

Evaluation of Transient Refrigerant Migration Modeling Approach on Automotive Air Conditioning Systems

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ABSTRACT

Automotive air conditioning systems are subject to constantly changing operation conditions and steady state simulations are not sufficient to describe the actual performance. The refrigerant mass migration during transient events such as clutch-cycling or start-up has a direct impact on the transient performance. It is therefore necessary to develop simulation tools which can accurately predict the migration of the refrigerant mass. To this end a dynamic model of an automotive air conditioning system is presented in this paper using a switched modeling framework. Model validation against experimental results demonstrates that the developed modeling approach is able to describe the transient behaviors of the system, and also predict the refrigerant mass migration among system components during compressor shut-down and start-up (stop-start) cycling operations. To further investigate the potential of the dynamic modeling tools, two simulation examples of evaluating the system performance are given in the paper: (i) impact of system component variations on the refrigerant mass migration; and (ii) control implementation for the system start-up performance improvement.

INTRODUCTION

Automotive air conditioning systems in actual vehicles are subject to constantly changing operation conditions, many start-ups over their lifetime and, for clutch cycling systems,

repetitive stop-start operations of the compressor. To evaluate the performance of air conditioning systems under these transient conditions requires expensive experimental measurements. Computer simulations can reduce the time and expenses required for experimentation but need to represent the dynamics of the real system.

The correct prediction of the refrigerant mass migration during the dynamic operations is paramount, since refrigerant mass migration affects the transient performance of air conditioning systems [1]. The redistribution of refrigerant mass either at start-up or during clutch cycling results in energy losses. Both, Wand and Wu [2] and Peuker and Hrnjak [3], provide experimental data demonstrating that compressor energy, and hence energy losses, can be reduced by preventing refrigerant mass migration during the compressor shut-down period. Both investigations show the importance of predicting the movement of refrigerant mass accurately. Therefore, dynamic simulations matching only pressure and temperature developments are not sufficient to faithfully represent real systems.

Peuker and Hrnjak [4] present experimentally measured data of pressure, temperature and the migration of refrigerant mass for an automotive fixed orifice tube system during the compressor stop-start operation. Their data provides a baseline to validate the capability of dynamic simulations to predict the refrigerant mass migration. A dynamic model of

the automotive air conditioning system used by Peuker and Hrnjak [4] has been developed using the modeling framework of Li and Alleyne [5].

In this paper, the dynamic model of the automotive air conditioning system is described and model validation against experimental data from [4] demonstrates the validity of the developed model in capturing the refrigerant mass migration during the stop-start operation. Simulation examples are presented to show the potential benefits of the dynamic modeling tools in representing the real vehicle system dynamics. Finally, a conclusion section summarizes the main points of the paper.

EXPERIMENTAL SYSTEM

The experimental system used for transient modeling and validation is an R134a automotive air conditioning system consisting of the following components: compressor, condenser, fixed orifice tube, evaporator, and accumulator. The components are installed into the experimental facility [4] with the same difference in vertical height as in the vehicle. The schematic of the experimental system is shown in Figure 1. As seen in Figure 1 ball valves are installed on either side of each component and, by closing these ball valves simultaneously, the refrigerant mass can be trapped in each section. The published experimental data in [4] are used for the validation of the dynamic model presented in this paper. The reader is referred to Appendix for brief descriptions of the experimental system components, and detailed specifications are provided in [6]. The accuracy and repeatability of the refrigerant mass measurement are verified by Peuker [6], and more information about the experimental system set-up, the refrigerant mass measurement method, experimental results and analysis can be found in [4][6].

