time for E7-2.

Figure F.88: Reservoir and main vessel pressure vs. time for E7-2.
Experiment E7-3

Figure F.89: Reservoir hydrostatic pressure vs. time for E7-3.

Figure F.90: Reservoir temperature vs. time for E7-3.
Figure F.91: Pressure drops across the throttle and feeder tubes vs. time for E7-3.

Figure F.92: Reservoir and main vessel pressure vs. time for E7-3.
Experiment E7-4

Figure F.93: Reservoir hydrostatic pressure vs. time for E7-4.

Figure F.94: Reservoir temperature vs. time for E7-4.
Figure F.95: Reservoir and main vessel pressure vs. time for E7-4.

Figure F.96: Flow distribution for E7-4.
Appendix G

Calculations of Mass Flow Rate

The mass flow rate, \( \dot{m} \), of a liquid flowing out of a container can be described as

\[
\dot{m} = \rho \cdot A \cdot \frac{dh}{dt}
\]  

(G.1)

where \( \rho \) is the liquid density, \( A \) is the cross-sectional area, and \( \frac{dh}{dt} \) is the rate of change of liquid level in the container.

We can use the liquid hydrostatic pressure measurement to determine \( \frac{dh}{dt} \). Since

\[
P = \rho \cdot g \cdot h
\]  

(G.2)

by differentiating it with respect of time, we get

\[
\frac{dP}{dt} = \rho \cdot g \cdot \frac{dh}{dt}
\]  

(G.3)

Thus, once we know the rate of hydrostatic pressure change, we can determine the mass flow rate. Appendix H shows the calculation to get \( \frac{dP}{dt} \) from the experimental data. The calculation for the reservoir’s cross-sectional area is discussed in Appendix I.

Another method of getting \( \frac{dh}{dt} \) is by using a stop watch or any time-measuring device to record the liquid level change rate.
Slope Determination through Least-Squares Regression (Chapra, 1988)

As discussed in Appendix G, we are interested in determining the slope, $\frac{dP}{dt}$, in reservoir hydrostatic pressure vs. time plots. To fit a line to the plots, I use Least-Squares Regression. Since I expect the line to be a straight line, linear regression, which fits a straight line to a set of paired data: $(x_1, y_1), (x_2, y_2), \ldots, (x_n, y_n)$, is used. The mathematical expression for the straight line is

$$y = a_0 + a_1 x + e$$  \hspace{1cm} (H.1)

where $a_0$ is the intercept at the y-axis, $a_1$ is the slope of the straight line, and $e$ is the error.

By linear regression approximation, the slope, $a_1$, is

$$a_1 = \frac{n \sum x_i y_i - \sum x_i \sum y_i}{n \sum x_i^2 - (\sum x_i)^2}$$  \hspace{1cm} (H.2)

and the summations are from $i = 1$ to $n$.

The y-axis intercept, $a_0$, is given by

$$a_0 = \bar{y} - a_1 \bar{x}$$  \hspace{1cm} (H.3)

where $\bar{y}$ and $\bar{x}$ are the means of $y$ and $x$, respectively.

The error, $e$, which is the discrepancy between the true value of $y$ and the approximate value, is expressed as
\[ e = y - a_n - a_1 x \]  

(H.4)
Appendix I

Reservoir Cross-Sectional Area Calculation

The coils in the reservoir complicate mass flow rate calculation in equation (G-1) since the cross-sectional area is not uniform throughout the height. In order to simplify the calculation, I approximate the reservoir with another constant cross-sectional area cylinder.

The coils occupy the space in the reservoir up to a height of 7.25 inches. The outer diameter of the coils is 0.375 inches. Therefore, if all the coils are packed adjacently without space in between, there will be approximately

\[
\frac{7.25}{0.375} \approx 19 \text{ coils}
\]

in the reservoir. Assume the spacing between the coils is about the size of the coil diameter, then there are approximately 10 coils.

The volume occupied by a single coil is approximated by

\[
(\text{coil cross-sectional area}) \cdot 2 \cdot \pi \cdot (\text{coil radius})
\]

\[
= \pi \cdot \left(\frac{0.375}{2}\right)^2 \cdot 2 \cdot \pi \cdot \left(\frac{5.25}{2}\right)
\]

\[
= 1.8216 \text{ in}^3.
\]

Therefore the total volume taken by the 10 coils is 18.216 in$^3$.

