A Model of Heat Exchangers and Automotive AC System with Refrigerant-Oil Mixtures

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ABSTRACT

We present the method to simulate refrigerant-oil mixtures (Thome 1995) via a one-dimensional (1D) transient multi-physics system-level simulation tool, GT-SUITE. The impact of oil on bubble point and evaporation enthalpy is first validated, followed by the comparison of the predicted heat transfer coefficient and friction factor with the data by Zurcher et al. (1997). A system model of an automobile air-conditioning (AC) system is developed and validated using experimental data by Liu and Hrnjak (2016). The predicted evaporator capacity and system coefficient of performance (COP) matches very well with the data (both have an average error of 1.4% and the largest error of 4.2%). The simulation provides the detailed state of the refrigerant-oil mixture and the oil concentration along the refrigerant path. The local heat transfer rate from the simulation gives insights into the effectiveness of the evaporator area for engineering design.

1. INTRODUCTION

Lubricant oil typically enters a refrigerant system from the compressor. This oil is carried by refrigerants in the entire system, which can have a significant impact on the performance of the components of the system. The presence of oil increases bubble point temperatures and changes thermodynamic and transport properties of the fluid, which alters the flow pattern, heat transfer and pressure drop of the system.

Many studies have been conducted to investigate the impact of lubricant oil on various aspects of refrigerant systems, including thermodynamic properties, thermo-hydraulic performance and system-level thermal performance. For the thermodynamic properties, Thome (1995) proposed a method to treat the refrigerant-oil mixture as zeotropic to calculate bubble point temperatures, local oil concentrations and enthalpy changes during evaporation. Idrissi et. al. (2003 and 2004) developed a model and validated it against experimental data to include the oil solubility curves in the properties of the refrigerant-oil mixtures. They also provided a review of the thermodynamic impacts of oil (Idrissi and Bonjour 2008). For the thermohydraulic characteristics of refrigerant-oil mixtures, pressure drop increases with an increase in oil concentration, but no such clear trend is observed for the heat transfer coefficient (Bandarra-Filho *et al.* 2009, Shen and Groll 2005). Both simulation and experimental studies on heat exchangers and systems have been conducted in the past for component as well as system design. Radermacher *et al.* (2006) presented a method to model the oil retained in the suction line and evaporator of an air-conditioning (AC) system. Semi-empirical correlations were used in their model to predict the interfacial friction factors within 26% relative errors compared to the experimental data. Stevanovic and Hrnjak (2017) developed a three-dimensional (3D) evaporator model to predict oil retention. The model was validated using experimental data (Li and Hrnjak 2015, Jin and Hrnjak 2016). Liu and Hrnjak (2015 and 2016) also reported oil effects on the system performance of an automobile AC. Previous simulations, especially those validated by experimental data, provide a more detailed understanding of the oil impact and are useful to guide the component and system design.

In the present study, Thome's method (1995 and 2004) is used to calculate bubble point and thermodynamic properties; and the method developed by his group (1997, 2004, 2009) based on including multipliers in the pure refrigerant coefficient is used to calculate heat transfer and friction coefficients. The model is implemented in a one-dimensional (1D) commercial software GT-SUITE, where the models of heat exchangers, expansion devices and compressors are available as templates in the built-in library. This paper first presents a validation of the model by comparing with the data published in the literature, then reports the impact of oil on a typical automobile AC system and its evaporator. The simulation provides detailed information of the impact of oil in individual components as well as the entire system, which is useful in designing and operation of the AC systems.

2. SIMULATION METHODOLGY

2.1 General Flow Solution

The basic building block of a GT-SUITE model is a flow volume. Pipes and flow splits in the system are modeled as flow volumes while orifices are modeled as boundaries of these volumes. Pipes are further discretized into sub-volumes. Heat exchangers are represented as a collection of pipes and flow splits. For each flow volume, conservation of mass, momentum and energy equations are solved. These equations are solved in a 1D manner, i.e., all quantities are averaged across the flow direction.

The software supports explicit as well as implicit time marching. Explicit marching takes smaller time steps (based on the acoustic restriction) and is suited when wave dynamics is important. Implicit marching takes much bigger time steps and typical time steps are of the order of 0.1 seconds. Implicit time marching is generally suited for systems with slower transients, and AC systems typically fall in this category. Therefore, we have used implicit marching in this study. The software has an extensive library to model heat transfer and pressure drop. The multipliers to the heat transfer and friction coefficients are added to those calculated using the correlations for pure refrigerants (Thome 2004). The details are documented in GT-SUITE Flow Manual (GT-SUITE 2020).

