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Refrigerant Migration Modeling during Shut-down and Start-up Cycling Transients

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ABSTRACT

Refrigerant mass migration and redistribution have been regarded as factors influencing the performance of air conditioning and refrigeration systems. This paper presents a dynamic model of an R134a automotive A/C system that is able to capture the shut-down and start-up cycling dynamics including the refrigerant mass migration transients. Building upon recent work (Li and Alleyne, 2010), the heat exchangers are developed using a switched moving-boundary modeling framework, which maintains constant model structures but accommodates different model representations. Thermosys, a Matlab/Simulink toolbox, is introduced to simulate system transient behaviors. Model validations against experimental data show that the developed dynamic model can well predict the migration performance of the refrigerant mass across the components during shut-down transients, and the resulting refrigerant redistribution transients at start-up operations. The qualitative accuracy of the validation results demonstrates the potential of this dynamic modeling approach for control design and diagnosis in improving system start-up performance.

1. INTRODUCTION

Refrigerant mass migration and redistribution across the components have been regarded as factors influencing the transient and cycling performance of air conditioning and refrigeration systems (Rubas and Bullard, 1995). The experimental study on an R134a automotive A/C system (Peuker and Hrnjak, 2009) show that during the shut-down period 48% of the total refrigerant mass migrates from the high-pressure components (condenser and liquid tube) to the low-pressure components (evaporator and accumulator). The end system states after shut-down transients is the initial condition for the start-up operations, and therefore the distribution of the refrigerant mass across the system components at the end of the shut-down period has a direct influence on the system start-up performance. Peuker and Hrnjak (2009) report a 28% reduction in compressor energy during the first 25 seconds of the start-up without cooling capacity loss when refrigerant charge migration is prevented during the shut-down period. Therefore, to accurately model system cycling performance it is necessary to predict the migration of the refrigerant mass across the components during shut-down transients and the resulting refrigerant redistribution performance after start-up operations.

An advanced switched moving-boundary modeling approach (Li and Alleyne, 2010) is presented to describe the transient behaviors in heat exchangers under compressor shut-down and start-up operations. Switching schemes between different model representations based on the mass conservation principle are introduced to accommodate the transitions of heat exchanger dynamic states. Model validation results prove the validity of the modeling approach. In this study, we aim to develop a dynamic model of the automotive A/C system (Peuker and Hrnjak, 2009) using the switched modeling framework to capture the refrigerant migration transient during the shutdown-startup operations.

The rest of the paper is organized as follows. The experimental system used for modeling and validation are briefly introduced in Section 2. Section 3 presents a dynamic system model to describe the shutdown-startup cycling behaviors along with the refrigerant mass migration dynamics. The model validation is provided in Section 4. The results demonstrate the capabilities of the developed model for capturing the refrigerant mass migration performance and system cycling dynamics. A conclusion section summarizes the main point of the paper.

2. EXPERIMENTAL SYSTEM

The experimental system used for modeling and validation in this study is an R134a automotive A/C system containing the following components: compressor, condenser, fixed orifice tube (FOT), evaporator, accumulator, and a full suite of sensors for temperature, pressure and mass flow rate. The components are installed into the experimental facility (Peuker, 2006) with the same difference in vertical height as in the vehicle. The schematic of the experimental system is shown in Figure 1. The ball valves are installed around each component, and the refrigerant and oil masses are trapped in five sections: compressor, condenser, liquid tube, evaporator and accumulator by closing the ball valves around each section simultaneously. The collected refrigerant charge distribution data during the shut-down and start-up transients are used for refrigerant migration validation in this paper. The reader is referred to Peuker and Hrnjak (2009) for more information about the refrigerant mass measurement approach, and the physical parameters. A detailed description of the experimental facility is given in Peuker (2006).

