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ABSTRACT

The adiabatic pressure drop of two-phase flow of refrigerant-oil mixtures in a small rectangular channel (d_h=148.0 microns) has been investigated. Tests have been performed with R134a with varying concentrations of a POE 32 oil. Pressure drop tests of pure refrigerant in these size channels have found reasonable agreement to the homogeneous pressure drop models, although flow visualization studies show that such flows are not always in regimes that can be considered homogeneous. The fluid properties of the liquid phase (density, viscosity, etc.) are difficult to characterize because they are governed by the amount of refrigerant that remains dissolved in the oil. Fluid property data measured for this refrigerant-oil pair is used to determine the actual properties of the liquid phase. A separated flow model that was developed from pressure drop of four different pure refrigerants to account for the varying fluid properties (Field and Hrnjak 2006) was compared to the data with the actual liquid phase properties, showing moderate success especially in the high quality region.

1. INTRODUCTION

With the increased popularity of microchannel heat exchangers for use in the air conditioning and refrigeration industry, the question of two-phase refrigerant flows is of interest. While there have been many notable studies of two-phase flow in microchannels, the study of refrigerant flows in microchannels is still underrepresented in the literature. Furthermore, with the end application in mind of a complete refrigeration system, the inclusion of oil in circulation with the refrigerant is imperative to consider. The effect of a miscible oil on refrigerant in circulation can be considered to be entirely on the fluid properties of the liquid phase, if the liquid phase is assumed to be well mixed and the oil is assumed to remain in liquid phase. Therefore, knowing the fluid properties of the local refrigerant-oil mixture combined with a two-phase pressure drop correlation that properly accounts for fluid properties could result in correct prediction of the two-phase pressure drop. A separated flow model that was developed for channels of this size using four different pure fluids, to provide a wide span of fluid properties (Field and Hrnjak 2006). Two-phase refrigerant-oil flow pressure drop is measured in this study, and the data are compared to the predications of this model.

2. TWO-PHASE PRESSURE DROP BACKGROUND

2.1. Homogeneous Models

Homogeneous models for pressure drop treat the two-phase mixture as a single-phase flow. To do this, a two-phase Reynolds number, Re_{avg}, is calculated based on an average fluid viscosity:

$$\mathrm{Re}_{avg} = \frac{Gd_h}{\mu_{avg}}$$

(1)

and is then used with a single-phase correlation for friction factor vs. Reynolds number (i.e. the Moody chart or it's equivalent equational form) to yield the average friction factor, f_{avg}. This is then used to predict the pressure drop in the same way as single phase: $\Delta P/\Delta L = f_{avg}\, G^2/(2\, d_h\, \rho_{avg})$, where the average density of the two-phase flow calculated from:

$$\frac{1}{\rho_{avg}} = \frac{x}{\rho_v} + \frac{1-x}{\rho_l}$$

(2)

The difference between homogeneous models lies in the definition of the average viscosity, μ_{avg}, found in Equation (1). Many different models have been proposed in the literature. The viscosity model developed by Dukler *et al.* (1964), calculates the average viscosity weighted on a volume basis:

$$\mu_{avg} = \rho_{avg} \left(\frac{x}{\rho_v} \mu_v + \frac{1-x}{\rho_l} \mu_l \right) \tag{3}$$

The underlying assumption in all of the homogeneous pressure drop models is that the two-phase flow can be treated as a single-phase mixture. This is a valid assumption in certain flow regimes, but is certainly not universally applicable. None the less many studies of microchannel flows, including this one, have found one or another of the homogeneous models to give decent predictions of pressure drop. Several early studies, e.g. Ungar and Cornwell (1992) and Tripplett *et al.* (1999), even concluded by recommending the use of homogeneous pressure drop models for microchannels based on their ease of calculation.