2.2 Fluid Properties

The solver has an extensive library for general and flexible modeling of fluid properties; it supports different forms of equation of state, including analytical formulations (such as ideal/real gas approximations and polynomial/algebraic correlations) and tabulated data. The solver fully supports multiple fluid species (Tiwari & Framke 2019, Tiwari & Harrison 2019) and multiple phases with phase transition (Moore at al. 2019, Tiwari & Harrison 2018). The routines for property calculation of pure refrigerants are built upon NIST REFPROP. Mixture properties are calculated by appropriate averaging (assuming that the mixture is homogeneous).

The presence of oil increases bubble point of a refrigerant-oil mixture (Thome 1995) compared to the corresponding pure refrigerant due to an increase in the local oil concentration in the liquid phase. To account oil enthalpy during mixture enthalpy calculations, the solver pre-computes and stores heat release curves (enthalpy change between the saturation liquid and saturation vapor of the mixture at a given pressure) for different oil concentrations.

2.3 Compressor Model

A map-based approach is used to model the compressor performance. Two maps are needed: efficiency map and speed map. Efficiency map includes compressor speed, pressure ratio and isentropic efficiency; and the speed map includes compressor speed, pressure ratio and refrigerant mass flowrate. Triangle interpolation method is used during the system simulation to eliminate any errors from the original data points. During the simulation of a transient operation, the pressure ratio is obtained from the flow solution. A combination of the compressor speed and the pressure ratio determines the mass flowrate and the isentropic efficiency of the compressor.

2.4 Expansion Device

The expansion valve is modeled as a round hole with a controllable diameter, i.e., an orifice connection for the portion in contact with the flow and a PID controller to target the superheat at the evaporator outlet of the AC system. The orifice is modeled as a planar object with no volume and the momentum equation is solved to compute the mass flow rate and the velocity (see GT-SUITE Flow Manual).

3. MODEL VALIDATION

3.1 Thermodynamic and Transport Properties

The comparison of bubble point and evaporation enthalpy of R134a-oil mixture predicted by GT-SUITE with

Thome's data (2004) is presented in Fig. 1 for the oil circulation ratio of 3%. Additionally, the comparison for oil circulation ratio of 5% is also conducted. For the oil circulation ratio of 5% and 3%, the difference in bubble point between Thome's data and GT prediction is within 0.04ºC and 0.09ºC, respectively; the difference in local oil concentration is within 0.006% and 0.44%, with an average error of 0.002% and 0.05%, respectively; the percentage difference in the evaporation enthalpy is within 1.6% and 1.4%, respectively. This comparison demonstrates correct implementation of the methodology proposed by Thome (1995 and 2014) in GT-SUITE to capture the thermodynamic properties in the presence of oil.

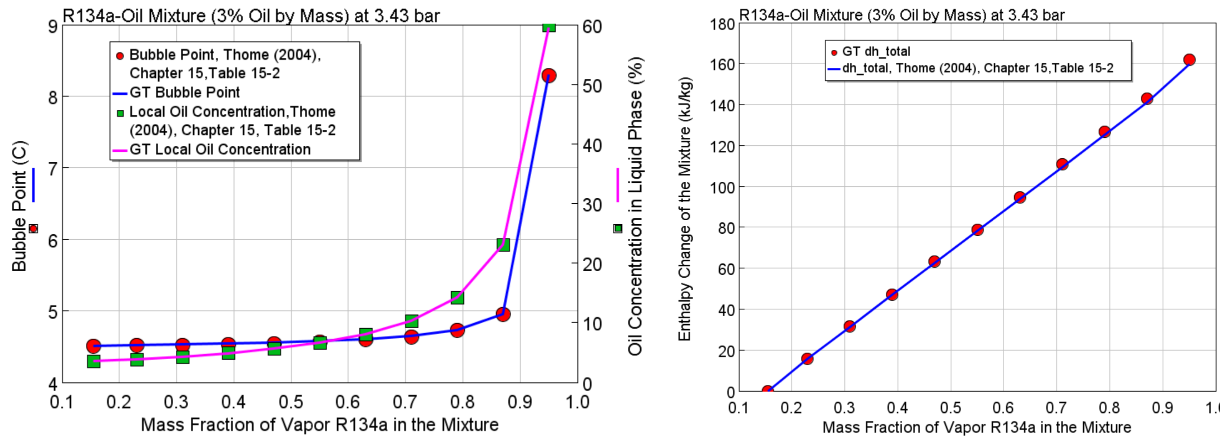


Figure 1: Comparison of the prediction with Thome's (2004) data for R134a and Mobil Arctic EAL68 (POE)

Table 1: Transport properties compared to Thome (2004) Chapter 15 Table 15-4.

