

INVESTIGATION OF OIL RETENTION AND PRESSURE DROP IN  
SUCTION LINES USING R134a WITH AB ISO 32

BY

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THESIS

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## **Abstract**

In refrigeration systems, a small quantity of oil, meant for compressor lubrication, is removed from the compressor during operation, by the high velocity refrigerant vapor exiting it, which forms a mixture with the oil. This circulating refrigerant/lubricant mixture affects the system performance in terms of heat transfer coefficient and pressure drop. The circulating liquid mixture possesses the maximum viscosity in the suction line to compressor, due to the superheated vapor, and it is for this reason that studies are performed to evaluate oil retention and pressure drop in suction lines. The aim of the current work is to study the oil retention and pressure drop in suction lines for a mixture of R134a as refrigerant and AB ISO 32 as the lubricating oil. The oil retention and pressure drop were evaluated in horizontal and vertical suction lines with an internal diameter of 10.2 mm and length approximately 2 m. The system operating conditions were kept fixed at a saturation temperature of 13°C and a superheat of 15°C. The mass flux was varied from 30 kg/m<sup>2</sup>-s – 140 kg/m<sup>2</sup>-s, and three OCRs of 1%, 3% and 5% were tested. The experimental results of the current work were compared to results of R134a/POE100 by Ramakrishnan and Hrnjak (2012) and R134a/POE32 by Sethi and Hrnjak (2011). Results showed that AB32 yields a higher oil retention than POE100 and POE32. An analytical model was also proposed to predict the experimental results. Based on the prediction of the results by the model and also the observations made during experimentation it was concluded that vapor R134a was miscible in AB32 oil. Flow visualizations in the liquid line before the evaporator and the horizontal and vertical suction lines were also carried out to study the different flow regimes.

*To My Grandparents*

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## Nomenclature

|                  |   |                                    |                    |   |                                                |
|------------------|---|------------------------------------|--------------------|---|------------------------------------------------|
| a                | = | constant                           | $\dot{m}$          | = | mass flow rate, kg/s                           |
| b                | = | constant                           | MO                 | = | mineral oil                                    |
| c                | = | constant                           | ISO                | = | International Organization for Standardization |
| A                | = | inner area of pipe, m <sup>2</sup> | OCR                | = | oil circulation ratio                          |
| AB               | = | alkyl benzene                      | P                  | = | pressure, Pa                                   |
| D                | = | diameter, m                        | P <sub>sat</sub>   | = | saturation pressure, Pa                        |
| EES              | = | Engineering Equation Solver        | PAG                | = | polyalkylene glycole oil                       |
| f <sub>i</sub>   | = | interfacial friction factor        | POE                | = | polyol ester oil                               |
| f <sub>s</sub>   | = | smooth pipe friction factor        | r                  | = | radial distance from axis                      |
| g                | = | gravity, m/s <sup>2</sup>          | R                  | = | pipe radius, m                                 |
| GWP              | = | global warming potential           | Re                 | = | Reynolds number                                |
| G                | = | mass flux, kg/m <sup>2</sup> -s    | SUS                | = | saybolt universal seconds                      |
| HFC              | = | hydrofluorocarbon                  | T <sub>bub</sub>   | = | bubble temperature, °C                         |
| HFO              | = | hydrofluoroolefins                 | u                  | = | velocity                                       |
| j*               | = | dimensionless superficial velocity | w <sub>local</sub> | = | local oil concentration in liquid film         |
| K                | = | constants                          | We                 | = | Weber number                                   |
| L                | = | length of suction pipe, m          | x                  | = | quality                                        |
| m <sub>oil</sub> | = | mass of oil, kg                    | z                  | = | axial distance                                 |

## Greek Symbols

|            |   |                                        |          |   |                              |
|------------|---|----------------------------------------|----------|---|------------------------------|
| $\alpha$   | = | void fraction                          | $\rho$   | = | density, kg/m <sup>3</sup>   |
| $\delta$   | = | liquid film thickness, m               | $\sigma$ | = | surface tension, N/m         |
| $\delta^+$ | = | dimensionless film thickness           | $\tau$   | = | shear stress, Pa             |
| $\mu$      | = | dynamic viscosity, Pa-s                | $\tau_i$ | = | interfacial shear stress, Pa |
| $\nu$      | = | kinematic viscosity, m <sup>2</sup> /s |          |   |                              |

## Subscripts

|   |   |                        |   |   |                  |
|---|---|------------------------|---|---|------------------|
| c | = | refrigerant vapor core | z | = | axial coordinate |
| l | = | liquid film            |   |   |                  |
| r | = | radial coordinate      |   |   |                  |
| v | = | refrigerant vapor      |   |   |                  |

## **CHAPTER 1: INTRODUCTION**

### **1.1 Overview**

Hydrofluorocarbons (HFCs) have been chosen as an alternative to chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs). HFCs are highly polar compounds, traditional non-polar lubricants such as mineral oils (MO) and alkylbenzene oils (AB) oils are immiscible with them for all practical purposes (Mehendale, 1998). Refrigeration oils like Polyol ester (POE), Polyalkyl ester (PAG), and other polar oils have been developed for usage with HFCs to have high miscibility with them. However, these newly developed oils have some drawbacks with respect to MO and AB oils: hygroscopicity, lubricity deterioration (for rotary compressors) and high cost. POEs can easily hydrolyze, therefore it is very necessary to control the amount of moisture contained in their manufacturing process. For rotary compressors having sliding parts (vane and rolling piston), POEs do not fare well. MO and AB oils are also very stable with HFCs, so they do not present any risk of corrosion problems inside refrigeration systems.

In refrigeration systems with reciprocating compressors, a small amount of lubricating oil leaves the compressor with the refrigerant vapor. This small amount of lubricant has to be recirculated, otherwise it would start accumulating in some part of the system, eventually leading to compressor failure due to lubricant starvation. For miscible refrigerant-lubricant combinations, the lubricant trapped in the system would be diluted with the low viscosity refrigerant, thereby enhancing the lubricant return to the compressor. However, Sunami et al. (1994) have demonstrated good oil return performance for an immiscible combination of AB and HFC. For any system, the suction lines to the compressor have conditions of high quality and low

temperature, which gives rise to the highest viscosity of the liquid mixture. This condition of high viscosity leads to the highest oil retention at the suction line for the system.

The focus of this work is on evaluating the oil retention and pressure drop in horizontal and vertical suction lines in an R134a system. The lubricant used for the current work is an alkylbenzene oil, having a viscosity of 32cSt. This AB oil has a lower density than R134a and is assumed to be completely immiscible in it over the range of operating temperatures. The mechanism of oil transport in the liquid line is quite different between miscible and immiscible refrigerant oil mixtures. For a miscible combination, the flow has a single phase, whereas for immiscible case, there is two – phase flow. Flow visualization of the liquid line was done to evaluate the oil circulation there.

## **1.2 Literature Review**

### **1.2.1 Oil Retention Studies**

In order to verify the data presented in ASHRAE Handbook (1973) and to establish conditions for minimum tonnage requirement for oil entrainment up suction risers, Jacobs *et al.* (1976) investigated the oil retention phenomenon in vertical suction risers. Using the then commonly available refrigerants R12 and R22 and naphthenic oils with 150 and 300 SUS viscosities respectively, the authors experimented with a wide range of suction and discharge conditions. Oil was injected into a 2.5cm diameter vertical test section at the bottom and oil transport was observed through the sight glass following the point of injection. Oil transport up the vertical test section was controlled by varying the vapor refrigerant mass flow rate. The amount of oil accumulated in the test section was directly proportional to the refrigerant vapor mass flux. The refrigerant vapor mass flux was reduced until the test section was flooded with oil. Experimental

data generated was correlated by using the flooding correlation by Wallis, 1969. The authors suggested a conservative estimate based on the experimental data for the dimensionless flooding correlation as

$$(j_g^*)^{1/2} = 0.85 \quad (1.1)$$

A more useful form of the above relation in terms of minimum refrigerant mass flux is given by

$$G = ((j_g^*)^{1/2})^2 [\rho_v g D (\rho_l - \rho_v)]^{0.5} \quad (1.2)$$

The authors also presented the data in the form of charts with minimum tonnage requirement to ensure oil return. A major drawback of this correlation is that it does not take into account the effect of oil concentration and viscosity on return of oil in vertical suction lines.

Mehendale and Radermacher (1998) performed an experimental and theoretical investigation of annular film flow reversal in a vertical pipe. This work focused on the mechanism of oil transport by refrigerants. Flow visualization experiments were conducted to identify conditions for oil fluid motion breakdown. Various refrigerant–lubricant pairs were studied, namely, R410A/POE, R22/MO, R407C/POE, R410A/MO and R407C/MO. It must be noted that the first three combinations of refrigerant and lubricant formed a miscible mixture, whereas the last two combinations formed an immiscible mixture. The experiments were carried out using an 8 mm inside diameter pipe to find out the critical refrigerant flow rate for oil flow reversal. All experiments were performed for three different refrigerant conditions, namely, superheated vapor, subcooled liquid and two-phase flow. The oil injection method was used to introduce oil into the test section, which was separated from the refrigerant at the end. During the tests when the refrigerant mass flow rate was reduced below a critical value required for oil flow reversal to

occur, the authors observed that the stable upward moving oil film initially became unstable, started oscillating and ultimately reversed its flow direction. Critical fluxes for the superheated refrigerant vapor conditions were found to be higher than the predictions of Jacobs *et al.* (1976). In the two-phase refrigerant flow case, the immiscible lubricant separated from the refrigerant in the form of a film or rivulet and the oil flow reversal occurred at a definite refrigerant flow rate. In case of the miscible refrigerant-lubricant pairs, oil did not separate out into a separate film and hence no flow reversal occurred. An analytical model to predict the onset of oil film reversal was developed as a part of this work. The model predicted the experimental results to within +7% and -4% of the experimental data. The model was also able to predict the thickness of the lubricant film for a given refrigerant mass flow rate. This helped to estimate the amount of oil circulating in the system outside the compressor at any given time. It was observed that the oil film was thicker in case of the immiscible refrigerant-lubricant pairs, thereby suggesting that the compressor would lose more oil.

Sunami *et al.* (1994) developed alkylbenzene (AB) oils with extremely low viscosities for usage as refrigeration oils with HFC refrigerants in systems with high-pressure dome type rotary compressors. The authors have demonstrated a good oil return performance at low temperatures. Since AB oils are immiscible in HFCs, their antiwear characteristics in rotary compressors are better than conventional mineral oils and POEs. Another observation the authors made was that the new low viscosity AB oils do not dissolve vapor HFCs in high-pressure chambers, as a result, it is possible to reduce the refrigerant charge in the system.

The viscosity of the lubricant-refrigerant mixture determines oil return to the compressor. Since AB oils are immiscible in HFCs, the authors measured the low temperature viscosities of the oils. They observed that viscosity characteristics of linear alkylbenzene (LAB) are better than

those of branched alkylbenzene (BAB), in fact the kinematic viscosity of LAB at  $-40^{\circ}\text{C}$  was about the same as that of naphthenic mineral oil at room temperature ( $56\text{ mm}^2/\text{s}$  at  $40^{\circ}\text{C}$ ). The oil return performance of low-viscosity AB was determined to be better at  $-20^{\circ}\text{C}$  and higher, while POE had a better oil return rate at  $-30^{\circ}\text{C}$  and below. The solubility of gaseous R134a in AB and POE was also studied. R134a was soluble with POE at high temperatures and pressures, while AB displayed a solubility of less than 50% by mass of that in POE. Since the dissolution of refrigerant gas in oil can cause a pressure drop in high-pressure dome type rotary compressors, AB oils that are immiscible in HFCs can be expected to prevent a drop in cooling efficiency.

Sumida *et al.* (1998) studied the oil return characteristics of alkylbenzene in split air conditioners with R410a. Accumulation of oil in the liquid line was studied by observing the flow pattern of the refrigerant oil mixture and the minimum velocity to keep the flow smooth. The experimental setup had two separate loops for oil and refrigerant circulation. The refrigerant-oil mixture generated at the outlet of the condenser entered the test section. To observe the flow pattern of the refrigerant-oil mixture, glass tubes were installed in the test section at three directions, namely vertical upward, horizontal and vertical downward. Oil accumulation in the liquid line was measured by using shut-off valves, detaching the sampling tubes and weighing the refrigerant and oil weight respectively. Flow patterns for the liquid refrigerant-oil mixtures were observed as being dispersed in the vertical, lines, and two-phase flow in the horizontal line. The alkylbenzene, which had a lower density than the liquid refrigerant, was on top in the horizontal line. In addition to studying the oil accumulation in the liquid line, the authors also observed the oil level in the compressor with the help of sight glasses. The authors observed that the minimum liquid refrigerant velocity in the vertical lines depended on the diameter of the oil droplets

entrained. This study demonstrated that a split air conditioner with R410a and AB had reliable oil return characteristics similar to R410a and POE systems.

Sundaresan *et al.* (1996) studied the oil return characteristic of R407C and R410A with miscible and immiscible lubricants, namely, POE and MO, in a 3 ton residential split air conditioning system, and compared the results to a R22/MO pair as a baseline. This study utilized two scroll compressors, one for medium pressure refrigerants (R22 and R407C) and one for high pressure refrigerant (R410A). The liquid level in the compressor was studied with the help of a sight glass. Tests are conducted with the aim of measuring the oil return characteristics and measure oil level in the compressor as a function of time. The steady state oil level in the compressor all the refrigerant/lubricant pairs was compared. Interestingly, for the R410/MO pair did not reach a steady state oil level in the compressor, if the compressor was run for long enough time, all the lubricant would leave the compressor, thereby starving it and causing compressor failure. The steady state liquid level for the miscible POE oil was observed to be always higher than the immiscible MO, which in other words suggests that POE outperforms MO. Another interesting observation that the authors made was that when they performed cyclic tests, R407C/MO pair continuously lost oil where as R410/MO pair did not. This indicated that oil migration is not a function of miscibility alone. Judging by the steady state liquid level in the compressor, the authors concluded, that for the immiscible refrigerant/lubricant pairs the lubricant never returned to the compressor. The authors also studied oil transport inside the condenser, for the miscible refrigerant/lubricant pairs only a single phase was observed, whereas for the immiscible pairs two phases were observed, with the lighter oil floating on top of the heavier refrigerant. The oil rolled over the faster moving refrigerant along the top tube wall. In the suction lines the authors observed a thicker oil film in case of the immiscible refrigerant/lubricant pairs when compared to

the miscible pairs. The authors state that the thickness of the oil film depends on the surface tension difference between the oil and the refrigerant vapor. The main conclusions of the study were that R407C/POE and R410/POE had reliable oil return characteristics, similar to R22/MO, whereas R407C/MO and R410A/MO had unreliable oil return characteristics. Both the immiscible refrigerant/lubricant pairs continuously removed oil from the compressor and could eventually lead to its failure.

Biancardi *et al.* (1996) conducted experimental and analytical investigations to study the lubricant circulation and return for HFC/POE and HFC /MO pairs in a residential HVAC system and compared the results to a HCFC/MO pair. Operating conditions for poor lubricant return to the compressor were defined for a heat pump system for R407C and R22 with miscible and immiscible lubricants. Both heating and cooling modes were studied and it was found out that poor oil return corresponded to low vapor velocities in the vertical suction lines. This study was carried out by comparing two immiscible R407C/MO pairs with four different miscible R407C/POE pairs, with R22/MO as the baseline for comparison. Both the heating and cooling modes were studied by using  $\frac{3}{4}$  inch (19.05mm) diameter suction lines to create minimum velocity conditions for oil return. The test facility designed had the capability to inject additional quantities of lubricant into the test section than what would normally flow due to compressor discharge. Oil injection tests showed that presence of excessive lubricant could result in localized oil pooling and trapping. Results for flow velocities for worst case oil management suggest velocities of 0.51m/s at 0.25-0.5 % oil concentrations for the heating mode and 1.78-1.9m/s for the cooling mode. Both R22 and R407C showed approximately the same minimum velocities with miscible lubricants and immiscible lubricants also showed good oil return characteristics. An interesting conclusion of the study was that the immiscible R407C/MO pairs exhibited good,

if not better, oil return as compared to miscible combinations and were able to operate at lower minimum velocities than their miscible counterparts could.

Lee *et al.* (2000) performed an experimental investigation on the flow characteristics of refrigerant/oil mixtures in vertical suction lines. The work concentrated on the estimation of the mean oil film thickness and observed the flow pattern in vertical upward flow in a R134a suction line. Three immiscible oils, namely, one mineral (MO ISO 10) and two alkylbenzenes (AB ISO 8 and AB ISO 10) were tested with R134a in the test facility. The test facility consisted of two separate loops, one for oil and the other for refrigerant. Within the range of refrigerant and oil mass flow rates investigated in the study the only flow patterns that were observed during the upward flow of the refrigerant/oil mixture in the vertical suction line were churn and annular flow respectively. For the purpose of the study the oil concentration was varied from 0.1-5%, refrigerant mass flow rates of 0.10, 0.37 and 0.57 g/s and oil volume flow rates of 4, 12 and 20 ml/hr respectively were investigated. To present the results the authors defined a non-dimensional variable called the mean oil film thickness ratio (MOFTR) as the ratio of the oil film thickness to the inside radius of the tube. The results showed that the oil with poor miscibility and higher viscosity created a thicker oil film in the tube and delayed the oil return. At low refrigerant mass flow rates, it was observed that the MOFTR increased due to increased oil viscosity. However, at high refrigerant mass flow rate (0.57 g/s) the MOFTR was not influenced by the oil type and viscosity. At a very low refrigerant flow rate (0.1 g/s), churn flow regime was observed and the MOFTR was 2-3 times higher than other cases with higher refrigerant mass flow rates. The authors recommended that the churn flow regime should be avoided as the oil on the wall moves downward and could cause possible oil return failure especially in some heat exchangers.

Cremaschi and Radermacher (2004) performed an experimental and analytical investigation of oil retention and transport in various components of a vapor compressor system for stationary applications. The oil injection/extraction experimental methodology was used to study oil retention in various components, more specifically, the condenser, evaporator, suction and the liquid lines. The authors tested with both miscible and immiscible refrigerant/lubricant combinations. The refrigerant/oil mixtures used were R22/MO, R410A/MO, R410A/POE, R134a/POE and R134a/PAG respectively, in order to cover the widest range of mass fluxes, core Reynolds numbers and liquid mixture film to refrigerant vapor viscosity ratios. One of the key focus areas of the study was to perform a parametric analysis of oil retention under varying conditions of oil mass fractions (OMF), refrigerant mass flow rates and oil film viscosities. The refrigerant mass flux was varied in the range from 106 to 400 kg/m<sup>2</sup>-s and the OMF was varied from 1 to 8% by weight. The reason for choosing such a range was its applications in stationary and automotive air conditioning systems. The inner diameters of the suction lines were varied from 13-19 mm in order to study the effect of changing pipe diameter on the oil retention. A reduction in the pipe diameter promoted oil transport in both the horizontal and vertical suction lines. It was observed that for an increase of mass flux from 181 kg/m<sup>2</sup>-s to 206 kg/m<sup>2</sup>-s the oil retention decreased by about 20% in the horizontal test section. A reduction in oil retention in the vertical suction line was observed only if the OMF was higher than 2% by weight. At an OMF of 5% by weight oil retention was observed to decrease by 22% in the horizontal suction line and by 19% in the vertical suction line when the mass flux was increased from 150 kg/m<sup>2</sup>-s to 206 kg/m<sup>2</sup>-s. It was also observed that poorly miscible mixtures had substantially higher oil retention mass in each component of the system. At an OMF of 5% by weight, R410A/MO had an oil retention mass in the suction line of about 31% higher than that of R410A/POE. The significant

increase of oil retention was due to a difference in the liquid film viscosity and degree of miscibility between refrigerant and oil. The R410A/MO mixture, which is an immiscible mixture, had 33% to 71% higher oil retention in the system as compared to other miscible refrigerant/oil mixtures tested. Oil retention volume also depended on the pipe orientation and it was observed that vertical suction lines could have up to 50% more oil retained compared to horizontal suction lines at the same operating conditions, thus highlighting the effects of gravity on oil retention. The authors made some recommendations based on their work. Some of the main recommendations were using high refrigerant mass flow rates, suction lines with inner diameters larger than 19 mm are unfavorable and adopting soluble and miscible refrigerant/oil pairs decreases liquid film viscosity in the suction lines thereby decreasing oil retention.

Kesim *et al.* (2000) studied the minimum refrigerant velocity required to carry oil vertically upwards in suction lines. The authors developed an equation for minimum refrigerant velocity to carry oil up vertical suction lines by using an analytical model. Navier-Stokes equations and continuity equation were used on the oil side in order to determine the velocity profile in the thin oil layer. Using suitable boundary conditions the velocity distribution equation was determined, which was used to find out the volumetric flow rate of oil. The pressure drop gradient on the refrigerant side was determined using the Darcy-Weisbach formula for frictional pressure drop. In order to determine the friction factor, the authors used the Blasius solution for turbulent flow in smooth pipes. In order to completely solve for the average refrigerant vapor velocity satisfying the oil return criteria, a relation for the interfacial shear stress at the refrigerant/lubricant interface had to be used. The refrigerant velocity determined was converted into a cooling capacity of the system, which is more useful for practical purposes. The minimum cooling capacity for oil entrainment in vertical suction and discharge lines was calculated for R134a and

copper tubing of different diameters and presented in the form of tables. Different copper tubes based of varying wall thickness were used to present the results. Saturated suction temperatures were varied between -35 and 5°C, while keeping the condensing temperature constant at 40°C. It must however be noted, Sethi and Hrnjak (2011) stated that Blasius solution is inappropriate for turbulent flow in smooth pipes. By performing flow visualizations, they observed that at the point of flow reversal in vertical suction lines the refrigerant/lubricant interface was extremely wavy.

### **1.2.2 Study of Flow of a Mixture of Two Immiscible Liquids**

Vedapuri (1999) performed a detailed study on oil-water flows in inclined pipes. The oil-water distribution across a cross-section of a pipe was studied, by varying the input water percentages between 20-80% and at six different pipe inclinations. The mixture velocities were varied from 0.2 to 2.0 m/s. The author observed three different flow patters across the entire test matrix; namely, semi-stratified, semi-mixed and semi-dispersed (the work has diagrams of the above flow patters describing the exact meaning of each). At low mixture velocities, however, a concurrent flow was observed in the water phase. Flow pattern transitions were seen to have a dependence on the percentage of water present in the mixture. At water percentages of 40% and 60%, better mixing of the oil/water mixture was observed. The flow pattern changes from semi-stratified to semi-mixed and semi-dispersed respectively as the mixture velocity goes up. A mechanistic model was developed to predict the hold up of three phases; namely, oil layer (on top), mixed layer (oil-water emulsion, in the center) and a water layer (at the bottom), in the mixture.