2.2 Separated Flow Models
Separated flow models use a form of two-phase multiplier to predict pressure drop. The Lockhart-Martinelli two-phase multiplier is the most typical form of separated flow model. The multiplier, ϕ_l^2, is formulated in the following manner:

$$\left(\frac{\Delta P}{\Delta L} \right)_{tp} = \phi_l^2 \left(\frac{\Delta P}{\Delta L} \right)_l \tag{4}$$

and then taken to be a function of the Lockhart-Martinelli parameter, X.

Chisholm (1967) developed the theoretical basis for this and proposed the following simplified form of the correlation:

$$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{5}$$

where C is as a constant that varied from 5 to 20, based upon whether the liquid or vapor phase is in the laminar or turbulent regime, as determined by the superficial phase velocities.

Following Chisholm's analysis, the parameter C can be viewed as the interaction parameter between the liquid and vapor phases. Improvements of separated flow models have been directed at determination of C. Mishima and Hibiki (1996) developed a correlation for C based on pressure drop data taken in tubes of diameter 1 to 4 mm. The working fluids were mostly air and water, although an ammonia-vapor data set were also used. The fit for C depended only on an exponential function of diameter, neglecting any other fluid property variation. Unfortunately, by testing different fluids in the same channel, it can be clearly demonstrated that C depends on more than just the channel diameter, although this was one of the earliest attempts to find a separated flow model for microchannels.

Another effort at correlating C was done by Lee and Lee (2001). Their channels were rectangular, with d_h varying from 0.78 to 6.7 mm, and their working fluids were air and water. In developing a correlation for C, they begin from the dimensional analysis of Suo and Griffith (1964) who selected the dimensionless groups λ and ψ as significant for the two-phase intermittent flow in capillary tubes.

$$\psi = \frac{\mu_l V}{\sigma} \qquad \lambda = \frac{\mu_l^2}{\rho_l \sigma d_h} \tag{6}$$

The parameter ψ is the ratio of the viscous to surface tension effects in the flow. The characteristic velocity, V, was taken by Sou and Griffith to be U_B, the bubble velocity. Lee and Lee took the characteristic velocity to be j_l, the superficial velocity of the liquid phase. Either choice of characteristic velocity allows ψ to vary with refrigerant quality. The parameter λ is based on fluid properties and geometric parameters, and thus remains constant given a fluid and channel geometry.

Lee and Lee included in their correlation the Reynolds number based on the characteristic velocity if all the refrigerant were flowing as a liquid, $Re_{lo} = G d_h / \mu_l$. Note that Re_{lo} also remains constant with a certain channel geometry, mass flow rate and fluid properties. Lee and Lee's correlation for C then has the form:

$$C = A \lambda^q \psi^r \, Re_{lo}^s \tag{7}$$

where the coefficients *A*, *q*, *r* and *s* were determined by regressing the measured pressure drop data for each of the four flow regimes, vv, vt, tv, and tt. However, the coefficients *q* and *r* were found to be zero for every regime except laminar-laminar in their model, reducing the dependence of *C* to only Re_{lo} in every regime except that one.

Tu and Hrnjak (2004) used the form of *C* found in Equation (7) to develop a correlation for his R134a/vapor data that was taken in five different channels of hydraulic diameters varying from 70 to 305 μm. However, instead of using j_l as a characteristic velocity in ψ, he took the bubble velocity, U_B, as the characteristic velocity. Rather than the {vv, vt, tv, tt} division used frequently in separated flow correlations, they used the flow map divisions from Akbar *et al.* (2003) to divide the data into an inertial-dominated regime (annular flow) or a surface tension-dominated regime (slug/plug).

Lee and Mudawar (2005) measured pressure drop across microchannel evaporators with $d_h = 350$ μm. They developed a correlation for *C* in the following form:

$$C = a_1 \, Re_{lo}^{a_2} \, We_{lo}^{a_3} \tag{8}$$

where $a_1, a_2,$ and a_3 were determined by regressing their data, based on laminar or turbulent regime for the vapor phase (the liquid phase was always found to be laminar in their experiments, so the two flow regimes considered were vv and vt). We_{lo} is the liquid-only Weber number, $We_{lo} = G^2 d_h/(\sigma \rho_l)$, which, just like Re_{lo}, remains constant for given fluid properties and channel geometry.

