

Modeling of a Reversible Air Conditioning-Heat Pump System for Electric Vehicles

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ABSTRACT

This paper presents a simulation model for a reversible air conditioning and heat pump system for electric vehicles. The system contains a variable speed compressor, three microchannel heat exchangers, an accumulator, and two electronic expansion valves. Heat exchangers are solved by discretizing into cells. Compressor and accumulator models are developed by fitting data with physical insights. Expansion valves are modeled by isenthalpic processes. System performance is calculated by connecting all parts in the same way as the physical system and solved iteratively. The model is reasonably validated against experimental data from a separate experimental study. Future improvement is needed to take into account maldistribution in outdoor heat exchanger working as an evaporator in HP mode. Charge retention in components also requires further study.

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INTRODUCTION

Electric vehicles (EV) have been growing in popularity in past few years. EV's do not generate as much waste heat as conventional internal combustion engine (ICE) cars. As a result, heating requirement for cabin air needs to be fulfilled by other heat sources. Compared to positive temperature coefficient (PTC) heaters, heat pump has higher second law efficiency, and can provide much more heating capacity for the cabin by consuming the same amount of electric power. Furthermore, heat pumping function can be achieved by adding only a few extra components to the mobile air conditioning system, which will generally be installed on an EV anyway. Currently, a few EV models are using heat pump for cabin heating, including Nissan Leaf, Renault Zoe, Kia Soul, and BMW i3. In a separate experimental study, a benchtop heat pump system was built in the lab by using the same heat exchangers and system configuration as the Nissan Leaf heat pump system. The system configuration and two working modes were shown in Figure 1, and component photos shown in Figure 2. The system performance characteristics were studied in a variety of operating conditions, especially for heating function. However, in real vehicle application, the heat pump system operates in a large parameter space, and detailed experimentation to cover the entire parameter space is costly and time-consuming. A simulation tool based on existing experimental data would largely help understanding of system physics, performance in different operating conditions, and modification of system design.

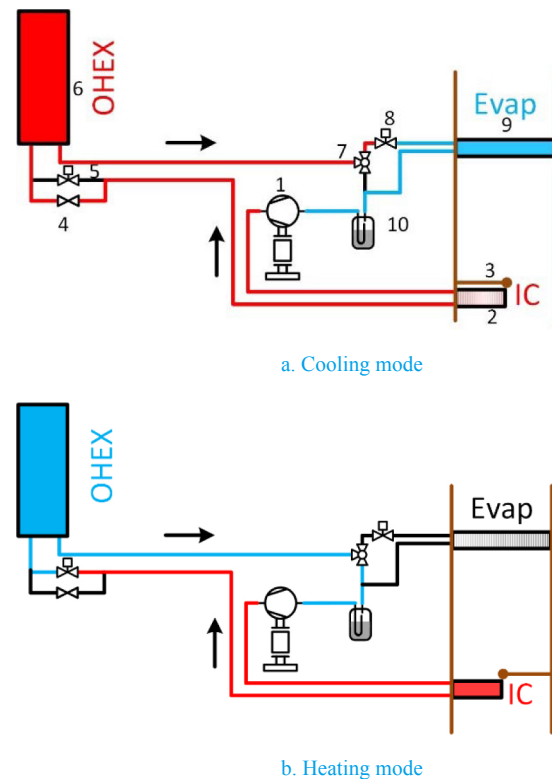


Figure 1. Reversible air conditioning/heat pump system architecture and two basic working modes 1 - compressor, 2 - inner condenser, 3 - air door, 4 - bypass valve, 5 - outdoor heat exchanger, 6 - EEV for heating, 7 - 3-way valve, 8 - EEV for cooling, 9 - evaporator, 10 - accumulator (low side receiver)

In this study, a steady state simulation model was built to predict system performance for different operating conditions. The programming language used is Engineering Equation Solver (EES), which has built-in functions for calculation of thermodynamic and transport properties of different operating fluids. The refrigerant used is R134a. The system model contains a compressor model, an accumulator model, and microchannel heat exchanger models, while the expansion valve was modeled by a simple isenthalpic process, and subcooling at condenser exit was arbitrarily set as an input. Compressor model was based on curve fitting of volumetric and isentropic efficiencies obtained from experimental data. Accumulator model was constructed to predict compressor suction vapor quality when there is liquid accumulating in the accumulator. In the current state of the study, accumulator model was disabled due to lack of good charge retention prediction in components, instead, the compressor suction quality was arbitrarily set to 0.98 to represent a realistic situation. Microchannel heat exchanger models were built based on finite volume method. The heat exchangers were divided into small segments and calculated consecutively using finite volume method. In the system model, all the components were connected by using pressure and enthalpy as connecting state property parameters. Exit property of the upstream component became input property of the downstream component. With appropriate iteration algorithm based on Newton-Raphson method and stopping criteria, the system was solved iteratively until all connecting states won't need further modification.