### 1.2.3 Oil Retention Investigations at ACRC, University of Illinois

Crompton *et al.* (2004) studied the oil retention in copper tubes having small diameters and varying internal geometries. The different internal geometries of the copper tubes tested by the authors were, internally smooth, axially microfinned and helically microfinned. All tubes had outer diameters of 9.53 mm (3/8"). The entire range of mass fluxes tested were from 75 to 300 kg/m<sup>2</sup>-s, with an emphasis on 75 and 150 kg/m<sup>2</sup>-s. A range of miscible and immiscible refrigerant/lubricant mixtures were examined, namely, R134a/POE, R134a/PAG, R134a/AB, R22/AB and R410a/POE. The quality was varied between 0-100% and the oil concentration between 0-4.3 with the exception of R134a/PAG where the variation was as high as 0-15%. The technique used by the authors to measure oil holdup was the direct measurement method. After steady state conditions are reached at a desired mass flux and quality, valves on each end of the test sections were closed simultaneously to trap the mixture. Afterwards, the test section was removed from the loop and weighed. The initial reading was for the combined weight of the refrigerant and the oil. Then the section was evacuated and weighed. The difference between the weights of the test section before and after vacuuming gives the oil holdup under the test conditions. This procedure is a standard method referred to as ASHRAE 41.4. The current study also employs the same method to measure oil retention in the suction lines. The authors observed that at mid-range qualities, the oil holdup reached a minimum, and as the quality increases the oil holdup increases due to an increase in the mixture viscosity. At lower mass fluxes (75 kg/m<sup>2</sup>-s) oil retention increases. The internal geometry of the tube does not have a significant effect on the oil holdup for low to mid-range qualities. However, for the R134a/AB mixture the effect of tube type is more pronounced on the oil retention with helically microfinned sections showing more oil retention than axial or smooth sections at high qualities. The authors performed flow

visualizations for each of the refrigerant/lubricant combinations to study the flow patterns at the different operating conditions. An analytical model for oil holdup was also prepared and the modeling results were compared with the experimental values.

Zoellick and Hrnjak (2010) studied oil retention and pressure drop in horizontal and vertical suction lines. They used R410A/POE32 as their refrigerant/lubricant combination for investigation. Studies were performed for two different suction line inner diameters of 7.2 mm and 18.5 mm respectively. An experimental setup was designed to have separate loops for refrigerant and lubricant circulation. The setup was designed in a way to allow the user to perform tests over varying OCR, mass flux and superheat conditions. The current study to measure oil retention and pressure drop for R134a/AB mixture has also been performed on the same setup with slight modifications. The authors used the direct measurement technique, as described by Crompton *et al.* (2004), to determine oil retention in horizontal and vertical suction lines. Oil retention and pressure drop was studied at three different OCRs and superheats, of 1%, 3%, and 5% and 5°C, 10°C, and 15°C respectively. The ranges of mass flux tested were 100-250 kg/m<sup>2</sup>-s for the 7.2 mm pipe and 60-100 kg/m<sup>2</sup>-s for the 18.5 mm pipe respectively. The authors observed that oil retention went up substantially when liquid film reversal started. The mass flux at which recirculation began was observed to be above the critical mass flux condition as described by Jacobs *et al.* (1976). The Jacob's limit predicts the onset of flow reversal and change of flow regime to churn from annular. The authors observed a hysteresis loop at the Jacob's mass flux. When the mass flux was increased above the Jacob's flux, transition to annular flow did not happen until a 30% higher mass flux than when it was being decreased. The OCR has a major effect on the oil retention in the suction lines. An increase from 1% to 3% in the OCR results in a 20-50% increase in the oil retention at all the different mass fluxes. The

vertical suction line always retained 10% more oil than the horizontal suction line due to the effects of gravity in the former. An increase of 5°C in the superheat increased the oil retention by 15% due to an increase in the viscosity of the oil rich liquid film.

Sethi and Hrnjak (2011) studied oil retention and pressure drop in horizontal and vertical suction lines. They employed R1234yf and R134a as refrigerants and used POE32 as the lubricating oil to study the oil retention and pressure drop. The authors also studied the effect of pipe inclination on oil retention. All tests were conducted using 10.2 mm internal diameter pipes which were 2 m long. The system conditions of 13°C saturation temperature and 15°C superheat were maintained for all the tests. Oil retention was measured in horizontal, vertical and inclined pipes. The method of direct measurement was used to measure the amount of oil retained. Transparent suction lines, which are a part of the current setup, were used to record high-speed videos of the flow to study the flow regimes. The authors also developed an analytical model to predict oil retention and pressure drop in suction lines using the experimental data obtained. The authors, while studying the effect of pipe inclination determined that inclined pipes retained more oil than either horizontal or vertical pipes. The authors also observed that R1234yf showed similar oil retention when compared to R134a at similar operating conditions, but pressure drops were 20-30% higher in case of R1234yf for both horizontal and vertical lines.

Ramakrishnan and Hrnjak (2012) studied oil retention and pressure drop in horizontal and vertical suction lines. They employed three refrigerants, namely, R1234yf, R134a and R410A and studied the effect of POE100 and POE32 on oil retention and pressure drop. Operating conditions maintained were identical to those used by Sethi and Hrnjak (2011). The authors studied R1234yf in order to determine if it is a suitable drop-in replacement for R134a in automotive systems. The tests were carried out using the same test setup as used by Sethi and

Hrnjak (2011) and the test sections were also unaltered in order to compare the experimental data. The authors also carried extensive flow visualization studies in both the horizontal and vertical lines to determine the flow regimes at various mass fluxes. An analytical model was also developed based on the experimental data. The results showed that POE100, which was three times more viscous than POE32, yielded up to 30% more oil retention and up to 60% more pressure drop, at similar operating conditions. When comparing different refrigerants, the authors observed that R1234yf showed up to 15% more oil retention and up to 60% more pressure drop than R134a.

#### **1.2.4 Oil Retention and Pressure Drop Modeling**

Prior to experimentation, researchers prefer to model oil retention and pressure drop in suction lines to have an idea of the amount of oil circulating in the system, using which they could prevent compressor failure due to lubricant starvation. It has been discussed above that annular flow regime is preferred during system operation as it leads to lower oil retention as compared to the churn flow regime. It has also been observed that vertical suction lines retain more oil than horizontal suction lines. It is for these reasons that analytical models are developed for annular flow in vertical suction lines, in order to size them in a better way thereby minimizing oil retention.

Extensive work has been done on modeling of oil retention in vertical suction lines. Many researchers have developed models for oil retention by applying the Navier-Stokes and continuity equations to the oil rich liquid film and the refrigerant vapor core. Equations generated are solved by using relevant boundary conditions and performing a momentum balance at the refrigerant/lubricant interface. Different authors considering their experimental data have proposed correlations.

Wallis (1969) proposed a correlation for the friction factor, which has now become one of the most widely used during oil retention modeling. The interfacial friction factor, used during the momentum balance at the refrigerant/lubricant interface, was expressed by Wallis (1969), as a function of film thickness for annular flow regimes in vertical tubes. Cremaschi (2004) proposed an interfacial friction factor for both the horizontal and vertical suction lines. He correlated the friction factor to the vapor core Reynolds number, the non-dimensional film thickness and the Weber number of the refrigerant/lubricant mixture. The model predicted the experimental oil retention data to within  $\pm 31\%$ . Zoellick and Hrnjak (2010) used an approach similar to van Rossum (1959), to propose a correlation relating the non-dimensional film thickness with the liquid film Reynolds number. They also employed the Wallis (1969) correlation to relate the interfacial friction factor at the refrigerant/lubricant interface with the film thickness. The model predicted their experimental oil retention data to within  $\pm 20\%$ .

### **1.3 Objectives of Current Study**

The aim of the current work is to experimentally measure oil retention and pressure drop in horizontal and vertical suction lines for an immiscible mixture of R134a and AB32 lubricant. The results obtained were compared with the results for R134a/POE32 studied by Sethi and Hrnjak (2011) and R134a/POE 100 studied by Ramakrishnan and Hrnjak (2012). This was done in order to study the effect of mixture viscosity on oil retention and compare miscible and immiscible refrigerant/lubricant pairs. Flow visualization studies were performed to study flow regimes and transitions in the horizontal and vertical suction lines. Similar to earlier work undertaken at the ACRC at the University of Illinois at Urbana-Champaign, 10.2 mm internal diameter suction lines measuring approximately 2m in length were used for this study as well. Flow visualization of the liquid/liquid immiscible mixture of R134a and AB32 before the

evaporator was also performed to better understand their behavior. An analytical model was also proposed to predict oil retention and pressure drop in vertical suction lines.

## **CHAPTER 2: EXPERIMENTAL SETUP**

### **2.1 Description of the Experimental Facility**

Zoellick and Hrnjak (2010) developed an experimental facility to study oil retention in horizontal and vertical suction lines, with internal diameters of 7.2 mm and 18.5 mm, in order to simulate the suction line a typical R410A air conditioning system. Sethi and Hrnjak (2011) modified the system by installing an R22 condensing unit to assist in heat removal from the refrigerant at higher refrigerant mass flow rates, thereby improving the range of refrigerant mass fluxes over which data could be recorded. The setup in its present state consists of horizontal and vertical suction lines made of clear PVC, each measuring approximately 2 m in length. The internal diameters of the suction lines are kept unchanged at 10.2 mm similar to Sethi and Hrnjak (2011) and Ramakrishnan and Hrnjak (2012). A schematic of the setup in its present state is shown in Figure 2.1. The refrigerant considered for the current study was R134a, while the lubricant considered was nominally 32 cSt AB oil. A 10 cm long clear transparent section was installed after the mixing section (where the liquid refrigerant and oil mix after emerging from their respective loops) and before the entry to the evaporator. This was done to perform flow visualization, to study the phenomena of mixing of the immiscible refrigerant/lubricant pair of R134a/AB32.

The direct measurement method was employed to determine the amount of oil retained in the horizontal and vertical suction lines. Ball valves having orifices equal to the internal diameter of the suction lines were installed at each end of the suction lines to facilitate in the measurement. Once equilibrium was attained at the desired operating condition, the ball valves were simultaneously closed to trap the refrigerant/lubricant mixture to be weighed. Oil retained was

measured using the direct measurement technique in accordance with ASHRAE 41.4. Differential pressure transducers were used to measure the pressure drops across the horizontal and vertical sections.

Pure, subcooled liquid refrigerant was circulated through the test facility by using a Magnetek gear pump. The refrigerant mass flow rate was varied using a variable speed drive. A bypass valve was also installed on the gear pump to achieve very low refrigerant flow rates. A subcooler was provided to increase the subcooling of the liquid refrigerant and reduce the probability of vapor formation, which could lead to fluctuations in the flow rate measurements. The mass flow rate and density of the pure refrigerant was measured by a MicroMotion CMF25 Coriolis flow meter ( $\pm 0.05$  % flow rate measurement accuracy,  $\pm 0.5$  kg/m<sup>3</sup> density measurement accuracy) located downstream of the refrigerant gear pump.

The oil stored in the oil tank was pumped by means of a Micropump GJ series gear pump. Minute quantities of vapor refrigerant are dissolved in the oil while the oil circulates in the test sections. A MicroMotion CMF 10 Coriolis flow meter ( $\pm 0.05$  % flow rate measurement accuracy,  $\pm 0.5$  kg/m<sup>3</sup> density measurement accuracy) was used to measure the flow rate and density of the mixture. It has been suggested by Mehendale and Radermacher (1998) that miscibility of vapor R134a in AB oil is less than 2%, so for all practical purposes it was assumed that there is no refrigerant dissolved in the oil. The oil pump was driven by a fixed frequency AC motor and therefore, to control the flow rate of the oil a ball valve and a bypass valve were installed. A subcooler was installed before the oil pump as well to avoid any fluctuations in the mass flow rate. The present system was designed for miscible refrigerant/lubricant combinations and hence the oil subcooler was necessary in those conditions to maintain a stable flow rate. The OCR was varied by changing the refrigerant and oil mass flow rates.

As mentioned earlier the system had two separate loops for refrigerant and oil circulation. The liquid refrigerant and oil were mixed in a T-junction before entering the evaporator. The transparent section, mentioned earlier, was installed immediately after the T-junction to study the mixing behavior of the immiscible liquid refrigerant with the oil. This technique is better than the injection-separation method as it simulates the actual conditions in a real system by ensuring proper mixing as well as equilibrium between the refrigerant and oil streams before they enter the evaporator. A brazed plate counter flow heat exchanger with 12 plates is used as the evaporator. The cold refrigerant/lubricant mixture is heated up by hot water in the evaporator. Superheated refrigerant vapor enters the suction lines located after the evaporator. The superheat conditions are controlled by varying the flow rate of the hot water. A development length of 100 diameters was provided before entry to the horizontal suction line to ensure thermal and hydrodynamic equilibrium between the vapor refrigerant and the oil mixture. The saturation temperature and the evaporator exit temperature were maintained to within  $\pm 3\%$  or  $\pm 1^\circ\text{C}$  of the testing temperature to main the desired concentration of oil in the liquid phase.

A Helical separator was provided after the exit to the vertical suction line to separate the refrigerant/lubricant mixture. The vapor refrigerant was passed on to the brazed plate counter flow heat exchanger with 12 plates , which acted as a condenser, to condense the vapor refrigerant back to liquid, from where it was sent to a receiver to be pumped again. The oil was sent back to the oil tank. The condenser served as an evaporator to the R22 condensing unit installed by Sethi and Hrnjak (2011). A variable speed drive was used to control the capacity of the R22 condensing unit, which in turn was used to maintain the system saturation temperature.

A Honeywell TJE absolute pressure transducer (range 0-3477 kPa, accuracy  $\pm 8.6$  kPa) was used to measure the system saturation pressure at the inlet to the horizontal suction line. Honeywell Z

type differential pressure transducers (range  $0 \pm 69 \text{ kPa}$ , accuracy  $\pm 0.1 \text{ kPa}$ ) were used to measure the horizontal and vertical pressure drops. An electronic mass balance with an uncertainty of  $\pm 0.03 \text{ g}$  was used to weigh the suction lines when conducting oil retention measurements.

A Yokogawa HR 1300 data-logger was used to read output electrical signals from thermocouples, flow meters and pressure transducers, which were then interfaced with a computer running a LabVIEW program that displayed the real time data. Zoellick and Hrnjak (2010), who developed this facility, have provided a detailed description of the testing procedure involved.

## **2.2 Testing Conditions**

For the present study an immiscible refrigerant/lubricant mixture of R134a/AB32 was used. In order to compare the results of this study with Sethi and Hrnjak (2011), and Ramakrishnan and Hrnjak (2012) the operating conditions of  $13^\circ\text{C}$  saturation temperature and  $15^\circ\text{C}$  superheat were employed. The tests were run at OCRs of 1%, 3%, and 5% respectively. The total mass flux was varied from 30-140  $\text{kg/m}^2\text{-s}$ . The lowest mass flux of 30  $\text{kg/m}^2\text{-s}$  corresponded to the Jacob's flux. It was not possible to run the system at the Jacob's flux due to the minimum flow rate restriction of the system. A high speed camera was used to perform flow visualizations to determine the various flow regimes at the different mass fluxes. The text matrix is tabulated in Table 2.1.

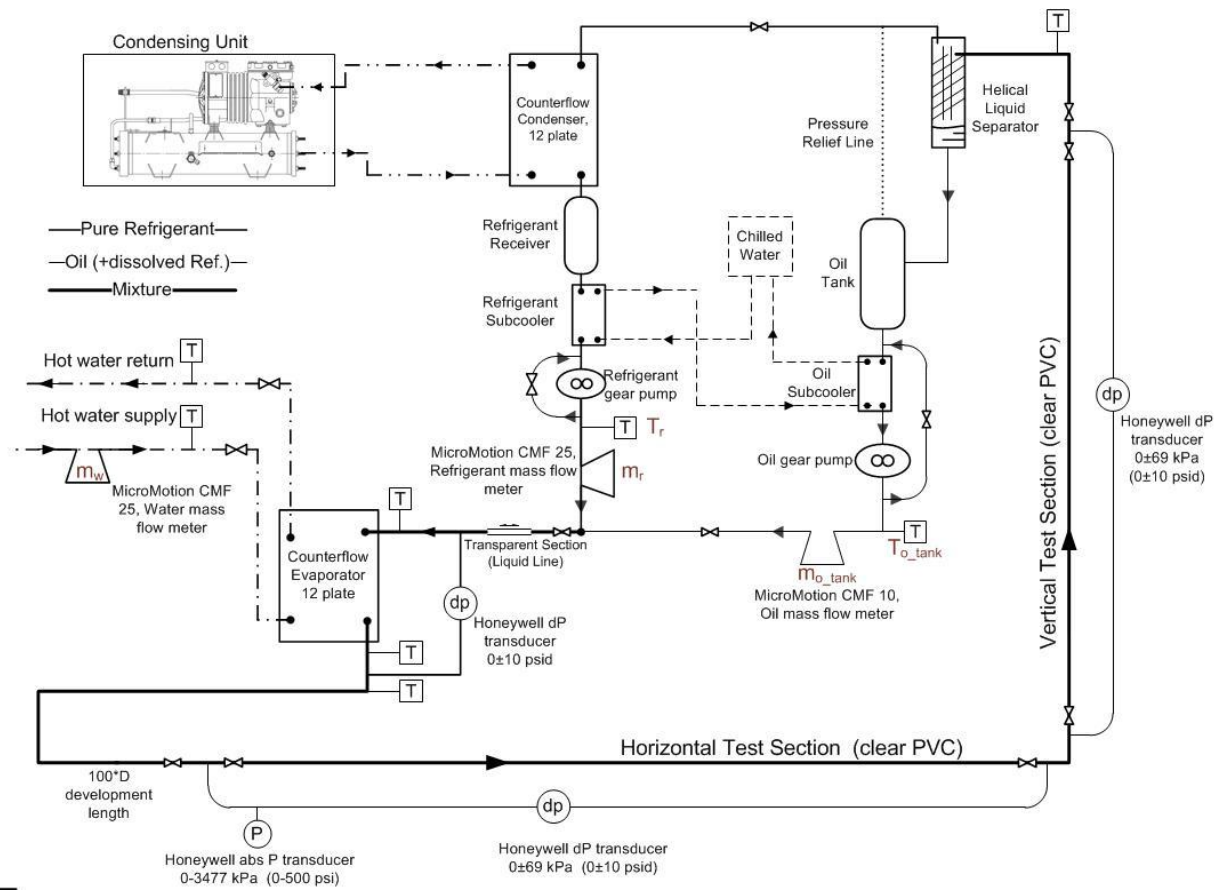


Figure 2.1 Test apparatus schematic

Table 2.1 Test conditions for R134a/AB32

| <b>Total Mass Flux</b>   | <b>Superficial Vapor Velocity</b> | <b>Total Mass Flow Rate</b> |
|--------------------------|-----------------------------------|-----------------------------|
| <b>kg/m<sup>2</sup>s</b> | <b>m/s</b>                        | <b>g/s</b>                  |
| 30 (Jacob's Flux)        | 1.5                               | 2.5                         |
| 52                       | 2.5                               | 4.2                         |
| 62                       | 3                                 | 5.1                         |
| 83                       | 4                                 | 6.7                         |
| 103                      | 5                                 | 8.4                         |
| 124                      | 6                                 | 10.1                        |
| 144                      | 7                                 | 11.8                        |

## **CHAPTER 3: EXPERIMENTAL RESULTS**

### **3.1 Flow Visualization**

Videos of flow of the refrigerant/lubricant mixture through tubes were captured using a high speed camera in order to carry out flow visualization studies to identify the regime of the flow at different mass flow rates and OCRs. Videos were taken at three different locations in the system, namely, liquid line before the evaporator, vertical and horizontal suction lines respectively. The videos were captured at mass fluxes varying between 50-140 kg/m<sup>2</sup>-s and OCRs of 5%, 3% and 1% respectively.

#### **3.1.1 Liquid Line Flow Visualization**

Figure 3.1 shows one frame of a video with its different aspects. A light source was placed behind the transparent section, which can be seen as a lighter area toward the right of the frame. The dark lines at the top and the bottom are the tube walls. The vertical markings below the lower tube wall are the gradations of a ruler used to measure the length of the section being videoed. The flow visualizations were carried out for R134a/AB32 mixture. In the liquid line, both the refrigerant and lubricant are present in its liquid state. The AB32 oil has a lower density than the liquid R134a. Hence, in the frame the layer on the top is the layer of pure oil (it may contain some vapor refrigerant dissolved in it) and the layer below is the liquid refrigerant layer. Some figures like the one shown may have a three dimensional interface between the two liquid phases, the interface may have various shades of gray. Two globules can be seen, one at the starting of the frame and one at the end, these are oil droplets that have separated from the pure oil layer and are floating in the refrigerant layer lower than the interface.

Figure 3.2 shows a matrix of pictures of the flow visualization experiments conducted in the liquid line to the evaporator. The matrix covers the entire test matrix with for the R134a/AB32 refrigerant lubricant combination. The Jacob's flux of  $30 \text{ kg/m}^2\text{-s}$  was not covered due to the mass flow rate restrictions of the system. At a very low mass flux the sub-cooling of the liquid refrigerant was insufficient and as a result too much vapor was present in the mass flow rate meter to give accurate readings for the mass flow rate. It may be noted that the pictures in which the tube is inclined are by virtue of the slight inclination of the tube. It can however be assumed that the tube was horizontal and the apparent inclination is due to the tube being slightly bent and the orientation of the camera.

The flow patterns observed in Figure 3.2 will be explained based on the flow patterns classifications for Oil-Water Flows given by Oglesby (1979). At a mass flux of  $50 \text{ kg/m}^2\text{-s}$  it can be seen that the refrigerant/lubricant layer is wavy and segregated with no mixing at the interface. It can be observed that as the OCR goes up the thickness of the oil layer increases due to the presence of an increased amount of oil. As the mass flux increases the segregated interface gradually becomes wavier, i.e. the amplitude and frequency of the waves increases. At a mass flux of  $100 \text{ kg/m}^2\text{-s}$  second the presence of oil globules in the flow can be observed. As the mass flux increases to  $140 \text{ kg/m}^2\text{-s}$  the number of oil globules in the flow is quite high as compared to  $100 \text{ kg/m}^2\text{-s}$ . The oil globules were observed to be present in the lower pure liquid refrigerant layer. From the study of the videos it was observed that the oil globules moved much faster than the overlying pure oil layer. This was attributed to the slip velocity at the refrigerant/lubricant interface because of the difference in the density and viscosity of the oil and refrigerant. Another observation is that the number of oil globules also increases with the OCR due to an increased amount of oil in the bulk flow. This is by virtue of the increased mass flow rate as the flow

becomes turbulent and hence more chaotic. The flow pattern observed at a mass flux of 140 kg/m<sup>2</sup>-s is the semi-segregated flow with mixing at the interface.