Field and Hrnjak (2006) studied two-phase pressure drop of four pure refrigerants in the same $d_h=140$ μm channel investigated here. The refrigerants selected, R134a, R410A, R290, and R717, represented a wide span of fluid properties to widen the parameter base of the two-phase pressure drops measurements. None of the previously mentioned pressure drop correlations were found to be produce good predictions of the two-phase pressure drops. A new pressure drop correlation was developed based on an investigation of the relevant parameters of those refrigerants and the separated flow model correlation of *C* was given the following form:

$$C = \beta_1 \, Re_v^{\beta_2} \, \psi_{U_B}^{\beta_3} \tag{9}$$

where ψ_{U_B} is ψ, from Equation (6) with the characteristic velocity taken as the bubble velocity, $U_B=1.2(j_l+j_v)$, following both Suo and Griffith (1964) and Tu and Hrnjak (2004). The data were divided into flow regimes using the flow map of Akbar *et al.* (2003), and the β coefficients regressed for each regime are shown in Table 1. This correlation resulted in good agreement with measured two-phase pressure drop of the pure fluids, although the prediction was better at higher qualities which corresponded to annular flow, dominated by inertial forces.

Table 1: Coefficients from Field and Hrnjak (2006) correlation of *C* (Equation 9).

Dominant Forces	Flow map boundaries	Flow Regime	β_1	β_2	β_3
Surface Tension	$We_{vs} \leq 11.0 We_{ls}^{0.14}$	Intermittant	1.008e-5	1.4591	-0.6428
Inertial	$We_{vs} > 11.0 We_{ls}^{0.14}$	Annular	0.0146	0.4794	-0.6888

3. EVAPORATING REFRIGERANT-OIL FLOW

In considering oil effects on the flows, it is the local oil concentration that gives rise to the properties of the refrigerant flow at any given point in the flow. Defining the local concentration of refrigerant in the liquid phase or the mass fraction of refrigerant, ω, in the following manner:

$$\omega = \frac{\dot{m}_l}{\dot{m}_o + \dot{m}_l} \tag{10}$$

On the other hand, *OCR*, the oil circulation rate on a "sample basis", given by ASHRAE Standard 41.4, is the ratio of oil flow rate to total flow rate, liquid vapor and oil:

$$OCR = \frac{\dot{m}_o}{\dot{m}_l + \dot{m}_v + \dot{m}_o} \tag{11}$$

and the quality of oily flow is considered equivalent to the thermodynamic quality and defined as:

$$x_{oily} = x = \frac{\dot{m}_v}{\dot{m}_l + \dot{m}_v + \dot{m}_o} \tag{12}$$

Combining all these expressions, we can determine the local concentration of refrigerant as a function of quality and *OCR*,

$$\omega = 1 - \frac{OCR}{1 - x} \tag{13}$$

and the oil fraction in the liquid, given by (1-ω) and which depends only on oil circulation rate and quality, can be plotted as shown in Figure (1):
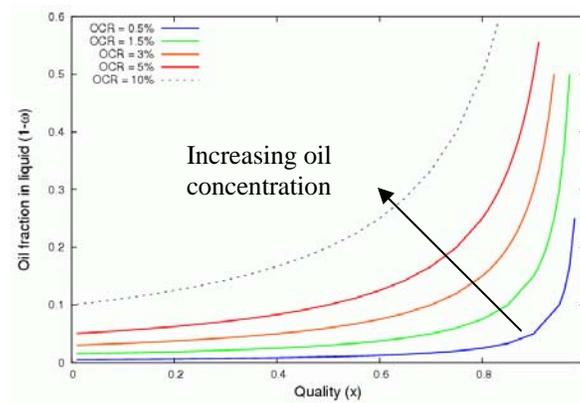