The total volume of liquid the reservoir can take up to a height of 7.25 inch is therefore

\[
(\pi \cdot (4)^2 \cdot 7.25) - 18.216 = 346.21 \text{ in}^3
\]

and thus the constant area cylinder which approximates the reservoir has a cross-sectional area of

\[
\frac{346.21}{7.25} = 47.75 \text{ in}^2.
\]
Appendix J

Computer Simulation and Codes

All computer programs in this thesis were written and run in MATLAB. The R-22 property table used was generated by NIST's database program (Ely 1989). In order to read the values from the Baroczy correlation multiplier and correction factor figures, I took data points on the figures and put them in matrix form. Two-dimensional interpolation was used to get the values from the matrices. The friction factor in a pipe was evaluated by using Colebrook formula (Munson, 1990).

```% Program: Integrated Henry-Fauske and Flow Distribution Model
% written by,  
% Sheit Sheng Chen  
% Massachusetts Institute of Technology  
% 1995

% This program calculates the mass flow rate based on  
% Henry-Fauske model. The flow rate is then used as an input  
% to the flow distribution model to predict the mass flow  
% rate in each feeder tube and pressure drop across  
% the feeder tubes.

% external files used:  
% R22 : R22 property table  
% M : Baroczy correlation multiplier table  
% C2 : Baroczy correlation correction factor table for mass  
% flux = 2e6 lbm/(hr*ft^2)  
% C3 : Baroczy correlation correction factor table for mass  
% flux = 3e6 lbm/(hr*ft^2)

% external functions used:  
% eval_f_factor() : evaluates the friction factor  
% eval_Omega() : evaluates the correction factor in Baroczy correlation  
% eval_Phi() : evaluates the multiplier in Baroczy correlation  
% solve_model() : solves the 6-equation, 6-unknowns refrigerant flow distribution model

% variables:  
% Dt : diameter of throttle (inch)
```
% f : friction factor in feeder tube
% FBa : Baroczy friction factor for feeder tube "a";
%   FBa = f*Phi*Omega at a
% FBb : Baroczy friction factor for feeder tube "b";
%   FBb = f*Phi*Omega at b
% FBc : Baroczy friction factor for feeder tube "c";
%   FBc = f*Phi*Omega at c
% FBd : Baroczy friction factor for feeder tube "d";
%   FBd = f*Phi*Omega at d
% FBf : Baroczy friction factor for feeder tube "e";
%   FBf = f*Phi*Omega at e
% G_SI : average mass flux in feeder tube in SI unit (kg/m^2/s)
% G_eg : average mass flux in feeder tube in British
%   unit (lbm/hr/ft^2)
% hh : height of the distributor feeder tube inlet plenum (meter)
% hg : enthalpy of vapor R-22 (Btu/lbm)
% hl : enthalpy of liquid R-22 (Btu/lbm)
% hlg : guessed liquid enthalpy (Btu/lbm)
% hlo : stagnation liquid enthalpy (Btu/lbm)
% K : entrance loss factor
% La : feeder tube length for tube "a" (meter)
% Lb : feeder tube length for tube "b" (meter)
% Lc : feeder tube length for tube "c" (meter)
% Ld : feeder tube length for tube "d" (meter)
% Le : feeder tube length for tube "e" (meter)
% mdot : total mass flow rate (lbm/s)
% mdot_a : mass flow rate in feeder tube "a" (kg/s)
% mdot_b : mass flow rate in feeder tube "b" (kg/s)
% mdot_c : mass flow rate in feeder tube "c" (kg/s)
% mdot_d : mass flow rate in feeder tube "d" (kg/s)
% mdot_e : mass flow rate in feeder tube "e" (kg/s)
% mdot_SI : total mass flow rate in SI unit (kg/s)
% mdotf : average mass flow rate in a feeder tube (kg/s)
% Mu : viscosity of liquid R-22 (N*s/m^2)
% Omega : Baroczy correction factor
% P : pressure of R-22 (psia)
% P12_a : pressure drop from section 1 to 2 in feeder tube "a"
% P23_a : pressure drop from section 2 to 3 in feeder tube "a"
% P34_a : pressure drop from section 3 to 4 in feeder tube "a"
% Pdrop_SI : model predicted pressure drop across the feeder
%   tubes in SI unit (Pa)
% Pdrop_eg : model predicted pressure drop across the feeder
%   tubes in British unit (psi)
% Phi : Baroczy multiplier
% Po : stagnation pressure (psia)
% Po_index : row number of Po in R-22 property table file
clear;