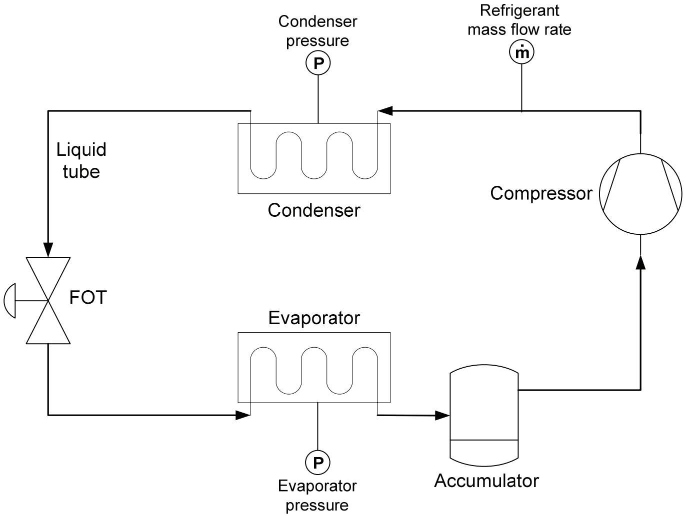


Figure 1: Schematic of the experimental A/C system

3. SYSTEM MODELING

This section is divided into two parts. Using the modeling framework (Li and Alleyne, 2010), a dynamic A/C system model is presented to capture the shut-down and start-up cycling behaviors. Secondly, the method to model the refrigerant mass migration is introduced. The reader is encouraged to examine Nomenclature for relevant variable definitions.

3.1 Component Modeling

The A/C automotive system shown in Figure 1 includes a compressor, a FOT, heat exchangers (condenser and evaporator), an accumulator and pipe models connecting each component. Using the switched moving-boundary modeling approach presented in Li and Alleyne (2010), the heat exchangers are developed with different model representations to accommodate the creation and destruction of dynamic states during the transients. Specifically, based on the experimental transient data analysis (Peuker and Hrnjak, 2009), the condenser is developed to consist of four different model representations (see Figure 2), and two different evaporator representations shown in Figure 3 are needed. The major heat exchanger modeling assumptions are given below.

- The heat exchanger is assumed to be a long horizontal tube;
- The refrigerant flow through the heat exchanger is a one-dimensional fluid flow;

- The refrigerant pressure is uniform within the heat exchanger;
- The header dynamics in the heat exchanger is ignored.

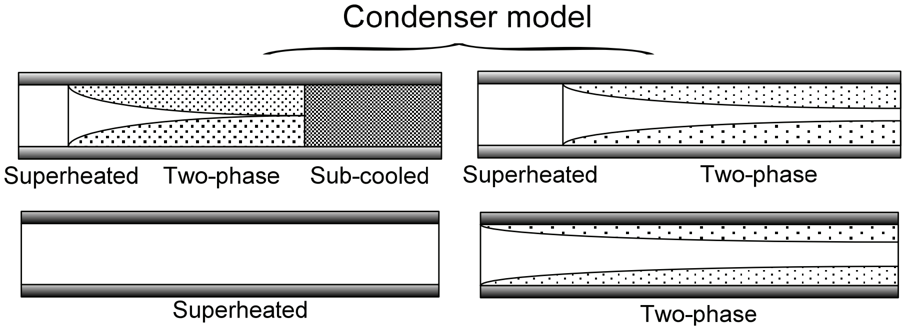


Figure 2: Switched moving-boundary condenser model structure

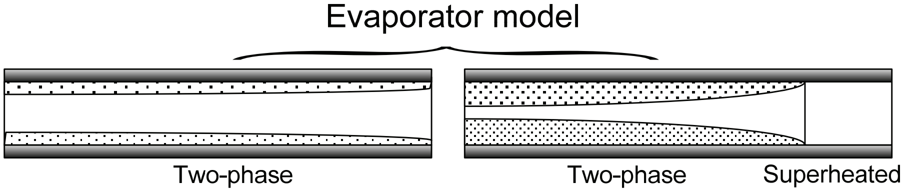


Figure 3: Switched moving-boundary condenser model structure

The dynamic state vectors in Equations (1)-(2) represent the condenser and evaporator conditions at each instant in time. The uniform state vector, independent of model representation, enables the heat exchanger models to retain a constant structure formulated in a nonlinear descriptor form (see Equation (3)). Another advantage with this uniform state vector is that it provides consistency in numerical simulation for different model representation switches. The refrigerant mass conservation is the major concern to choose the switching criteria among different model representations. The switching schemes along with pseudo-state variables discussed in Li and Alleyne (2010) are applied here to simulate the cycling transients.

