REFRIGERANT AND OIL CHARGE DISTRIBUTION IN REFRIGERATION SYSTEMS - MAC UNIT EXAMPLE

RASPODELA RASHLADNOG FLUIDA I ULJA U RASHLADNIM SISTEMIMA NA PRIMERU AUTOMOBILSKOG KLIMA-UREĐAJA

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The paper presents a semi-empirical model to predict refrigerant and lubricant inventory in both microchannel condenser and plate-and-fin evaporator of an air conditioning system.

In the model, heat exchanger was discretized into finite volumes. Temperature, pressure and mass inventory were calculated by applying heat transfer, pressure drop and void fraction correlations to each of these volumes. Refrigerant and lubricant were treated as a zeotropic mixture with a temperature glide. As refrigerant evaporates or condenses, thermophysical properties were evaluated accordingly with the change of lubricant concentration. Oil was assumed to be retained in refrigerant oil mixture at concentration that was changing during phase change in condenser and evaporator.

Experimental data with R134a and PAG oil indicated 20% agreement with the model of refrigerant and oil retention in the evaporator. However, in the condenser, lubricant mass was consistently under-predicted while refrigerant mass was predicted same as in evaporator (within 15% error). Moreover, the lubricant under-prediction became more significant at higher Oil Circulation Ratio (OCR). The analysis showed that the lubricant was separated from the flow in the condenser header and started to accumulate in the bottom channels. The temperature profile in the infrared image supported this hypothesis, as the temperature of the bottom channels was much lower.

U radu je prikazan polu-empirijski model za predviđanje količine rashladnog fluida i lubrikanta (sredstva za podmazivanje) i u mikronanalnom kondenzatoru i u isparivaču sa orebrenim pločama jednog klima-uređaja (sistema za klimatizaciju).

Na modelu je razmenjivač toplote diskretizovan u konačne zapremine. Temperatura, pritisak i masa su izračunati primenom korelacija prenosa toplote, pada pritiska i udela praznine na svaku od ovih zapremina. Rashladni fluid i lubrikant su tretirani kao zeotropska mešavina sa klizanjem temperature. Budući da rashladni fluid isparava ili se kondenzuje, izvršena je procena termofizičkih svojstava u skladu sa promenom koncentracije lubrikanta. Pretpostavka je da se ulje zadržava u mešavini rashladnog fluida i ulja u koncentraciji koja se menjala tokom fazne promene u kondenzatoru i isparivaču.

Eksperimentalni podaci sa R134a i uljem PAG pokazuju slaganje od 20% sa modelom rashladnog fluida i zadržavanja ulja u isparivaču. Međutim, u kondenzatoru, masa lubrikanta je stalno bila predviđana u manjoj vrednosti, dok je masa rashladnog fluida bila predviđena u istoj vrednosti kao u isparivaču (u okviru greške od 15%). Takođe predviđanje manje vrednosti lubrikanta postalo je značajnije pri višem koeficijentu cirkulacije ulja (Oil Circulation Ratio - OCR). Analiza je pokazala da se sredstvo za podmazivanje odvojilo iz toka u kolektoru kondenzatora i počelo da se akumulira u kanalima na dnu. Temperaturni profil na infracrvenoj slici podržava ovu hipotezu, jer je temperatura u kanalima na dnu bila znatno niža.

1. INTRODUCTION

In a vapor compression AC system, refrigerant and lubricant charge is crucial for the reliability of compressor and the performance of heat exchangers. Over the years researchers have developed various ways to experimentally measure refrigerant and lubricant retention. For instance, Peuker and Hrnjak (2010) installed ball valves to isolate charge in each component and measured the mass respectively afterwards. The method was reported to have an uncertainty of 0.4% for refrigerant and 2% for lubricant, but took about 40 hours to finish 1 data set. Due to the time-consuming nature of such measurements, it is rather desired to have an experimentally validated model to generalize the problem and extend to various system and heat exchanger configurations.

Most of the methods to model refrigerant-lubricant mixture found in open literature can be divided into two categories. One is the "oil contamination approach", where oil is treated as a contaminant in an otherwise pure refrigerant. Pure refrigerant properties and correlations are used and the presence of oil is transformed into an independent contamination parameter. Although simple to implement, this approach is in lack of physical meaning. It precludes the physical properties of lubricant in the calculation and the contamination parameter is solely empirical. The other method, presented by Thome (1995), is the so called "thermodynamic approach" where refrigerant-oil mixture is treated as a zeotropic mixture with a temperature glide. This approach overcomes the negative aspects of the oil contamination but requires additional complexity for determining boiling temperature and enthalpy change. The "thermodynamic approach" is used in this work. Since the lubricant used in this study is essentially oil, the term "lubricant" and "oil" are used interchangeably.