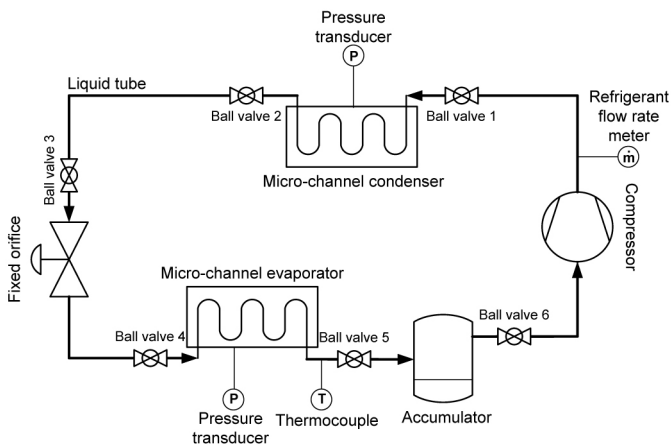


Figure 1. Schematic of the experimental system.

SYSTEM MODELING

This section is divided into three parts. Using the switched modeling framework from [5], the component models are introduced first for simulating the stop-start transients of the automotive air conditioning system. Second, methods to calculate the refrigerant mass in each component are presented, and the third part illustrates the simulation environment.

COMPONENT MODELING

The automotive air conditioning system as shown in Figure 1 is subdivided into four components, each of which is discussed below.

Heat exchangers

Most of the attention in the system modeling is paid to the heat exchangers, both condenser and evaporator. Since the compressor stop-start operations produce very large transients, a switched moving-boundary modeling framework is developed and presented in [5] to describe the refrigerant transient responses. In particular, the heat exchangers are developed to accommodate different model representations. The dynamic state vectors in Equations (1) and (2) represent the condenser and evaporator conditions at each instant in time. The pseudo-state technique [7] is applied to maintain the uniform state vector, independent of model representations. The state derivative equation is formulated in a nonlinear descriptor form given in Equation (3), and each model representation has its own coefficient matrix $Z(\bar{x}, \bar{u})$ and forcing function $f(\bar{x}, \bar{u})$ for storing thermodynamic variables, and mass and energy balance terms. An advantage of this approach is the tracking of the vapor and liquid refrigerant dynamic states in numerical simulations while ensuring refrigerant mass conservation during switches in system model representations.

$$\bar{x}_c = \begin{bmatrix} h_{c1} & P_c & h_{c3} & \zeta_{c1} & \zeta_{c2} & T_{c1,w} & T_{c2,w} & T_{c3,w} & \bar{\gamma}_c \end{bmatrix}^T \quad (1)$$

$$\bar{x}_e = \begin{bmatrix} \zeta_{e1} & P_e & h_{e2} & T_{e1,w} & T_{e2,w} & \bar{\gamma}_e \end{bmatrix}^T \quad (2)$$

$$Z(\bar{x}, \bar{u}) \cdot \dot{\bar{x}} = f(\bar{x}, \bar{u}) \quad (3)$$

Based on the switched modeling method and experimental study [4], the condenser is developed to consist of four different model representations in Figure 2, and two different model representations, as shown in Figure 3, are used for the evaporator. The dynamic states in Equations (1) and (2) are applied to describe the heat exchanger transients, where the mean void fraction $\bar{\gamma}_c$ and $\bar{\gamma}_e$ are used for switching purposes

[7]. The reader is referred to [5][7] for detailed descriptions about the derivations of air-side and refrigerant-side governing equations in the different model representations for the heat exchangers, as well as switching schemes to simulate the stop-start dynamics.

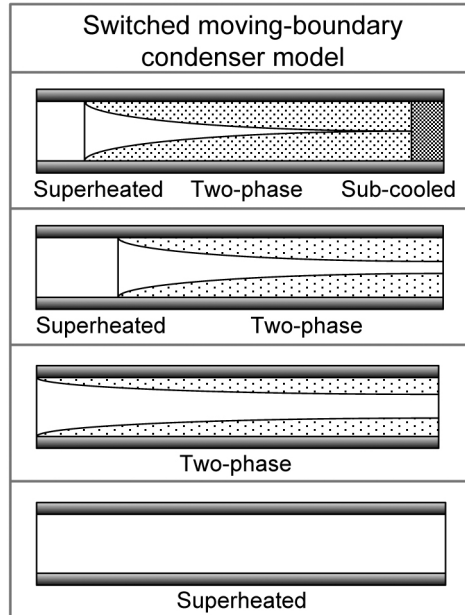


Figure 2. Condenser model structure.