% input variables

Dt = 0.080; % throat diameter
Po = 226.2; % stagnation pressure
To = 103.9; % stagnation temperature
Pv = 97.38; % main vessel pressure
La = 0.65786; % feeder tube "a" length
Lb = 0.65786; % feeder tube "b" length
Lc = 0.5715; % feeder tube "c" length
Ld = 0.50292; % feeder tube "d" length
Le = 0.50292; % feeder tube "e" length
Le = 0.50292; % feeder tube "e" length

RT = 0.0027432; % throttle discharge radius
Rf = 0.00124; % feeder tube radius
hh = 0.001; % feeder tube inlet plenum height
K = 0.8; % entrance loss factor

L_avg = (La+Lb+Lc+Ld+Le)/5; % average feeder tube length

if Po <= Pv
    error('stagnation pressure is lower or equal to vessel pressure')
end

% initialization

global C2 C3 M;
load R22;
load M;
load C2;
load C3;

% Initialize variables

T = R22(:,2); % temperature
rho_l = R22(:,6); % density (liquid)
P = R22(:,3); % pressure
hl = R22(:,8); % enthalpy (liquid)
hg = R22(:,9); % enthalpy (vapor)
Sl = R22(:,10); % entropy (liquid)
Sg = R22(:,11); % entropy (vapor)
v1 = R22(:,4); % specific volume (liquid)
vg = R22(:,5); % specific volume (vapor)
Mu = R22(:,12)*1e-7; % viscosity

vlo = interpl(T,v1,To); % interpolate to get stagnation specific volume
Slo = interpl(T,Sl,To); % interpolate to get stagnation entropy
hlo = interpl(T,hl,To); % interpolate to get stagnation enthalpy

%********************************************************************%
% PART 1: Henry-Fauske's model:
% predict the mass flow rate
%********************************************************************

if Po > 274.70
error('stagnation pressure is out of range on the R22\n property table file')
elseif Po > 250
    Po_start_search_index = 726;
    Po_stop_search_index = 800;
elseif Po > 230
    Po_start_search_index = 663;
    Po_stop_search_index = 727;
elseif Po > 200
    Po_start_search_index = 561;
    Po_stop_search_index = 664;
elseif Po > 180
    Po_start_search_index = 487;
    Po_stop_search_index = 562;
else
    Po_start_search_index = 1;
    Po_stop_search_index = 487;
end

if Pv < 83.61
    error('main vessel pressure is out of range on the R22\n property table file')
elseif Pv < 90
    Pv_start_search_index = 2;
    Pv_stop_search_index = 45;
elseif Pv < 100
    Pv_start_search_index = 44;
    Pv_stop_search_index = 107;
elseif Pv < 110
    Pv_start_search_index = 107;
    Pv_stop_search_index = 165;
elseif Pv < 120
    Pv_start_search_index = 165;
    Pv_stop_search_index = 219;
else
    Pv_start_search_index = 218;
    Pv_stop_search_index = Po_start_search_index;
end

for i = Po_start_search_index:Po_stop_search_index
    P_Po_difference(i) = abs(P(i)-Po);
end

[num, Po_relative_index] = min(P_Po_difference(Po_start_search_index:Po_stop_search_index));
Po_index = Po_relative_index + Po_start_search_index - 1;
if Po_index > 599 % to avoid T_gs > 100F since the range of Phi
    Po_index = 599 % is from 40 to 100F in M file
end

for i = Pv_start_search_index:Pv_stop_search_index
    P_Pv_difference(i) = abs(P(i)-Pv);
end
[num, Pv_relatv_index] = min(P_Pv_difference(Pv_start_search_index:Pv_stop_search_index));
Pv_index = Pv_relatv_index + Pv_start_search_index - 1;