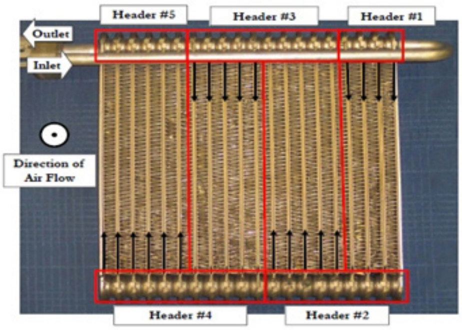
Cremaschi (2004) developed a model for lubricant retention in tube-and-fin evaporator and condenser. The model was validated against experiments in a residential heat pump, with an average error of about 20%. Crompton et al. (2004) investigated oil retention in smooth and micro fin tubes with outer diameter of 9.53mm (3/8"). Based on examination of void fraction and mixture visualization, two models were developed for oil hold-up prediction. The first was based on the test section's liquid volume fraction and was used at mid to low-range qualities. The second, with the Blasius turbulent flow formula as a basis, was used to predict holdup at high qualities. Alonso et al. (2010) measured oil holdup and heat transfer in 9 mm nominal diameter smooth, axially finned and 18° helically finned tubes for both evaporation and condensation. Result was similar to that found under adiabatic conditions in Crompton et al. (2004) and Piggott et al. (2001). However, oil holdup under condensation conditions was found to be lower than oil holdup under evaporation conditions at low mass fluxes where stratified flow exists (100kg/m^{2}s), which could be attributed to "washing" of the walls by pure refrigerant during condensation conditions. Oil was seen to inhibit condensation heat transfer in all cases, but to enhance evaporation heat transfer at lower and mid-range qualities, as was also pointed out by Shen and Groll (2003) in their literature review.

Padilla and Hrnjak (2013) examined the effect of various correlations in modeling pure refrigerant charge in microchannel evaporators and condensers. Validated by experimental data taken from three other different test facilities, their model predicted capacity within 10% and refrigerant charge within 25%.

2. EXPERIMENTAL MEASUREMENTS

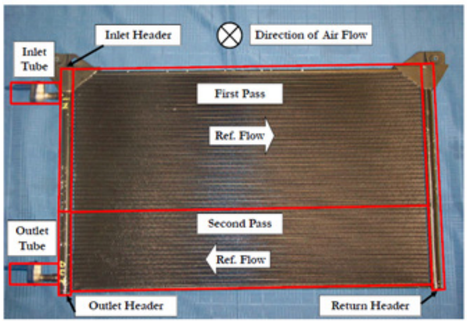
Following the method developed by Peuker and Hrnjak (2010), measurements are carried out on an automotive air-conditioning system (MAC). Components from a GM2007 model SUV are set up at the same relative height as the real system. The compressor is a piston-cylinder reciprocating compressor with fixed displacement with no oil separator. A two-pass design microchannel heat exchanger is used as condenser, with 31 channels in the first pass and 17 in the second. Each channel has 12 ports with a hydraulic diameter of 1mm. The headers are vertical with a diameter of 17 mm. There is no designated passes for subcooling. An external receiver is installed after the condenser, with an internal diameter of 13.08 mm and a length of 560 mm. A thermal expansion valve (TXV) is used as throttling device. A plate-and-fin heat exchanger is used as evaporator. It has 4 passes, 19 plates and horizontal headers with a diameter of 35 mm. Both evaporator and condenser have multi louver fin surface on the air side.

Due to the large size of the environment chambers, tubes used to connect the components are longer than real system. As a result, the experimental system volume is larger than original system and the largest difference in tubing length is from condenser to evaporator, or the liquid line. The volume is determined with two methods: one is based on combining the volume calculated from tube geometry with the internal volumes of components provided by manufacturers; the second is to charge a known amount of gas and calculate volume based on density, also referred to as the isothermal gas method. The two methods agree well, but the isothermal gas method has a slightly larger value than geometrical method. As expected, this is due to the “inactive volume” such as pressure taps and charge ports. Since the inactive volume can hold refrigerant mass as well, the values of isothermal gas method are used.



4 pass; 19 plates; air side area: 4.53 m²
core depth: 7.3 cm; width: 25.45 cm; height: 22.86 cm

Figure 1: Plate-and-fin evaporator



48 channel; 1 mm port; air side area:8.43m²;
core depth: 1.8 cm; width: 71.8 cm; height: 43.58 cm.