Figure 1: Variation of local oil fraction with increasing vapor quality

Fluid property measurements (density, viscosity, solubility and surface tension) have been made for several refrigerant-oil combinations. In particular, the properties of R134a and this POE32 combination have been measured at concentrations all the way from pure refrigerant to pure oil (Seeton and Hrnjak, personal communication). Knowing the local oil fraction from Equation 13, shown in Figure 1, and applying that to the property measurements, the local fluid properties of the liquid phase of a flow can be determined. Density and viscosity of the refrigerant-oil mixtures, as functions of quality are plotted below in Figures 2 and 3.
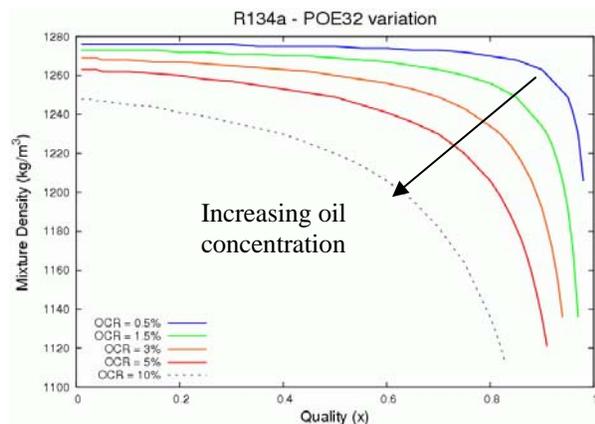

Figure 2: Varying density of liquid refrigerant-oil mixture due to oil concentration
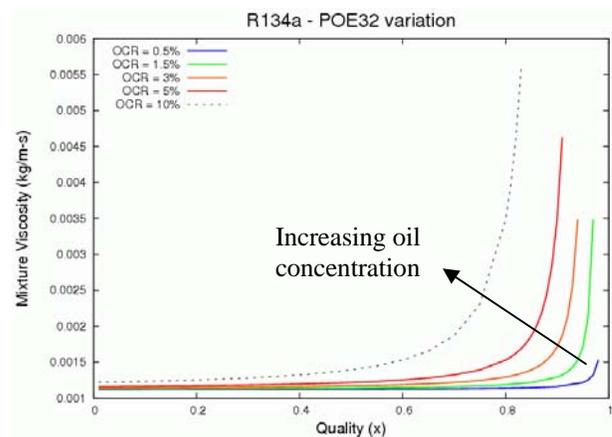

Figure 3: Varying viscosity of liquid refrigerant-oil mixture due to oil concentration

4. TWO-PHASE PRESSURE DROP OF OILY FLOW

4.1 Experimental Facility
A schematic of the test facility is shown in Figure 4. It is arranged in a once-through system driven by temperature-induced pressure differential. Refrigerant and oil were mixed in the reservoir tank to provide four different oil concentrations. The refrigerant-oil pair is fully miscible at the working temperature, so the liquid in the reservoir tank is assumed to remain fully mixed throughout the test, and the volume of fluid that was removed in the course of the test was sufficiently small so as to produce less than a 2.2% variation in the initial concentration for most of the

runs. Since the variation of initial oil concentration is small, no vapor space corrections have been made in the data presented, although the data were collected in such a way as to make such calculations possible. The mixture is heated in the reservoir tank to provide a driving pressure difference. Liquid refrigerant is removed from the bottom of the tank and is subcooled in the piping to room temperature before it reaches the mass flow meter. The mass flow meter (Rheotherm model, TU1/16) measures liquid flow rate based on an energy balance and needs to be supplied with subcooled liquid. In addition, the flow meter was calibrated with each fluid. Each experimental run had from 10-15 ^{o}C of subcooling at the mass flow meter. The temperature and pressure of the refrigerant were measured before the expansion/metering valve. Immediately after the metering valve, the refrigerant piping entered a vacuum dome, which minimized the convective losses from the refrigerant lines, heater and the channel. A pre-heater supplied a measured heat flux, $\dot{q}$, to the flow to control the vapor quality of the refrigerant entering the channel.

