

An Experimentally Validated Model for Predicting Refrigerant and Lubricant Inventory in MAC Heat Exchangers

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ABSTRACT

The paper presents a semi-empirical model to predict refrigerant and lubricant inventory in both evaporator and condenser of an automotive air conditioning (MAC) system.

In the model, heat exchanger is discretized into small volumes. Temperature, pressure and mass inventory are calculated by applying heat transfer, pressure drop and void fraction correlations to these volumes respectively. Refrigerant and lubricant are treated as a zeotropic mixture with a temperature glide. As refrigerant evaporates or condenses, thermophysical properties are evaluated accordingly with the change of lubricant concentration.

Experimental data is used to validate the model. As a result, refrigerant and lubricant mass is predicted within 20% in the evaporator. However, in the condenser, lubricant mass was consistently under-predicted while refrigerant mass was predicted within 15% error. Moreover, the lubricant under-prediction becomes more significant at higher Oil Circulation Ratio (OCR). The analysis showed that the lubricant is separated from the flow in the condenser header and starts to accumulate in the bottom channels. The temperature profile in the infrared image supports this hypothesis, as the temperature of the bottom channels is much lower. After correcting this by counting in the bottom lubricant mass, the model predicts condenser lubricant mass consistently within 15% error.

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INTRODUCTION

Study has shown that in a typical MAC system, only less than 25% of the total lubricant charge is retained in the compressor at a steady state operating condition [1]. At certain condition and charge, a combined 70% of total lubricant charge resides in the evaporator and condenser [1]. Numerous studies have shown that the presence of oil can significantly change the thermophysical properties of the working fluid, thus affect system performance by altering heat transfer and pressure drop characteristics in the heat exchangers. For better design of heat exchangers with lower oil retention, it is of great interest to investigate how oil is distributed inside the heat exchangers.

Since the lubricant used in this study as well as in most refrigeration system is essentially oil, the term "lubricant" and "oil" are used interchangeably. Over years researchers have developed various ways to experimentally measure refrigerant retention [2][3][4] and lubricant retention [2][3][5][6][7] in vapor compression systems and heat exchangers. These results provided valuable data quantifying the total amount of mass retention and greatly helped to improve the understanding. However, such experiments are usually very time-consuming. It was reported that a whole set of data could take over 40

hours of lab work [2]. So it is rather desired to have an experimentally validated model to generalize the problem and extend to various system and heat exchanger configurations.

In literature, most of the methods to model refrigerant-lubricant mixture 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 essence. It precludes the physical properties of lubricant in the calculation and the contamination parameter is solely empirical. The other method, proposed by Thome [8] in 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.

Using theoretical analysis combined with empirical correlations has shown success to model oil retention in adiabatic tubes. Cremaschi [7] described a separate flow model to simulate oil retention in the suction line. The model assumed an annular flow regime and applied force balance analysis to the refrigerant-rich vapor core and the oil-rich liquid film annuli. With interfacial friction factor and mixture properties determined from empirical correlations, the model well predicted lubricant retention and pressure drop. As was validated by their experimental data on a residential heat pump, the reported accuracy was 21% in average. Sethi and Hrnjak [9] developed a similar model to simulate oil retention in suction lines with R1234yf and POE lubricant. With flow visualization of tubes in different angles and measurement in an independent lab with different measuring techniques, their model predicted 90% data within 20% error.

Modeling refrigerant and lubricant retention in heat exchangers is inherently much more complicated because of the change in flow regime and the addition of energy transport. As a result, heat exchanger models rely more heavily on correlation methods. Using various heat transfer, pressure drop and void fraction correlations, Cremaschi [7] modeled the tube-and-fin evaporator and condenser with an average error of 21% and 23%, respectively. Crompton and Hrnjak [10] investigated oil retention in smooth and micron 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. However, the results from round tube may not be necessarily applicable to microchannel tubes. Padilla and Hrnjak [11] examined the effect of various correlations in modeling 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%.

MODEL DESCRIPTION

The thermodynamic approach was used in this study to model oil and refrigerant retention in the evaporator and condenser. All simulation and properties evaluation were carried out in Engineering Equation Solver (EES). Both heat exchangers in this study, a plate-and-fin evaporator and a microchannel condenser, were discretized into small elements. The number of element was 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 were 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. As outlined in [12], the maximum possible heat transfer in one element is defined as

$$Q_{\max} = C_{\text{air}} (T_{\text{ref},\text{in}} - T_{\text{air},\text{in}}) \quad (1)$$

The actual heat transfer rate in one element is given as

$$Q = \varepsilon C_{\text{air}} (T_{\text{ref},\text{in}} - T_{\text{air},\text{in}}) \quad (2)$$

And the effectiveness is then defined as the ratio of the actual to the maximum possible heat transfer.