Absolute Quality	Apparent Quality	Mixture Density (kg/m^3)			Mixture Thermal Conductivity (W/m-K)		
x_{tot}	x_{ref}	Thome (2004)	GT simulation	Difference	Thome (2004)	GT simulation	Difference
0.000	0.000	1276.0	1273.7	2.3	0.0940	0.0944	0.0004
0.279	0.294	50.5	50.2	0.3	0.0718	0.0719	0.0001
0.558	0.587	25.8	25.6	0.2	0.0487	0.0495	0.0008
0.744	0.783	19.5	19.3	0.2	0.0335	0.0345	0.0010
0.930	0.979	18.0	14.9	3.1	0.0194	0.0200	0.0006
Absolute Quality	**Apparent Quality**	**Mixture Specific Heat (J/kg-K)**			**Liquid Viscosity (cP)**		
x_{tot}	x_{ref}	**Thome (2004)**	**GT simulation**	**Difference**	**Thome (2004)**	**Arrhenius Mixing Law**	**Linear Mixing Law**
0.000	0.000	1363.0	1359.8	0.2%	0.398	0.384	20.2
0.279	0.294	1237.2	1235.9	0.1%	0.458	0.442	27.7
0.558	0.587	1110.9	1112.0	0.1%	0.628	0.608	44.9
0.744	0.783	1027.0	1029.2	0.2%	1.137	1.115	76.9
0.930	0.979	960.5	940.4	2.1%	35.83	35.4	186.4

A comparison of transport properties is listed in Table 1. It compares mixture density, mixture thermal conductivity, mixture specific heat, and liquid viscosity for different oil mass fractions. In the table, absolute quality (x_{tot}) refers to the mass fraction of the refrigerant vapor to the total mass of refrigerant and oil, while apparent quality (x_{ref}) refers to the ratio of refrigerant mass to the total refrigerant mass. Excellent agreement further validates the implementation of the method. Note that liquid viscosity is chosen for comparison here rather than mixture viscosity because the heat transfer and friction correlations use phasic viscosities and not mixture viscosities. Also note that Thome (2004) applied Arrhenius mixing rule to calculate the viscosity of the liquid phase. If a linear mixing rule based on mass percentage is used, the viscosity of the liquid phase could be a few orders of magnitude larger. Overall, the transport properties calculated in GT simulation are consistent with the ones published by Thome (2004).

3.2 Thermo-Hydraulic Performance

Heat transfer coefficient predicted in this study is compared with the test data of Zurcher et al. 1997 in Fig.2. The tests were conducted for flow boiling of R134a with various oil circulation ratios at mass flux = 300kg/m^2-s. The simulation model is set to calculate the heat transfer coefficient at 3.43bar, i.e. the saturation pressure of R134a at 4.4 ºC. For this study, we used the Yoshida correlation (1983) implemented in the solver. Fig. 2 shows that it successfully predicts the trends with oil presence: (1) the heat transfer coefficient decreases with the increase in system oil percentage; and (2) the peak in heat transfer coefficient shifts to lower refrigerant quality as oil increases. The maximum heat transfer coefficient occurs around absolute quality of 0.74 and 0.69 for 3% and 5% oil, respectively. The maximum heat transfer coefficient for 3% and 5% oil is 5945W/m^2-K and 4936 W/m^2-K, respectively.

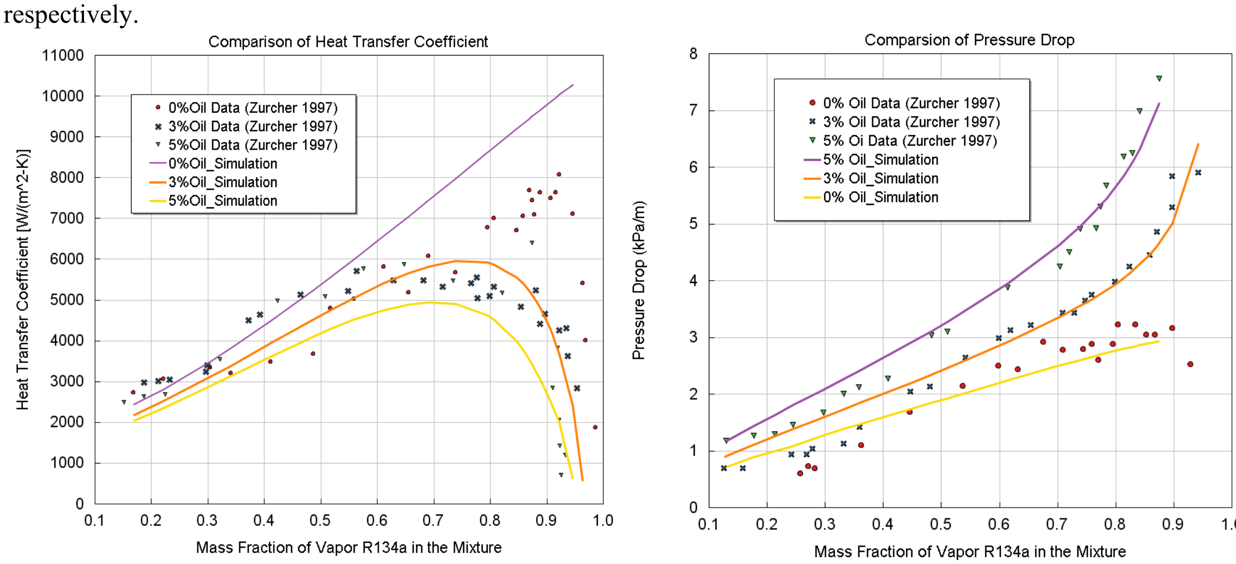


Figure 2: Predicted heat transfer coefficient (left) and pressure drop (right) of the R134a-oil mixture compared to data in Zurcher et al. (1997).