Saturation pressure as well as differential pressure drop was measured inside the channel via two pressure taps (see Figure 5). Outlet temperature and pressure was measured, and then the refrigerant was collected in a receiver which was maintained at ambient temperature and sat on a digital balance. The balance reading was recorded by the data logger and used to double-check the mass flow rate of the refrigerant.
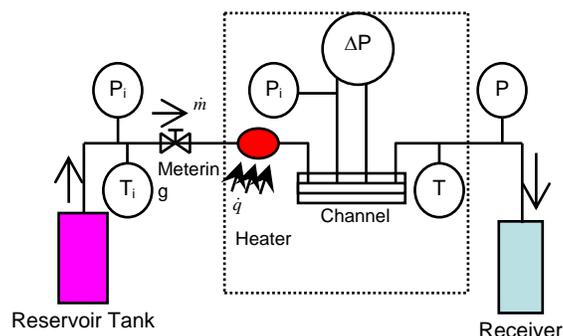

Figure 4: Facility schematic

The absolute pressure measurements were made with transducers (Setra, model 206, range 0-1723 kPa), with an accuracy of 0.2% of full scale. The differential pressure measurements were made with a differential transducer, (Sensotec, model number Z/5556-01, range 0-34.5 kPa), with an accuracy of 1% of full scale. Temperature measurements were made with type-T thermocouples, with an accuracy of 0.2 ^{o}C. A propagation of error analysis was preformed, with the measurement and calculated uncertainties listed in Table 2.

Table 2: Experimental Uncertainties

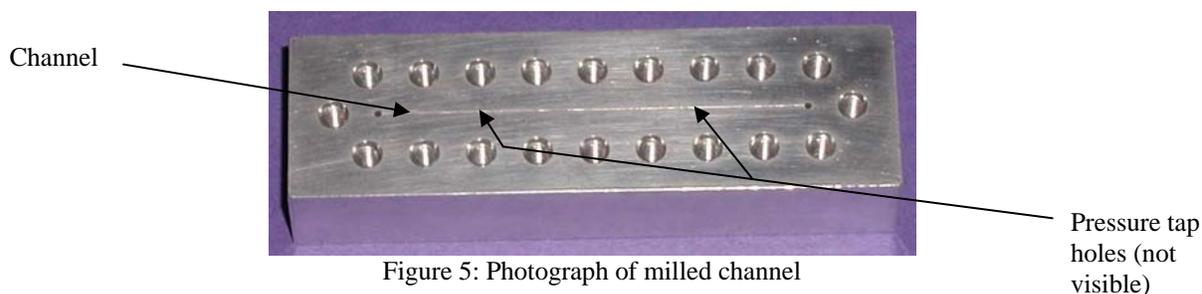| Parameter:	$\dot{m}$	$\dot{q}$	Temp.	Abs. Pressure	Diff. Pressure	Quality	$\Delta P/\Delta L$
Uncertainty:	±5%	±1%	±0.2 ^{o}C	±3.5 kPa	±0.34 kPa	±5.25%	±0.17 kPa

Channel

Pressure tap holes (not visible)

Figure 5: Photograph of milled channel

4.2 Test Section

The test section was made from two pieces of aluminum bolted together. The channel was milled into the bottom piece, with inlet and outlet ports at both ends of the channel and two pressure tap holes positioned in the middle of the channel. The top piece is attached by means of bolts to form the rest of the channel. A photograph of the

bottom piece containing the milled channel is shown in Figure 5. The holes around the perimeter of the piece are threaded for the bolts from the top of the channel. The test section was made in this way to allow for a detailed measurement of the channel size. It had been noted in prior experiments with pure refrigerant that as much as a 10% deviation in the measurement of the channel dimensions resulted in a 50% error in single-phase pressure drop relations. The channel dimensions were measured by a Sloan Dektak stylus surface profilometer. Measurements were made at eleven crossectional locations and averaged to determine overall channel dimensions. The maximum crossectional variation in measurement of depth was 2.5 μm, and of width was 14 μm. The average channel depth was thus measured to be 100.4 μm and the width was measured to be 281.1 μm, giving a hydraulic diameter of 148.0 μm.