$$x_c = \left[h_{c1} \quad P_c \quad h_{c3} \quad \zeta_{c1} \quad \zeta_{c2} \quad T_{c1w} \quad T_{c2w} \quad T_{c3w} \quad \bar{\gamma}_c \right] \tag{1}$$

$$x_e = \left[\zeta_{e1} \quad P_e \quad h_{e2} \quad T_{e1w} \quad T_{e2w} \quad \bar{\gamma}_e \right] \tag{2}$$

$$Z(x,u)\dot{x} = f(x,u) \tag{3}$$

The compressor and FOT are mass flow devices and developed with semi-empirical modeling approaches (Rasmussen, 2005). For the fixed orifice tube (FOT) expansion valve, the mass flow rate is calculated in Equation (4), where the flow coefficient *C* is assumed to be a function of pressure differential, P_c-P_e, and the refrigerant density ρ is a function of valve inlet refrigerant quality.

$$\dot{m}_{fot} = C\sqrt{\rho\left(P_c - P_e\right)} \tag{4}$$

The refrigerant-side governing equations in the accumulator model are derived by applying the mass and energy conservation principles, and given as follows.

$$\begin{bmatrix} \dfrac{\delta\rho_{ac}}{\delta P_{ac}} & \dfrac{\delta\rho_{ac}}{\delta\bar{\gamma}_{ac}} \\ \dfrac{\delta h_{ac}}{\delta P_{ac}} - \dfrac{1}{\rho_{ac}} & \dfrac{\delta h_{ac}}{\delta\bar{\gamma}_{ac}} \end{bmatrix} \begin{bmatrix} \dfrac{dP_{ac}}{dt} \\ \dfrac{d\bar{\gamma}_{ac}}{dt} \end{bmatrix} = \begin{bmatrix} \dfrac{\dot{m}_{i\_ac} - \dot{m}_{o\_ac}}{V_{ac}} \\ \dfrac{\dot{Q}_{ac} + \dot{m}_{i\_ac}(h_{i\_ac} - h_{ac}) - \dot{m}_{o\_ac}(h_g - h_{ac})}{\rho_{ac}V_{ac}} \end{bmatrix} \tag{5}$$

3.2 Refrigerant Migration Modeling

There are normally two ways to model the refrigerant mass migration during the shut-down and start-up cycling operations. With the known inlet and outlet refrigerant mass flow conditions around each section of the A/C system, the refrigerant charge distribution in each section is computed from Equation (6). Another way is based on the refrigerant information in each component during transients, such as mean void fraction $\bar{\gamma}$ and refrigerant density ρ. Examples of refrigerant mass calculation in the liquid tube and evaporator are given in Equations (7)-(8).

$$M = M_{initial} + \int \left(\dot{m}_i - \dot{m}_o \right) dt \tag{6}$$

$$M_{lt} = \rho_{lt} V_{lt} = f(P_c, h_{lt}) V_{lt} \tag{7}$$

$$M_{evap} = \zeta_{e1} V_{evap} \rho_{TP} + (1 - \zeta_{e1}) V_{evap} \rho_{SH} \tag{8}$$

4. MODEL VALIDATION

4.1 Simulation Environment

To validate the A/C automotive system model described above, the system is implemented in Thermosys (Rasmussen, 2005) to simulate the cycling dynamics with compressor shut-down and start-up operations. The inputs to each component model are generally the outputs of other component models. For instance, the refrigerant inlet and outlet mass flow rates are the switched evaporator model inputs, yet they themselves are the outputs of the FOT and compressor model, respectively. The reader is referred to Li and Alleyne (2010) for the refrigerant-side and air-side heat transfer correlations used in transients and more information about the solution procedures for the system model.

4.2 System Transients

The model validation scenario here includes shutdown-startup step changes in compressor speed (see Figure 4) while maintaining the air flow rates. The operating conditions for the validation are summarized in Table 1 along with the condenser and evaporator air inlet temperature conditions. The compressor shut-down time period is 3 minutes. The pressure and temperature measurements are taken every 1.5 seconds.