The Yoshida correlation used here is a corrected form of the original equation with two multipliers: one is fixed value of 1.7964 to correct the baseline of pure refrigerant and the other is the multiplier based on viscosity ratio and local oil concentration for the oil impact (Thome 2004). As mentioned earlier, GT-SUITE provides various options to model heat transfer and friction coefficients. Additional flow boiling correlations already implemented in GT-SUITE, including Shah (1982), Gungor (1987), Yan (1999), Klimenko (1990), and Kandlikar (2000) are applied, and it is found that the corrected Yoshida correlation provides the best results.

The simulated pressure drop also provides consistent results compared to the test data of Zurcher 1997, as shown in Fig. 2. The tests were conducted to study the flow boiling of R134a with various oil concentration for mass flux of 200kg/m^2-s at 3.43bar, i.e. the saturation pressure of R134a at 4.4 ºC. The model is setup under the same operating conditions. Here also, we applied two correlations for pressure drop: Friedel (1979) and Muller-Steinhagen-Heck correlation (1986), and found that Friedel correlation with a multiplier provided best match. The model form here also has two multipliers: one for oil effect (1.2 and 1.5 for 3% and 5% oil, respectively) and the other for the viscosity ratio as proposed by Thome (2004).

The simulation correctly predicts the trend of the change in the pressure drop as the refrigerant evaporates. The pressure drop increases almost linearly with vapor quality for pure refrigerant. When oil is presented, as shown in Fig.2, the increase in pressure drop becomes much larger than that of pure refrigerant, especially when the quality is larger than a critical value (~0.7 or ~0.8 for 3% oil and 5% oil, respectively). For example, when absolute quality is 0.87, the predicted pressure drop is 2.9kPa/m, 4.60kPa/m and 7.13kPa/m, respectively, for pure R134a, R134a with

3% oil, and R134a with 5% oil. Such a large increase in pressure drop due to the oil may significantly contribute to the pressure drop in the suction line as observed in the experiments (Liu and Hrnjak 2016).

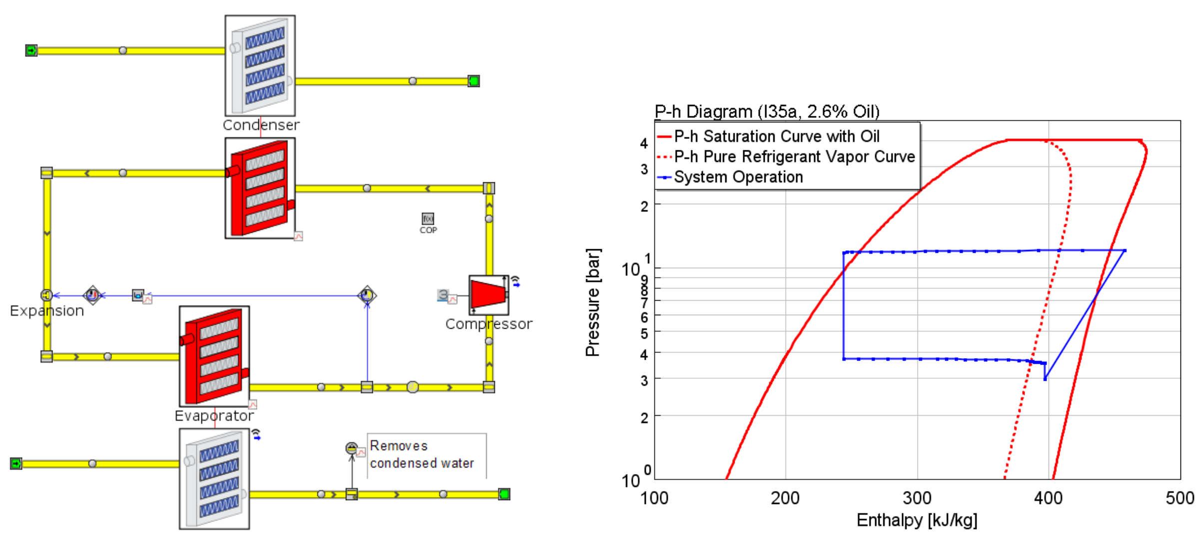
4. SYSTEM SIMULATION

4.1 Model Setup

We now consider a realistic refrigerant-oil system model. We developed this model in GT-SUITE to simulate the performance of an automobile AC system using R134a with oil presence as tested by Liu and Hrnjak (2016). For comparison with the test data, the simulation is performed under two operating conditions (I35a and M35a from SAEJ2765), and six different the oil circulation ratios of PGA46 oil in R134a for each operating condition.