$$\varepsilon = Q / Q_{\max} \quad (3)$$

The effectiveness is a function of the NTU and heat capacity ratio C_r , as defined in Equation 4.19. When two-phase flow is present, the heat capacity ratio is zero.

$$C_r = \frac{\min(C_{\text{ref}}, C_{\text{air}})}{\max(C_{\text{ref}}, C_{\text{air}})} = \frac{C_{\min}}{C_{\max}} \quad (4)$$

Since the flow for condenser and evaporator has a cross flow arrangement, Equation 4.20 is used for single phase flow and Equation 4.21 is used for two-phase flow.

$$\varepsilon = 1 - \exp \left[\left(\frac{1}{C_r} \right) \cdot NTU^{0.22} \cdot \left(\exp \left[-C_r \cdot NTU^{0.78} \right] - 1 \right) \right] \quad (5)$$

$$\varepsilon = 1 - \exp(-NTU) \quad (6)$$

where NTU is defined as the ratio of overall heat transfer coefficient and heat capacity rate

$$NTU = \frac{UA}{C_{\min}} \quad (7)$$

The overall heat transfer coefficient, UA, is defined in terms of the total thermal resistance to the heat transfer between the air and refrigerant.

$$\frac{1}{UA} = \frac{1}{\eta h_{\text{air}} A_{\text{air}}} + \frac{\delta_{\text{wall}}}{k_{\text{wall}} A_{\text{ref}}} + \frac{1}{h_{\text{ref}} A_{\text{ref}}} \quad (8)$$

Thermophysical properties were 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 was assumed to be in liquid phase only and vapor phase was approximated as pure refrigerant. The vapor quality of the mixture was then defined as

$$x_{mix} = \frac{\dot{m}_{ref,vap}}{\dot{m}_{ref,vap} + \dot{m}_{ref,liq} + \dot{m}_{oil}} \quad (9)$$

Since OCR was defined as the oil concentration in the liquid line where all circulating fluid was in liquid phase and a homogeneous mixture, OCR was expressed as

$$OCR = \frac{\dot{m}_{ref,liq}}{\dot{m}_{ref,liq} + \dot{m}_{oil}} \quad (10)$$

In the range of operation condition, oil was assumed to be miscible with liquid refrigerant [13]. As phase change occurs, oil concentration became diluted during condensation and enriched during evaporation. Based on mass conservation, the local oil concentration was related with local vapor quality and OCR by rearranging and combining [equation \(9\)](#) and [equation \(10\)](#) as

$$c_{oil} = \frac{\dot{m}_{oil}}{\dot{m}_{oil} + \dot{m}_{ref,liq} + \dot{m}_{ref,vap}} = \frac{OCR}{1 - x_{mix}} \quad (11)$$

[Equation \(11\)](#) shows that the oil concentration would reach maximum at the maximum mixture vapor quality. This maximum concentration was determined by the solubility, as defined by the refrigerant concentration in the liquid mixture at the condition where pure refrigerant would have been superheated.

$$x_{mix,max} = \frac{\dot{m}_{ref,vap}}{\dot{m}_{ref,vap} + \dot{m}_{oil}/c_{oil,max}} = 1 - \frac{OCR}{c_{oil,max}} \quad (12)$$

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, void fraction can be expressed as a function of vapor quality, vapor density, liquid density and slip ratio [29].

$$\alpha = \frac{1}{1 + \frac{1-x}{x} \left(\frac{\rho_v}{\rho_l} \right) S} \quad (13)$$

Depending on different theoretical model or experimental data curve fit, the slip ratio S could be correlated differently. The homogeneous model, in which liquid and vapor are assumed to be flowing at the same velocity, has a slip ratio 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.

MIXTURE PROPERTIES

The change of mixture properties with oil concentration would be one of the primary differences between modeling pure refrigerant and refrigerant-oil mixture. This section listed the equations used in the model that represented such physical essence. The liquid mixture density is approximated by the rule given by Jensen and Jackman [14]

$$\frac{1}{\rho_{mix}} = \frac{c_{oil}}{\rho_{oil}} + \frac{1-c_{oil}}{\rho_{ref,liq}} \quad (14)$$

The liquid viscosity is given by Yokozeki [15] as

$$\ln \mu_{mix} = \xi_{ref} \ln \mu_{ref} + \xi_{oil} \ln \mu_{oil}$$

where

$$\xi_i = \frac{W_i^k \psi_i}{\sum_j W_j^k \psi_j}; k = 0.58 \quad (15)$$

the Yokozeki factor ξ_i is the modified component mole fraction. ψ and W_i are the mole fraction and the molecular mass of the component i , respectively. k is a constant specific to a particular mixture with a 0.58 recommended for universal use. The mole fraction of oil in the liquid phase can be obtained from

$$\psi_{oil} = \frac{c_{oil} \left(\frac{W_{ref}}{W_{oil}} \right)}{1 - c_{oil} + c_{oil} \left(\frac{W_{ref}}{W_{oil}} \right)} \quad (16)$$