Table 1: System operating conditions for validation

Input	**Step Time for Shut-down**	**Before Shut-down**	**Step Time for Start-up**
Compressor speed	130s	900 rpm	310s
Cond. air mass flow rate	0.525 kg/s		
Evap. air mass flow rate	0.156 kg/s		
Cond. air inlet temperature	35°C		
Evap. air inlet temperature	35°C		

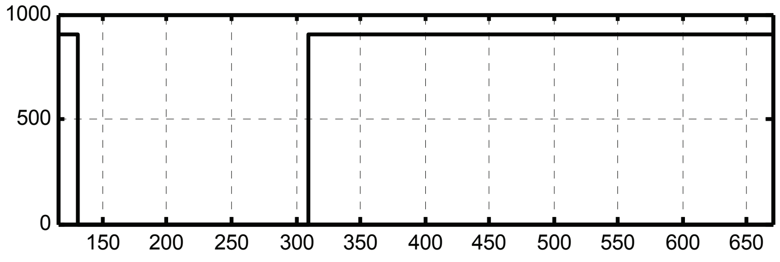
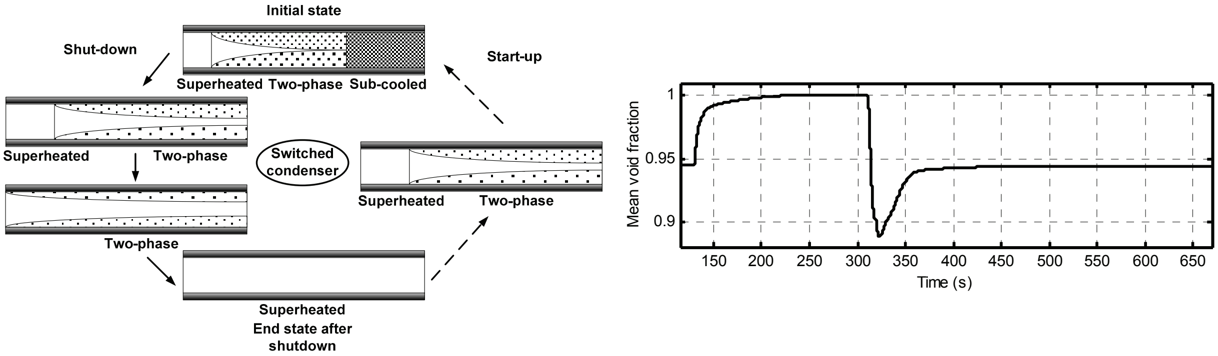


Figure 4: System input in terms of compressor speed for validation

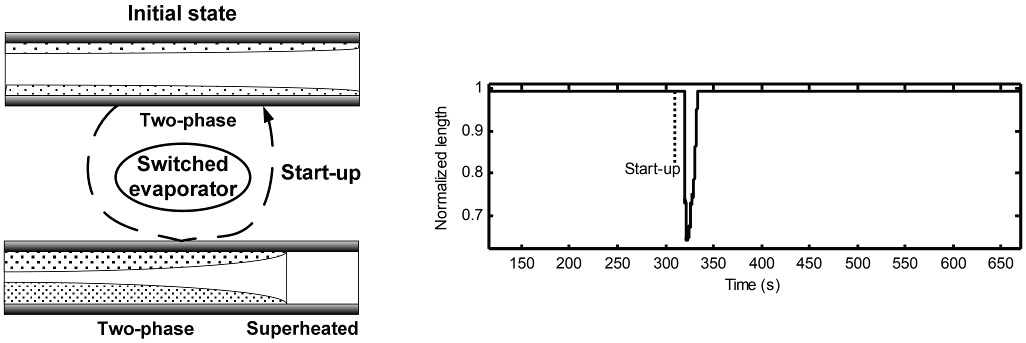
Figure 5(a) describes the dynamic switching of different model representations in the condenser model structure for this test scenario. The condenser model switches from the initial 3-zone (superheated, two-phase and sub-cooled) model to the 1-zone (superheated) model during shut-down transients, which can also be seen from the dynamic behaviors of mean void fraction $\bar{\gamma}_c$ in Figure 5(b). It takes around 1 minute for the condenser to be filled with superheated vapor (the mean void faction value increases above 0.999) after shut-down operations, which coincide with the experimental observation (Peuker and Hrnjak, 2009).