### **3.1.2 Horizontal Suction Line Flow Visualization**

Within the scope of the test matrix, two flow regimes were observed in the horizontal suction lines, namely, stratified flow regime and annular flow regime respectively. The flow was observed to be stratified at low mass fluxes and transitioned into annular flow at high mass fluxes. Figures 3.3, 3.4, and 3.5 show pictures of various horizontal flow visualization experiments for three different refrigerant/lubricant combinations of R134a/POE100 (Ramakrishnan and Hrnjak (2012)), R134a/POE32 (Sethi and Hrnjak (2011)) and R134a/AB32 (present study). Similar flow patterns were observed for all refrigerant/lubricant mixtures at all operating conditions. Figure 3.6 compares the flow in the horizontal suction line for R134a/AB32 mixture over the entire range of mass fluxes and OCRs. For R134a/POE 100 and R134a/AB32 annular flow regime exists at a mass flux of 100 kg/m<sup>2</sup>-s, where as for R134a/POE32 the oil film was seen to ‘climb-up’ the tube wall at the same mass flux. A video of R134a/POE32 wasn’t available at a mass flux of 120 kg/m<sup>2</sup>-s, but the flow can be clearly seen to be annular at 140 kg/m<sup>2</sup>-s. An important observation of the flow visualization study was that the oil film was purely annular only at high mass flux values. Lower mass fluxes are characterized by thickening of the lower part of the annular film, which progressively increases as the mass flux is lowered further. This phenomenon comes into being by virtue of reduction in the vapor core momentum, which is insufficient to spread the oil film uniformly across the entire inner surface of the tube. At a mass flux of 80 kg/m<sup>2</sup>-s all the refrigerant/lubricant mixtures were observed to have stratified wavy flows with slight oil wetting observed on the tube sidewalls. At even lower mass fluxes of 60 and 50 kg/m<sup>2</sup>-s the oil film was present only at tube bottom with

almost no oil wetting along the sidewalls. Flow variations with varying OCR were also observed. It can be seen that as the OCR increased from 1% to 5% the size of the waves of the oil mixture on the tube walls in the annular flow regime also increased. Oil entrainment in the refrigerant vapor core also increases as size of the ripples increase. In the stratified wavy flow regime, higher OCR flows showed thicker oil films, thus leading to higher oil retention. For all the three OCRs and refrigerant/lubricant mixtures, the oil film thickness remained approximately the same at mass fluxes of  $80 \text{ kg/m}^2\text{-s}$ ,  $60 \text{ kg/m}^2\text{-s}$  and  $50 \text{ kg/m}^2\text{-s}$ , this was experimentally confirmed in the present study and also by Ramakrishnan and Hrnjak (2012) and Sethi and Hrnjak (2011).

Figure 3.7 shows the Baker's (1954) flow map for horizontal flows. This flow map predicts the influence of OCR on the flow regime, indicating that stratified wavy flows transition into annular flows more quickly for 1% OCR as compared to higher OCRs of 3% and 5%. These trends were consistent with the flow visualizations and can be seen in Figure 3.6.

### **3.1.3 Vertical Suction Line Flow Visualization**

Figures 3.8, 3.9, and 3.10 show pictures of various vertical flow visualization experiments for three different refrigerant/lubricant combinations of R134a/POE100 (Ramakrishnan and Hrnjak (2012)), R134a/POE32 (Sethi and Hrnjak (2012)) and R134a/AB32 (present study). Figure 3.11 compares the flow in the vertical suction line for R134a/AB32 mixture over the entire range of mass fluxes and OCRs. In the vertical suction line, the flow regime was observed to be annular at all the mass fluxes tested. As mentioned earlier, in the present study it was not possible to take data at the Jacob's flux, hence the churn flow regime, as mentioned by Ramakrishnan and Hrnjak (2012) and Sethi and Hrnjak (2011), was not observed. The annular film had surface waves, which were seen to be thicker at lower mass fluxes, and as the mass flux was increased these surface waves became thinner and increased in frequency. Hence, at lower mass fluxes enhanced

oil retention was observed as compared to the higher mass fluxes. Ramakrishnan and Hrnjak (2012), observed the onset of flow reversal for R1234yf/POE100 at 5% OCR and 50 kg/m<sup>2</sup>-s mass flux. They said that at this point some of the oil film was seen to travel in the downward direction. In the present study, however, no film reversal was observed at 50 kg/m<sup>2</sup>-s (the lowest mass flux tested) for any OCR. It was noted by Sethi and Hrnjak (2011), even before the churn flow regime was achieved at the Jacob's flux, appreciable increase in the oil film thickness was observed. Increase in the oil retention was a result of this increased oil film thickness. Hence, a new critical flux based on onset of film flow reversal at the tube wall was proposed. Such a critical flux has also been proposed in the present study.

### **3.2 Lubricant Retention and Pressure Drop**

Experimental data for oil retention and pressure drop in horizontal and vertical suction lines is presented in this section. As mentioned the present study was focused on taking data for R134a/AB32. A comparison between the experimental data for R134a/AB32, R134a/POE100 (Ramakrishnan and Hrnjak (2012)) and R134a/POE32 (Sethi and Hrnjak (2011)) has been done in this section.

#### **3.2.1 Lubricant Retention and Pressure Drop for R134a/AB32**

Figure 3.12 presents the variation of oil retention with mass flux, for vertical and horizontal suction lines. All the oil retention data has been presented in terms of oil retained per unit length of pipe (g/m). Figure 3.13 shows the pressure drop variation with mass flux, for vertical and horizontal suction lines. The data has been presented for three different OCRs of 1%, 3% and 5% and the mass flux was varied from 50 kg/m<sup>2</sup>-s to 140 kg/m<sup>2</sup>-s. A smooth pipe friction factor correlation by Knudsen and Katz (1958) was used to predict the pressure drop in suction lines in

the absence of oil. It was observed that the oil retention at different mass fluxes and OCRs in both horizontal and vertical suction lines was higher than the pressure drops predicted by the smooth pipe correlation. This suggests that the presence of oil in the suction line increases the pressure drop. This can be attributed to the increase in the vapor velocity of the refrigerant due to the presence of an oil film. The Jacob's flux of  $30 \text{ kg/m}^2\text{-s}$  was not tested due to mass flow rate limitations of the system.

#### **3.2.1.1 Vertical Suction Line**

It can be seen from Figure 3.12 that the oil retention in vertical suction lines exceeds the retention in horizontal lines; this variation can be attributed to the influence of gravity in the vertical line. The general trend shows an enhanced oil retention with decreasing mass fluxes. For 5% OCR the oil retention increased from  $6.97 \text{ g/m}$  to  $11.84 \text{ g/m}$  when the mass flux was decreased from  $140 \text{ kg/m}^2\text{-s}$  to  $50 \text{ kg/m}^2\text{-s}$ . The oil retention was seen to increase with the increase in OCR at a particular mass flux. At a mass flux of  $140 \text{ kg/m}^2\text{-s}$  the oil retention increased by 31.2 % when the OCR was increased from 1% to 3% and increased by 16.7% when the OCR was increased from 3% to 5%. It can also be seen from the Figure 3.12 that the oil retention in vertical and horizontal suction lines was very similar at a mass flux of  $140 \text{ kg/m}^2\text{-s}$ . The variation between the oil retained in the horizontal and vertical suction lines increased as the mass flux was decreased, with the vertical suction line showing a higher retention in each case.

Figure 3.13 shows the pressure drop variation in the vertical and horizontal suction lines. It was observed that the pressure drop decreased till a mass flux of  $60 \text{ kg/m}^2\text{-s}$  for all OCRs after which it increased when the mass flux was reduced to  $50 \text{ kg/m}^2\text{-s}$  for 1% and 5% OCR, however for an OCR of 3% it decreased further (though only 10 %). Ramakrishnan and Hrnjak (2012) mentioned that the pressure drop starts to increase with a decrease in mass flux at the point

where flow reversal initiates, they confirmed this through their experimental data and flow visualizations. However, for the present study no flow reversal was observed even at  $50 \text{ kg/m}^2\text{-s}$ . At the Jacob's flux the pressure drop is expected to be higher than at a higher mass flux where the flow reversal initiates as observed by Sethi and Hrnjak (2011) and Ramakrishnan and Hrnjak (2012). This happens as the flow regime transitions from the annular to churn flow. The pressure drop was seen to increase with an increase in the OCR. It was mentioned in the previous section that higher OCR values were characterized by a thicker annular oil films. Due to the increased thickness of the oil film, the superficial vapor velocity of the refrigerant increases as the refrigerant has a lesser area to flow through, thus increasing the pressure drop. At higher OCRs the ripples were also observed to be thicker than at lower OCRs. Thicker ripples would lead to an increased interaction area between the vapor refrigerant and the oil film, thus enabling a higher momentum transfer leading to an increased pressure drop. The pressure drop increased by 72% when the OCR was increased from 1% to 3%. This percentage increase in the pressure drop with an increase in the OCR was seen to decrease with the decrease in the mass flux.

### **3.2.1.2 Horizontal Suction Line**

Figure 3.12 shows the variation in oil retention data with mass flux for horizontal and vertical suction lines. The oil retention decreases with the increase in the mass flux, attain a minimum, then increases again before finally decreasing again when the mass flux is further increases. At an OCR of 3%, the oil retention was  $6.18 \text{ g/m}$  at a mass flux of  $50 \text{ kg/m}^2\text{-s}$ , which reduced to  $5.89 \text{ g/m}$  at  $60 \text{ kg/m}^2\text{-s}$ , and finally to  $5.72 \text{ g/m}$  at  $80 \text{ kg/m}^2\text{-s}$ . However, this trend did not continue with when the mass flux was increased further. At  $100 \text{ kg/m}^2\text{-s}$ , the oil retention increased to  $6.42 \text{ g/m}$ , reducing to  $6.05 \text{ g/m}$  at  $120 \text{ kg/m}^2\text{-s}$  and finally reducing further to  $5.56 \text{ g/m}$  at  $140 \text{ kg/m}^2\text{-s}$ . The minimum oil retention was observed at  $80 \text{ kg/m}^2\text{-s}$ . From the flow

visualization it was seen that the flow regime transitioned from annular (at a higher mass flux) to stratified-wavy at  $80 \text{ kg/m}^2\text{-s}$ . The flow is stratified-wavy at mass fluxes of  $50 \text{ kg/m}^2\text{-s}$  and  $60 \text{ kg/m}^2\text{-s}$ , with almost identical film thicknesses, which is consistent with the oil retention data showing only a 7-8% increase with decreasing mass flux. This trend was consistent across all the OCRs, with the oil retention increasing with the increase in OCR. The oil retention increased by 37% when the OCR was increased from 1% to 3% and by 19% when the OCR was increased from 3% to 5%. Another observation was that the percentage increase in the oil retention with an increased in OCR was higher when operating at lower mass fluxes.

Pressure drop variation for horizontal and vertical suction lines is presented in Figure 3.13. With the reduction in the mass flux, the pressure drop was found to decrease, because the flow regime transitioned from annular at high mass fluxes to stratified-wavy at lower mass fluxes. In the horizontal suction lines, the pressure drop is dominated by the frictional component, which depends only on the superficial vapor velocity of the refrigerant core. At lower mass fluxes the superficial vapor velocity is lower, which reduces the frictional pressure drop. This trend was observed to be similar across all the OCRs. However, at a particular mass flux, the pressure drop increased with an increase in the OCR. The pressure drop increased by 66% when the OCR was increased from 1% to 3% and by 14% when the OCR was increased from 3% to 5% at a mass flux of  $140 \text{ kg/m}^2\text{-s}$ .

### **3.3 Effect of Lubricant Viscosity on Oil Retention and Pressure Drop**

The effect of lubricant viscosity on the oil retention and pressure drop in horizontal and vertical suction lines was studied. In the present study, comparisons were made with the experimental data obtained for R134a/AB32 with R134a/POE100 obtained by Ramakrishnan and Hrnjak

(2012) and R134a/POE32 obtained by Sethi and Hrnjak (2011). The results for all the three refrigerant/lubricant mixtures are presented at different OCRs. Variations of pressure drop and oil retention with mass flux are represented per unit length of the suction lines. The operating conditions for all the test data presented were the same, at a saturation temperature of 13°C and superheat of 15°C.

### **3.3.1 Horizontal Suction Line**

The data for comparison between R134a/POE100, R134a/POE32 and R134a/AB32 has been presented separately for oil retention and pressure drop, with data at the same OCR being plotted on the same graph. Figures 3.14, 3.15 and 3.16 show the oil retention data against mass flux at OCRs of 1%, 3% and 5% respectively. It was observed that the oil retention for AB32 was between 20-30% greater than POE100 and between 15-40% greater than POE32 over the entire range of mass fluxes and OCRs. Ramakrishnan and Hrnjak (2012) mentioned that POE100 gave higher oil retention data than POE32 due to a higher viscosity of the oil rich liquid film. Since, the oil retention of AB32 exceeds both POE100 and POE32 it can be stated that the viscosity of the oil rich liquid film in case of AB32 exceeds those of POE100 and POE32 respectively. So when AB32 is used in a system, the R134a refrigerant vapor would require a higher superficial vapor velocity by operating the system at a higher mass flux as the refrigerant vapor cannot carry away the mixture with the same ease as it would in the case of POE100 and POE32.

Figures 3.17, 3.18 and 3.19 show the pressure drop data against mass flux at OCRs of 1%, 3% and 5% respectively. For an OCR of 1% the pressure drop data of POE100 and POE32 was about 20% greater than AB32 data. This can be attributed to the unstable mass flow rates that were observed in the suction lines at an OCR of 1%. However, at OCRs of 3% and 5% the pressure drop data of AB32 exceeded that of POE32 by 13-18% and of POE100 by 2-5%. This

was again a consequence of increased liquid film viscosity in case of AB32. When the viscosity of the liquid film is higher the oil film has a greater thickness, which leads to an increase in the superficial vapor velocity of the refrigerant core, thus increasing the pressure drop.

### **3.3.2 Vertical Suction Line**

Figures 3.14, 3.15 and 3.16 show the oil retention data against mass flux at OCRs of 1%, 3% and 5% respectively. It was observed that the oil retention for AB32 was between 18-35% greater than POE100 and between 42-50% greater than POE32. Similar to the horizontal suction line, the increased oil retention can be attributed to an increased in the viscosity of the liquid film.

Figures 3.17, 3.18 and 3.19 show the pressure drop data against mass flux for OCRs of 1%, 3% and 5% respectively. The vertical pressure drop values of AB32 were between 13-15% higher than POE32, but the pressure drop values of POE100 exceeded those of AB32 by 12-30%. It has been shown earlier that the oil retention in the vertical suction line for AB32 exceeded both for POE32 and POE100. This was attributed to an increased thickness of the liquid film in case of AB32 as compared to either POE32 and POE100. Due to the increased thickness of the oil film the superficial vapor velocity of the refrigerant vapor would increase, thereby resulting in an increase in the pressure drop in the vertical suction line. The pressure drop values of AB32 exceed those of POE32; these results are compliant with the above flow physics. On the other hand the pressure drop values of POE100 exceed those of AB32, this does not agree with the flow physics. The only possible explanation for this can be inaccurate vertical pressure drop data.

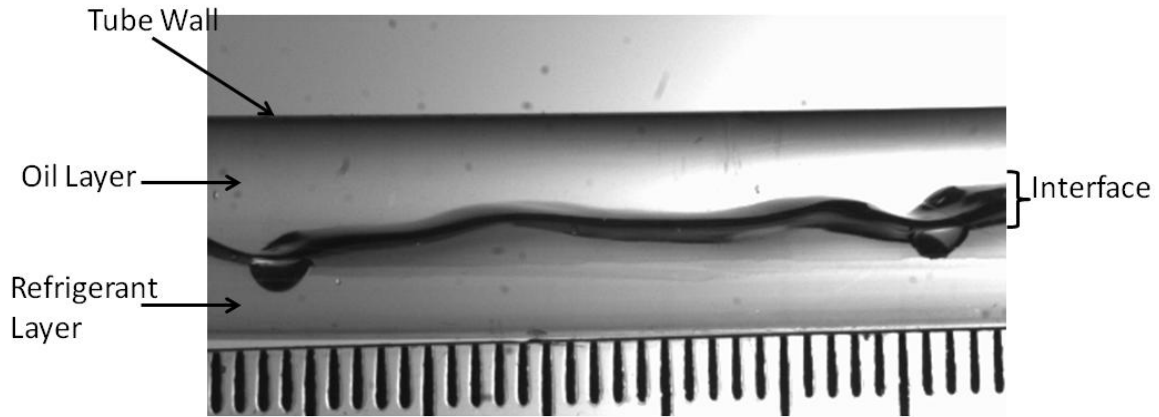


Figure 3.1 Flow visualization in the liquid line

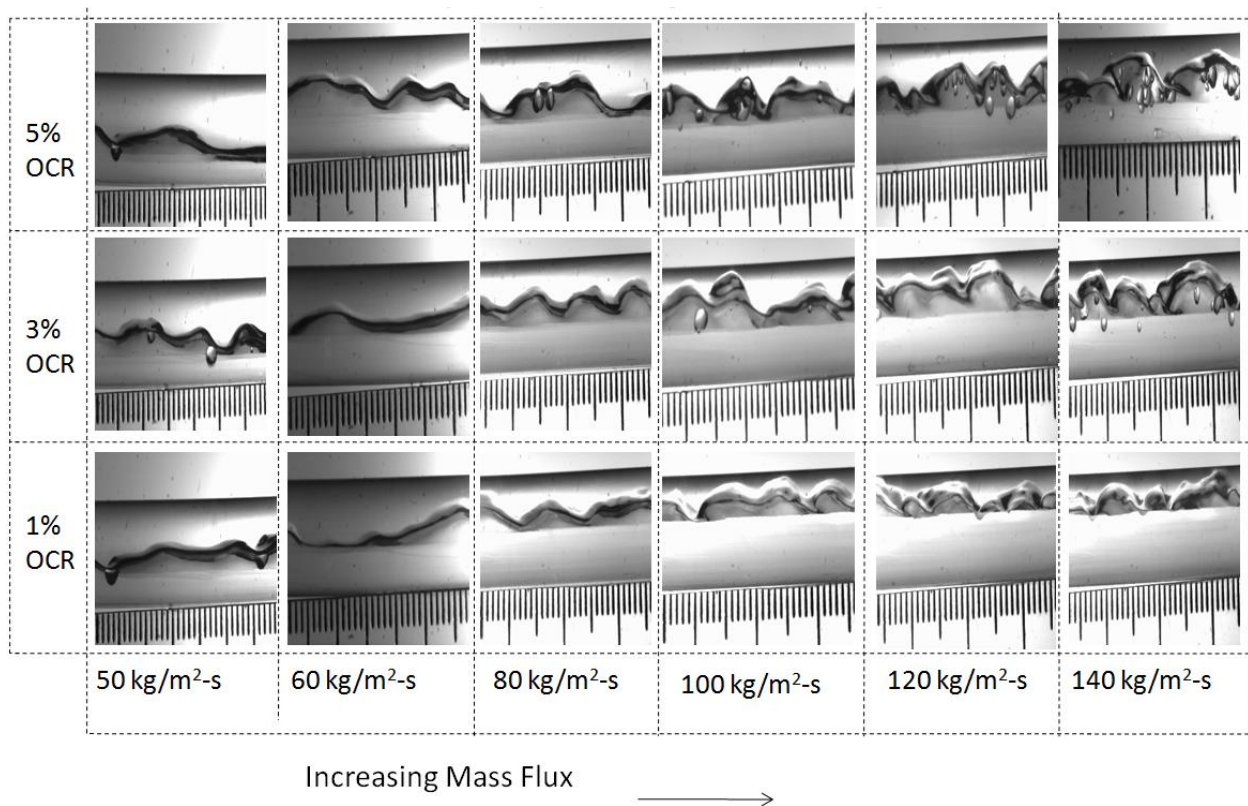


Figure 3.2 Liquid line flow visualization for R134a/AB32

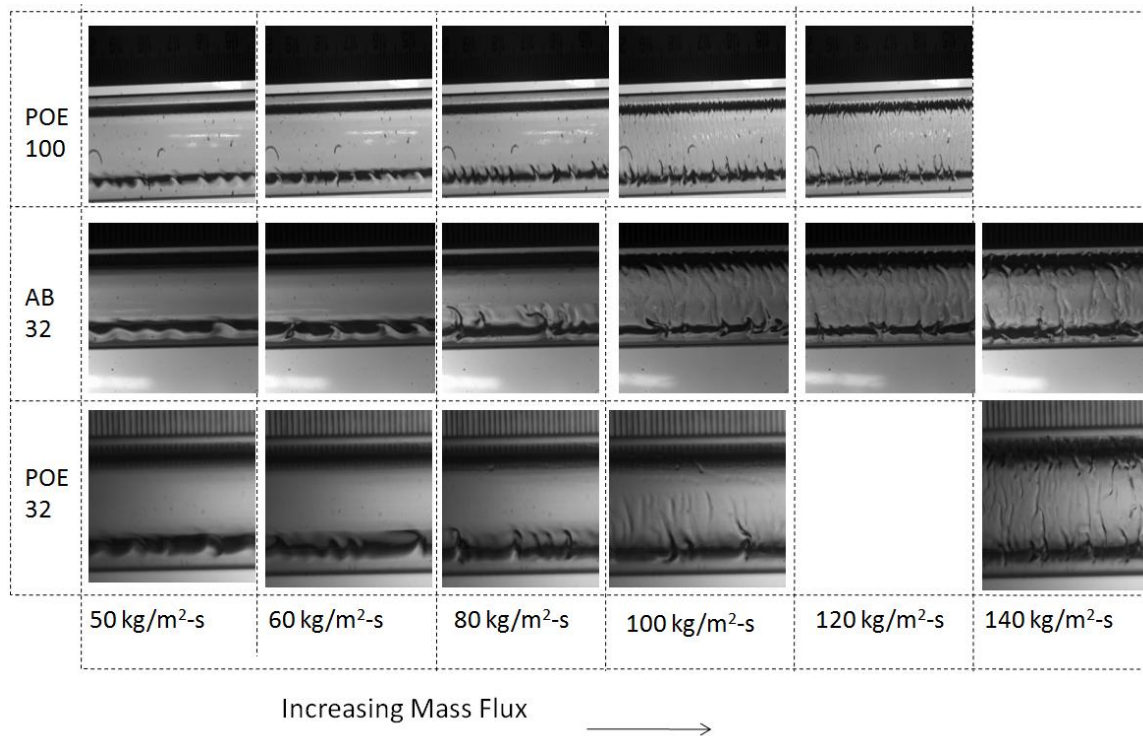


Figure 3.3 Horizontal suction line flow visualization comparison between POE100, AB32 and POE32 with R134a at 1% OCR