4.3 Pressure Drop Results

The four oil concentrations tested were 0.47%, 1.53%, 2.66%, and 5.18%. These are equivalent to OCR values in refrigeration system. Pressure drop data were taken three mass fluxes, 335, 450, and 600 kg/m^2-sec. Figures 6 – 8 show the measured pressure drops for the mass fluxes for the different oil concentrations.
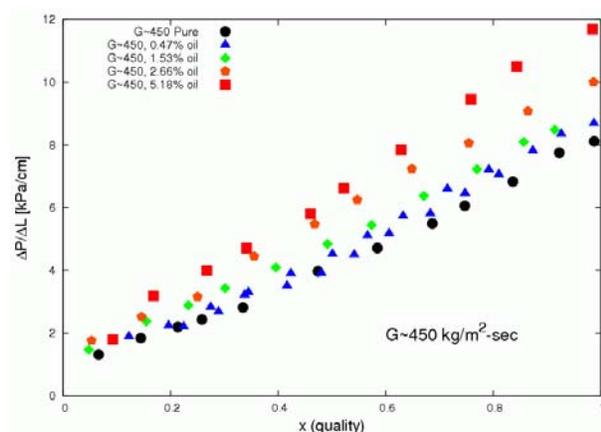

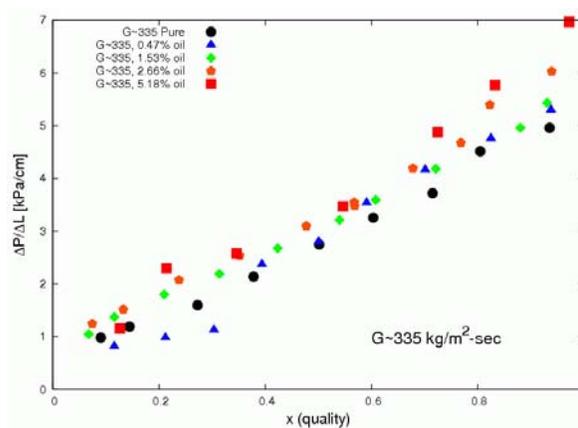Figure 6: Pressure drops for concentrations of R134a-POE32 at medium mass flux (G=450 kg/m^2-sec)

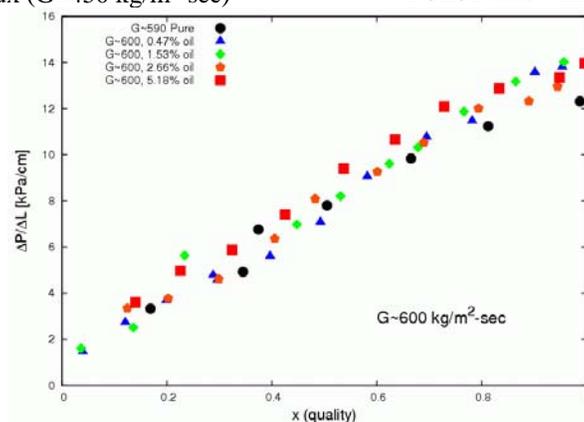Figure 7: Pressure drops for concentrations of R134a-POE32 at low mass flux (G=335 kg/m^2-sec)

Figure 8: Pressure drops for concentrations of R134a-POE32 at high mass flux (G=600 kg/m^2-sec)

Figure 6 shows the medium mass flux of 450 kg/m^2-sec. The data from this mass flux are exactly what might be expected: an increasing pressure drop with increasing oil concentrations, with greatest effect seen at highest quality. This is because the high qualities have a greater concentration of oil in the remaining liquid. The low mass flux of 335 kg/m^2-sec, seen in Figure 7, shows this same trend at the high qualities, but for the lower quality data this is not as apparent, and there appears to be less of a dependence of pressure drop on oil concentration. For the high mass flux of 600 kg/m^2-sec, shown in Figure 8, there is almost no pressure drop variation with oil concentration. This could be explained by the fact that a higher mass flux means more kinetic energy in the flow and therefore a lower dependence of pressure drop on viscosity of the flow.