(a) (b)

Figure 5: (a) Switching routines in the condenser during transients; (b) Mean void fraction dynamics in the condenser during cycling operations

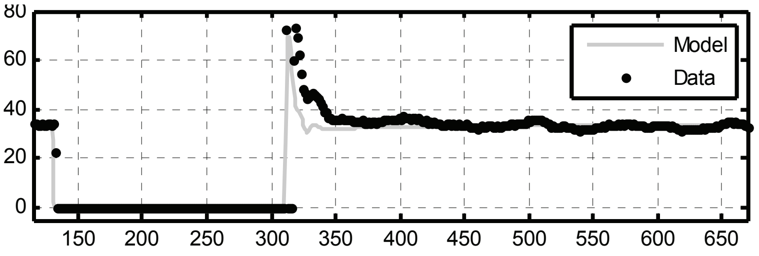
Interestingly, the evaporator model stays in the 1-zone (two-phase) model representation during the shut-down period, and a switch occurs from 1-zone (two-phase) to 2-zone (two-phase and superheated) after system starts up (see Figure 6(a)). The experimental studies given by Peuker and Hrnjak (2009) demonstrate the occurrence of superheated vapor at the evaporator outlet after start-up. The 2-zone (two-phase and superheated) evaporator model lasts around 25 seconds and then switches back to 1-zone (two-phase) model, as seen from Figure 6(b).



(a) (b)

Figure 6: (a) Switching schemes in the evaporator during start-up transients; (b) Normalized length of the two-phase zone in the evaporator during cycling operations

The plots in Figure 7 compare experimental data with various system model outputs, and the results demonstrate the capability of the system model to capture the shut-down and start-up dynamics.



(a)

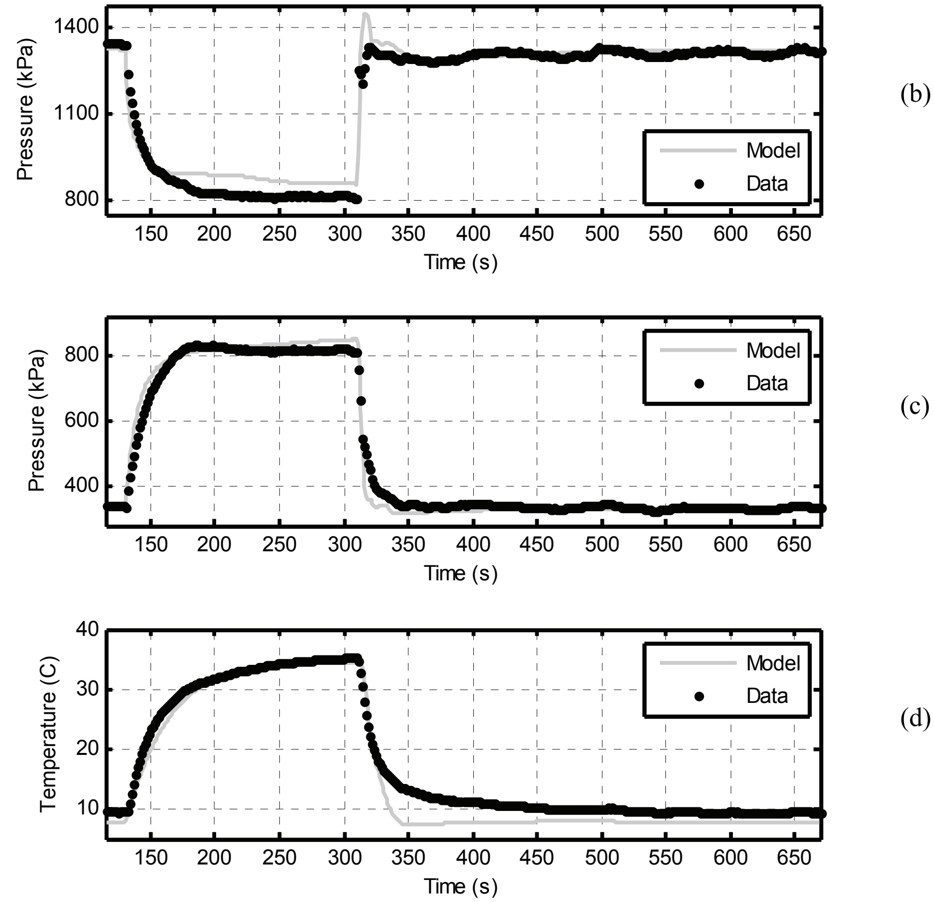
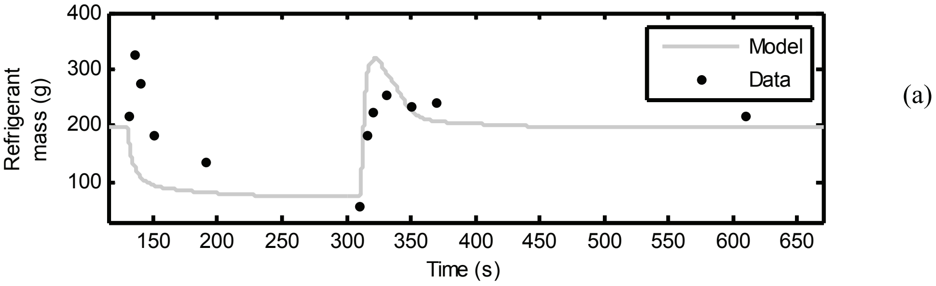