For condenser and evaporator modeling, the geometry of the heat exchangers (Liu 2015, Liu and Hrnjak 2016) is used as an input to the model. The correlations for heat transfer and pressure drop are used to calculate their thermal and hydraulic performance. One of the salient features of GT-SUITE is that its extensive library allows to set different correlations for different regions in the system depending on the refrigerant state (according to the P-h diagram). For heat transfer coefficient, we use the same modified Yoshida correlation that we described in Section 3.2. Note that multipliers are used to account for the difference in geometry and operating conditions used to develop the published correlations. Similarly, a friction multiplier is added to adjust the friction factor calculated directly from the correlation for the refrigerant pressure drop in each heat exchanger. Moreover, a discharge coefficient is included to model the area change at the connections between the heat exchanger and the pipe.

For the compressor, the 12-test data around the compressor (Liu and Hrnjak 2016), namely the compressor speed, pressure ratio, mass flowrate and efficiency, are used to develop a compressor performance map. Two multipliers are applied to adjust the mass flowrate and the isentropic efficiency in its performance map. For the expansion valve, the degree of superheat from the tests (Liu 2015) is used as a target for the PID controller to adjust the opening of the valve. The pressure drop in the suction line is modeled as a function of flowrate. The data reported in Liu and Hrnjak (2016) are used to develop this relationship. The reported refrigerant charge is also used in the simulation for all 12 sets of test data.



(a) (b)

Figure 3: (a) System model layout in GT-SUITE with built-in templates of heat exchangers, compressors etc. (The fluids on the two sides of the heat exchanger are represented in different colors: blue for air and red for refrigerant flow.), and (b) P-h diagram of the simulated system performance with 2.6% oil under the I35a condition.

Fig. 3 shows the system model in GT-SUITE (with built-in templates) and an example of the P-h diagram from the system simulation with 2.6% oil under I35a. When the model is used to simulate the performance of the system reported by Liu and Hrnjak (2016), the inputs to the model are (1) the oil circulation ratio, (2) compressor speed, (3) degree of superheat at evaporator outlet, (4) air flowrate, temperature and humidity at the evaporator inlet, and (5) air flowrate and temperature at the condenser inlet. These inputs remain the same as reported in Liu and Hrnjak (2016) for each set of test data.

4.2 Simulation Results of System Performance

The simulation results under the testing conditions reported by Liu (2015) and Liu and Hrnjak (2016) are presented in this section. (Note that since the degree of superheat at the evaporator outlet is used to adjust the expansion valve, we have verified that the targeted superheat is reached for each data point in the simulation.) Comparison of simulation results with the data is presented in Figs. 4 to 6. Heat transfer rates in the evaporator and condenser are shown in Fig. 4.; compressor power consumption and system COP are shown in Fig 5., and pressure drop in the evaporator and mass flowrate of the refrigerant-oil mixture are shown in Fig. 6.

The heat transfer rate in the evaporator is predicted within 2% of the test data for the 12 data points except one (I35a with 1% oil), as shown in Fig. 4. For I35a with 1% oil, the percentage error compared to the test data is 4.2%, which is within the measurement uncertainty reported by Liu and Hrnjak (2016). The 6 data points under the condition of M35a reveal the trend of monotonic decrease of heat transfer rate with an increase in the oil circulation ratio in the system. The same trend is followed by 5 data points forI35a except the one with 1% oil. The accuracy of this data point might be questionable. Fig. 4 also shows that the simulation over-predicts the heat rejection from the condenser by 6% in average and the largest error is 10% for the test under the condition of I35a with 1% oil.