Figure 2: Microchannel condenser

During experiment, valves located at both inlet and outlet of each heat exchanger is closed simultaneously to isolate refrigerant and lubricant mass at steady state condition. Using liquid nitrogen as cooling medium, refrigerant mass is recovered and then measured. Lubricant mass is measured by mix and sample technique developed by Peuker and Hrnjak (2010): by adding a known amount of pure refrigerant and sampling the concentration. The detailed measuring principle and procedure can be found in Peuker and Hrnjak (2010). Measurements are carried out with two common refrigerant and oil combinations: R134a with PAG46 and R1234yf with PAG46. The testing condition is mainly focused on idle condition, i.e. minimum compressor speed. So the I35-dry condition from SAE J2765 (SAE Standard, 2008) is investigated as the design condition. Charge tests are performed at this condition to determine the refrigerant charge for maximized COP for both R134a and R1234yf. The same type of lubricant (PAG 46) is used in all tests but varying charge amounts are employed to change the OCR. The tested OCR ranges from 2% to 4%. The system is also operated at higher compressor speeds (L35) to investigate the effect of higher mass flux. Detailed results are presented in Jin and Hrnjak (2013).

Except for the retained mass of refrigerant and lubricant, steady state operation parameters such as temperature, pressure and mass flow rate were also recorded. These measurements at the inlet of condenser or evaporator, together with OCR, were used as inputs for the model to initialize the calculation.

3. MODEL DESCRIPTION

The thermodynamic approach is used in this study to model oil and refrigerant retention in the evaporator and condenser. All simulation and properties evaluation are carried out in Engineering Equation Solver (EES). Both heat exchangers in this study, a plate-and-fin evaporator and a microchannel condenser, are discretized into small elements. The number of element is determined from a sensitivity analysis such that any further increment in elements had trivial effect on the calculation results (<0.5%). In each element, heat transfer and pressure drop correlations are applied so that the calculated temperature and pressure at the outlet could be used as the inlet condition of the next element.

To model the heat transfer between the air and the refrigerant, the Effectiveness - Number of Transfer Unit (ε-NTU) method is used. In equation (1), the overall heat transfer coefficient, UA, is expressed in terms of the total thermal resistance to the heat transfer between the air and refrigerant. The heat transfer coefficients h for different heat exchanger configurations are determined from the designated heat transfer correlations.

$$\frac{1}{UA}=\frac{1}{\eta h_{air}A_{air}}+\frac{\delta_{wall}}{k_{wall}A_{ref}}+\frac{1}{h_{ref}A_{ref}} \tag{1}$$

Thermophysical properties are also evaluated in each element. As a binary mixture, the temperature moves along the dew point line during condensation while evaporation moves along the bubble point line. The vapor pressure of oil is usually orders of magnitude smaller than that of refrigerant. So oil is assumed to be in liquid phase only and vapor phase is approximated as pure refrigerant. The measured pressure at the inlet and proper pressure drop correlations are used to evaluate the pressure in each element. As phase change occurs, oil concentration became diluted during condensation and enriched during evaporation. The local oil concentration is related with local vapor quality x_{mix} and OCR in Equation (2). At steady state, OCR, the ratio of oil mass flow rate over total mass flow rate, is a constant throughout the system. Following the method listed by Radermacher et al. (2006), other thermophysical mixture properties such as density, surface tension, conductivity, specific heat, bubble point temperature and enthalpy change are evaluated. A complete list of equations used in model can be found in Jin and Hrnjak (2014).

$$c_{oil} = \frac{\dot{m}_{oil}}{\dot{m}_{oil} + \dot{m}_{ref,liq}} = \frac{OCR}{1 - x_{mix}} \tag{2}$$

The void fraction model was used to calculate the charge inventory of refrigerant and oil. In a given volume containing both vapor and liquid, void fraction gave the information about the volume taken by each phase. It could be expressed as volume ratio (volumetric void fraction) or area ratio (cross sectional void fraction). In general, as used by Thome (2004), void fraction can be expressed as a function of vapor quality, vapor density, liquid density and slip ratio, as shown in Equation (3). Depending on different theoretical model or experimental data curve fit, the slip ratio *S* can be correlated differently. The homogeneous model, in which liquid and vapor are assumed to be flowing at the same velocity, has a slip ratio *S* of unity. But usually in most models, vapor velocity is higher. The void fraction has a direct impact on the model result of the charge inventory, as will be discussed and compared in later sections. As a general guideline, the selection of void fraction model should represent the effect of fluid properties, flow regime and flow channel geometry.

$$\alpha = \frac{1}{1 + \frac{1-x}{x}\left(\frac{\rho_v}{\rho_l}\right)S} \tag{3}$$