Figure 7: (a) Refrigerant mass flow rate; (b) Condenser pressure; (c) Evaporator pressure; (d) Evaporator air outlet temperature

4.3 Refrigerant Migration Transients

Refrigerant migration and redistribution are involved with the compressor shut-down and start-up cycling transients. The refrigerant mass measurement method (Peuker and Hrnjak, 2009) is used to determine the refrigerant migration for the A/C automotive system. In this validation scenario, the refrigerant charge distribution in each section of the system is measured at t = 130s, 135s, 140s, 150s, and 190s in shut-down transients, and at t = 310s, 315s, 320s, 330s, 350s, 370s, 490s and 610s after system starts up. The refrigerant migration modeling method in Equation (6) is implemented in this study to calculate the refrigerant mass in each component. During the shut-down period, the refrigerant mass migrates from the high-pressure components (condenser and liquid tube) to the low-pressure components (evaporator and accumulator) through the FOT expansion valve. Figure 8 presents the migration comparisons between experimental data and model in the high-pressure components. It can be seen that the majority of the migration occurs one minute after shut-down operations, and the model can well predict the steady-state and transients of the migration.



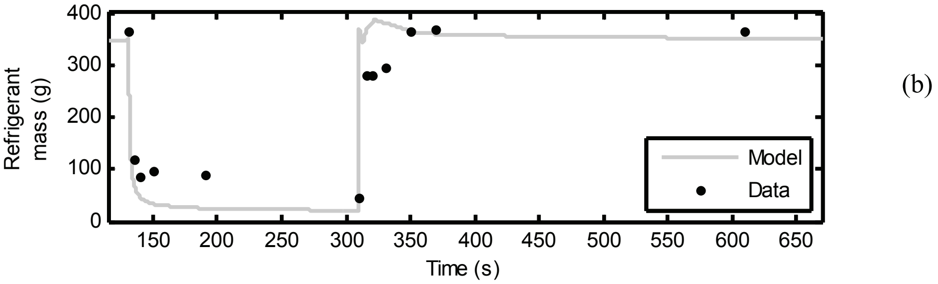


Figure 8: Refrigerant mass migration in the high-pressure components (a) condenser; (b) liquid tube

Prior to the start-up operations, over 80% of the total refrigerant mass of the system is located in the low-pressure components (evaporator and accumulator), as shown in Figure 9(a). During the start-up transients, the refrigerant charge in each component is redistributed, and the refrigerant mass migrates from the low-pressure components to the high-pressure components through the compressor. The plot in Figure 9(b) shows the migration performance in the evaporator, and the mass redistribution dynamics after the start-up is well described by the system model.