% 1.2 Henry-Fauske model iteration: select the critical mass
% flux where the difference between the guessed and model
% predicted critical pressure ratio is the smallest.

for i= Pv_index:Po_index
  derivative(i) = (Sl(i+1) - Sl(i-1))/(P(i+1) - P(i-1));
  x(i) = (Slo - S1(i))/(Sg(i) - Sl(i));
  N(i) = x(i)/0.14;
  Gsquare(i) = 1/((vg(i) - vlo)*N(i)*derivative(i)/(Sg(i) - Sl(i)));
  Pratioguess(i) = P(i)/Po;
  Pratio(i) = 1 - vlo*Gsquare(i)/(2*Po);
  Pdifference(i) = abs(Pratioguess(i) - Pratio(i));
end

[minvalue, Pt_relatv_index] = min(Pdifference(Pv_index:Po_index));
Pt_index = Pt_relatv_index + Pv_index - 1;

% 1.3 Output the mass flow rate and related variables to a file

diary OUTPUT.txt

fprintf('Henry-Fauske Model result\n')
fprintf('Po = %g.\n', Po)
fprintf('To = %g.\n', To)
fprintf('Pv = %g.\n', Pv)
fprintf('throat pressure is %g.\n', P(Pt_index))
mdot = (sqrt(Gsquare(Pt_index)*0.2234))*pi*(Dt/2)^2
fprintf('derivative is %g.\n',derivative(Pt_index))
fprintf('x is %g.\n', x(Pt_index))
fprintf('N is %g.\n', N(Pt_index))
fprintf('Gsquare is %g.\n',Gsquare(Pt_index))
fprintf('Pratioguess is %g.\n',Pratioguess(Pt_index))
fprintf('Pratio is %g.\n',Pratio(Pt_index))
mdot_SI = mdot*0.4526 % mass flow rate in SI unit, i.e. kg/s

diary off

% PART 2: Quality Iteration at Throttle Discharge

% 2.1 Initialize the relevant variables

i= Pv_index:Po_index;
hl_gs = hl(i);
hlg_gs = hg(i) - hl(i);
v1_gs = v1(i);
vgs = vg(i);
Mu_gs = Mu(i);
T_gs = interp1(P,T,P(i));
dP_gs = P(i)-Pv;
x_gs = (hlo-hl_gs)./(hlg_gs);
mdotf = (mdot/5)*0.4536;
G_SI = mdotf/(pi*(Rf)^2);
G_eg = G_SI*737.5;
if G_eg > 3.5e6
    fprintf('Warning: G_eg is too big')
end
vgs = (v1_gs + x_gs.*(vggs-vl_gs)).*1/16.0185;
Re_gs = G_SI*2*Rf./Mu_gs;

% 2.2 Evaluate the friction factor, multiplier, and correction factor
%------------------------------------------

j = zeros(1,Po_index-Pv_index+1); % preallocate the vectors to make
f = zeros((Po_index-Pv_index+1),1); % the for loop go faster
f((Po_index-Pv_index+1),1)=1000;
Phi((Po_index-Pv_index+1),1)=1000;
Omega((Po_index-Pv_index+1),1)=1000;
for j= 1:(Po_index-Pv_index+1)
    if x_gs(j) < 0.001
        break;
    else
        f(j,1) = evalfjfactor(Re_gs(j));
        Phi(j,1) = evalPhi(T_gs(j),x_gs(j));
        Omega(j,1) = evalOmega(T_gs(j),x_gs(j),G_eg);
    end
end

% 2.3 select the quality that gives the smallest difference
% between the guessed and Baroczy model predicted pressure
% drop across the feeder tubes
%------------------------------------------

dP_Baroczy = f.*Phi.*Omega.*L_avg.*(G_SI^2).*vgs.*((0.145/1000)\ /
        (2*2*Rf)); % 1 kPa = 0.145 psi
dPdifference= abs(dP_gs - dP_Baroczy);