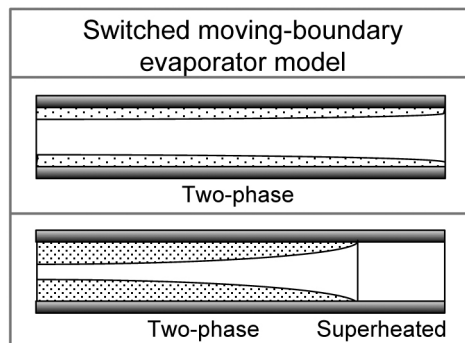


Figure 3. Evaporator model structure.

Accumulator

The accumulator is a key component in the automotive air conditioning system. Its primary function is to store excess refrigerant mass to ensure system capacity over a large range of operating conditions. Meanwhile, it captures two-phase refrigerant thus preventing damage to the compressor. Two dynamic states, refrigerant pressure P_{ac} , and refrigerant enthalpy h_{ac} , are defined to describe the refrigerant dynamics inside the accumulator. By applying mass and energy conservation principles, the governing equations are given below.

$$\frac{\delta \rho_{ac}}{\delta P_{ac}} \frac{dP_{ac}}{dt} + \frac{\delta \rho_{ac}}{\delta h_{ac}} \frac{dh_{ac}}{dt} = \frac{\dot{m}_{i,ac} - \dot{m}_{o,ac}}{V_{ac}} \quad (4)$$

$$-\frac{1}{\rho_{ac}} \frac{dP_{ac}}{dt} + \frac{dh_{ac}}{dt} = \frac{(UA)_{ac}(T_{amb} - T_{ac}) + \dot{m}_{i,ac}(h_{i,ac} - h_{ac}) + \dot{m}_{o,ac}(h_{i,ac} - h_{ac})}{\rho_{ac} V_{ac}} \quad (5)$$

where the term $(UA)_{ac}(T_{amb} - T_{ac})$ represents the heat transfer rate from the ambient air to the refrigerant inside the accumulator. The heat transfer is considered since it influences the refrigerant mass migration of the accumulator.

Fixed orifice tube and compressor

Since the dynamics of the mass flow devices (fixed orifice tube and compressor) are generally an order of magnitude faster than those of the heat exchangers [8], these two components are considered to be static models. The mass flow rate across the fixed orifice tube influences the refrigerant pressure in both heat exchangers as well as the refrigerant mass distribution during the stop-start transients. Therefore, an accurate valve model is necessary. Equation (6) is used to calculate the orifice mass flow rate, where the flow coefficient C_f is determined via a semi-empirical mapping approach [8], and the refrigerant density ρ_v is assumed to be a function of the refrigerant inlet conditions (i.e., liquid-vapor mixture, or vapor) to the orifice tube, as given in (7).

$$\dot{m}_v = C_f \sqrt{\rho_v (P_l - P_e)} \quad (6)$$

$$\rho_v = \rho(P_l, h_{i,v}) \quad (7)$$

The compressor model illustrated in [5] has been demonstrated to predict the refrigerant mass flow after system start-up with sufficient accuracy to be useful for simulation purposes. The compressor mass flow rate is computed by:

$$\dot{m}_k = V_k \omega_k \rho_k \eta_{vol} \quad (8)$$

where V_k is the cylinder volume, ω_k is the compressor speed, and ρ_k is the refrigerant inlet density. The volumetric efficiency η_{vol} is found using a performance mapping approach similar to the one given in [8].