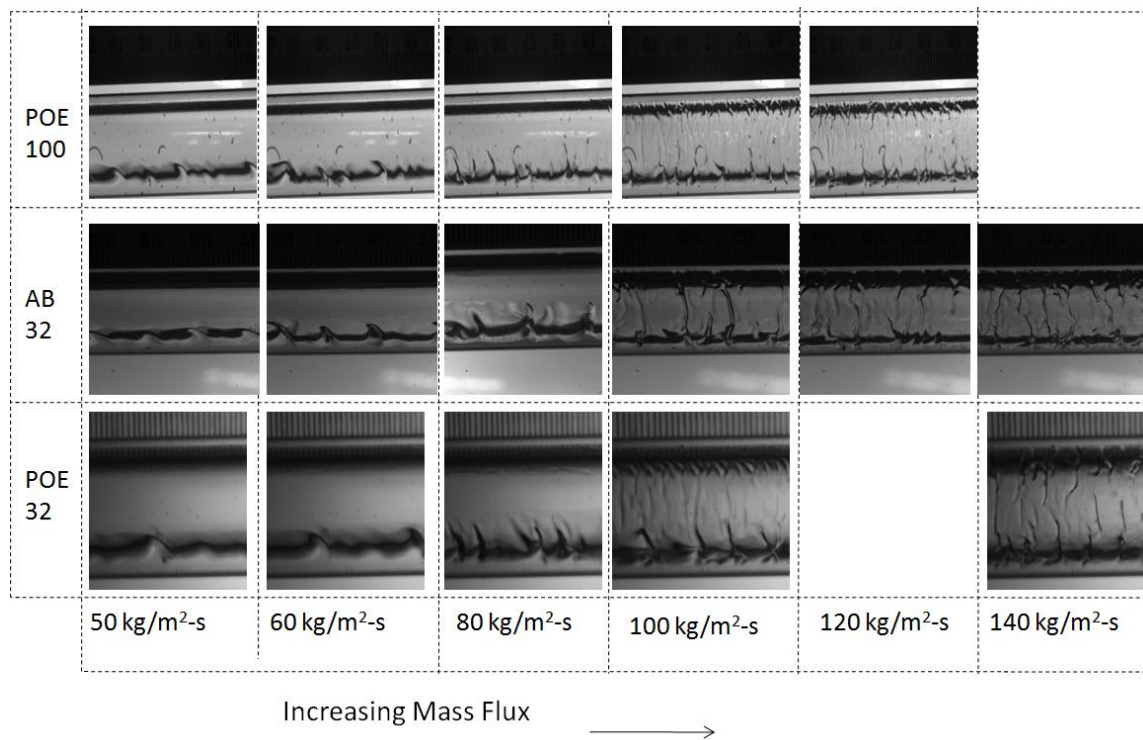


Figure 3.4 Horizontal suction line flow visualization comparison between POE100, AB32 and POE32 with R134a at 3 % OCR

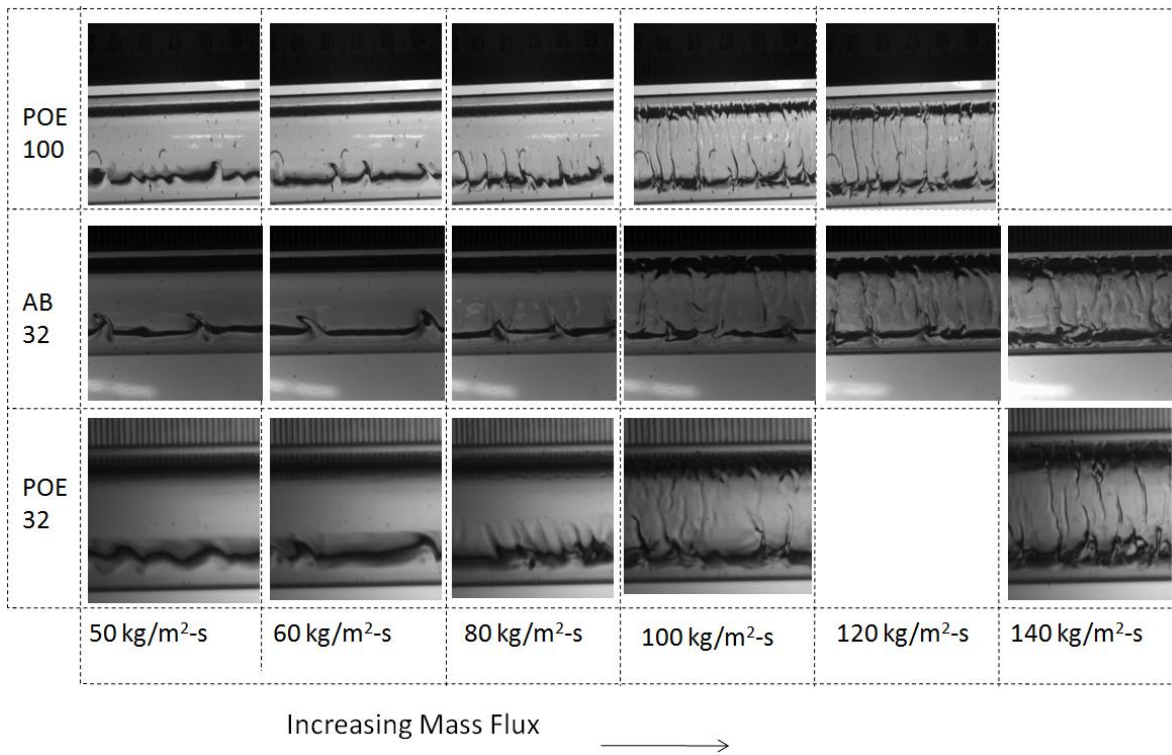


Figure 3.5 Horizontal suction line flow visualization comparison between POE100, AB32 and POE32 with R134a at 5 % OCR

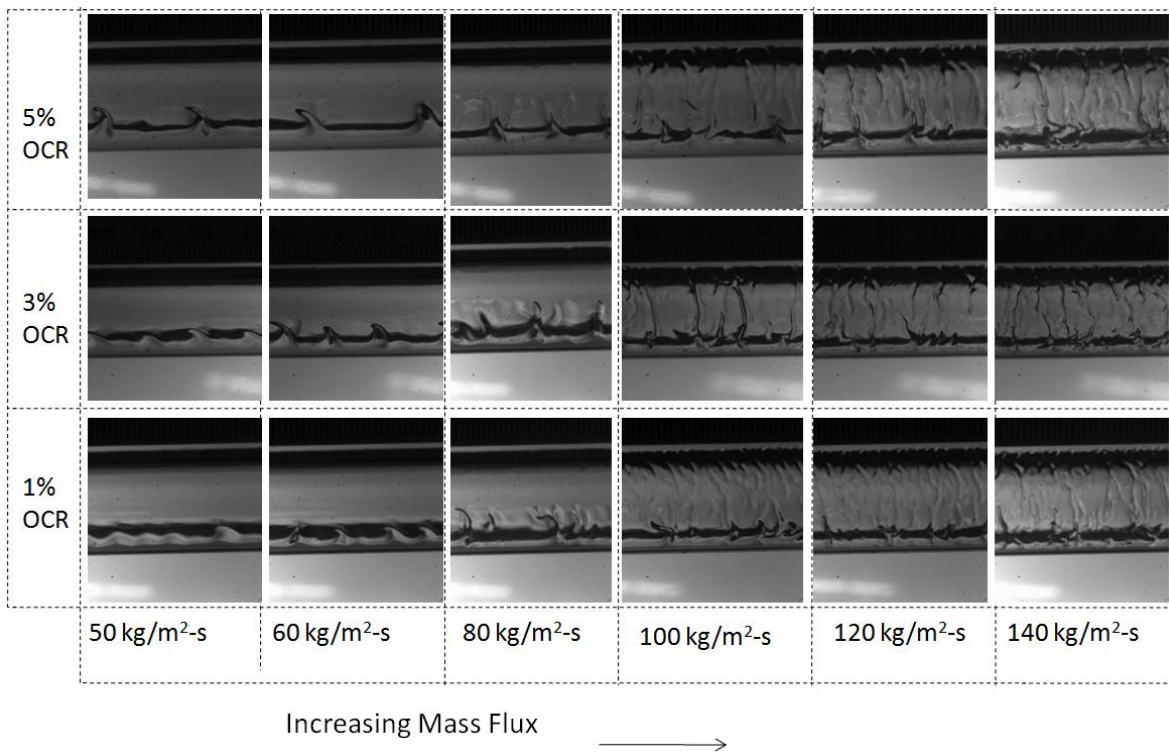


Figure 3.6 Horizontal suction line flow visualization for R134a/AB32

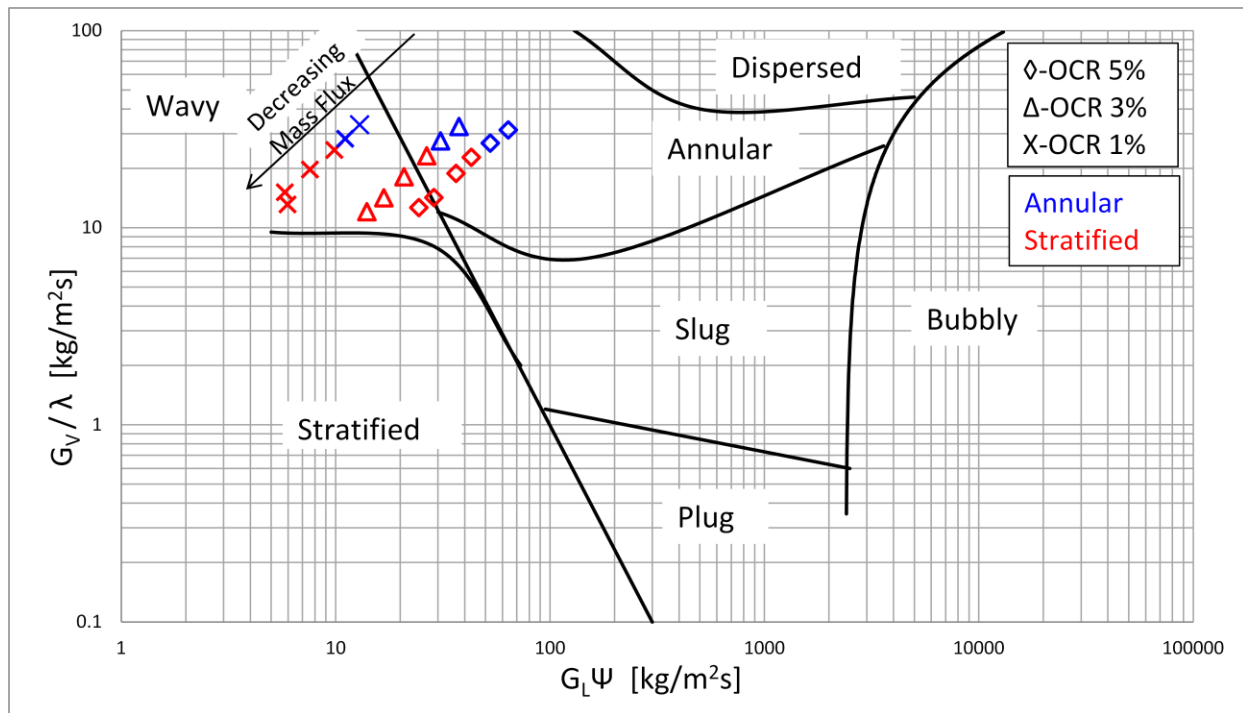


Figure 3.7 Bakers flow map for R134a/AB32 in a 10.2 mm internal diameter horizontal suction line

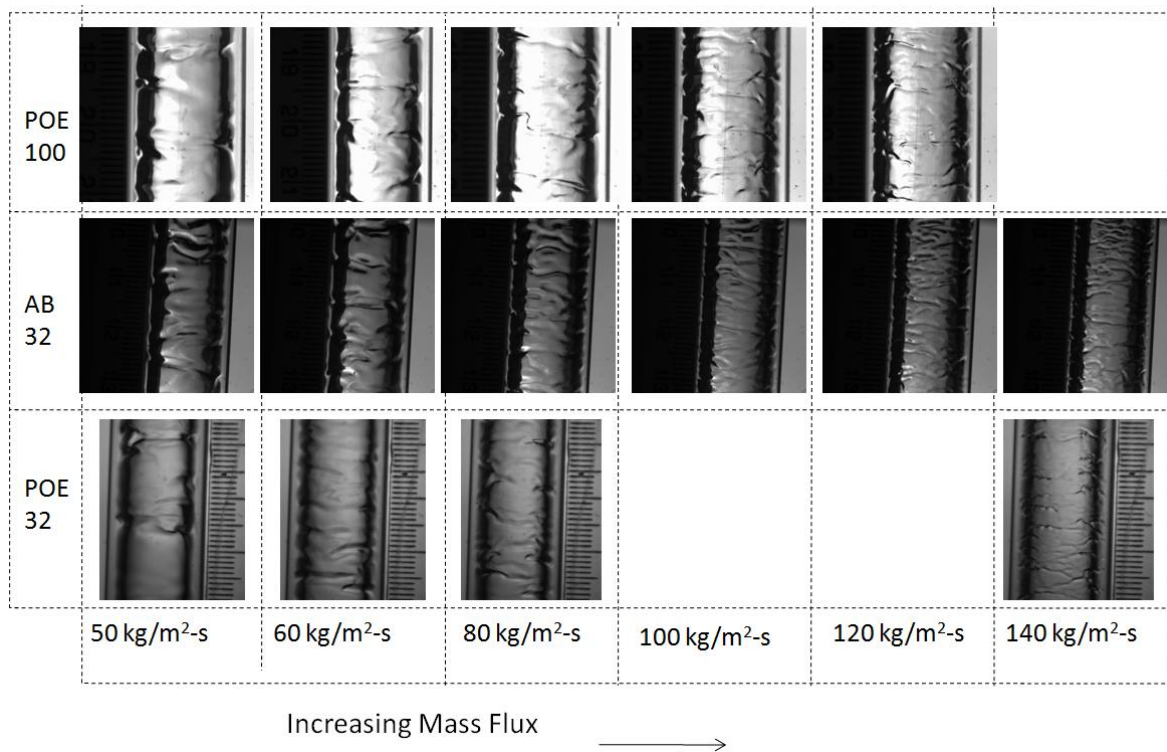


Figure 3.8 Vertical suction line flow visualization comparison between POE100, AB32 and POE32 with R134a at 1% OCR

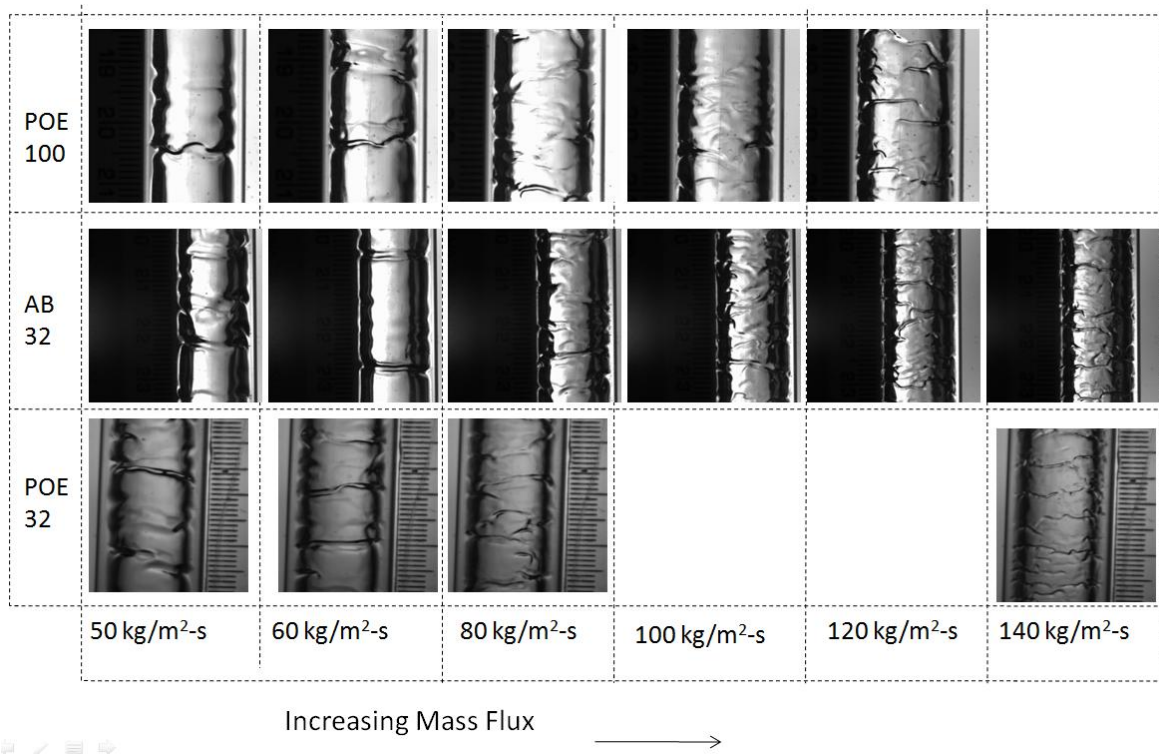


Figure 3.9 Vertical suction line flow visualization comparison between POE100, AB32 and POE32 with R134a at 3% OCR

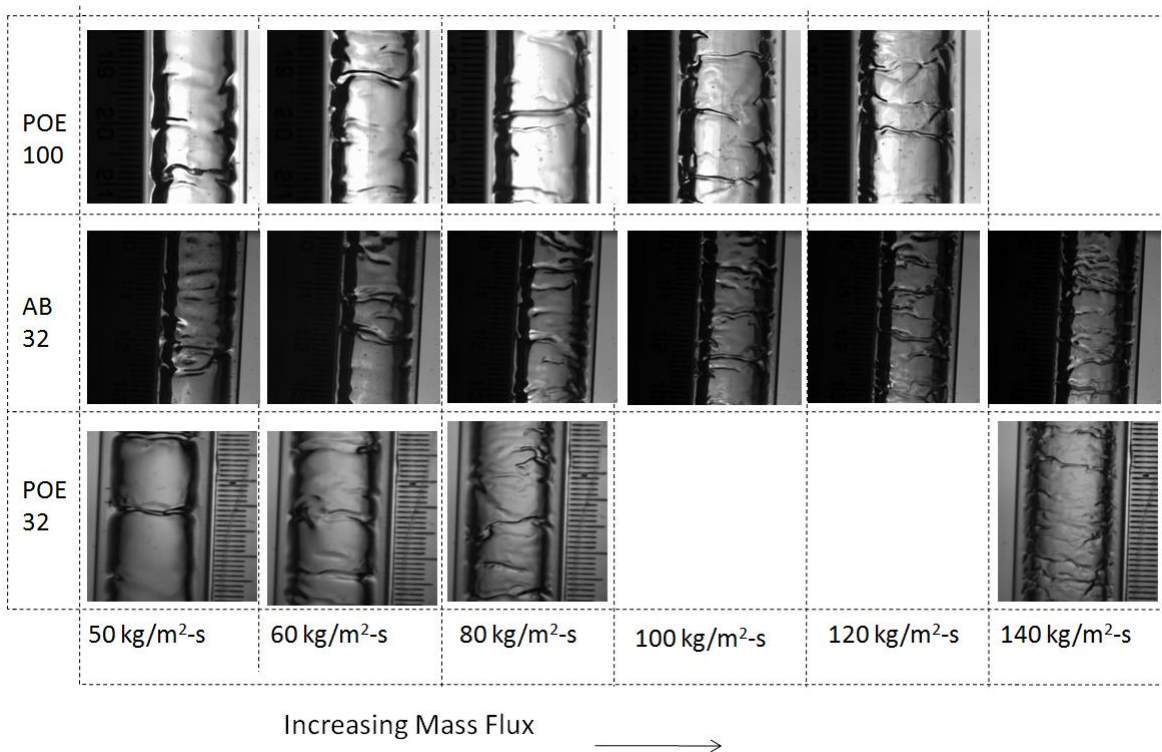


Figure 3.10 Vertical suction line flow visualization comparison between POE100, AB32 and POE32 with R134a at 5% OCR