[min_value, PT_relatv_index] = min(dPdifference);
PT_index = PT_relatv_index + Pv_index - 1;
PT = P(PT_index);
x_T = x_gs(PT_relatv_index);

%*****************************
% PART 3: Flow Distribution Model
%*****************************

% 3.1 Initialize with the values calculated in the quality iteration

% Define variables

v_T = v_gs(PT_relatv_index);
Re_T = Regs(PT_relatv_index);
dP_T = dP_Baroczy(PT_relatv_index);
dP_difference_T = dP_difference(PT_relatv_index);
Vl = (mdot*0.4536/(2*pi*RT*hh))*v_T;
rho = l/v_T;

FBA = f(PT_relatv_index)*Phi(PT_relatv_index)*Omega(PT_relatv_index);
FBB = FBA;
FBC = FBA;
FBD = FBA;
FBE = FBA;

dumvar1 = (Vl*RT/(2*pi*Rf))^2;  % define dummy variable 1

dumvar2 = (Vl*RT*hh/(pi*Rf^2))^2;  % define dummy variable 2

term1a = 0.5*rho*(dumvar1 + K*dumvar2 + FBA*(La/(2*Rf))*dumvar2);
term1b = 0.5*rho*(dumvar1 + K*dumvar2 + FBB*(Lb/(2*Rf))*dumvar2);
term1c = 0.5*rho*(dumvar1 + K*dumvar2 + FBC*(Lc/(2*Rf))*dumvar2);
term1d = 0.5*rho*(dumvar1 + K*dumvar2 + FBD*(Ld/(2*Rf))*dumvar2);
term1e = 0.5*rho*(dumvar1 + K*dumvar2 + FBE*(Le/(2*Rf))*dumvar2);
term2 = 0.5*rho*Vl^2;

% 3.2 Call the function that solves the flow distribution model equations

[dP_a, Ta, Tb, Tc, Td, Te]=solve_model(term1a, term1b, term1c, term1d, term1e, term2);

% 3.3 send the results to the output file

mdot_a = rho*Vl*RT*hh*Ta
mdot_b = rho*Vl*RT*hh*Tb
mdot_c = rho*Vl*RT*hh*Tc
mdot_d = rho*Vl*RT*hh*Td
mdot_e = rho*Vl*RT*hh*Te
\( P_{12_a} = 0.5 \cdot \rho \cdot (\text{dumvar1} \cdot T_a ^2 - (V_1) ^2) \)
\( P_{23_a} = 0.5 \cdot K \cdot \rho \cdot \text{dumvar2} \cdot T_a ^2 \)
\( P_{34_a} = \text{FBa} \cdot (L_a/(2 \cdot R_f)) \cdot \rho \cdot \text{dumvar2} \cdot 0.5 \cdot T_a ^2 \)

\( P_{\text{dropSI}} = P_{12_a} + P_{23_a} + P_{34_a} \)

\( P_{\text{drop_eg}} = (P_{\text{dropSI}}/1000) \times .145 \) \( \% \) pressure drop across feeder\tube in psi.

\( P_{\text{drop_eg}} = (P_{\text{dropSI}}/1000) \times .145 \)

\( P_T_m = P_{\text{drop_eg}} + P_v \)

\% pressure at throttle discharge\predicted by flow distribution model.

diary off

External functions called from the main program:

function \( y = \text{eval}_f\_\text{factor}(x) \)

\% This function returns the friction factor for a given Reynolds
\% number in a smooth pipe (roughness = 0) based on Colebrook
\% Formula which is valid for the entire nonlaminar range
\% of the Moody chart.
\% written by Sheit Sheng Chen, MIT, 1995.
\% input: Reynolds number
\% output: friction factor
\%---------------------------------------------------------------------
\% \( f_{\text{lower_bound}} = 0.014 \); \% this is the range of interest
\% \( f_{\text{upper_bound}} = 0.030 \);
\% \( \text{Re}_{\text{lower_bound}} = 1e4 \);
\% \( \text{Re}_{\text{upper_bound}} = 3e5 \);

\( \text{Re} = x; \)

if \( \text{Re} < 1e4 \mid \text{Re} > 3e5 \)
    error('Reynolds number is out of range')
end