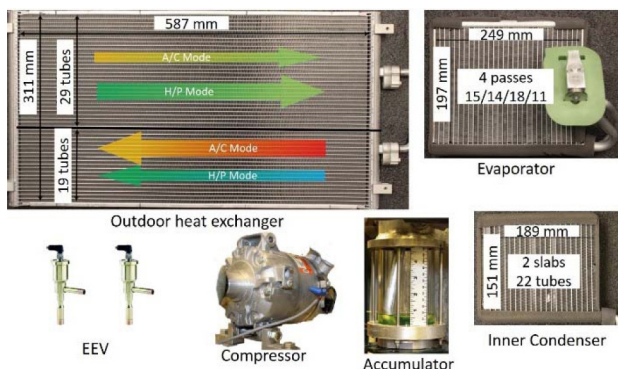


Figure 2. Major components used in the reversible system

MODEL DEVELOPMENT

Component Models

Compressor Model

In the literature, the term “compressor model” comes in a wide range of different types of models, from purely physical models with details of compressor structure to purely statistical models obtained solely from experimental data and contained no physical insights [1]. Physical model can capture compressor characteristics better, but will require a lot of details of the specific compressor being studied, and cannot be generalized easily, hence not suitable for refrigeration system simulation. While purely statistic model will lose physical insights which are valuable for understanding how the system works. In this study, the compressor model was built with statistical curve

fitting based on some level of physical understandings. The following equation was used to correlate compressor volumetric efficiency and compression pressure ratio between discharge and suction. The clearance volume ratio and compression polytropic exponent were found by curve fitting. With given suction state and obtained volumetric efficiency, refrigerant mass flow rate throughout the refrigerant circuit can be calculated.

$$\eta_{volm} = 1 - \epsilon_{cv} \left[\left(\frac{P_{dis}}{P_{suc}} \right)^{1/\eta_{poly}} - 1 \right] \quad (1)$$

Another property that needs to be determined by the compressor model is the compressor discharge enthalpy, which can be calculated with a proper isentropic efficiency equation. Compressor manufacturers typically provide isentropic efficiency in a curve fitting form. For this study, the manufacturer equation was not available, while a large database was generated in the related experimental study. Therefore, the isentropic efficiency curve fitting was obtained from that database. It was found that a linear curve fitting between compression pressure ratio and the ratio of isentropic efficiency to volumetric efficiency gave the best result.

Based on these two efficiency curve fittings, refrigerant mass flow rate and compressor discharge state can be determined by using suction state and discharge pressure as input variables. When implementing system model, discharge pressure needs to be given by condenser model, which has an arbitrarily set exit subcooling level.

Accumulator Model

The original Nissan Leaf system used an accumulator at compressor suction to store extra refrigerant charge, especially for heating mode, as system demand for charge was found much less than that for cooling mode. From the experimental study, in heating mode, the bottom of the J-tube inside the accumulator was almost always soaked inside liquid, not only returning oil to compressor suction, but also brings some liquid into the compressor. The experimental compressor inlet quality was derived with calibrated compressor shell heat loss. For modeling of the accumulator, the liquid flow of refrigerant and oil mixture was driven by the pressure difference across the pinhole at the bottom of the J-tube. This pressure difference has three components: dynamic pressure loss, frictional pressure drop, and hydraulic pressure difference, as shown in Equation (2).

$$\Delta P = \Delta P_{dyn} + \Delta P_f + \Delta P_{hydr} \quad (2)$$

The dynamic term calculated in Equation (3) is the difference between static pressure and dynamic pressure. As liquid fraction is typically very low, here the refrigerant mass flux through the J-tube was calculated with total refrigerant mass flow rate.

$$\Delta P_{dyn} = \frac{G_{ref,Jtube}^2}{2\rho_{ref,g}} \quad (3)$$

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The friction term was obtained by using Churchill's single phase friction factor correlation [2], in Equations (4) and (5). An improved Churchill correlation is available in the literature [3], and will be implemented in future study to improve evaporating pressure drop prediction.