With the fluid property measurements and the local oil concentrations known as functions of quality (see Figures 1, 2 and 3), the exact liquid phase fluid properties for this refrigerant-oil combination being tested are known. These fluid properties can be applied into the separated flow model of Field and Hrnjak (2006), Equation 9, giving pressure drop predictions based on the pure refrigerant model developed in these same channels. Figure 9 shows the ratio of predicted to measured two-phase pressure drop with the fluid properties of each data point modified according to the local oil concentration. It can be seen that the predictions for the high quality region is quite good, whereas the low quality flow is under predicted. This is slightly surprising, since the greatest effect of oil is in the higher qualities, since the oil concentration is higher at high qualities.
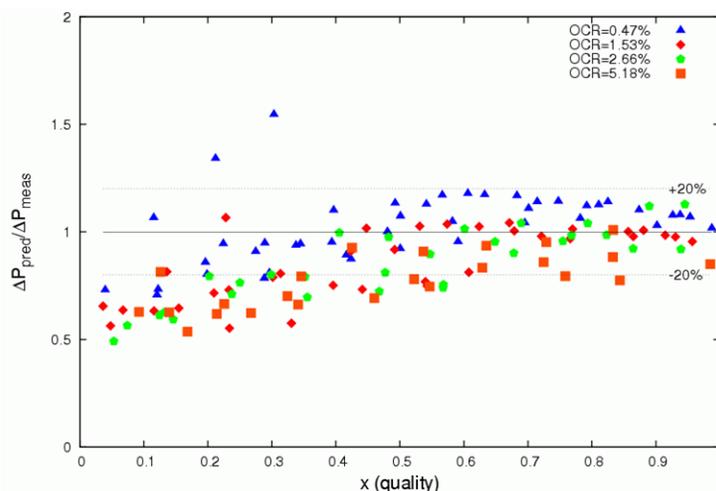

Figure 9: Ratio of predicted to measured pressure drop for refrigerant-oil mixtures. Prediction from Field and Hrnjak (2006), given in Equation (9), accounting for fluid property variation from oil.

5. CONCLUSIONS

In this study, the two-phase pressure drop of refrigerant-oil mixtures have been measured. For the medium mass flux and for low mass flux at high qualities, the findings are as expected: oil causes increase in pressure drop, with increased effect at higher qualities. For the high mass flux, there was a lack of effect on oil concentration which may be explained by the increased kinetic energy in the flow and thus a lower dependence on viscosity. Using knowledge of the local oil concentration, the liquid properties were applied directly into a model that had been developed from a wide span of fluid properties, with success in the higher qualities.

NOMENCLATURE

	Symbols	**units**
C	Two-phase interaction parameter	–
d_h	hydraulic diameter	m
f	friction factor	–
G	Mass flux	kg/m^2-sec
j	Superficial velocity	m/s
L	Length	m
$\dot{m}$	Mass flow rate	kg/s
P	Pressure	Pa
$\dot{q}$	Heat flux	W
Re	Reynolds number	–
U_B	Bubble velocity	m/s
We	Weber number	–
x	Vapor quality	–

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	Greek Letters	**units**		**Subscripts**
ϕ^2	Two-phase multiplier	–	avg	Averaged property
λ	Dimensionless group	–	l	Liquid phase
μ	Viscosity	Ns/m^2	lo	Liquid only
ρ	Density	kg/m^3	meas	Measured value
σ	Surface tension	N/m	o	Oil
ω	Refrigerant fraction in liquid	kg/kg	pred	Predicted value
ψ	Dimensionless group	–	s	Superficial (modifies *l* or *v*)
			tp	Two-phase
			v	Vapor phase

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