\( f = f_{\text{lower_bound}}:0.0002:f_{\text{upper_bound}}; \)
\( \text{difference} = \text{abs}(\text{sqrt}(f) + 2. \cdot \log10((2.51)/(\text{Re} \cdot \text{sqrt}(f)))); \)

[\( \text{num} \), \( f_{\text{relatv_index}} \)] = min(difference);
\( f_{\text{factor}} = 0.014 + (f_{\text{relatv_index}} - 1) \times 0.0002; \)
\( y = f_{\text{factor}}; \)

function \( y = \text{eval}_\text{Omega}(x1,x2,x3) \)

\% This function evaluates the correction factor in the Baroczy's
\% correlation.
\% written by Sheit Sheng Chen, MIT, 1995.
% inputs:  x1:  temperature (degree F)
%        x2:  quality
%        x3:  mass flux (lbm/(hr*ft^2))
% output:  y:  correction factor
% The correction factor tables C2 (For mass flux = 2e6 lbm/(hr*ft^2))
%, and C3 (For mass flux = 3e6 lbm/(hr*ft^2) ) have to be placed
% in the same directory as eval_Omega.m

\begin{verbatim}
T = x1;
x = x2;
G_e = x3;

G_e_lower_bound = 0.75e6;
G_e_upper_bound = 4.00e6;

if G_e < G_e_lower_bound | G_e > G_e_upper_bound
    error('G_e is out of bound')
end

if 0.75e6 < G_e & G_e <= 1.5e6
    Omega = 1;
y = Omega;
    return;
elseif 1.5e6 < G_e & G_e <= 2.5e6
    tab = C2;
else
    tab = C3;
end

T_array = tab(:,1);
[row_num,column_num] = size(tab);

x_lower_bound = tab(1,2);
x_upper_bound = tab(1,column_num);
T_lower_bound = tab(2,1);
T_upper_bound = tab(row_num,1);

if T < T_lower_bound | T > T_upper_bound
    error('correction factor chart temperature out of range')
elsif x < x_lower_bound | x > x_upper_bound
    error('correction factor chart quality out of range')
end

i = zeros(1,row_num);  % preallocate the vectors to make the
T_array_diff = zeros(1,row_num);  % for loop go faster

% perform 2-D interpolation

for i = 2:row_num
    T_array_diff(i) = T - T_array(i);
    if T_array_diff(i) == 0
        Tmark1 = i;
        Tmark2 = i + 1;
        break;
    \end{verbatim}
```matlab
elseif T_array_diff(i) < 0
    Tmark1 = i - 1;
    Tmark2 = i;
    break;
end

Omega_1 = interp1(tab(1,:),tab(Tmark1,:),x);
Omega_2 = interp1(tab(1,:),tab(Tmark2,:),x);
Omega_interpr = [Omega_1, Omega_2];
T_interpr = [tab(Tmark1,1), tab(Tmark2,1)];
Omega = interp1(T_interpr,Omega_interpr,T);
y = Omega;

function y = eval_Phi(xl,x2)
% This function evaluates the multiplier in the Baroczy’s correlation.
% written by Sheit Sheng Chen, MIT, 1995.
% inputs:   x1: temperature (degree F)
%            y: multiplier
% output:   y: multiplier
% The Multiplier table M (For mass flux = 1e6 lbm/(hr*ft^2)) has
% to be placed in the same directory as eval_Phi.m
% T_array_diff = zeros(1,row_num); % preallocate the vectors to make the for loop go faster
% perform 2-D interpolation
```

---

The document contains code snippets that involve interpolation functions and a function `eval_Phi` for evaluating multipliers. The code includes conditional checks and array manipulations typical of numerical analysis or computational physics tasks. The `eval_Phi` function requires temperature and quality inputs and outputs a multiplier based on a multiplier table placed in the same directory as the script.
for i = 2:row_num
    T_array_diff(i) = T - T_array(i);
    if T_array_diff(i) == 0
        Tmark1 = i;
        Tmark2 = i + 1;
        break;
    elseif T_array_diff(i) < 0
        Tmark1 = i - 1;
        Tmark2 = i;
        break;
    end
end