3.1 Evaporator Model

To reasonably represent the reality but keep the model simple and realistic for calculation, the following assumptions were made for the evaporator:

1. Miscible and homogeneous liquid mixture
2. Mixture properties for liquid in correlation
3. Negligible pressure drop in header
4. Uniform distribution of mass flow
5. Bottom headers have liquid level filled up to the channel inlets, as pointed out by Tuo and Hrnjak (2013)

The air side of evaporator has a multi louver fin surface, which has been extensively studied over the past two decades. Park and Jacobi (2009) generalized a correlation from the experimental database of nine independent laboratories for 126 sample heat exchangers. Their correlation is selected to predict air side heat transfer coefficient.

On the refrigerant side, usually refrigerant enters into the evaporator as a two-phase flow, the quality of which is determined isenthalpically from the inlet condition of the expansion valve. For the protection of compressor, it is usually designed to have refrigerant leaving the evaporator as superheated vapor. The degree of superheat is controlled by the thermal expansion valve. As a result, the evaporator is divided into two-phase region and superheat region in this model, with a dividing criterion of the maximum vapor quality. Heat transfer correlations for both regions are applied accordingly. The correlation developed by Yan and Lin (1999) with R134a in a small brazed plate evaporator is selected in this study because of the similar geometry. The correlations were originally developed from pure refrigerant. However, as concluded by Shen and Groll (2003), at certain condition oil can enhance nucleate boiling and increase wetted surface. In addition, the possible foaming can be a significant factor to increase wetted surface. But on the other hand, the high mixture viscosity and mass transfer resistance were considered primary negative factors that impair the flow boiling. Although various studies were carried out, but most correlations are focused on tubes and no correction factor has been found for oil effect on plate heat exchanger. The mixture properties were used instead.

3.2 Condenser Model

In general, the same model was applied to condenser, only with differences in the geometry thus the selection of correlations. The first 4 assumptions for evaporator were also made for the condenser. On the air side, same correlation by Park and Jacobi (2009) was used because of the similar air side fin geometry.

Table 1: Summary of Evaporator Model Correlations

Item	Main Correlation	Reference
Air Heat Transfer	$j = C_1 j_{\mathrm{Re}} j_{low} j_{louver} \alpha^{C_2} N_{LB}{}^{C_3} \left(\frac{F_l}{L_p}\right)^{C_4} \left(\frac{F_d}{F_p}\right)^{C_5} \left(\frac{L_l}{F_l}\right)^{C_6} \left(\frac{F_l}{T_p}\right)^{C_7} \left(1-\frac{\delta_f}{L_p}\right)^{C_8} \left(\frac{L_p}{F_p}\right)^{C_9}$	Park and Jacobi (2009)
Refrigerant Heat Transfer	$Nu_r = 1.926 \mathrm{Pr}_l^{1/3} Bo_{eq}^{0.3} \mathrm{Re}_{eq}^{0.5} \left[(1-X_m) + X_m \left(\frac{\rho_l}{\rho_v}\right)^{0.5}\right]$	Yan and Lin (1999)
Refrigerant Pressure Drop	$\delta P \propto \frac{KE}{Volume} = \frac{G^2}{2\rho}$	Jassim et al. (2006)
Refrigerant Void Fraction	$S = \left(\rho_l / \rho_v\right)^{1/3}$	Zivi (1964)
	$\alpha = \left(1 + 0.025 Xtt\right)^{-2}$	Mandrusiak and Carey (1988)
	$\alpha_{eff} = \beta \alpha_{\mathrm{hom}}$ $\beta = 1 - e^{(ax^b)}$ $\beta = \frac{\rho_l \left(1 - \frac{m_{tp}}{m_{sp}}\right)}{\frac{m_{tp}}{m_{sp}} \alpha_{\mathrm{hom}} \left(\rho_v - \rho_l\right)}$	Jassim et al. (2006)
	$S = 1$	Homogeneous Model

On the refrigerant side however, correlations were selected differently as a result of the different heat transfer mechanism and tube geometry. It was assumed that the refrigerant undergoes four heat transfer process in the condenser: a single-phase superheated vapor process, a superheated condensing process (Kondou and Hrnjak, 2013), a two-phase condensing process and a single-phase subcooled liquid process. So in the model, the condenser was divided into these four zones according to the mixture quality. For the single-phase zones, superheat vapor and subcooled liquid, the Dittus-Bolter heat transfer correlation and Churchill pressure drop correlation were used. In the superheated condensing zone, correlations by Kondou and Hrnjak (2013) were used. In the two-phase zone, heat transfer coefficient and pressure drop were calculated by correlations presented by Cavallini et al. (2006). The correlation was developed for microchannel with hydraulic diameters ranging from 0.4 to 3 mm, which was reported to be the best fit by Padilla and Hrnjak (2013) in their model for microchannel condensers.