The liquid mixture surface tension is given by Jensen and Jackman [14]

$$\sigma_{mix} = \sigma_{ref,liq} + (\sigma_{oil} - \sigma_{ref,liq})\sqrt{c_{oil}}$$

where

$$\sigma_{oil} = (35 - 0.15T[^{\circ}C]) / 1000, \quad (17)$$

The mixture conductivity is given by Filippov and Novoselova (1955)

$$k_{mix} = k_{ref,liq}(1 - c_{oil}) + k_{oil}c_{oil} - 0.72(k_{ref,liq} - k_{oil})(1 - c_{oil})c_{oil}$$

where

$$k_{oil} = 0.1172(1 - 0.0054T) / s$$

$$s = \rho_{oil} / \rho_{water} = 0.93 \quad (18)$$

The mixture specific heat is based on a linear relationship based on mass fraction recommended by Thome [8]

$$Cp_{mix} = c_{oil}Cp_{oil} + (1 - c_{oil})Cp_{liq,ref} \quad (19)$$

The author [30] recommended using [equation \(20\)](#) for oil specific heat

$$Cp_{oil} = 4.186 \frac{0.338 + 0.00045 \cdot (1.8 \cdot T[^{\circ}C] + 32)}{\sqrt{s}} \quad (20)$$

[Equation \(20\)](#) holds for

$$-18[^{\circ}C] < T < 204[^{\circ}C]$$

Takaishi and Oguchi presented an empirical equation for predicting the bubble point temperature from R-22 mixed with alkyl benzene (AB) oil in 1987. Thome [8] verified and generalized the method for other refrigerant and oil mixture including R134a/PAG 46. The relationship is given by

$$T_{bub} = \frac{A(c_{oil})}{\ln P_{sat}[MPa] - B(c_{oil})} \quad (21)$$

Where P_{sat} is the saturation pressure in MPa and T_{bub} is the bubble point temperature in K. $A(c_{oil})$ and $B(c_{oil})$ are given by [Equations \(22\)](#) and [\(23\)](#)

$$A(c_{oil}) = a_0 + a_1c_{oil} + a_2c_{oil}^3 + a_3c_{oil}^5 + a_4c_{oil}^7 \quad (22)$$

$$B(c_{oil}) = b_0 + b_1c_{oil} + b_2c_{oil}^3 + b_3c_{oil}^5 + b_4c_{oil}^7 \quad (23)$$

Thome [8] recommended using accurate pure refrigerant vapor pressure equation to determine the values of a_0 and b_0 while keeping other constants unchanged. The values of the other empirical constants are universal for all automotive refrigerant types:

$$a_1 = 182.52; a_2 = -724.21; a_3 = 3868.0; a_4 = -5268.9$$

$$b_1 = -0.72212; b_2 = 2.3914; b_3 = -13.779; b_4 = 17.066$$

The rest two constants a_0 and b_0 depend on the specific refrigerant and saturation pressure through the following procedures:

- Two pressures just above and below P_{sat} are set and the corresponding saturation temperature are found for pure refrigerant vapor.
- The two sets of values for T_{bub} and P_{sat} are set in the equation with oil concentration set to zero.
- Since all values from a_1 to a_4 and b_1 to b_4 are constant, two unknowns a_0 and b_0 can be solved from two equations

Thome concluded that the enthalpy change dh of a mixture during phase change consists of three contributions:

1. Latent heat to the fraction of the liquid vaporized (dx_{mix});
2. Sensible heat to the fraction of liquid in the liquid phase ($1 - x_{mix}$) changed into another bubble point temperature;
3. Sensible heat to the fraction of liquid in the vapor phase (x_{mix}) changed into another bubble point temperature.

The enthalpy change, also known as heat release enthalpy curve, can be expressed as

$$dh = i_{fg} \cdot d_{mix} + (1 - x_{mix}) \cdot Cp_{mix} \cdot dT_{bub} + x_{mix} \cdot Cp_{ref,vap} \cdot dT_{bub} \quad (24)$$

Where h_{LV} is the latent heat of vaporization of pure refrigerant; Cp_{mix} is the specific heat of refrigerant-oil mixture; $Cp_{ref,vap}$ is the specific heat of the pure refrigerant vapor; dT_{bub} is the change of bubble temperature of the refrigerant-oil mixture. Mixing heat of refrigerant and oil is neglected in the equation since it's usually small and typically not available for refrigerant-oil mixture.

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 [27], as indicated in [Figure 1](#).