Phi_1 = interp1(M(1,:),M(Tmark1,:),x);
Phi_2 = interp1(M(1,:),M(Tmark2,:),x);
Phi_interpr = [Phi_1, Phi_2];
T_interpr = [M(Tmark1,1), M(Tmark2,1)];
Phi = interp1(T_interpr,Phi_interpr,T);

y = Phi;

function [u,v,w,x,y,z] = solve_model(x1,x2,x3,x4,x5,x6)
%---------------------------------------------------------------
% This function solves the 6-equation, 6-unknowns refrigerant flow
% distribution model
% written by Sheit Sheng Chen, MIT, 1995.
%---------------------------------------------------------------
mpa('term1a',int2str(x1));
mpa('term1b',int2str(x2));
mpa('term1c',int2str(x3));
mpa('term1d',int2str(x4));
mpa('term1e',int2str(x5));
mpa('term1f',int2str(x6));
eqn1 = 'Ta + Tb + Tc + Td + Te - 2*pi ';
eqn2 = 'dP - term1a*(abs(Ta)*Ta) + term2 ';
eqn3 = 'dP - term1b*(abs(Tb)*Tb) + term2 ';
eqn4 = 'dP - term1c*(abs(Tc)*Tc) + term2 ';
eqn5 = 'dP - term1d*(abs(Td)*Td) + term2 ';
eqn6 = 'dP - term1e*(abs(Te)*Te) + term2 ';
[u,v,w,x,y,z]=solve(eqn1,eqn2,eqn3,eqn4,eqn5,eqn6,'dP, Ta, Tb, Tc, \ Td, Te');

u = str2num(u);
v = str2num(v);
w = str2num(w);
x = str2num(x);
y = str2num(y);
z = str2num(z);
R22 file:

R22 file is the R-22 property table. The temperature ranges from 40.1 °F to 120.0 °F in an increment of 0.1 °F.

M file:

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C2 file:

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C3 file:

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### Appendix K

**Experimental Data and Model Predicted Results**

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<th>stagnation pressure (psia)</th>
<th>degree subcooled pressure (degree F)</th>
<th>vessel pressure (psia)</th>
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*Table K.1: Experiment stagnation conditions*
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Table K.2: Measured and model predicted mass flow rates.
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Table K.3: Henry-Fauske's model predicted conditions at throat.
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<td>117 0.1267 1.8028e+04 8.4946</td>
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<td>118.3074 0.0871 2.9273e+04 10.1518</td>
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Table K.4: Flow distribution model predicted values at throttle discharge and feeder tubes.
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Table K.5: Flow distribution model predicted values at feeder tubes.
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<th>flow</th>
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<td>distribution</td>
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<td>( which</td>
<td>drop across</td>
<td>model</td>
<td></td>
</tr>
<tr>
<td>measured</td>
<td>throttle</td>
<td>predicted</td>
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</tr>
<tr>
<td>pressure</td>
<td>( after</td>
<td>pressure</td>
<td></td>
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<td>offset</td>
<td>drop across</td>
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</tr>
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<td>(psi)</td>
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2-20 | -4 | 95.35 | 69.2861 |
3-1  | -3.71 | 111.99 | 99.0811 |
3-2  | -3.42 | 109.67 | 92.3413 |
3-5  | -4.3 | 133.68 | 117.1086 |
3-6  | -4.39 | 88.36 | 77.7805 |
3-7  | -4.69 | 146.96 | 133.5926 |
5-1  | -4.98 | 135.48 | 122.5425 |
5-3  | -4.39 | 77.94 | 87.6683 |
6-3  | -4.10 | 85.49 | 94.7083 |
6-4  | -4.10 | 149.28 | 141.8527 |
7-2  | -4.20 | 142.12 | 131.4464 |
7-3  | -3.91 | 86.21 | 101.8688 |

Table K.6: Measured and model predicted pressure drop across throttle.
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<th>feeder tubes zero offset (psi)</th>
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Table K.7: Measured and model predicted pressure drop across the feeder tubes.
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Table K.8: Model predicted feeder tube inlet plenum sector angles.
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Table K.9: Model predicted feeder tube mass flow rate.