Shen and Groll (2003) presented a critical review of the influence of oil on refrigerant heat transfer and pressure drop. The authors compared different heat transfer correlations that can be used in condensation and evaporation of refrigerant and oil mixtures and concluded that when a lubricant impacts both evaporation and condensation adversely, condensation tends to be less affected. The difference is explained by the concept of oil excess layer which forms at the location where phase change occurs. The oil excess layer reduces heat transfer by its insulation effect. During evaporation, the oil excess layer exists at both solid-liquid interface and liquid-vapor interface. But during condensation, the excess oil layer only exists at the solid-liquid interface since refrigerant is continuously condensing at the liquid-vapor interface and diluting the mixture. Since oil does not significantly change heat transfer mechanism during condensation, the correlation for pure refrigerant is used, only by replacing pure refrigerant properties with oil refrigerant mixture properties.

Five void fraction models were used in the condenser. Zivi's (1964) correlation was used since it provided good prediction to refrigerant charge as reported by Padilla and Hrnjak (2013). The homogeneous correlation was also compared. However, both correlations did not include the effect of mass flux and the change of other properties which became more significant due to the presence of oil. To include these changes, the correlation proposed by by Premoli et al. (1971) was used. Another correlation was proposed by Niño et al. (2002). It was developed from the void fraction measurement in microchannel tubes. The vapor Webber number *We*, as the ratio of vapor inertial force with the interfacial force, was correlated to reflect the force vapor needs to break barriers of liquid and promote annular flow.

Instead of using pure refrigerant properties, mixture properties were used in the two correlations. However, compared with measurement and visualization conducted by Burr et al. (2005), the correlations still overestimate void fraction at high quality regions. The reason may due to the fact that the data used for the empirical correlations does not cover such a high viscosity as the oil-rich mixture had. So the last correlation used was a modified Zivi's correlation, with constant as a correction factor. Admittedly a simple constant may not be the best way to correct the void fraction, as the correction factor may also be a function of oil concentration and etc. However, only such results were available in literature in public domain. The range of conditions and refrigerant and oil combination was limited so the current factor is an averaged value. Further experiments would be beneficial.

Table 2: Summary of Condenser Model Correlations

Item	Main Correlation	Reference
Air Heat Transfer	$j = C_1 j_{\mathrm{Re}} j_{low} j_{louver} \alpha^{C_2} N_{LB}{}^{C_3} \left(\frac{F_l}{L_p}\right)^{C_4} \left(\frac{F_d}{F_p}\right)^{C_5} \left(\frac{L_l}{F_l}\right)^{C_6} \left(\frac{F_l}{T_p}\right)^{C_7} \left(1-\frac{\delta_f}{L_p}\right)^{C_8} \left(\frac{L_p}{F_p}\right)^{C_9}$	Park and Jacobi (2009)
Refrigerant Heat Transfer	$h = \rho_l c_{P,l} \left(\tau / \rho_l\right)^{0.5} \big/ T^+$	Cavallini et al. (2006)
Refrigerant Pressure Drop	$\left(dP/dz\right)_f = \Phi_{LO}^2 2 f_{LO} G^2 \big/ (D_h \, \rho_L)$	Cavallini et al. (2006)
Refrigerant Void Fraction	$\alpha_{Burr} = 0.8763 \alpha_{Zivi}$	Burr et al. (2005)
	$S = 1 + B_1 \left(\frac{y}{1+yB_2} - yB_2\right)^{1/2}$	Premoli et al. (1971)
	$S = \left(\rho_l / \rho_v\right)^{1/3}$	Zivi (1964)
	$S = 1$	Homogeneous
	$\alpha = \left[1 + \left(Xtt + \frac{1}{We^{1.3}}\right)\left(\frac{\rho_l}{\rho_v}\right)^{0.9}\right]^{-0.06}$	Niño et al. (2002)

4. EXPERIMENTAL VALIDATION

4.1 Evaporator Validation Results

The comparison between model and experiment are listed in Figure 3, 4 and Table 3. The capacity is predicted within 10% error and both refrigerant and oil mass are predicted within 20% error. The average and standard deviations are calculated as Equation (4), where n is the number of data.