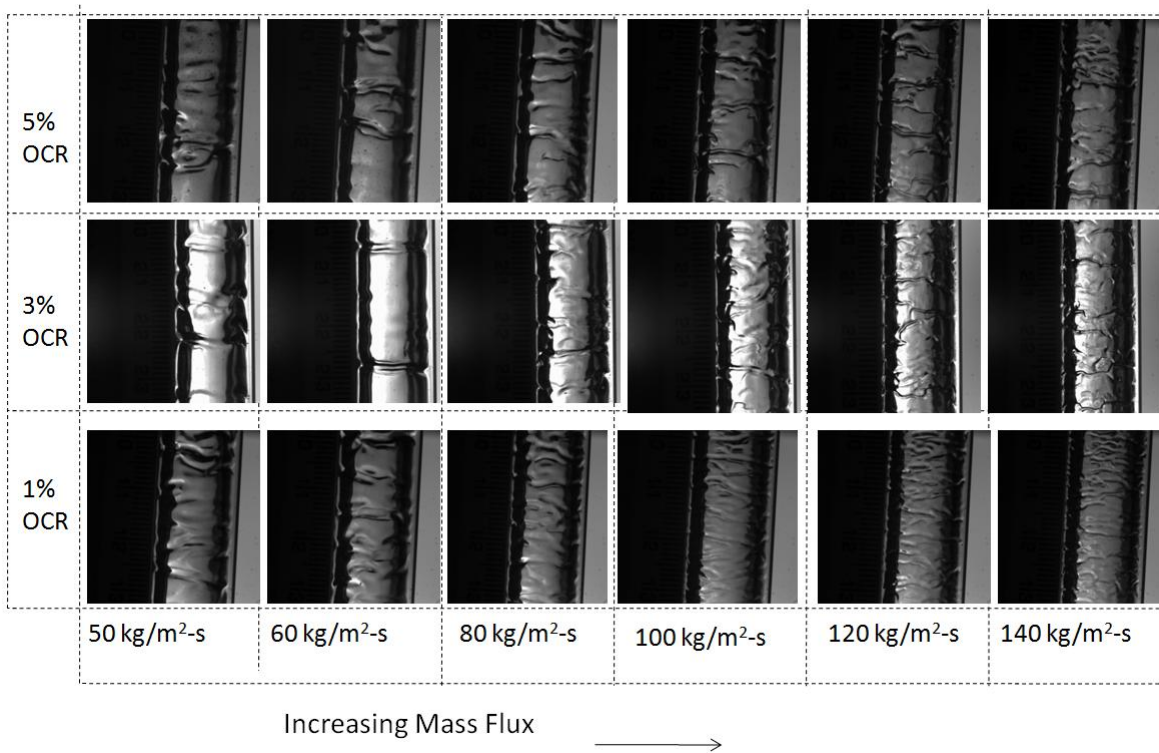


Figure 3.11 Vertical suction line flow visualization for R134a/AB32

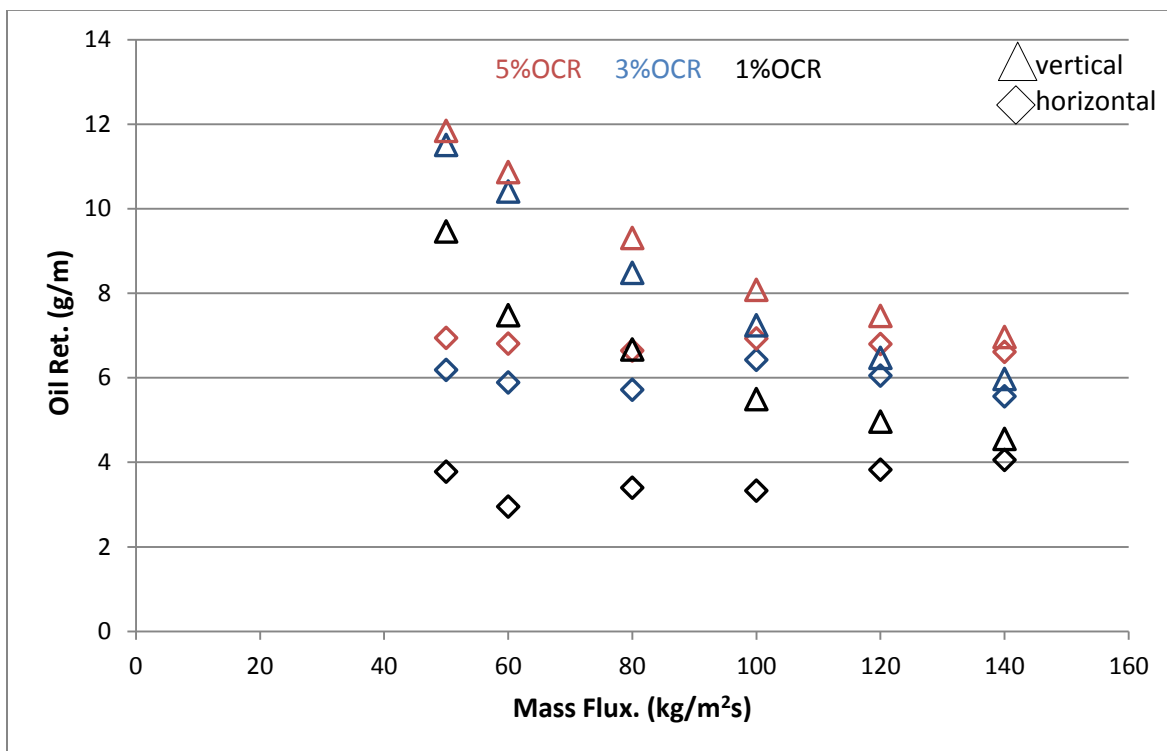


Figure 3.12 Variation of oil retention with mass flux for R134a/AB32

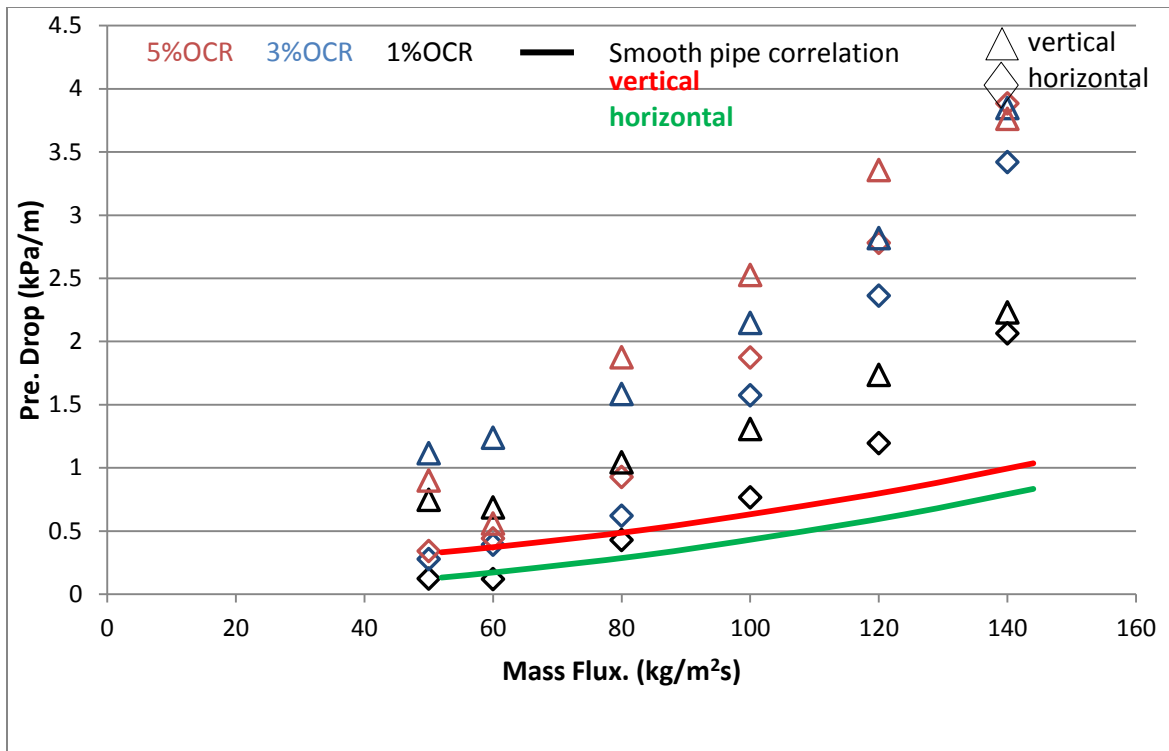


Figure 3.13 Variation of pressure drop with mass flux for R134a/AB32

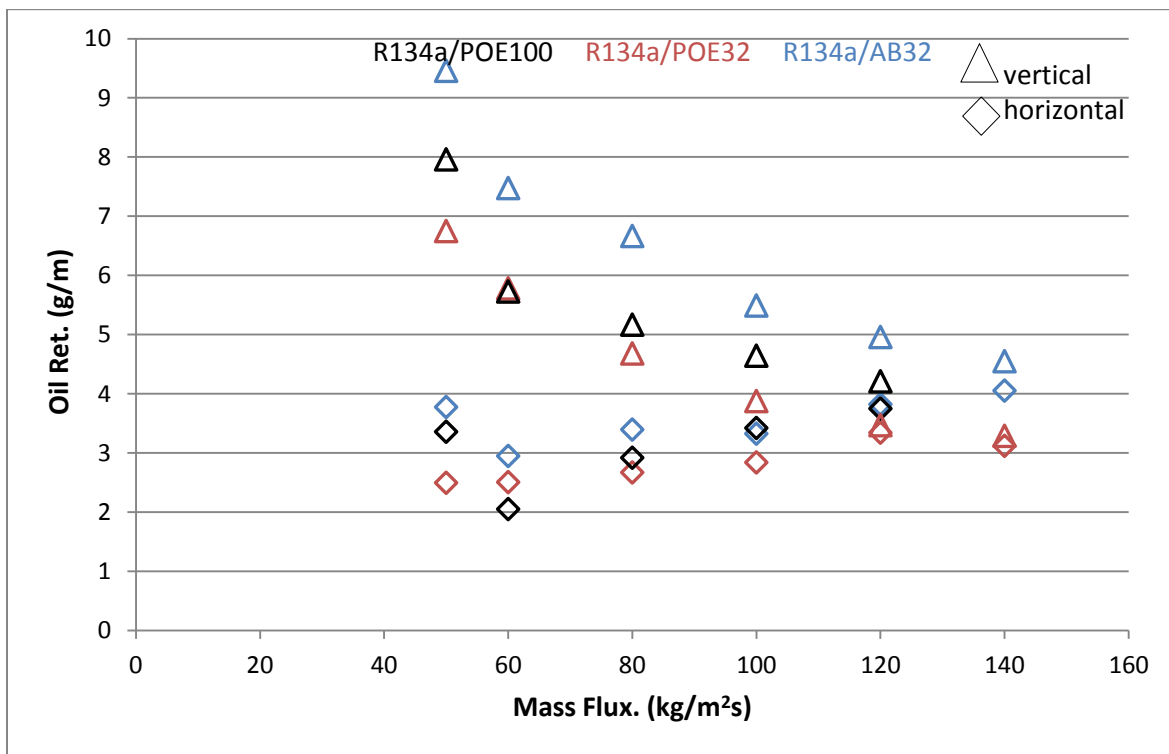


Figure 3.14 Variation of oil retention with mass flux for POE100, POE32 and AB32 with R134a at 1% OCR

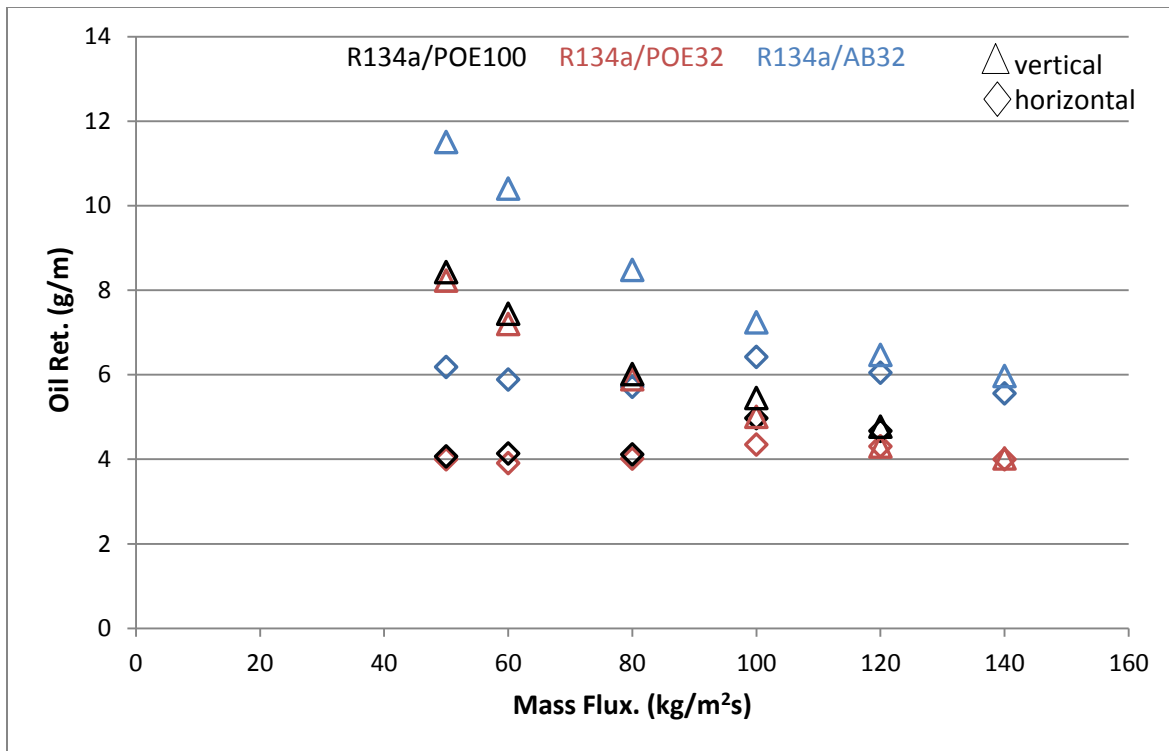


Figure 3.15 Variation of oil retention with mass flux for POE100, POE32 and AB32 with R134a at 3% OCR

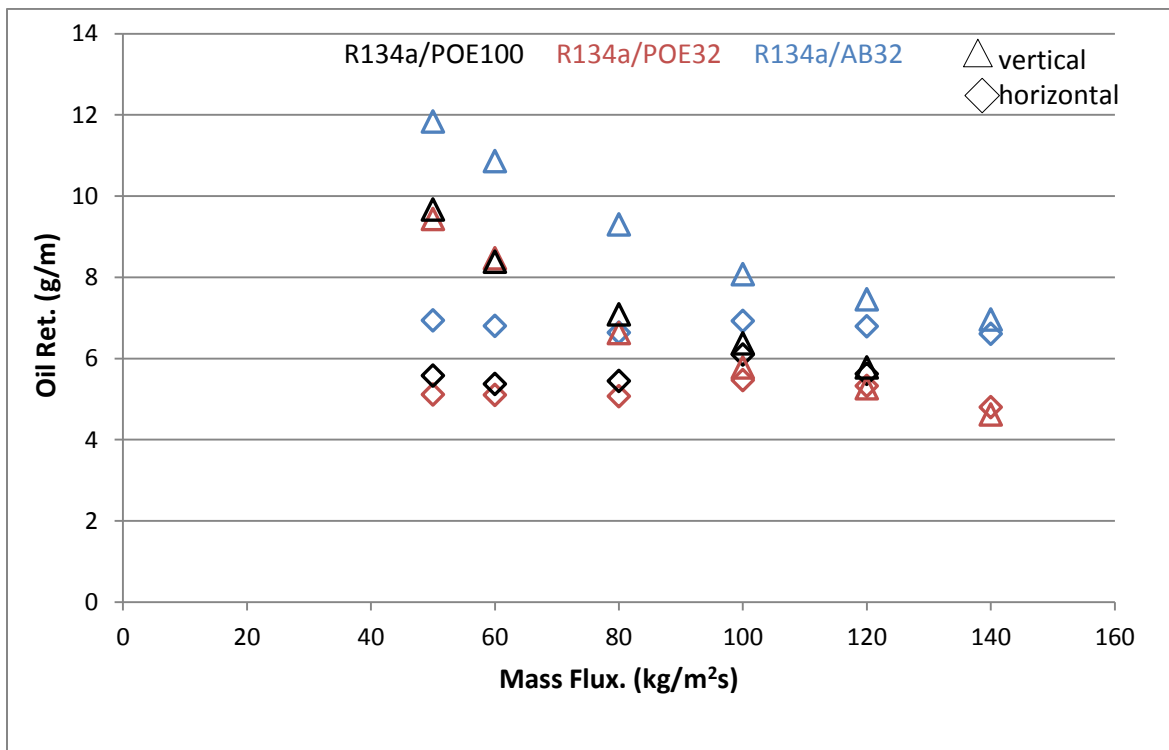


Figure 3.16 Variation of oil retention with mass flux for POE100, POE32 and AB32 with R134a at 5% OCR

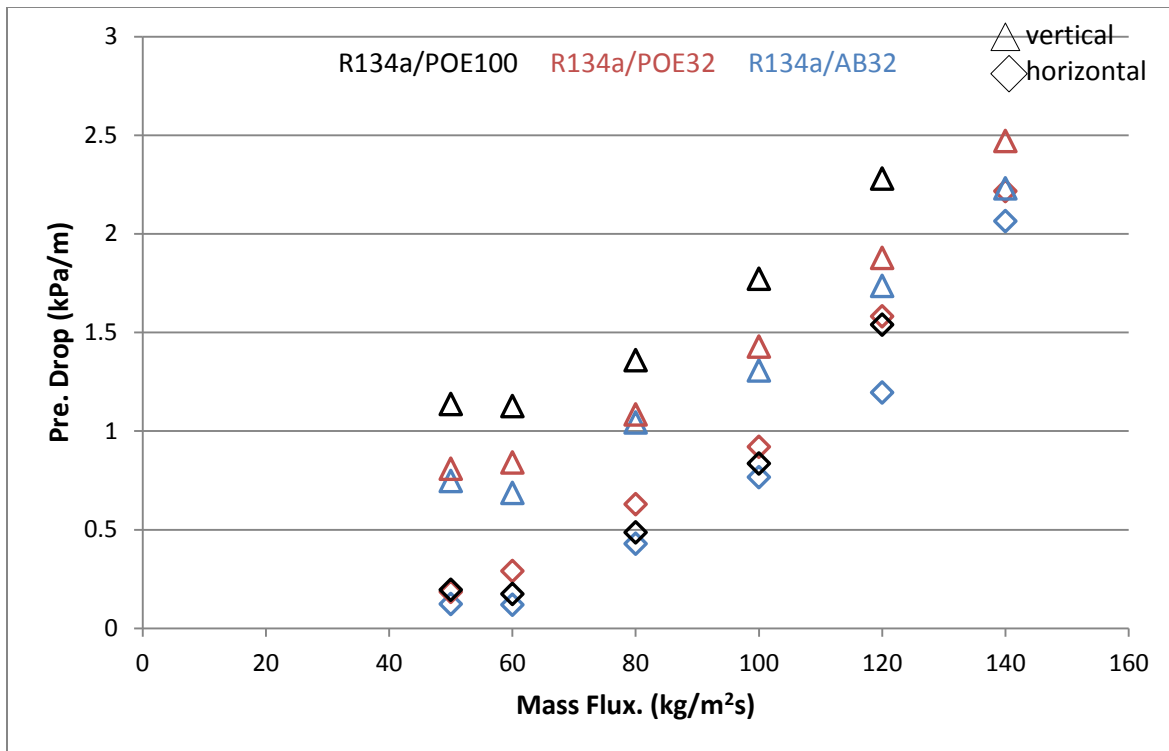


Figure 3.17 Variation of pressure drop with mass flux for POE100, POE32 and AB32 with R134a at 1% OCR

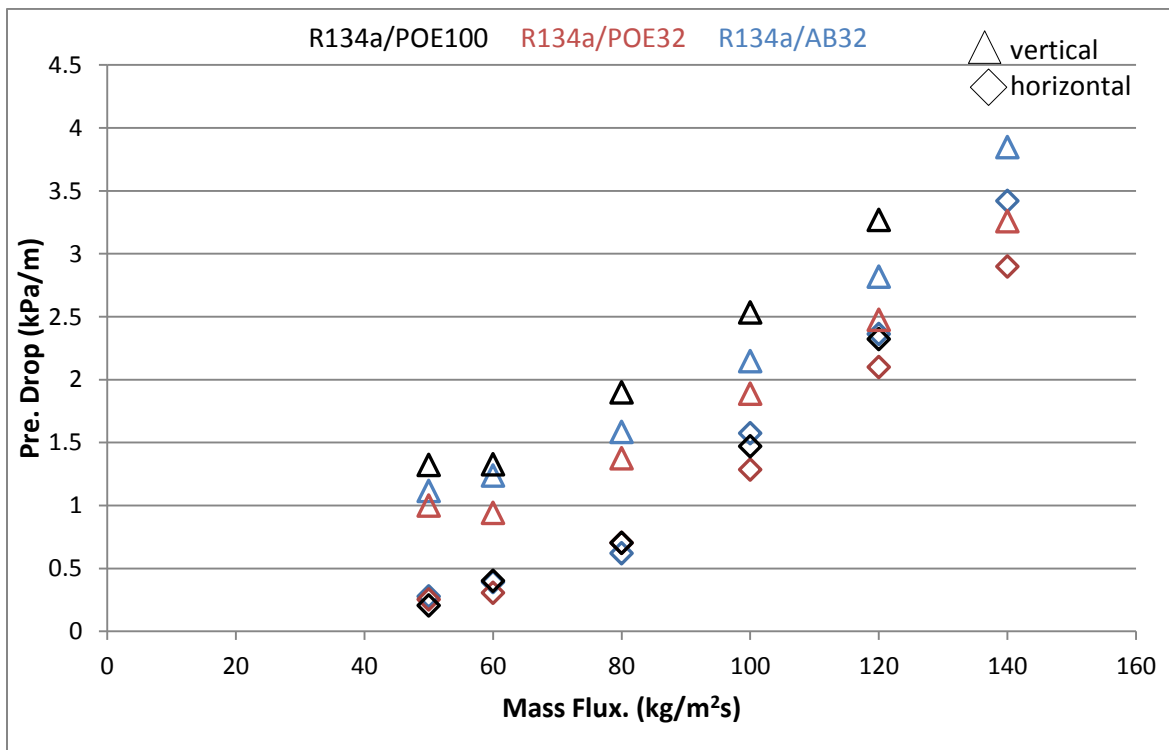


Figure 3.18 Variation of pressure drop with mass flux for POE100, POE32 and AB32 with R134a at 3% OCR

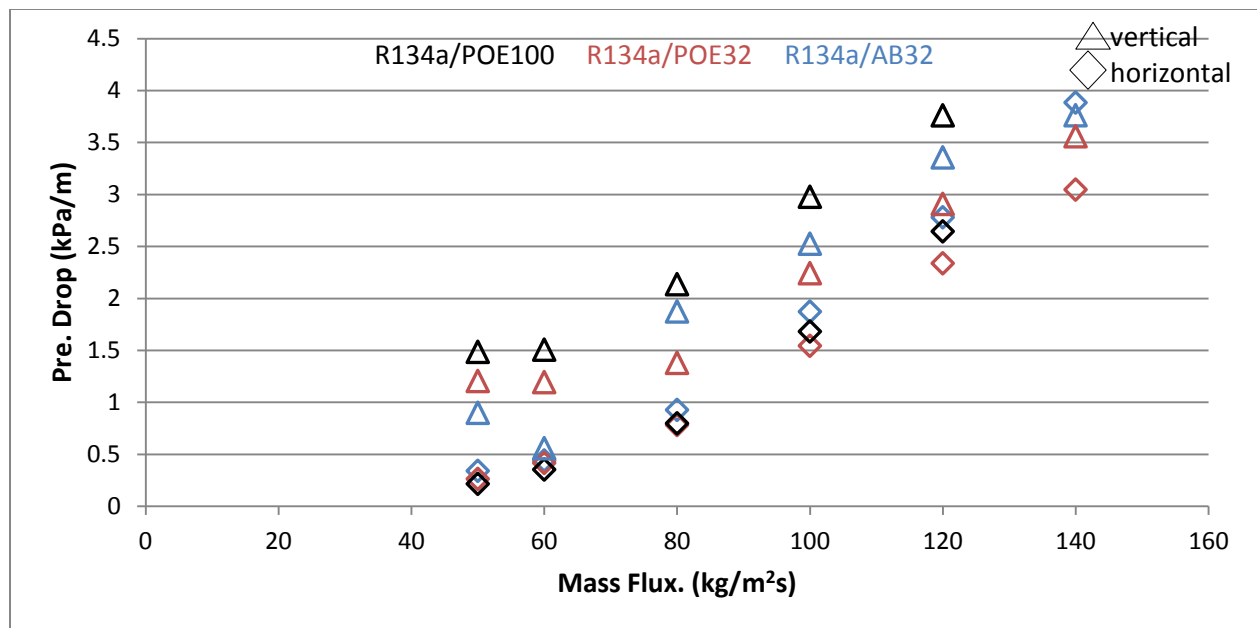


Figure 3.19 Variation of pressure drop with mass flux for POE100, POE32 and AB32 with R134a at 5% OCR

## **CHAPTER 4: MATHEMATICAL MODEL OF OIL RETENTION AND PRESSURE DROP IN VERTICAL SUCTION LINES**

It has been discussed in an earlier chapter, that analytical models are developed for annular flow in vertical suction lines, in order to size them in a better way thereby minimizing oil retention. The aim of this chapter is to present a model, which takes into account the refrigerant mass flux, OCR, lubricant viscosity, and superheat to predict oil retention and pressure drop in vertical suction lines. As a further extension to the model, a criterion for predicting minimum refrigerant mass flux for design of vertical suction lines is also presented.