REFRIGERANT MASS CALCULATION

Approaches are explored here to evaluate the refrigerant mass distribution behaviors in the stop-start operation. Poggi et al. [9] presents a summary of available refrigerant mass calculation methods in the literature. The authors in [9] also

point out that the major challenge is the mass evaluation in two-phase components, such as heat exchangers. For general numerical simulations, the refrigerant mass migration in each system component can be obtained by Equation (9) with known refrigerant inlet and outlet mass flow rate conditions. An alternative way to evaluate the refrigerant mass distribution in heat exchangers during transients is using the mean void fraction $\bar{\gamma}_c$ and $\bar{\gamma}_e$, which is an integral form of a local void fraction. The Zivi local void fraction correlation [10] is applied in this study. The two-phase and single-phase (vapor or liquid) refrigerant mass calculations in the heat exchangers are given in Equations (10) and (11).

$$M_{\text{component}} = M_{\text{steady-state}} + \int (\dot{m}_i - \dot{m}_o) dt \quad (9)$$

$$M_{\text{two-phase, exchangers}} = (\bar{\gamma} \rho_g + (1 - \bar{\gamma}) \rho_f) \zeta_{\text{two-phase}} V_{\text{exchangers}} \quad (10)$$

$$M_{\text{single-phase, exchangers}} = \rho_{\text{single-phase}} \zeta_{\text{single-phase}} V_{\text{exchangers}} \quad (11)$$

The total refrigerant mass in the automotive air conditioning system (see Figure 1) is described below.

$$M_{\text{total}} = M_{\text{exchangers}} + M_{\text{accumulator}} + M_{\text{liquid_tube}} + M_{\text{compressor}} + M_{\text{pipes}} \quad (12)$$

where the calculation of the single-phase refrigerant mass in components, such as superheated vapor in pipes, is based on the refrigerant density and component volume information.

SIMULATION ENVIRONMENT

To validate the refrigerant mass migration during the compressor stop-start operations, the component models described above, along with pipe models connecting each component, are implemented in Thermosys [8], a Matlab/Simulink toolbox for simulation. The pipe models calculate pressure drops between any two components. The inputs to each model are generally the outputs of other component models. For instance, the refrigerant inlet and outlet mass flow rates are the evaporator model inputs, yet they themselves are the outputs of the fixed orifice tube and compressor model, respectively. Numerical solution procedures on the switched heat exchanger models are given in [7]. Detailed information about the air-side and refrigerant-side heat transfer calculations in transients can be found in [5].

MODEL VALIDATION

As mentioned earlier, the refrigerant mass migration experimental data presented in [4] are used for transient validation in the paper. The model validation study, consists

of three stop-start step changes in compressor speed (see Figure 4) while maintaining the air flow rates across the heat exchangers. The steps in the system inputs for the validation are summarized in Table 1 along with the condenser and evaporator air inlet temperature conditions. The first compressor stop-start operations are identical to the experimental scenario studied in [4]. The three compressor shut-down time periods are 3 minutes, 2 minutes and 1 minute respectively.

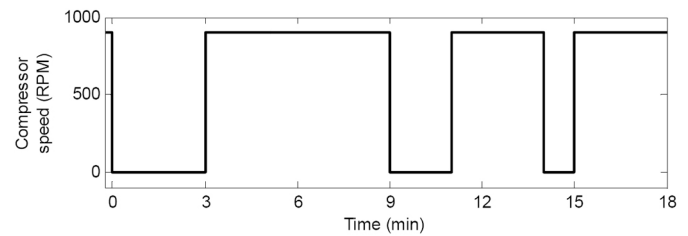


Figure 4. Compressor speed input for validation.

Table 1. System operating conditions for validation.

Input	Step time for shut-down	Before shut-down	Step time for start-up
Compressor speed	130 s / 670 s / 970 s	900 rpm	310 s / 790 s / 1030 s
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		

The plots in Figures 5, 6, 7, 8, 9 compare the experimental results with the system model outputs and demonstrate good agreement. During the compressor shut-down period, the refrigerant migrates from the high-pressure to the low-pressure components through the fixed orifice tube, which results in the heat exchanger pressure changes as seen in Figures 5 and 6. The evaporator air outlet temperature, as shown in Figure 7, increases towards the ambient temperature (35°C) as the thermal mass of the heat exchanger and refrigerant mass inside approach an equilibrium condition. In addition, the system model predicts the transients of the refrigerant mass flow rate across the compressor at start-up shown in Figure 9.