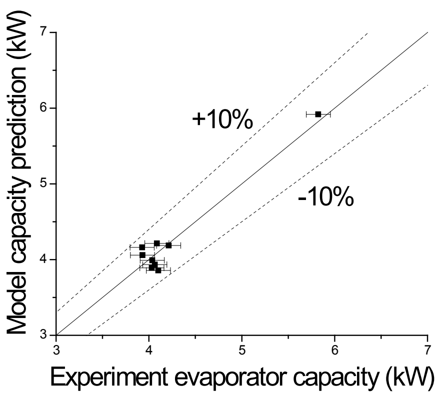


Figure 3: Evaporator capacity validation

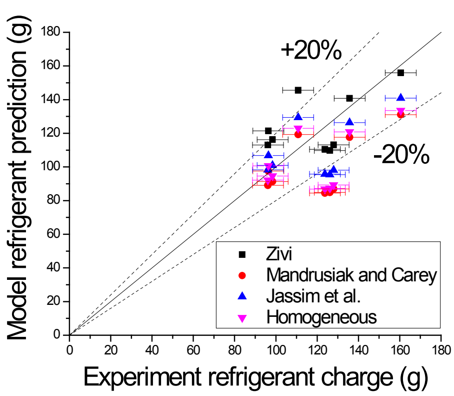


Figure 4: Evaporator refrigerant retention validation

4.2 Condenser Validation Results

As shown in Figure 6 and 7, the capacity and refrigerant mass are reasonably predicted. The average error is less than 10% and 15%, respectively. However, in Figure 8 and Table 4, oil retention is insensitive to the OCR change, resulting in significant under prediction. Such insensitivity indicates there might be a problem in the previous assumptions. Since the model was validated in the evaporator and refrigerant was reasonably predicted in the condenser, the discrepancy in oil prediction indicates that some of the assumptions may not be valid in the condenser. It is possible that oil was trapped at some location so that the even distribution assumption no longer applies. The trapped oil also complicated the relationship between oil retention and OCR, which made the condenser model less sensitive to the input change of OCR.

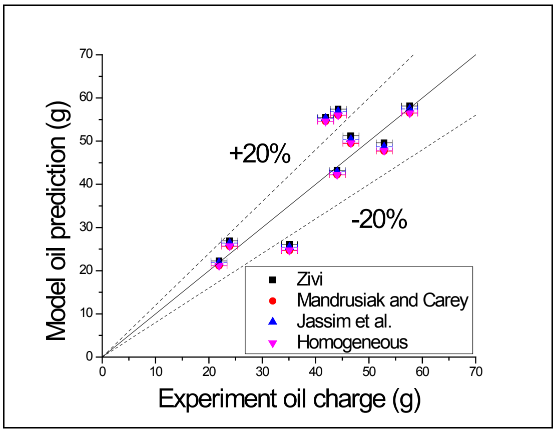


Figure 5: Evaporator oil mass retention validation

Table 3: Evaporator model deviations

Void Fraction Model	$\sigma_{average}$ (%) Ref.	$\sigma_{average}$ (%) Oil	$\sigma_{standard}$ (%) Ref.	$\sigma_{standard}$ (%) Oil
Zivi	15.0	13.4	17.4	18.0
Mandrusiak and Carey	16.9	13.3	20.5	17.4
Jassim et al.	13.6	13.1	15.8	17.8
Homogeneous	15.8	13.2	19.3	17.4

Both gravity and refrigerant shear stress in the vertical header point downwards, it is hypothesized that the inlet header may function as an oil separator and the extra oil starts to fill up the bottom channels in the first pass, as shown in Figure 8. Since oil does not undergo phase change, the temperature of which should be cooled down faster than the rest of the two-phase tubes.