### **4.1 Model Outline**

A semi-empirical mathematical model was developed in order to predict oil retention and pressure drop in vertical suction lines. The underlying principle of the model has been the basis of many works published earlier. Navier-Stokes and continuity equations were applied to the annular liquid film and vapor core to obtain the liquid film mass flow rate by using appropriate assumptions and boundary conditions. A momentum balance at the refrigerant/lubricant interface was used to correlate the pressure drop to the interfacial shear stress. An interfacial friction factor was then used to correlate interfacial shear stress to the liquid film thickness. This liquid film thickness was then used to predict oil retention. As a part of the current work a new interfacial friction factor correlation has been developed, taking into account data from R134a/POE32, R134a/POE100 and R134a/AB32 studies.

#### **4.1.1 Assumptions Made for Modeling**

- The flow was assumed to be steady, fully developed, adiabatic and incompressible in nature.

- The annular liquid film in the vertical section was assumed to have a constant thickness over the entire tube length.
- Due to the circular geometry of the smooth pipe, the flow is assumed to be axisymmetric.
- The effect of oil droplet entrainment into the vapor core was neglected.

## 4.2 Analytical Derivation of the Model

Figure 4.1 shows a schematic of the flow problem being addressed in this chapter taking all the above mentioned assumptions into account. The Navier-Stokes equations in cylindrical coordinates will be used to model the flow physics of the problem. The schematic shows the direction of the  $r$ ,  $\theta$ , and  $z$  coordinates. The upward flow direction in the vertical suction line is represented by  $z$ , the radial direction is represented by  $r$ , and the  $\theta$  coordinate is neglected as the flow is assumed to be axisymmetric.

### 4.2.1 Navier-Stokes Equations Solved for Liquid Film

The general form of the continuity equation in the cylindrical components is given by

$$\frac{\partial \rho}{\partial t} + \frac{1}{r} \frac{\partial}{\partial r}(\rho r v_r) + \frac{1}{r} \frac{\partial}{\partial \theta}(\rho v_\theta) + \frac{\partial}{\partial z}(\rho v_z) = 0 \quad (4.1)$$

For steady, incompressible flow the continuity equation can be simplified and rewritten as

$$\frac{\partial}{\partial r}(r v_r) = 0 \quad (4.2)$$

Integrating equation (4.2), we have

$$r \cdot v_r = \text{Const.} \quad (4.3)$$

In order to evaluate the constant of integration, the no slip condition can be applied at the inner wall of the tube.

$$R \cdot v_r = 0 \quad (4.4)$$

As a consequence of the above, it can be said that, radial velocity is zero throughout the liquid film

$$v_r = 0, \forall R - \delta \leq r \leq R \quad (4.5)$$

The momentum equation in the flow direction, z, after applying the assumptions, can be written, as

$$\frac{\mu_l}{r} \frac{d}{dr} \left( r \frac{dv}{dr} \right) = \frac{dP}{dz} + \rho_l g_z \quad (4.6)$$

Integrating equation (4.6), we have

$$\mu_l r \left( \frac{dv}{dr} \right) = \left( \frac{dP}{dz} + \rho_l g_z \right) \frac{r^2}{2} + C \quad (4.7)$$

In the above expression, C is the constant of integration.

For a Newtonian fluid, Newton's Law of viscosity can be applied, after simplification as

$$\tau_{rz} = \tau = -\mu_l \frac{dv}{dr} \quad (4.8)$$

Equation (4.8) can be substituted into equation (4.7), on simplification we get

$$-\tau \cdot r = \left( \frac{dP}{dz} + \rho_l g_z \right) \frac{r^2}{2} + C \quad (4.9)$$

The interfacial shear stress,  $\tau_i$ , at the refrigerant vapor core-liquid annular film interface can now be used as a boundary condition and substituted in equation (4.9), thus  $\tau(r = R - \delta) = \tau_i$ , leads to

$$-\tau_i \cdot (R - \delta) = \left( \frac{dP}{dz} + \rho_l g_z \right) \frac{(R - \delta)^2}{2} + C \quad (4.10)$$

Subtracting equation (4.9) from equation (4.10), we have

$$\tau = \tau_i \frac{(R - \delta)}{r} - \frac{1}{2} \left( \frac{dP}{dz} + \rho_l g_z \right) \left( \frac{r^2 - (R - \delta)^2}{r} \right) \quad (4.11)$$

Equation (4.8) is now substituted into equation (4.11), this yields

$$-\mu_l \frac{dv}{dr} = \tau_i \frac{(R - \delta)}{r} - \frac{1}{2} \left( \frac{dP}{dz} + \rho_l g_z \right) \left( \frac{r^2 - (R - \delta)^2}{r} \right) \quad (4.12)$$

Integrating equation (4.12) with respect to  $r$  will result in the velocity profile across the annular film

$$-\mu_l v = \tau_i (R - \delta) \ln r - \frac{1}{2} \left( \frac{dP}{dz} + \rho_l g_z \right) \left( \frac{r^2}{2} - (R - \delta)^2 \ln r \right) + C_1 \quad (4.13)$$

Boundary condition, no slip at inner tube wall  $v(r = R) = 0$ , is used to eliminate  $C_1$

$$C_1 = -\tau_i (R - \delta) \ln R + \frac{1}{2} \left( \frac{dP}{dz} + \rho_l g_z \right) \left( \frac{R^2}{2} - (R - \delta)^2 \ln R \right) \quad (4.14)$$

This value of  $C_1$  can be back substituted into equation (4.13), which on rearranging yields the velocity profile

$$v = \frac{1}{\mu_l} \left[ \left( \tau_i (R - \delta) + \left( \frac{(R - \delta)^2}{2} \left( \frac{dP}{dz} + \rho_l g_z \right) \right) \right) \ln \frac{R}{r} - \frac{1}{4} \left( \frac{dP}{dz} + \rho_l g_z \right) (R^2 - r^2) \right] \quad (4.15)$$

Integrating the liquid film velocity across the annular film cross section yields the mass flow rate of the film

$$\dot{m}_l = \int_{R-\delta}^R 2\pi v \rho_l r dr \quad (4.16)$$

Substituting for  $v$  from equation (4.15) into equation (4.16) and integrating, we have

$$\begin{aligned} \dot{m}_l = \frac{2\pi\rho_l}{\mu_l} & \left[ \left( \tau_i (R - \delta) + \left( \frac{(R - \delta)^2}{2} \left( \frac{dP}{dz} + \rho_l g_z \right) \right) \right) \times \left( \frac{(R^2 - (R - \delta)^2)}{4} - \frac{(R - \delta)^2}{2} \ln \frac{R}{R - \delta} \right) \right] \\ & - \left[ \frac{\pi\rho_l}{8\mu_l} \left( \frac{dP}{dz} + \rho_l g_z \right) (R^2 - (R - \delta)^2)^2 \right] \end{aligned} \quad (4.17)$$

The above expression for mass flow rate is constituted of three variables, namely, interfacial shear stress ( $\tau_i$ ), pressure gradient ( $dP/dz$ ) and liquid film thickness ( $\delta$ ). The equations for pressure gradient and interfacial shear stress will be solved for in the subsequent sections.

#### 4.2.2 Refrigerant Vapor Core Momentum Balance

Figure 4.1 shows the components of the momentum balance on the refrigerant vapor core. Doing a force balance on the refrigerant vapor core, we have

$$\frac{dP}{dz} + \rho_v g_z + \frac{\pi\tau_i D_c}{A_c} = 0 \quad (4.18)$$

The void fraction,  $\alpha$ , is given by the equation

$$\alpha = \frac{A_c}{A} = \left( \frac{D_c}{D} \right)^2 = \left( \frac{D - 2\delta}{D} \right)^2 \quad (4.19)$$

Substituting equation (4.19) into equation (4.18), we have

$$\frac{dP}{dz} + \rho_v g_z + \frac{4\tau_i}{D\sqrt{\alpha}} = 0 \quad (4.20)$$

The interfacial shear stress ( $\tau_i$ ), is related to the pressure drop ( $dP/dz$ ), by equation (4.20). The interfacial friction factor correlation is used to relate the interfacial shear stress to the annular film thickness.

#### 4.2.3 Developing a Correlation for the Interfacial Friction Factor

As discussed in previous sections, a correlation for the interfacial friction factor acts as a closure equation for the above mentioned set of equations. Literature review has shown that annular flows have been analytically studied using interfacial friction factor correlations. Wallis (1969) proposed a correlation for the interfacial friction factor, which has come to become the basis for many studies previously conducted. The Wallis correlation under predicts the smooth pipe friction factor,  $f_i$ , for non-dimensional film thickness,  $\delta/D \geq 0.02$  and over predicts it for  $\delta/D \leq 0.005$  (Belt *et al.* 2009). Wongwises and Kongkiatwanitch (2001) proposed a correlation, which related  $f_i$  with the refrigerant vapor Reynolds number and the dimensionless liquid film thickness. This was also a widely used correlation, where it was shown to predict the experimental  $f_i$  values to within  $\pm 25\%$ . For the purpose of the current work a correlation with a similar form to Asali *et al.* (1985) has been proposed, which correlates the interfacial friction factor with the vapor core Reynolds number, the liquid film Reynolds number, the non-

dimensional film thickness and the mixture Weber number. Thus, the proposed correlation has the form

$$\frac{f_i}{f_s} = 1 + K \text{Re}_{lf}^a \text{Re}_v^b \delta_v^{+c} \text{We}_{mix}^d \quad (4.21)$$

With the non-dimensional parameters being

$$f_s = 0.046 \text{Re}_v^{-0.2} \quad (4.22)$$

$$\text{Re}_{lf} = \frac{G(1-x)D}{4\mu_l} \quad (4.23)$$

$$\text{Re}_v = \frac{GxD}{\mu_v} \quad (4.24)$$

$$\delta_v^+ = \frac{\delta}{\nu_v} v^* \quad (4.25)$$

Where  $v^*$  is given by the expression

$$v^* = \sqrt{\frac{\tau_i}{\rho_v}} \quad (4.26)$$

$$\text{We}_{mix} = \frac{G^2 D}{\sigma_{mix} \rho_l} \quad (4.27)$$

Cremaschi (2004), suggested a formula for  $\sigma_{mix}$ , which has been used in the present work to evaluate the mixture Weber number given by the equation (4.27).

$$\sigma_{mix} = \sigma_{ref,liq} + (\sigma_{oil} - \sigma_{ref,liq}) \sqrt{w_{local}} \quad (4.28)$$

EES is used to calculate the surface tension of the pure liquid refrigerant,  $\sigma_{ref,liq}$ . Sunami *et al.* (1994) measured the surface tension of branched and linear alkylbenzene lubricants. For the modeling the surface tension of pure AB oil was take to be equal to 29 mN/m. To compare the data of Sethi and Hrnjak (2011) and Ramakrishnan and Hrnjak (2012), the surface tension of POE was assumed to be 46 mN/m as quoted by Cremaschi (2004). The interfacial shear stress  $\tau_i$ , is related to  $f_i$ , by the following expression

$$\tau_i = \frac{1}{2} f_i \rho_v (v_v - v_l)^2 \quad (4.29)$$

Because average vapor velocity is much higher than average liquid film velocity, equation (4.29) can be modified to

$$\tau_i = \frac{1}{2} f_i \rho_v v_v^2 \quad (4.30)$$

Where

$$v_v = \frac{Gx}{\rho_v \alpha} \quad (4.31)$$

The coefficients in equation (4.21), namely, a, b, c, d, and K, were determined by performing a least square regression analysis using Microsoft Excel on the experimental data collected for the R134a/AB32 refrigerant/lubricant combination. The correlation resulting from the regression analysis is given as under

$$\frac{f_i}{f_s} = 1 + 1.305 \times 10^{40} \text{Re}_{lf}^{-0.98904} \text{Re}_v^{-10.3257} \delta_v^{+3.55778} \text{We}_{mix}^{4.97924} \quad (4.32)$$

It must be noted, the friction factor is only applicable for vertically upward flow in suction lines with annular flow only. The coefficients for equation (4.32), evaluated using Microsoft Excel, do not take into account the flow physics of the problem. These are just values obtained that fit the data set used to calculate them. It must however be noted, during the process of modeling the coefficients in the above correlation must make physical sense. For example, the coefficient for the liquid film Reynolds number,  $Re_{lf}$ , must be negative. This can be understood based on the results and comparing different refrigerant/lubricant mixtures. This is also consistent with the signs of the coefficients for different non-dimensionless numbers for the correlations proposed by Sethi and Hrnjak (2011) and Ramakrishnan and Hrnjak (2012).

During the process of modeling, it was assumed that R134a refrigerant vapor is not completely immiscible in AB32 liquid oil, in other words some vapor refrigerant is soluble in the oil. During modeling, the viscosity of the oil rich liquid mixture was taken to be lower than the viscosity of pure oil and the density was taken to higher than that of pure oil. This process yielded the correlation given in equation (4.32). Here, the coefficients of the various non-dimensionless numbers make physical sense. This assumption is also consistent with the observations during experimentation. During the process of vacuuming the test sections to calculate the amount of oil retained it was observed that refrigerant vapor is bubbled off from the oil. Hence, it showed that some refrigerant was dissolved in the oil.

From this analysis, it can be concluded that the assumption that AB32 is completely immiscible in R134a is incorrect. It was shown in chapter 3 that liquid R134a is immiscible in liquid AB32 oil based on the flow visualizations, but the modeling suggests that the same cannot be assumed for vapor R134a and liquid AB32 oil.

### 4.3 Calculations for Predicting Oil Retention and Pressure Drop

In order for the suggested model to compute oil retention and pressure drop, operating conditions like system saturation pressure, superheated evaporator outlet temperature, total mass flux and oil circulation ratio (OCR) have to be provided as inputs. Thermophysical properties of the refrigerant and lubricant are also required at the operating conditions. The OCR was calculated in a similar way as described in Zoellick and Hrnjak (2010), the only difference for the current case is that the refrigerant/lubricant mixture was immiscible. The oil tank was assumed to contain only pure lubricant. In some cases some liquid refrigerant was found settled at the bottom of the oil layer, but this was entirely pumped out by running the micro pump for sufficient time, and only then was OCR measurement started. The formula used was

$$OCR = \frac{\dot{m}_{oil}}{\dot{m}_{oil} + \dot{m}_{ref}} \quad (4.33)$$

The local oil concentration and the bulk vapor quality at the exit to the evaporator was also required to be calculated for the model. Even though superheated conditions exist, and the refrigerant/lubricant combination is assumed to be completely immiscible, there will always be minute quantities of refrigerant vapor dissolved in the liquid film (this was also seen when vacuuming the test sections at the end of the test to remove refrigerant, bubbling was observed from the oil, suggesting some dissolved refrigerant). This would cause the bulk vapor quality to be lower than unity. The local oil concentration was evaluated by using the method developed by Thome (1995). The set of equations used in the model as described below

$$T_{bub} = \frac{A(w_{local})}{\ln(P_{sat}) - B(w_{local})} \quad (4.34)$$

$$A(w_{local}) = a_0 + a_1 w_{local} + a_2 w_{local}^3 + a_3 w_{local}^5 + a_4 w_{local}^7 \quad (4.35)$$

$$B(w_{local}) = b_0 + b_1 w_{local} + b_2 w_{local}^3 + b_3 w_{local}^5 + b_4 w_{local}^7 \quad (4.36)$$

Where,

$$a_1 = 182.52$$

$$b_1 = -0.72212$$

$$a_2 = -724.21$$

$$b_2 = 2.3914$$

$$a_3 = 3868$$

$$b_3 = -13.779$$

$$a_4 = -5268.9$$

$$b_4 = 17.066$$

Equation (4.34) was used in conjunction with an equation for the pure refrigerant vapor pressure to evaluate the values of  $a_0$  and  $b_0$ . The vapor quality was calculated using the equation

$$w_{local} = \frac{OCR}{1 - x} \quad (4.37)$$

Once the film thickness and pressure gradient is obtained by solving the preceding equations, the amount of oil in the suction can be estimated by the equation as follows

$$m_{oil} = (w_{local}) \cdot 2\pi R \delta L \rho_l \quad (4.38)$$

#### 4.4 Validating the Proposed Model

Figures 4.2 and 4.3 show the validation of oil retention and pressure drop predictions using different sets of experimental data for the proposed model. The model was validated using three data sets, namely, the current experimental data for R134a/AB32, data from Sethi and Hrnjak (2011) for R134a/POE32 and data from Ramakrishnan and Hrnjak (2012) for R134a/POE100. It

can be seen that more than 90% of the data points were predicted within a range of  $\pm 40\%$  in case of oil retention data and  $\pm 50\%$  in case of pressure drop data.

Figures 4.4 and 4.5 show the variation of oil retention and pressure drop with respect to mass flux respectively for both the experimental and the model prediction data for R134a/AB32 refrigerant/lubricant combination. It can be seen that the model is able to predict the oil retention well for 3% and 5% OCRs at all mass fluxes, but for 1% OCR the model is unable to predict mass fluxes lower than  $80 \text{ kg/m}^2\text{-s}$ . The model over predicts the pressure drop data for all the OCRs, but the trends are very similar to the experimental results.

#### **4.5 Prediction of Critical Refrigerant Mass Flux**

It has been demonstrated by Sethi and Hrnjak(2011) and Ramakrishnan and Hrnjak (2012) that Jacob's critical flux limit is characterized by high oil retention. It has been mentioned in Chapter 3, for the present study it was not possible to operate at a low enough mass flux to attain the Jacob's mass flux due to the minimum flow rate restriction of the system. At the Jacob's flux (when the churn flow regime is observed) the oil retention is extremely high as compared to other higher mass fluxes (when the annular flow regime is observed). An ideal value of the critical mass flux would be one where the flow of the liquid in the annular film just starts to reverse. At this point the pressure drop is minimum and the oil retention is much lower than the churn flow regime, this has been demonstrated by Sethi and Hrnjak (2011) and Ramakrishnan and Hrnjak (2012).

##### **4.5.1 Development of the Model**

The model presented here follows the same approach as adopted by Sethi and Hrnjak (2011) and Ramakrishnan and Hrnjak (2012). As mentioned, a suitable limit for the critical mass flux would

be when the annular film of liquid in the vertical suction line just starts to reverse in its flow direction. The shear stress at the wall will become zero at this limit. The wall shear stress ( $\tau_w$ ) is obtained at the tube wall from the Equation (4.11) as follows

$$\tau_w = \tau_i \frac{(R-\delta)}{R} - \frac{1}{2} \left( \frac{dP}{dz} + \rho_l g_z \right) \left( \frac{R^2 - (R-\delta)^2}{R} \right) \quad (4.39)$$

At point of flow reversal,  $\tau_w$  becomes zero,  $\delta^2$ , can be assumed to be negligible, then equation (4.39) simplifies to

$$\tau_i \left( \frac{R-\delta}{R} \right) = \frac{1}{2} \left( \frac{dP}{dz} + \rho_l g_z \right) (2R-\delta) \quad (4.40)$$

As earlier, during the oil retention and pressure drop predictions, here too operating conditions need to be input into the model. The parameters needed are the OCR, the degree of superheat, diameter and length of suction line and thermophysical properties of the liquid vapor components. The model developed for the prediction of oil retention and pressure is modified to include the following equation in order to predict the critical mass flux condition.

$$\dot{m}_v = \pi \alpha \rho_v \frac{D^2}{4} v_v \quad (4.41)$$

$$x = \frac{\dot{m}_v}{\dot{m}_v + \dot{m}_l} \quad (4.42)$$

Where  $x$  is the inlet vapor quality,  $\dot{m}_v$  and  $\dot{m}_l$  are the vapor core and liquid film mass flow rates respectively. The critical refrigerant mass flux is given by

$$G_{critical} = \frac{\dot{m}_v + \dot{m}_l (1 - w_{local})}{\pi \frac{D^2}{4}} \quad (4.43)$$

The proposed critical mass flux limit is plotted in Figures 4.9 and 4.10 along with the vertical suction line oil retention and pressured drop data respectively.