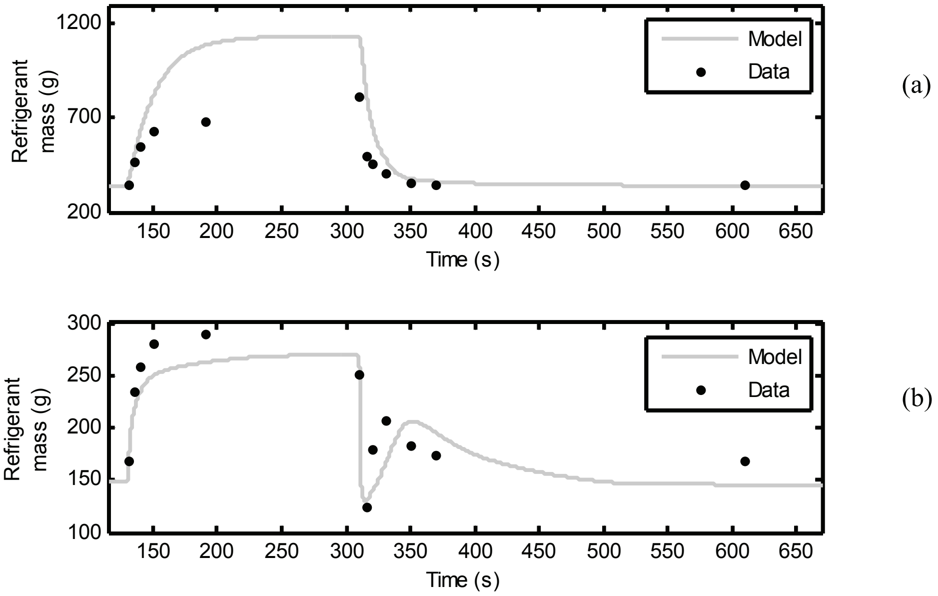


Figure 9: Refrigerant mass migration in the low pressure components (a) evaporator + accumulator; (b) evaporator

It is shown from Figures. 8-9 that the qualitative agreement between experimental data and model prediction is good. In the light of the gaps, further work is needed to quantify the model accuracy to capture the migration transients. As discussed in Peuker and Hrnjak (2009), the accumulator is the key component of the automotive system to store excess refrigerant, and a primary component to influence the refrigerant charge distribution during transients. Further work also involves the detailed investigation of mass migration in the accumulator model.

5. CONCLUSIONS

In this study, a dynamic R134a automotive system model is developed to capture refrigerant mass migration transients with compressor shut-down and start-up operations. Using the modeling framework (Li and Alleyne, 2010) and based on the experimental investigations (Peuker and Hrnjak, 2009), the moving-boundary heat exchangers are developed with switched model representations to accommodate the changing numbers of fluid zones due to the cycling transients. The refrigerant mass distribution in each section of the system is modeled during the shutdown-startup transients. One experimental model validation scenario is given. The results demonstrate that the developed model is able to describe the transient behaviors of the system, and also qualitatively predict the refrigerant mass migration performance during the shut-down and refrigerant redistribution after the start-up.

Future work will focus on the improvement of model accuracy to predict the refrigerant migration transients in the system. Given the relationship between refrigerant mass distribution and system start-up performance (Peuker and Hrnjak, 2009), the potential of utilizing compressor cycling control strategy to improve system performance will be another interesting topic.

NOMENCLATURE

Variables

C	flow coefficient	(dimensionless)
f	forcing function	
u	input	
V	volume	(m^3)
x	state vector	
Z	coefficient matrix	
h	refrigerant enthalpy	($kJkg^{-1}$)
h	refrigerant enthalpy	($kJkg^{-1}$)
M	refrigerant mass	(kg)
$\dot{m}$	mass flow rate	(kgs^{-1})
$\dot{m}$	mass flow rate	(kgs^{-1})
P	refrigerant pressure	(kPa)
$\dot{Q}$	heat transfer rate	(kW)
T	temperature	(°C)
$\bar{\gamma}$	mean void fraction	(dimensionless)
ζ	normalized zone length	(dimensionless)

Subscripts

$c1$	superheat zone in condenser
g	saturated vapor
lt	liquid tube
i	inlet
o	outlet
c	condenser
ac	accumulator
w	heat exchanger structure
e	evaporator
$e1$	two-phase in evaporator

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