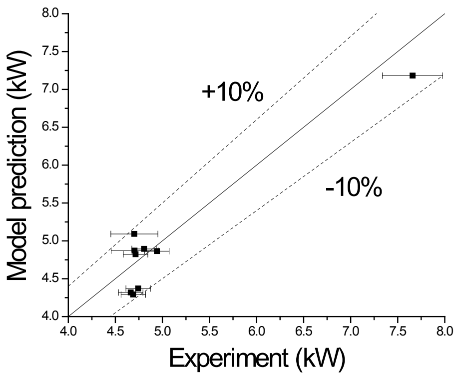


Figure 6: Condenser capacity validation

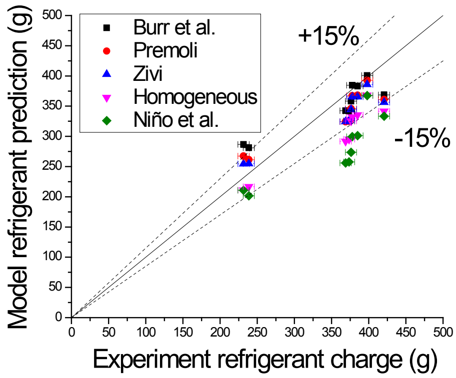


Figure 7: Condenser refrigerant mass validation

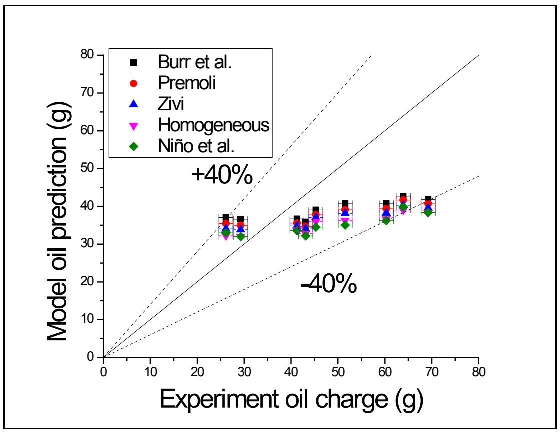


Figure 8: Condenser oil insensitive to OCR, indicating problem

Table 4: Condenser model deviations

Void Fraction Model	$\sigma_{average}$ (%) Ref.	$\sigma_{average}$ (%) Oil	$\sigma_{standard}$ (%) Ref.	$\sigma_{standard}$ (%) Oil
Modified Zivi	8.6	26.1	11.5	28.2
Premoli et al.	9.0	26.7	10.2	28.3
Zivi	8.5	27.0	9.4	28.7
Homogeneous	13.9	27.6	14.7	29.6
Niño	20.5	28.7	22.0	30.5

The infrared image supports this hypothesis, as shown in Figure 9. The channels above the dash line (first pass) show lower temperature, the number of which increases with total oil charge. In addition, the circled areas in Figure 9 also indicate a higher possibility for the channels directly facing the inlet tube to receive more oil droplets, as shown in Figure 10.

After identifying the phenomena, new assumptions were made:

1. First pass of condenser is divided into two parts: vapor channels and liquid channels
2. Uniform mass distribution among channels within each part

3. Same pressure drop across vapor channels and liquid channels
4. Total refrigerant and oil mass flow rates are conserved

The modified model counts in the number of liquid channels from infrared image and adjusted the mass flow in the vapor channels and liquid channels to match their pressure drop. After the correction, prediction of refrigerant retention has slightly higher deviation, as shown in Table 5, but oil prediction shows significant improvement. Overall, the modified model consistently predicts both refrigerant and lubricant within 15%, as shown in Figure 11 and Figure 12.

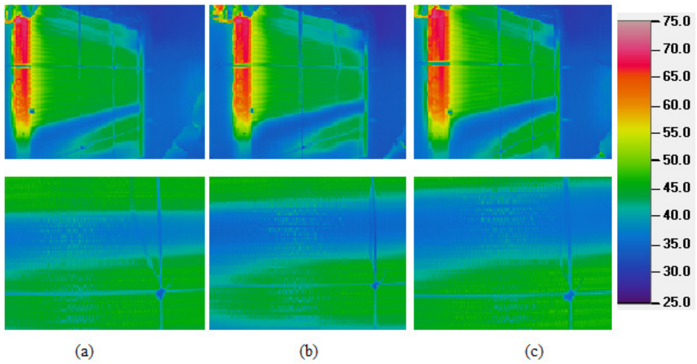


Figure 9: Infrared image for condenser: (a) total oil charge 14g (b) total oil charge 175g; (c) total oil charge 205g