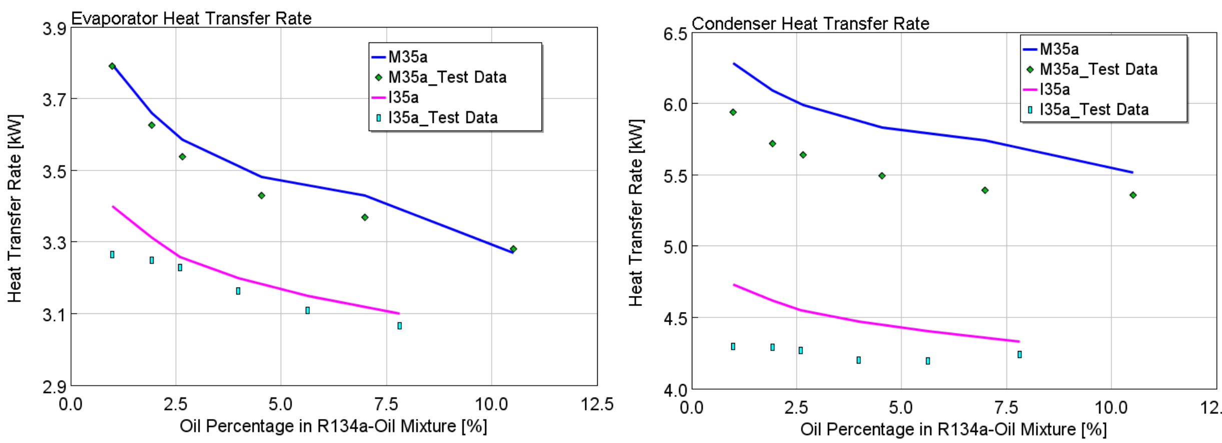


Figure 4: Simulation results of the 12 tests compared with the experimental measurements: heat transfer rate in evaporator and condenser.

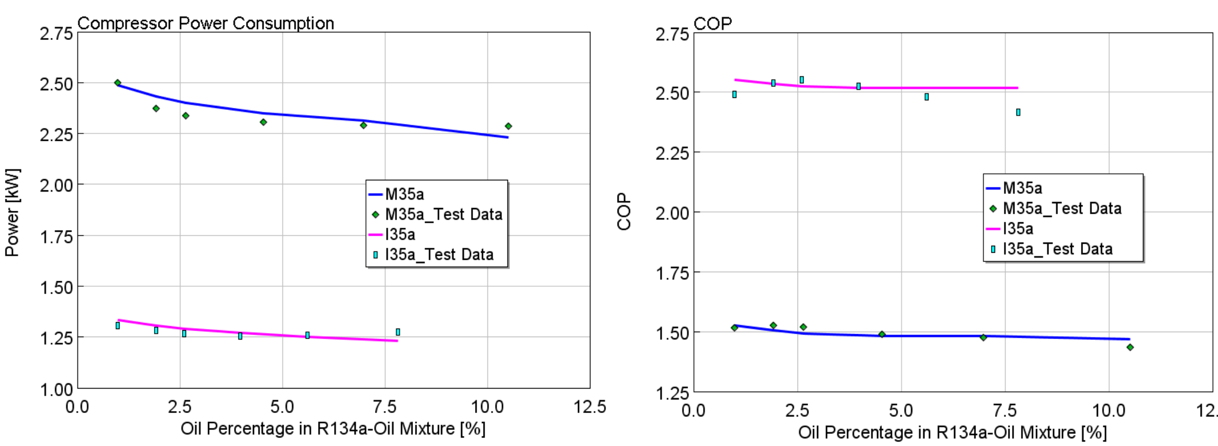


Figure 5: Simulation results of the 12 tests compared with the experimental measurements: compressor power consumption and system COP.

The system model predicts the power consumption of the compressor within 3.3% of the test data as shown in Fig.5. The average error in power consumption is 1.8%; the largest error is 3.3% for the test under I35a with 7.8% oil.

Consistent with the predication in compressor power, the simulated COP differs from the test data with an average error of 1.4%; the largest error is 4.2% for the test of I35a with 7.8% oil.

The predicted pressure drops of the refrigerant-oil mixture in the evaporator need to be improved. As shown in Fig. 6, the average difference between the simulated pressure drop and the test data is 2.4kPa for the 12 data points, but the maximum difference is 6.4kPa. The percentage difference is in the range of 1.6% to 23.3%. One encouraging trend is that this difference seems consistent with the difference in the mass flowrate of the refrigerant-oil mixture. When the mass flowrate is predicted close to the test data, e.g. I35a with oil circulation ratio of 1.9% and 2.6%, the pressure drop is predicted to be 1.9% and 1.6%, respectively. When the mass flowrate is significantly under-predicted, e.g. test data of M35a with 1% to 4.5% oil, the pressure drop is under-predicted by more than 14%. To improve the simulation results, the multipliers and discharge coefficients in the evaporator should be optimized. These parameters, as the characteristics of the heat exchangers, are fixed over the system simulation for the 12 data points. They are determined by a few iterations of trial and error. If improved accuracy of the simulation results is desired, an optimization of these parameters could be conducted using the built-in optimizer of GT-SUITE.