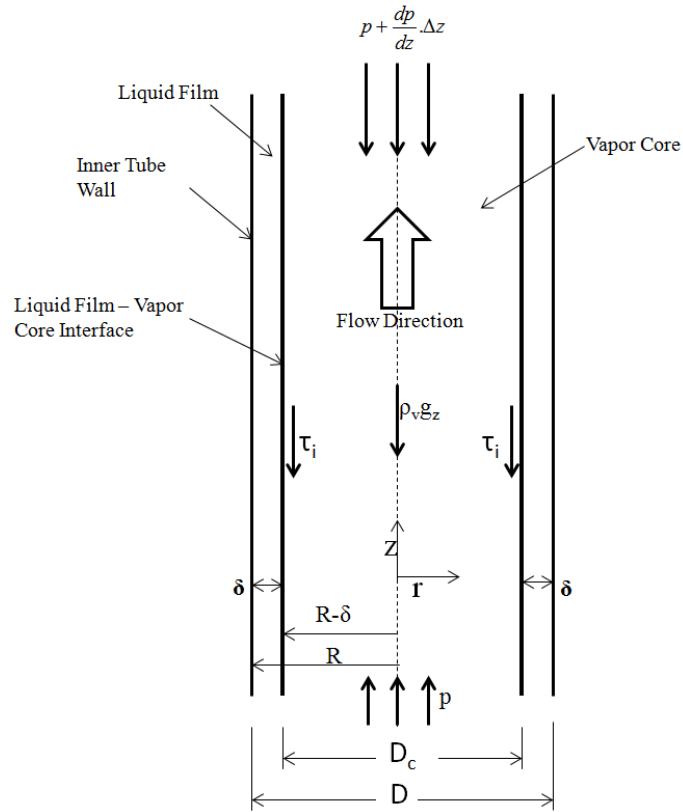


Figure 4.1 Schematic of the annular flow in the vertical suction line

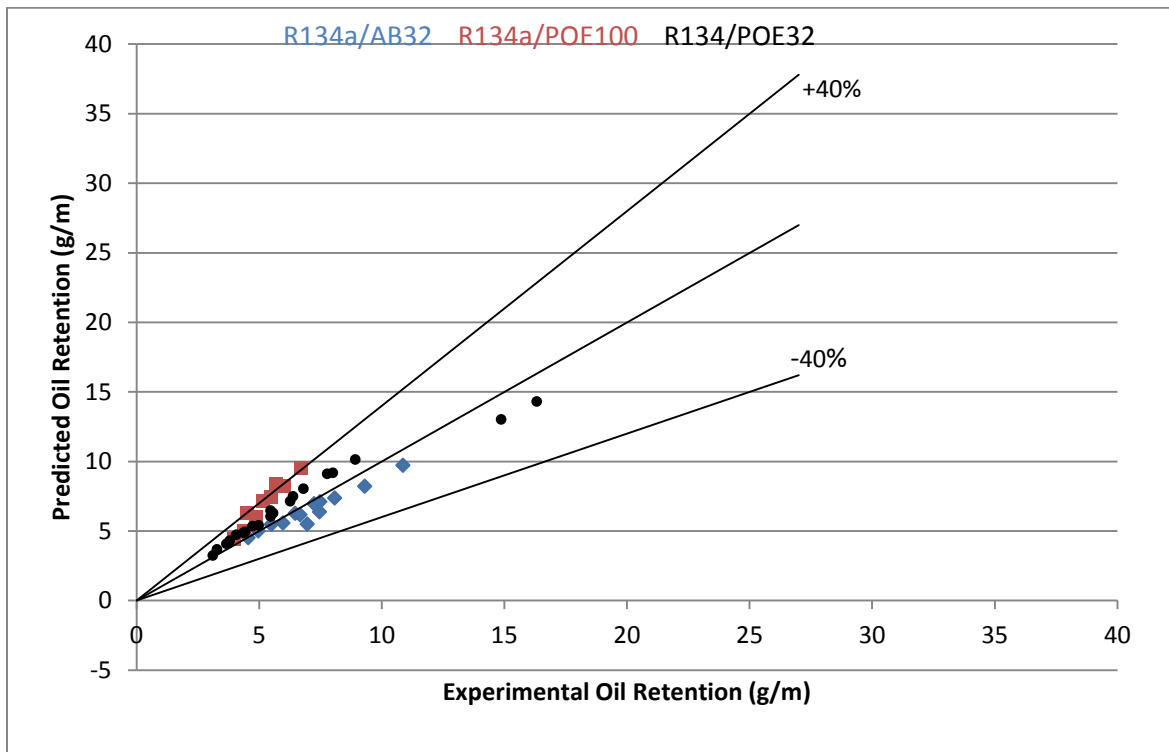


Figure 4.2 Validation of the proposed model for oil retention in vertical suction lines

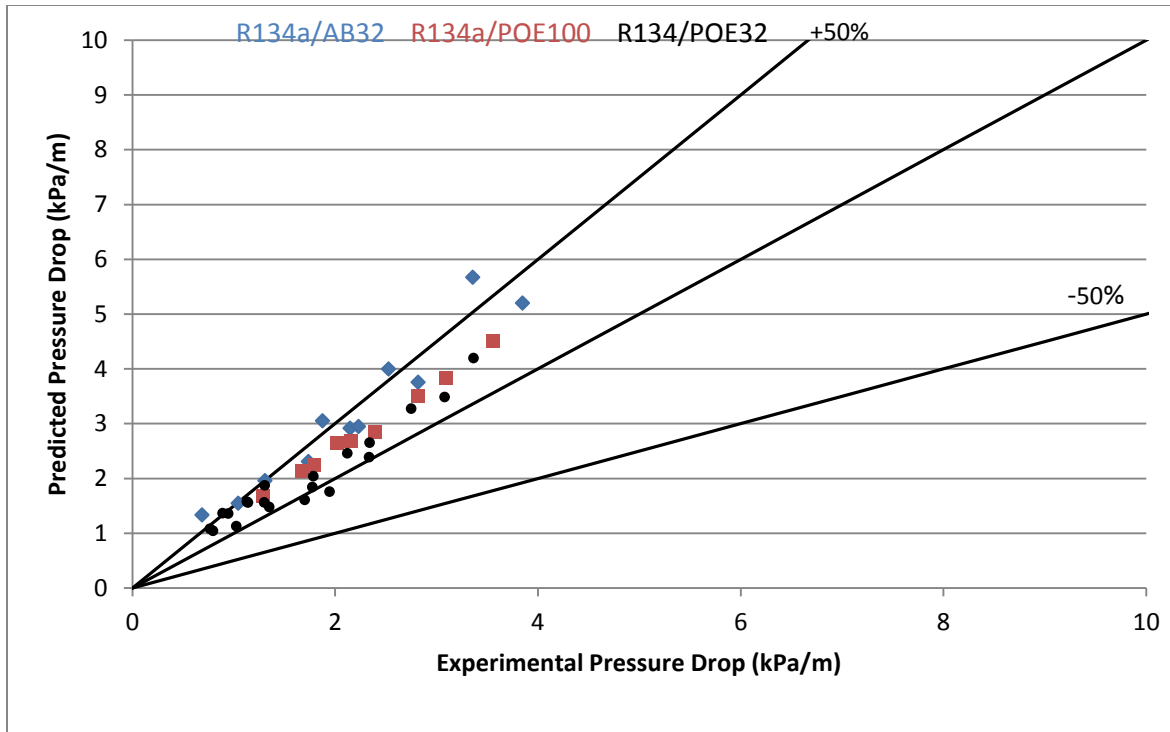


Figure 4.3 Validation of the proposed model for pressure drop in vertical suction lines

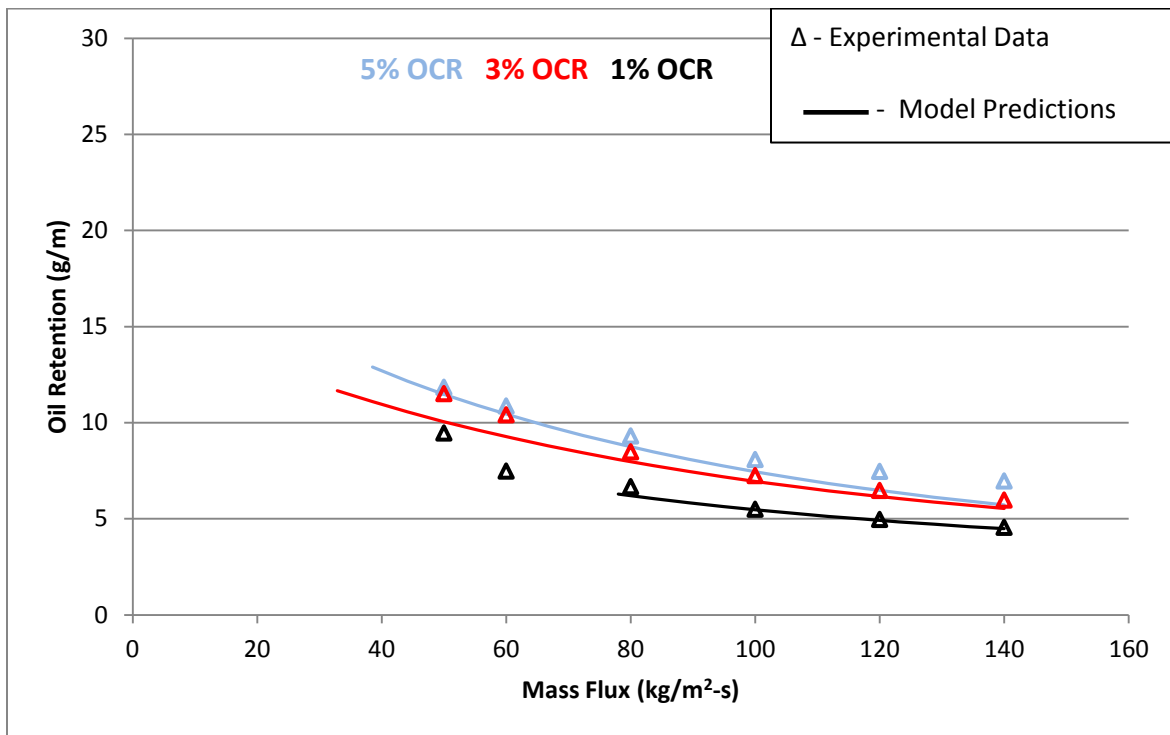


Figure 4.4 Experimental data and model predictions for variation of oil retention with mass flux for R134a/AB32

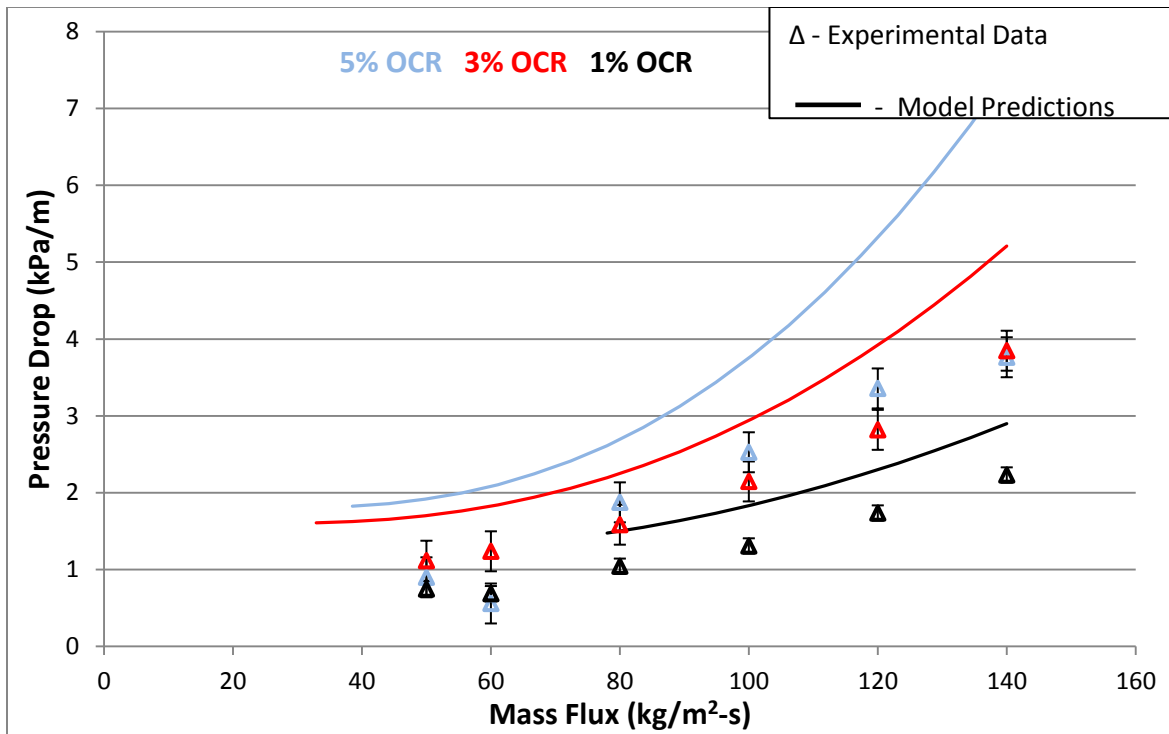


Figure 4.5 Experimental data and model predictions for variation of pressure drop with mass flux for R134a/AB32

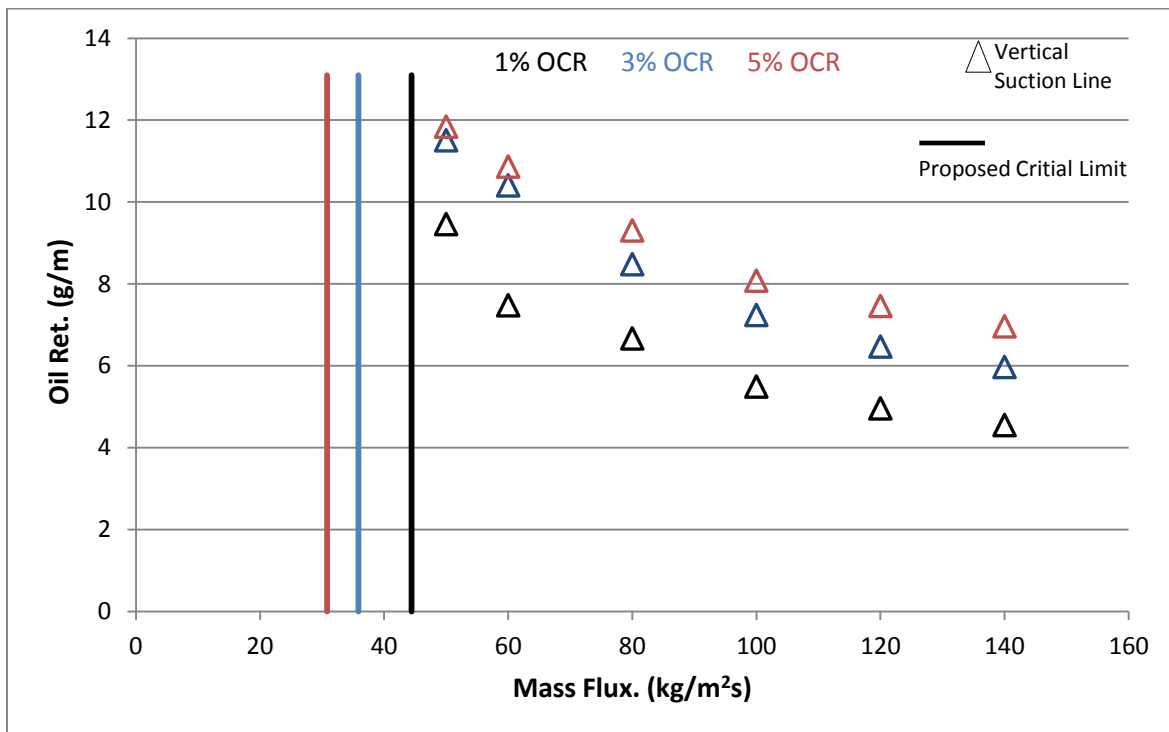


Figure 4.6 Oil retention for R134a/AB32 in the vertical suction line with the proposed critical mass flux limit

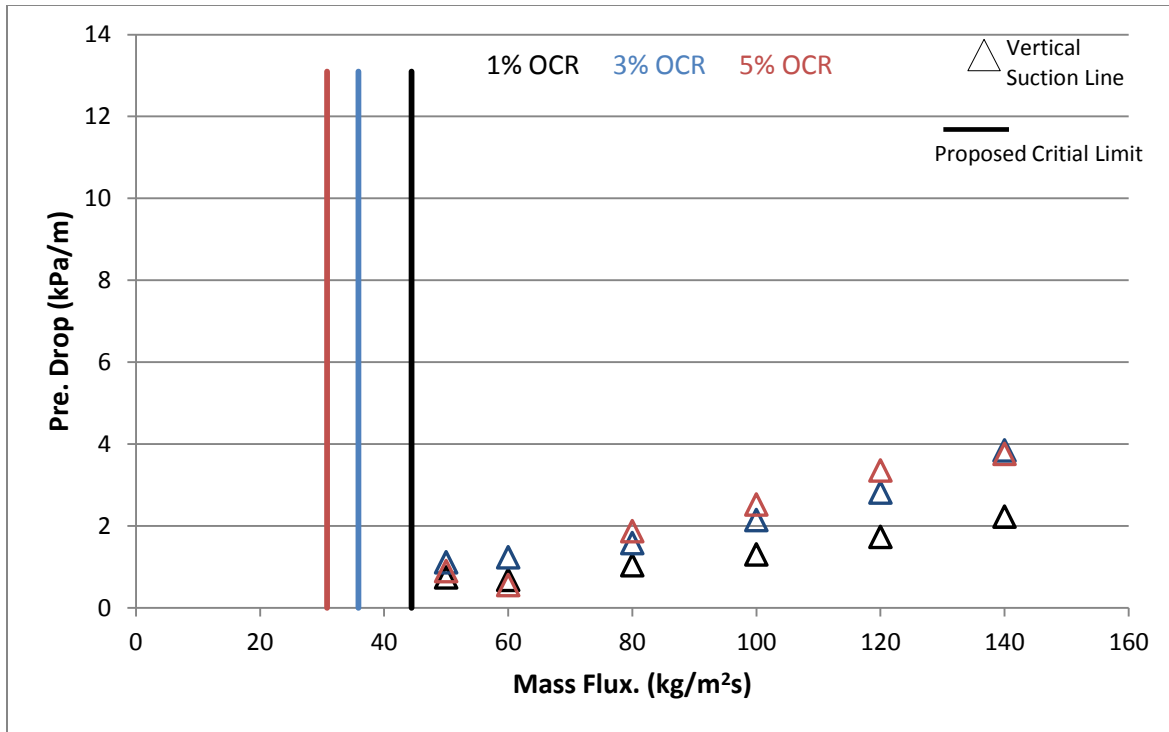


Figure 4.7 Pressure drop for R134a/AB32 in the vertical suction line with the proposed critical mass flux limit

## CHAPTER 5: CONCLUSIONS

In this work oil retention and pressure drop in horizontal and vertical suction lines was studied, and data was presented for the refrigerant/lubricant mixture of R134a/AB32. The experimental data obtained in the current work was compared to the data for R134a/POE100 studied by Ramakrishnan and Hrnjak (2012) and R134a/POE32 studied by Sethi and Hrnjak (2011). A flow visualization study was undertaken to study the flow of an immiscible mixture of R134a/AB32 in the liquid line before the evaporator and to determine the flow regimes in the horizontal and vertical transparent suction lines. An analytical model was proposed to predict the experimental oil retention and pressure drop data. The conclusions of the current work are as follows:

- From the liquid line before the evaporator it was observed that AB32 oil and liquid R134a refrigerant were completely immiscible. The lighter oil always flowed on top of the liquid refrigerant layer. At low mass fluxes, the oil/refrigerant layer was wavy with no mixing at the interface. As the mass flux was increased, the interface became gradually wavier and oil globules began to break away from the bulk oil layer and mix in the liquid refrigerant layer. The flow pattern at high mass fluxes is semi-segregated with mixing at the interface.
- The flow regime in the horizontal suction line was annular at high mass fluxes, and it transitioned into stratified-wavy regime at lower mass fluxes. It was concluded from the flow visualizations that the transition occurred at a mass flux of  $80 \text{ kg/m}^2\text{-s}$ .
- In the vertical suction line, the flow regime was annular at all mass fluxes. No flow reversal was observed at low mass fluxes.

- The oil retention in the horizontal suction line decreased initially, with an increase in mass flux, attained a minimum, then increased again before finally decreasing as the mass flux was further increased. Oil retention in the vertical suction line increased continuously with a decrease in the mass flux. It was also observed, that the vertical suction line oil retention always exceed the horizontal suction line oil retention due to the effect of gravity on the former.
- Horizontal pressure drop showed a decreasing trend with a reducing mass flux. In the vertical suction line as well the pressure drop decreased continuously with a decrease in the mass flux.
- Oil retention was observed to be continually higher in AB32 as compared to POE 100 and POE32. It was concluded that the viscosity of the annular oil layer in the R134a/AB32 mixture was higher than either the R134a/POE100 and R134a/POE32 mixtures.
- The assumption at the outset that R134a was completely immiscible with AB32 was not to be entirely correct. Even though the liquid mixture of R134a/AB32 was completely immiscible, vapor R134a was miscible with AB32. This was confirmed by the evolution of bubbles from oil during vacuuming. This was further confirmed when the semi-empirical model developed by taking the viscosity and density of the oil/refrigerant mixture to be different than the pure oil values, predicted the experimental results.
- The proposed model predicted the experimental data from the current study, data by Ramakrishnan and Hrnjak (2012) and data by Sethi and Hrnjak (2012) to  $\pm 40\%$  for oil retention and  $\pm 50\%$  for pressure drop.

- Based on the predictions of the model and the fact that vapor refrigerant evolves from liquid oil during experimentation, at the vacuuming stage, it can be concluded that R134a vapor refrigerant is miscible in liquid AB32 oil. However, the percentage miscibility is not known.
- As a part of the current work, a critical mass flux correlation was proposed, based on which vertical suction lines can be sized. The critical mass flux limit determined was at a mass flux at which the flow reversal of the annular liquid film initiates.

## REFERENCES

1. ASHRAE, 1973, *Handbook and Product Directory, 1973 Systems*
2. Baker, O., 1954, "Simultaneous Flow of Oil and Gas", *Oil and Gas Journal*, vol. 53, no. 12: p. 185-190.
3. Belt, R.J., Van't Westende, J.M.C. & Portela, L.M., 2009, "Prediction of the Interfacial Shear-Stress in Vertical Annular Flow", *International Journal of Multiphase Flow*, vol. 35, no. 7: p. 689-697.
4. Biancardi, F.R, Michels, H.H., Sienel, T.H. & Pandey, D.R., 1996, "Study of Lubricant Circulation in HVAC Systems. Volume I – Description of Technical Effort and results", *UTRC Technical Report*, United Technologies Research Center, East Hartford, CT.
5. Cremaschi, L., Hwang, Y. & Radermacher, R., 2005, "Experimental Investigation of Oil Retention in Air Conditioning Systems", *International Journal of Refrigeration*, vol. 28, no. 7: p. 1018-1028.
6. Cremaschi, L., 2004, "Experimental and Theoretical Investigation of Oil Retention in Vapor Compression Systems", *PhD Thesis*, University of Maryland, College Park, MD.
7. Crompton, J.A., Newell, T.A. & Chato, J.C., 2004, "Experimental Measurement and Modeling of Oil Holdup", *ACRC TR-226*, Air Conditioning and Refrigeration Center, University of Illinois at Urbana-Champaign, Urbana, IL
8. Jacobs, M.L., Scheideman, F.C., Kazem, S.M. & Macken, N.A., 1976, "Oil Transport by Refrigerant Vapor", *ASHRAE Transactions*, vol. 82, no. 2: p. 318-329.
9. Kesim, S.C., Albayrak, K. & Ileri, A., 2000, "Oil entrainment in Vertical Refrigerant Piping", *International Journal of Refrigeration*, vol. 23, no. 8: p. 626-631.