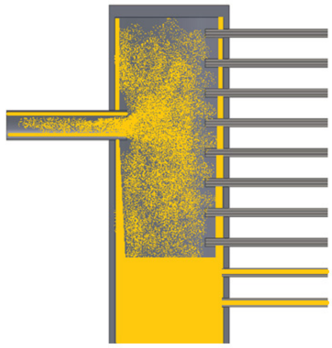


Figure 10: Schematics of oil separation in the inlet header of condenser

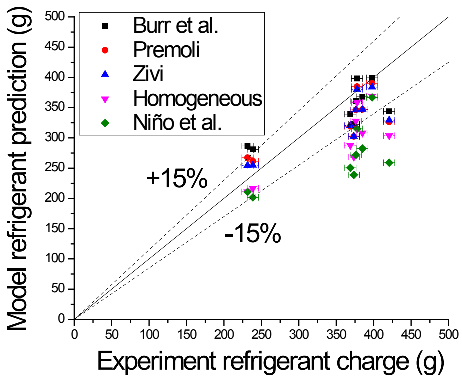


Figure 11: Condenser refrigerant mass modified model

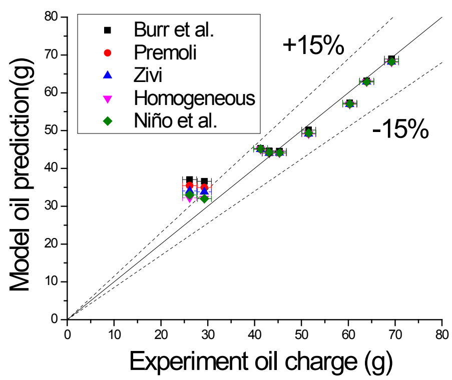


Figure 12: Condenser oil mass modified model

Table 5: Condenser modified model deviation

Void Fraction Model	$\sigma_{average}$ (%)		$\sigma_{standard}$ (%)	
	Ref.	**Oil**	**Ref.**	**Oil**
Modified Zivi	10.7	10.1	13.1	16.7
Premoli et al.	11.3	9.2	13.1	14.2
Zivi	10.3	8.1	12.1	12.1
Homogeneous	15.7	6.7	17.8	9.3
Niño	23.3	6.9	25.7	10.2

$$\varepsilon_{error} = \frac{\left| m_{experiment} - m_{model} \right|}{m_{experiment}} \times 100\%$$

$$\sigma_{average} = \frac{1}{n} \sum \varepsilon_{error} \tag{4}$$

$$\sigma_{standard} = \sqrt{\frac{\sum \varepsilon_{error}^2}{n}}$$

5. SUMMARY AND DISCUSSION

A semi-empirical model to predict refrigerant and lubricant inventory in both evaporator and condenser of an automotive air conditioning (MAC) system was developed. Experimental data from a TXV system with two combinations of refrigerant and lubricant: R134a and PAG 46, R1234yf and PAG 46 respectively, were used to validate the model.

Employing the "thermodynamic approach" and correlations independent of refrigerant and oil type, this model could be generalized to most common miscible refrigerants and oils and would be in principle similar for different channel geometries, whether it's a micro-channel, a plate or a round tube. Yet difference in flow regimes due to the size of flow passage might affect the results, which requires proper selection of correlations for void fraction. From the results with different void fraction models, although physically simplified, the homogeneous model gave a fair estimation. Zivi's (1964) correlation, derived from minimization of kinetic energy for two phases, seemed to be the best overall fit for both condenser and evaporator. Application of the only available data for microchannel void fraction that involves oil (Burr et al. 2005) did not show significant improvement in results.

Initial application of the model predicted both refrigerant and lubricant mass retention in the evaporator within 20% error. In the condenser, refrigerant mass was also predicted within 15%. However, condenser lubricant mass was consistently under-predicted and insensitive to OCR change. In analysis of the under-predicted oil, it was later hypothesized that the lubricant was separated from the flow in the condenser header and started to accumulate in the bottom channels. Additional experiments with infrared image supported this hypothesis: the number of lower temperature channels, presumably filled with oil rich liquid, increases with total oil charge. In addition, after modifying the model by counting in these liquid filled channels, both refrigerant and oil mass in the condenser can be modeled within 15% error. Although the number of liquid channels was not predicted so far, it was the first time in open literature that such phenomenon was documented.

NOMENCLATURE

A	Area	m^2
Bo	Boiling Number	
c	Oil concentration	
Cp	Specific heat	kJ/kg-K
Dh	Hydraulic diameter	m
G	Mass flux	kg/m^2s
h	Heat transfer coefficient	W/m^2-K
j	Colburn-j factor	
k	Conductivity	W/m-K
KE	Kinetic energy	kJ
Nu	Nusselt number	
OCR	Oil Circulation Ratio	
P	Pressure	kPa
PAG	Polyalkylene Glycol	
Pr	Prandtl number	
q	Heat flux	kW/m^2
Q	Capacity	kW
Re	Reynolds number	
s	Specific gravity	
S	Slip ratio	
T	Temperature	K
UA	Overall heat transfer coefficient	W/K
We	Webber Number	
x	Vapor quality	
Xtt	Lockhart-Martinelli parameter	

Subscript

air	Air side
hom	Homogeneous
l	Liquid
lo	Liquid only
mix	Mixture
oil	Lubricant, oil
ref	Refrigerant side
tp	Two-phase
v	Vapor

Greek Letter

α	Void fraction	
ε	Effectiveness, Error	
σ	Deviation	
μ	Viscosity	kg/m-s
ρ	Density	kg/m^3
Φ	Pressure drop multiplier	

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