$$\Delta P_f = f \frac{L_{tube}}{D_{i,tube}} \frac{G_{ref,tube}^2}{\rho_{ref,g}} \quad (4)$$

$$A = \left[2.457 \ln \left(\frac{1}{(7/Re_D)^{0.9} + 0.27(\varepsilon/D)} \right) \right]^{16}$$

$$B = \left(\frac{37530}{Re_D} \right)^{16}$$

$$f = 8 \left[\left(\frac{8}{Re_D} \right)^{12} + (A+B)^{-1.5} \right]^{\frac{1}{12}} \quad (5)$$

The hydraulic pressure difference is created by the liquid level above the pinhole inside the accumulator, and was calculated with Equation (6).

$$\Delta P_{hydr} = \rho_{ref,f} g h_{ll} \quad (6)$$

With total pressure difference determined, the liquid refrigerant mass flow rate through the pinhole was then calculated by balance between viscous shear stress and pressure difference in Equation (7), assuming a steady parabolic velocity profile. Note that when there is no liquid refrigerant accumulating in the accumulator, or the liquid level is not high enough to reach the pinhole at the bottom of the J-tube, then there will be no liquid entering compressor suction in principle.

$$\dot{m}_{ref,f} = \begin{cases} 0, & h_{ll} \leq 0 \\ \Delta P \frac{\rho_{ref,f}}{\mu_{ref,f}} \frac{\pi D_{pinhole}^4}{128 \delta_{pinhole}}, & h_{ll} > 0 \end{cases} \quad (7)$$

Compressor suction quality can now be calculated with Equation (8).

$$x_{cpr} = 1 - \frac{\dot{m}_{ref,f}}{\dot{m}_{ref,tot}} \quad (8)$$

Microchannel Heat Exchanger Model

Finite volume method is a prevalent way of modeling microchannel heat exchangers. The heat exchanger is divided into small segments along the refrigerant flow circuit. Starting from inlet of the heat exchanger, local heat transfer coefficient and friction factor at refrigerant side are calculated with inlet refrigerant state for each segment in order to calculate local heat transfer and pressure drop. The exit state of this segment is then determined by the amount of heat being transferred and the pressure loss. This exit state became the inlet state of the downstream segment. The computer program calculates consecutively for each segment until exit state of the heat exchanger is determined. Total heat transfer capacity is simply the

summation of heat transfer amount in all the segments. The above procedure works fine for single slab heat exchangers by using air inlet state for local heat transfer calculation. However, for multi-slab heat exchangers, air inlet state for air side downstream slab(s) may depend on refrigerant side downstream pass(es). For that situation, iteration is needed. Iteration starts with air inlet state for all elements set to heat exchanger air inlet state. After each calculation from refrigerant side inlet to outlet, air inlet states of elements at air side downstream slabs(s) are updated with air outlet states of corresponding upstream elements. Finally, iteration terminates when refrigerant exit state stopped changing.

In this study, finite volume method was used for all three heat exchangers. The following assumptions were made:

1. Steady state flow and heat transfer;
2. Thermodynamic equilibrium, or otherwise noted;
3. Uniform distribution among parallel tubes;
4. Heat transfer and pressure drop in headers neglected;
5. Thermal resistance by conduction through wall neglected;
6. Lubricant effect not included in current study;
7. Dehumidification not studied in current stage of study.

Outside geometry of all three heat exchangers is known.

Microchannel tube interior geometry of outdoor heat exchanger is indirectly measured by analysis of cross-section. Microchannel tube interior geometries of evaporator and inner condenser were derived by exterior measurements with reasonable guess values of tube wall thickness, port barrier thickness, and number of ports. These missing measurements will be updated in the future when heat exchangers can be cut open for measurements. Each heat exchanger is divided into numbers of elements with equal length. The number of elements is determined by sensitivity analysis. By doubling this number, change in heat exchanger capacity should be less than 0.5%. Heat transfer in each element is solved by using ε -NTU method with Equations (9), (10), (11), (12). Equation (11) works for two phase region, and Equation (12) for single phase region. C_{min} is the minimum of refrigerant side and air side heat capacity flow, and C_r is the ratio of C_{max} to C_{min} .

$$UA = \left(\frac{1}{htc_{air} A_{air} \eta_{tot}} + \frac{1}{htc_{ref} A_{ref}} \right)^{-1} \quad (9)$$

$$NTU = \frac{UA}{C_{min}} \quad (10)$$

$$\varepsilon = 1 - \exp(-NTU) \quad (11)$$

$$\varepsilon = 1 - \exp \left(\left(\frac{1}{C_r} \right) NTU^{0.22} \left(\exp(-C_r NTU^{0.78}) - 1 \right) \right) \quad (12)$$