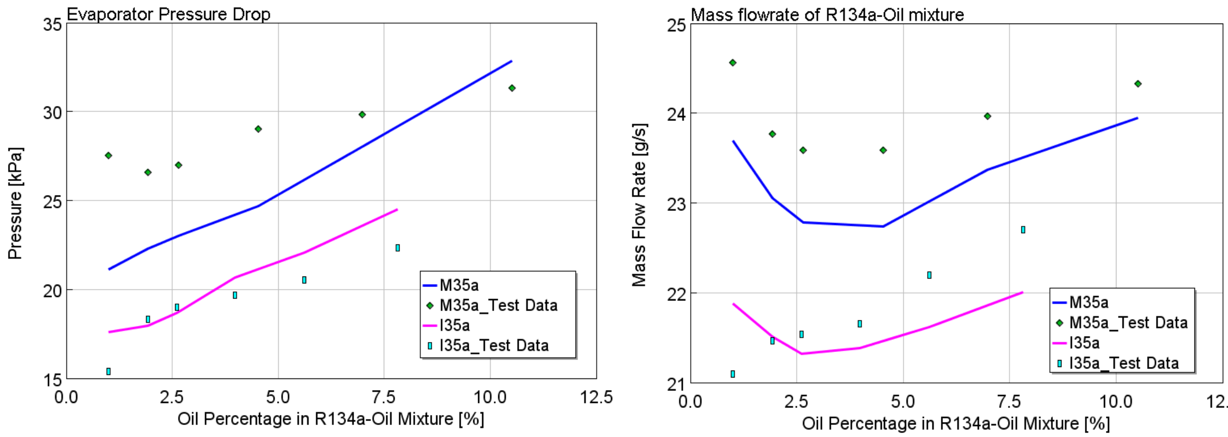


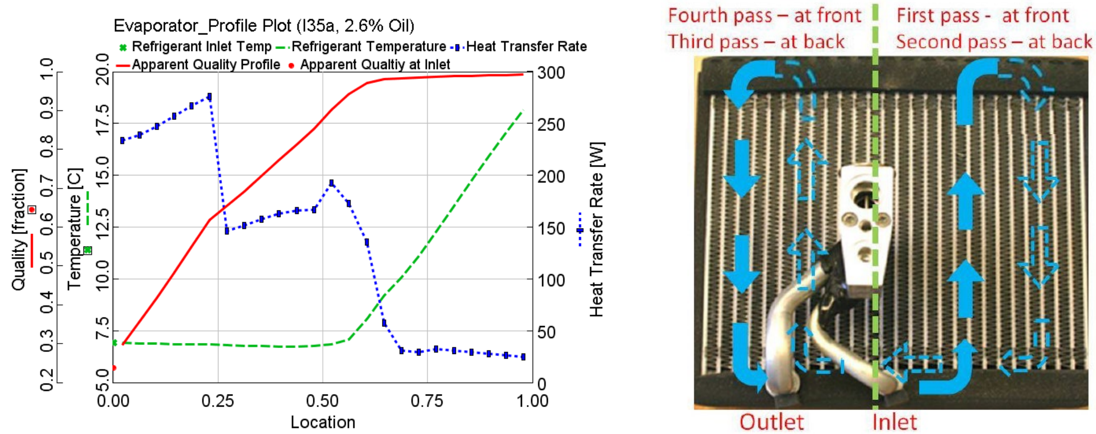
Figure 6: Simulation results of the 12 tests compared with the experimental measurements: refrigerant pressure drops in the evaporator and mass flowrate of the R134a-oil mixture.

4.3 Detailed Analysis of Evaporator Performance

In this section, we analyze in detail the results of the evaporator performance under the testing condition of I35a with 2.6% oil as an example to explore the oil impact. This case is selected because most of the performance for this test are well-predicted by the system model (see Figs. 4-6). The goal here is to demonstrate that the detailed state of the refrigerant-oil mixture inside the evaporator can be obtained to guide the system design with this 1D model.

The simulated results of the heat transfer rate, apparent quality and the temperature of the refrigerant-oil mixture are reported in Fig. 7(a) along the normalized location in the flow direction, that is, 0 and 1 represent inlet and outlet of the evaporator, respectively. In the simulation, the evaporator is discretized into 24 sub-volumes to capture the property change during the evaporation. The temperature of the refrigerant-oil mixture starts to increase after the normalized location of 0.52 as the apparent quality becomes close to 1, and correspondingly, the heat transfer rate decreases significantly after this location. As presented in Liu (2015), the refrigerant-oil mixture entered into the 15 tubes from the downstream of the airflow, and then turned into the 15 tubes upstream to the airflow. After that, the flow entered the 10 tubes upstream of the airflow on the left side of the picture in Fig. 7(b). The normalized location of 0.5 is where the flow turned from the 15-tube to the 10-tube at front upstream of the airflow. The simulation shows that the effective heat transfer occurs in the 15 tubes on the right side of Fig. 7(b) under this condition.

The system performance in the P-h diagram with respect to the saturation vapor line of both pure refrigerant and the refrigerant-oil mixture is shown in Fig.3(b). As discussed by Thome (1995), the evaporation enthalpy increases with the oil concentration in the liquid phase of the mixture. When the refrigerant-oil mixture reaches the saturation vapor line of the pure refrigerant, the apparent quality is close to 1 and it is difficult to evaporate the remaining refrigerant from the liquid phase of the mixture.