10. Lee, J.P., Hwang, Y., Radermacher & R., Mehendale, S.S., 2001, "Experimental Investigations on Oil Accumulation Characteristics in a Vertical Suction Line", *2001 ASME International Mechanical Engineering Congress and Exposition, November 11 - November 16, 2001*, American Society of Mechanical Engineers, New York, NY, United States, vol. 41, p. 63-69.
11. Lee, J.P., 2003, "Experimental and Theoretical Investigation of Oil Retention in Carbon Dioxide Air Conditioning System", *PhD Thesis*, University of Maryland, College Park, MD.
12. Mehendale, S.S., 1998, "Experimental and Theoretical Investigation of Annular Film Flow Reversal in a Vertical Pipe" *PhD Thesis*, University of Maryland, College Park, MD.
13. Mehendale, S.S. & Radermacher, R., 2000, "Experimental and Theoretical Investigation of Annular Film Flow Reversal in a Vertical Pipe: Application to Oil Return in Refrigeration Systems", *HVAC and R Research*, vol. 6, no. 1: p. 55-74.
14. Radermacher, R., Cremaschi, L. & Schwentker, R.A., 2006, "Modeling of Oil Retention in the Suction Line and Evaporator of Air-Conditioning Systems", *HVAC and R Research*, vol. 12, no. 1: p. 35-56.
15. Ramakrishnan, A., & Hrnjak, P., 2012, Investigation of Oil Retention and Pressure Drop in Suction Lines Using R1234yf, R134a and R410A with POE ISO 100, MS Thesis, University of Illinois at Urbana Champaign.
16. Sumida, Y., Nakayama, M., Suzuki, S., & Kawaguchi, S., 1998, Alkylbenzenes for Split Air Conditioners with R-410A Part 2: Oil Return Characteristics, International Compressor Engineering Conference.

17. Sunami, M., Takigawa, K., & Suda, S., 1994, New Immiscible Refrigeration Lubricant for HFCs, International Refrigeration and Air Conditioning Conference.
18. Sundaresan, S.G., Judge, J., Chu, W., & Radermacher, R., 1996, A Comparison of the Oil Return Characteristics of R22/Mineral Oil, and Its HFC Alternatives(R407C &R410A) with Mineral Oil and POE in a Residential Heat Pump, International Refrigeration and Air Conditioning Conference.
19. Sethi, A. & Hrnjak, P., 2011, Oil Retention and Pressure Drop of R1234yf and R134a with POE ISO 32 in Suction Lines, *ACRC TR-281*, Air Conditioning and Refrigeration Center, University of Illinois at Urbana-Champaign.
20. Takaishi, Y. & Oguchi, K. 1987, "Measurements of vapor pressures of R22/oil solution", *Proceedings of the 18th International Congress of Refrigeration*, p. 217-222.
21. Thome, J.R., 1995, "Comprehensive Thermodynamic Approach to Modeling Refrigerant-Lubricating Oil Mixtures", *HVAC&R Research*, vol. 1, no. 2: p. 110-125.
22. Thome, J.R., 2004, *Engineering Data Book III*, Wolverine Tube Inc., chap. 12: p. 1-34.
23. van Rossum, J.J., 1959, "Experimental Investigation of Horizontal Liquid Films. Wave Formation, Atomization, Film Thickness", *Chemical Engineering Science*, vol. 11, no. 1: p. 35-52.
24. Vedapuri, D., 1999, Studies on Oil Water Flows in Inclined Pipelines, MS Thesis, Ohio University.
25. Wallis, G., 1969, *One Dimensional Two-Phase Flow*, McGraw-Hill Book Company, New York.

26. Wongwises, S. & Kongkiatwanitch, W., 2001, “Interfacial Friction Factor in Vertical Upward Gas-Liquid Annular Two-Phase Flow”, *Int. Comm. Heat Mass Transfer*, vol. 28, no. 3: p. 323-336.
27. Zoellick, K.F. & Hrnjak, P.S., 2010, “Oil Retention and Pressure Drop in Horizontal and Vertical Suction Lines with R410A/POE ISO 32”, *ACRC TR-271*, Air Conditioning and Refrigeration Center, University of Illinois at Urbana-Champaign.

## APPENDIX A

### Repeatability Tests

Experiments were conducted in order to verify the accuracy and repeatability of the test setup after it was thoroughly cleaned prior to commencing experimentation. Four experimental tests were conducted over a period of four weeks, with R134a as the refrigerant and AB ISO 32 as the lubricant. The operating conditions selected for performing the repeatability tests were a mass flux of  $80 \text{ kg/m}^2\text{-s}$  and an OCR of 3%. The saturation temperature and superheat were maintained at  $13^\circ\text{C}$  and  $15^\circ\text{C}$  respectively. The operating conditions and the results obtained have been tabulated in Table A.1. The results indicate that the tests are repeatable to a good degree of accuracy. The slight variations are a result of errors during experimentation, for example, the mistimed closing of the ball valves at the ends of the test sections. The average oil retention in the horizontal suction line was  $5.37 \text{ g/m}$  and  $7.83 \text{ g/m}$  in the vertical suction line. Average values for the horizontal and vertical pressure drops were  $0.74 \text{ kPa/m}$  and  $1.61 \text{ kPa/m}$  respectively.

**Table A.1 Repeatability Test Data**

| $T_{\text{Sat}}$   | $G$                        | $T_{\text{SH}}$    | $T_{\text{TW}}$    | OCR    | $m_{\text{oil,hor}}$ | $m_{\text{oil,vert}}$ | $dP_{\text{hor}}$ | $dP_{\text{vert}}$ |
|--------------------|----------------------------|--------------------|--------------------|--------|----------------------|-----------------------|-------------------|--------------------|
| $[^\circ\text{C}]$ | $[\text{kg/m}^2\text{-s}]$ | $[^\circ\text{C}]$ | $[^\circ\text{C}]$ |        | $[\text{g/m}]$       | $[\text{g/m}]$        | $[\text{kPa/m}]$  | $[\text{kPa/m}]$   |
| 13.1               | 78.4                       | 28.1               | 26.9               | 0.030  | 5.71                 | 8.48                  | 0.62              | 1.58               |
| 13.1               | 83.9                       | 28.5               | 26.9               | 0.0299 | 5.56                 | 8.01                  | 0.88              | 1.71               |
| 13.0               | 82.3                       | 28.0               | 27.3               | 0.030  | 5.33                 | 7.95                  | 0.75              | 1.63               |
| 13.2               | 84.1                       | 27.6               | 26.5               | 0.0311 | 4.87                 | 6.86                  | 0.72              | 1.51               |

## APPENDIX B

### EES Model Code for calculating oil retention and pressure drop in vertical suction lines

#### {1. Defined Quantities}

G = 63.6 {Total mass flux kg/m<sup>2</sup> s}  
Psat = 0.455 [MPa] {System saturation pressure in MPa}  
w\_inlet = 0.0104 {Oil concentration ratio, OCR}  
T\_evap\_out = 28.7 {Evaporator outlet temperature in °C}  
D = 0.0102 {Inner tube diameter in m}  
L\_vert = 1.89 {Length of Vertical Suction Line in m}  
nu\_l\_cst = 30 {Viscosity of refrigerant oil mixture in cSt}  
rho\_l = 925 {Density of refrigerant oil mixture in kg/m<sup>3</sup>}

{Thome's (1995) Method for calculating local oil concentration}

#### {2. Determine local oil concentration in liquid at inlet of suction line}

##### {2.1 Interpolating two points just above and below P\_sat}

Pabove = Psat + .005  
Pbelow = Psat - .005  
Tabove = Temperature(R134a, P=Pabove, x=.1)  
Tbelow = Temperature(R134a, P=Pbelow, x=.1)

##### {2.2 Calculate a\_0 and b\_0 assuming zero w\_inlet}

Tabove + 273 = a\_0 / (ln(Pabove) - b\_0)  
Tbelow + 273 = a\_0 / (ln(Pbelow) - b\_0)

##### {2.3 a\_1 to b\_4 values are constants which are used along the calculated a\_0 and b\_0}

a\_1 = 182.52  
a\_2 = -724.21  
a\_3 = 3868  
a\_4 = -5268.9

b\_1 = -.72212  
b\_2 = 2.3914  
b\_3 = -13.779  
b\_4 = 17.066

#### {2.4 Finally calculate w\_local}

$$\begin{aligned}A_w\text{\_local} &= a_0 + a_1*w\_local + a_2*w\_local^3 + a_3*w\_local^5 + a_4*w\_local^7 \\B_w\text{\_local} &= b_0 + b_1*w\_local + b_2*w\_local^3 + b_3*w\_local^5 + b_4*w\_local^7 \\T_{\text{evap\_out}}+273 &= A_w\text{\_local} / (\ln(P_{\text{sat}}) - B_w\text{\_local})\end{aligned}$$

#### {3. Determining Quality at inlet to the suction lines}

$$w\_local*(1-x) = w_{\text{inlet}}$$

#### {4. Calculate Pure Vapor Density}

$$\rho_v = \text{Density}(R134a, T=T_{\text{evap\_out}}, P=P_{\text{sat}})$$

#### {5. Calculate Liquid Film Properties}

$$\begin{aligned}\mu_v &= \text{Viscosity}(R134a, T=T_{\text{evap\_out}}, P=P_{\text{sat}}) \\ \mu_r &= \text{Viscosity}(R134a, T=T_{\text{evap\_out}}, x=0) \\ \mu_l &= \rho_l * \nu_l\end{aligned}$$

$$\begin{aligned}\nu_v &= \mu_v / \rho_v && \{\text{Kinematic viscosity of the vapor}\} \\ \nu_l &= \mu_l / \rho_l * 10^{-6} && \{\text{Kinematic viscosity of the liquid}\}\end{aligned}$$

#### {6. Calculating oil retention and pressure drop in the vertical suction line}

$$G_l = G*(1-x)$$

$$G_v = G*x$$

$$\dot{m}_l = (G_l)*(3.14*D^2*0.25)$$

$$\dot{m}_l = a + b$$

$$a = (2*3.14*\rho_l/\mu_l)*((\tau_i*(0.5*D-\delta) + ((0.5*D-\delta)^2/2)*(dpdz + \rho_l*9.81))*(((0.5*D)^2 - (0.5*D-\delta)^2)/4 - ((0.5*D-\delta)^2/2)*\ln(1/(1-\delta_{\text{by\_R}}))))$$

$$b = (-1*3.14*\rho_l/(8*\mu_l))*(dpdz + \rho_l*9.81)*((0.5*D)^2 - (0.5*D-\delta)^2)^2$$

$$\delta_{\text{by\_R}} = \delta/(0.5*D)$$

$$dpdz + \rho_v*9.81 + (4*\tau_i/(D*(\alpha)^{0.5})) = 0$$

$$\alpha = ((D - 2*\delta)/D)^2$$

$$\tau_i = 0.5*f_i*\rho_v*v_v^2$$

$$v_v = G_v/(\rho_v*\alpha)$$

$$\delta_{\text{plus}} = \delta/(\nu_v)*(\tau_i/\rho_v)^{0.5}$$

### {6.1 Interfacial Friction Factor Correlation}

$$f_i/f_s = 1 + 1.30561 \cdot 10^{40} \cdot (\text{Re}_{lf})^{(0.989041469)} \cdot \text{Re}_v^{(10.32567502)} \cdot (\text{delta\_plus})^{(3.557783518)} \cdot (\text{Weber\_mix})^{(4.979239513)}$$

$$f_s = 0.046 \cdot \text{Re}_v^{(-0.2)}$$

### {6.2 Defining Various Non-Dimensional Terms in the Model}

$$\text{Re}_v = \rho_v \cdot v_v \cdot D / \mu_v$$

$$\text{Re}_{lf} = G \cdot (1-x) \cdot D / (4 \cdot \mu_l)$$

$$\sigma_r = \text{SurfaceTension}(\text{R134a}, T = T_{\text{evap\_out}})$$

$$\sigma_o = 29 \cdot 10^{(-3)}$$

$$\sigma_{\text{mix}} = (\sigma_r) + (\sigma_o - \sigma_r) \cdot (w_{\text{local}})^{(0.5)}$$

$$\text{Weber\_mix} = (G^2 \cdot D) / (\sigma_{\text{mix}} \cdot \rho_l)$$

### {6.3 Final Calculated Parameters}

$$\text{Oil\_retention\_predicted} = ((3.14 \cdot D \cdot \text{delta} \cdot \rho_l \cdot w_{\text{local}})) \cdot 1000 \quad \{\text{Oil Retention in g/m}\}$$

$$\text{PressureDrop\_Predicted} = \text{dpdz} \cdot (-1) / 1000 \quad \{\text{Pressure Drop in kPa/m}\}$$

## EES Model Code for calculating critical refrigerant mass flux

### {1. Defined Quantities}

G = 63.6 {Total mass flux kg/m<sup>2</sup> s}  
Psat = 0.455 [MPa] {System saturation pressure in MPa}  
w\_inlet = 0.0104 {Oil concentration ratio, OCR}  
T\_evap\_out = 28.7 {Evaporator outlet temperature in °C}  
D = 0.0102 {Inner tube diameter in m}  
L\_vert = 1.89 {Length of Vertical Suction Line in m}  
nu\_l\_cst = 30 {Viscosity of refrigerant oil mixture in cSt}  
rho\_l = 925 {Density of refrigerant oil mixture in kg/m<sup>3</sup>}

{Thome's (1995) Method for calculating local oil concentration}

### {2. Determine local oil concentration in liquid at inlet of suction line}

#### {2.1 Interpolating two points just above and below P\_sat}

Pabove = Psat + .005  
Pbelow = Psat - .005  
Tabove = Temperature(R134a, P = Pabove, x = .1)  
Tbelow = Temperature(R134a, P = Pbelow, x = .1)

#### {2.2 Calculate a\_0 and b\_0 assuming zero w\_inlet}

Tabove + 273 = a\_0 / (ln(Pabove) - b\_0)  
Tbelow + 273 = a\_0 / (ln(Pbelow) - b\_0)

#### {2.3 a\_1 to b\_4 values are constants which are used along the calculated a\_0 and b\_0}

a\_1 = 182.52  
a\_2 = -724.21  
a\_3 = 3868  
a\_4 = -5268.9

b\_1 = -.72212  
b\_2 = 2.3914  
b\_3 = -13.779  
b\_4 = 17.066

#### {2.4 Finally calculate w\_local}

A\_w\_local = a\_0 + a\_1\*w\_local + a\_2\*w\_local^3 + a\_3\*w\_local^5 + a\_4\*w\_local^7  
B\_w\_local = b\_0 + b\_1\*w\_local + b\_2\*w\_local^3 + b\_3\*w\_local^5 + b\_4\*w\_local^7

$$T_{\text{evap\_out}} + 273 = A_{\text{w\_local}} / (\ln(P_{\text{sat}}) - B_{\text{w\_local}})$$

### {3. Determining Quality at inlet to the suction lines}

$$w_{\text{local}}(1-x) = w_{\text{inlet}}$$

### {4. Calculate Pure Vapor Density}

$$\rho_v = \text{Density}(\text{R134a}, T = T_{\text{evap\_out}}, P = P_{\text{sat}})$$

### {5. Calculate Liquid Film Properties}

$$\mu_v = \text{Viscosity}(\text{R134a}, T = T_{\text{evap\_out}}, P = P_{\text{sat}})$$

$$\mu_r = \text{Viscosity}(\text{R134a}, T = T_{\text{evap\_out}}, x = 0)$$

$$\mu_l = \rho_l * \nu_l$$

$$\nu_v = \mu_v / \rho_v \quad \{\text{Kinematic viscosity of the vapor}\}$$

$$\nu_l = \mu_l / \rho_l * 10^{-6} \quad \{\text{Kinematic viscosity of the liquid}\}$$

### {6. Calculating oil retention and pressure drop in the vertical suction line}

$$\tau_i(0.5D - \delta) / \delta = 0.5(d\text{pdz} + \rho_l * 9.81)(D - \delta) \quad \{\text{Wall Shear Stress Equated to Zero}\}$$

$$d\text{pdz} + \rho_v * 9.81 + (4 * \tau_i / (D * (\alpha)^{0.5})) = 0$$

$$\tau_i = 0.5 * f_i * \rho_v * u_v^2$$

$$\dot{m}_l = a + b$$

$$a = (2 * 3.14 * \rho_l / \mu_l) * ((\tau_i(0.5D - \delta) + ((0.5D - \delta)^2 / 2) * (d\text{pdz} + \rho_l * 9.81)) * (((0.5D)^2 - (0.5D - \delta)^2) / 4 - ((0.5D - \delta)^2 / 2) * \ln(1 / (1 - \delta_{\text{by\_R}}))))$$

$$b = (-1 * 3.14 * \rho_l / (8 * \mu_l)) * (d\text{pdz} + \rho_l * 9.81) * ((0.5D)^2 - (0.5D - \delta)^2)^2$$

$$\delta_{\text{by\_R}} = \delta / (0.5D)$$

$$d\text{pdz} + \rho_v * 9.81 + (4 * \tau_i / (D * (\alpha)^{0.5})) = 0$$

$$\alpha = ((D - 2 * \delta) / D)^2$$

$$\tau_i = 0.5 * f_i * \rho_v * v_v^2$$

$$v_v = G_v / (\rho_v * \alpha)$$

$$\delta_{\text{plus}} = \delta / (\nu_v * (\tau_i / \rho_v)^{0.5})$$

#### {6.1 Interfacial Friction Factor Correlation}

$$f_i / f_s = 1 + 1.30561 * 10^{40} * (\text{Re}_{\text{lf}})^{(0.989041469)} * \text{Re}_v^{(10.32567502)} * (\delta_{\text{plus}})^{(3.557783518)} * (\text{Weber}_{\text{mix}})^{(4.979239513)}$$

$$f_s = 0.046 * \text{Re}_v^{(-0.2)}$$

### {6.2 Defining Various Non-Dimensional Terms in the Model}

$$\text{Re}_v = \rho_v v D / \mu_v$$

$$\text{Re}_{lf} = G(1-x)D / (4\mu_l)$$

$$\sigma_r = \text{SurfaceTension}(\text{R134a}, T = T_{\text{evap\_out}})$$

$$\sigma_o = 46 \cdot 10^{-3}$$

$$\sigma_{\text{mix}} = (\sigma_r) + (\sigma_o - \sigma_r)(w_{\text{local}})^{0.5}$$

$$\text{Weber}_{\text{mix}} = (G^2 D) / (\sigma_{\text{mix}} \rho_l)$$

### {6.3 Calculating Critical Mass Flux}

$$x = \dot{m}_v / (\dot{m}_v + \dot{m}_l)$$

$$\dot{m}_v = \rho_v 0.25 D^2 3.14 u_v \alpha$$

$$G_{\text{critical}} = ((\dot{m}_v + \dot{m}_l (1 - w_{\text{local}})) / (0.25 \cdot 3.14 D^2))$$

## APPENDIX C

### R134a/AB32 raw data – 10.2 mm pipe diameter

| P_sat   | T_sat    | Total Mass Flux        | OCR    | T_ref_evap_out | T_r_e_wall | Mass Oil (Ho) | Mass Oil (Vert) | Pressure Drop (Ho) | Pressure Drop (Vert) |
|---------|----------|------------------------|--------|----------------|------------|---------------|-----------------|--------------------|----------------------|
| ± 9 kPa | ± 0.3 °C | (kg/m <sup>2</sup> -s) |        | ± 0.5 °C       | ± 0.5 °C   | g             | g               | ± 0.1 kPa          | ± 0.1 kPa            |
| 454     | 13.0     | 55.4                   | 0.0121 | 27.8           | 25.5       | 7.14          | 17.22           | 0.23               | 1.10                 |
| 455     | 13.1     | 63.6                   | 0.0104 | 28.7           | 28.0       | 5.58          | 13.61           | 0.23               | 0.99                 |
| 454     | 13.0     | 83.0                   | 0.0103 | 27.9           | 27.4       | 6.42          | 12.14           | 0.81               | 1.64                 |
| 456     | 13.1     | 104.4                  | 0.0107 | 28.2           | 27.7       | 6.29          | 10.00           | 1.45               | 2.12                 |
| 455     | 13.1     | 119.3                  | 0.0104 | 27.8           | 27.0       | 7.23          | 9.03            | 2.26               | 2.70                 |
| 456     | 13.2     | 140.3                  | 0.0104 | 27.8           | 25.5       | 7.66          | 8.29            | 3.26               | 3.30                 |
| 456     | 13.2     | 52.2                   | 0.0303 | 28.2           | 26.2       | 11.69         | 21.75           | 0.53               | 2.11                 |
| 455     | 13.1     | 61.3                   | 0.0307 | 27.7           | 25.2       | 11.13         | 19.67           | 0.74               | 2.34                 |
| 456     | 13.1     | 78.4                   | 0.0300 | 28.1           | 26.9       | 1.08          | 16.03           | 1.17               | 3.00                 |
| 455     | 13.1     | 83.9                   | 0.0299 | 28.5           | 26.9       | 10.51         | 15.14           | 1.67               | 3.24                 |
| 454     | 13.0     | 82.3                   | 0.0300 | 28.0           | 27.3       | 10.07         | 15.03           | 1.42               | 3.09                 |
| 457     | 13.2     | 84.1                   | 0.0311 | 27.6           | 26.5       | 9.20          | 12.48           | 1.36               | 2.75                 |
| 455     | 13.1     | 100.1                  | 0.0298 | 27.8           | 26.6       | 12.14         | 13.68           | 2.89               | 4.06                 |
| 455     | 13.1     | 118.8                  | 0.0295 | 28.4           | 26.4       | 11.43         | 12.23           | 4.47               | 5.33                 |
| 453     | 13.0     | 140.8                  | 0.0301 | 27.8           | 23.5       | 10.51         | 11.28           | 6.47               | 7.27                 |
| 457     | 13.2     | 56.2                   | 0.0494 | 28.6           | 24.9       | 13.12         | 21.55           | 0.65               | 1.64                 |
| 454     | 13.0     | 63.1                   | 0.0510 | 27.7           | 23.9       | 12.86         | 19.77           | 0.84               | 1.02                 |
| 455     | 13.1     | 83.5                   | 0.0494 | 28.2           | 27.0       | 12.55         | 16.94           | 1.75               | 3.41                 |
| 454     | 13.0     | 100.4                  | 0.0495 | 29.5           | 28.0       | 13.10         | 14.70           | 3.54               | 4.60                 |
| 454     | 13.0     | 119.1                  | 0.0501 | 28.2           | 25.5       | 12.85         | 13.58           | 5.26               | 6.11                 |
| 458     | 13.3     | 139.8                  | 0.0508 | 27.6           | 23.6       | 12.50         | 12.68           | 7.34               | 6.85                 |