(a) (b)

Figure 7: (a) Performance of the evaporator along the normalized refrigerant flow direction from the simulation of I35a condition with 2.6% oil. The symbols in the line of the heat transfer rate represents the location of each subvolume. (b) Refrigerant flow configuration in the evaporator (picture from Liu 2015): 15 tubes on the right and 10 tubes on the left with the refrigerant-oil mixture turning upstream and downstream of air flow.

Table 2: Oil mass fraction, vapor quality and degree of superheat at the evaporator inlet and outlet

Operating Conditions	**Total Oil Mass Fraction, w_{tot} (%)**	**Liquid Phase Oil Mass Fraction, w_{loc} (%)**		**Apparent Quality, x_{ref}**		**Absolute Quality, x_{tot}**		**Superheat *wrt.* pure R134a at same press. (Evap.Outlet) (°C)**
		Evap. Inlet	**Evap. Outlet**	**Evap. Inlet**	**Evap. Outlet**	**Evap. Inlet**	**Evap. Outlet**	
M35a	1.0	1.3	80.0	0.234	0.997	0.231	0.988	15.0
M35a	1.9	2.5	81.3	0.240	0.995	0.235	0.976	16.3
M35a	2.6	3.5	82.0	0.244	0.994	0.237	0.968	16.8
M35a	4.5	6.0	82.4	0.251	0.990	0.240	0.945	17.3
M35a	7.0	9.2	81.9	0.257	0.983	0.239	0.915	16.8
M35a	10.5	14.0	82.4	0.278	0.975	0.248	0.872	17.2
I35a	1.0	1.3	75.1	0.231	0.997	0.229	0.987	11.2
I35a	1.9	2.5	76.5	0.235	0.994	0.231	0.975	12.2
I35a	2.6	3.4	77.3	0.238	0.992	0.232	0.966	12.8
I35a	4.0	5.2	77.5	0.242	0.988	0.232	0.949	13.0
I35a	5.6	7.3	77.7	0.246	0.983	0.232	0.928	13.1
I35a	7.8	10.2	78.3	0.252	0.976	0.232	0.900	13.6

Table 2 shows oil mass fraction, vapor quality and degree of superheat at the evaporator inlet and outlet for all the test cases considered. Note that for the test of I35a with 2.6% oil, the apparent quality is 0.992, i.e., 99.2% refrigerant is in vapor phase at the evaporator outlet. Under this condition, the mass of vapor refrigerant (absolute quality) is 96.6% of the total R134a-oil mass and the oil mass fraction in the liquid phase is 77.3%. When the refrigerant-oil mixture turned into the 10 tubes on the left, most of the refrigerant is already vaporized, thus most of the liquid phase contains oil. The local oil mass fraction is found less than 83% at the evaporator outlet. This oil percentage indicates the presence of oil in liquid R134a leaving the evaporator, even though there is over 10°C superheat with respect to pure R134a. The larger the oil circulation ratio in the system, the smaller the aboslute quality at the evaporator outlet under the same testing condition. Though the absoute quality, as commented by Thome (1995), is thermodynamically correct, the apparent quality based on the total mass of refrigerant provide direct measure to the vaporization of the refrigerant and it is pratically challenging, if not impossible, for the apparent quality to reach one because of the oil presence.

5. CONCLUSION

The refrigerant-oil mixture is modeled as a zeotropic mixture (Thome 1995) in the 1D simulation tool GT-SUITE. Thermodynamic and transport properties from the model are first verified against the data published by Thome (2004). The heat transfer coefficient and pressure drop during evaporation are also calculated and compared with published data by Zurcher et al. (1997). Validation tests showed excellent prediction of the thermo-hydraulic performance. The corrected Yoshida (1983) is found to provide the best prediction for the heat transfer coefficient of R134a with different oil circulation ratios.

The system model of an automotive AC with R134a and oil is developed using the built-in templates (including heat exchangers, the compressor and the expansion valve) in GT-SUITE. The simulation predicts evaporator capacity and COP very well (with an average error of 1.40% and 1.38%, respectively). The error in predicted pressure drop in the evaporator ranges from 1.6% to 23.3%. The error in pressure drop is closely related to the prediction in the mass flowrate of the refrigerant-oil mixture. The model can be improved by optimizing the parameters used to characterize the heat exchanger performance and the compressor by the built-in tool of GT-SUITE.

The model provides the detailed state of the refrigerant along the flow path. We analyzed evaporator performance and found that the effective heat transfer (82% capacity) was on the left 15 tubes of the evaporator (60% of area) under test I35a with 2.6% oil. The simulation manifested that there was always liquid R134a carried in the liquid with oil leaving the evaporator despite over 10ºC superheat at the evaporator outlet. This behavior may be caused by the change in evaporation enthalpy with the oil presence. The model can provide the local oil concentration and the system performance in the P-h diagram with respect to the saturation vapor line of both pure refrigerant and the refrigerant-oil mixture.

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