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Modeling Improvements and Analysis of Refrigerator System Approach Based on Experimental Performance

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ABSTRACT

This paper examines a set of simple equations describing a domestic refrigerator/freezer system and suggests several modeling improvements to enhance temperature analysis within the refrigerator system, based on experimental results. The experimental setup is described and limitations in the accuracy of the analysis are examined. Data are compared to predictions from a first generation model. Changes are made in the model to improve representations of heat exchanger geometry and flow regimes, and air side energy equations. The experimental data are re-examined in order to quantify the accuracy gained as model complexity was increased. For both models, parameters are estimated from the data Simscape environment of Matlab simulink and data are generated for obtaining the required results.

Keyword: Refrigerator, Heat System, Freezer Cabinets Compartment

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OVERVIEW

INTRODUCTION

Energy and environmental concerns, as reflected through government regulations, have prompted manufacturers to design more energy efficient refrigerator systems. The design process can be greatly enhanced by using accurate design and simulation models. Unlike design models, simulation models can be used to predict the performance of the system during off design conditions. This report describes a refrigerator system model which can run in either design or simulation mode. The results of the model are compared with data taken from a refrigerator system operating under a very wide range of conditions. In general there may be large uncertainties associated with the input parameters for the models. Consequently it may not be possible to discern the improvements in system performance associated with design changes from the errors in output variables that result from parametric uncertainty. Therefore, the model is also subjected to a detailed temperature analysis study to determine which parameters have the greatest effect on certain key output variables. The temperature analysis is conducted at both a local and global level. The error propagation characteristics of the model are examined to determine if the model can be used to evaluate design tradeoffs. Finally, possible design improvements for refrigerator are examined.

The refrigerator temperature analyzer to be modeled is used to measure the pressure, (draft) pressure measurement as well as differential temperature measurement within the refrigerator compartment.

LITERATURE REVIEW

Literature review is to identify works which are useful for understanding refrigeration system modeling and temperature analysis techniques. Refrigeration System Modeling Currently there is two steady state models which represent the state-of-art in the public domain. Most other models are simpler, offering fewer system configurations and modeling options, especially those used for transient system modeling. These steady state models, along with expansion devices and charge calculations, are discussed by Reeves et al., (1992). The Oak Ridge National Laboratory Modulating Heat Pump Design Tool is a steady-state heat pump computer model developed for the U.S. Department of Energy (Rice, 1991). The heat exchangers are modeled with multiple zones with variable pressure drop and heat transfer coefficients calculated from correlations. The model simulated both variable speed compressors and fans. The model allows the user to either specify or determine the charge in the simulation; numerous charge calculation methods are available. The charge option allows the model to run in either design or simulation mode. A refrigeration system model is currently being developed for the Environmental Protection Agency Refrigerator Analysis (ERA) program. It is an enhanced version of the first public domain refrigeration analysis program which was written by Arthur D. Little, Inc. for the U.S. Department of Energy (Arthur, 1982). Both versions are design models; they neither calculate system charge nor utilize a capillary tube mass flow rat~ equation. The enhanced version models multi zone heat exchangers with either user-specified or calculated overall heat transfer coefficients (Merriam et al., 1992). It also contains a detailed model of cabinet loads.

An integral part of any refrigeration system simulation model is the equation characterizing the expansion device. (Sami and Duong, 1988) present an analytical method for relating pressure drop to mass flow rate through a capillary tube, using mass, momentum and energy balances and an equation of state. The model accepts either sub cooled, saturated or two phase refrigerant conditions at the inlet of the capillary tube. Unlike many other capillary tube models, the model is not limited to adiabatic situations. The authors present graphs showing good agreement between experimental data and the results of the model. A more detailed review of capillary tube/suction line heat exchangers was conducted by (Purvis, 1992). Another important part of a simulation model is the charge calculation.

Damasceno et al., (1991) discuss various means of calculating total system charge for a heat pump. Four void fraction models were examined: Hughmark (2012). The authors stressed the importance of including all parasitic volumes, return bends and internal volumes when predicting total system charge. Using precise internal volumes and modified void fraction models, the authors were able to achieve good agreement between measured and predicted dependence of heat pump capacity on total system charge.

Heat Exchanger Modeling

A basic review of the heat transfer relations necessary for heat exchanger modeling is obtained from a text on refrigeration and air conditioning (Stoecker, 1982). Stoecker's text reviews the basic relations for conductance (VA), overall heat transfer coefficient (V), and ways of evaluating these quantities using resistance networks and convective heat transfer coefficients. Several analytical solutions for calculating heat transfer coefficients are presented in a report that summarizes the Oak Ridge National Lab (ORNL) heat pump model (Rice, 1983). Relations are included for both air side and refrigerant side heat transfer coefficients. The relations are closed form solutions of integrals presented in the write-up. This model also treats mass transfer onto a coil as a function of humidity ratio. A review of a text specifically written for heat exchanger design (Kays& London, 1984) provides a wealth of information for heat exchanger modeling using the E-NTU method. Analytical solutions for heat exchanger effectiveness are provided for many different geometries and configurations. Also, methods are reviewed for calculating these parameters analytically from basic relations for geometries that are not included in the text.

Heat transfer augmentation/degradation due to oil concentration is examined by Eckels and Pate for in-tube evaporation and condensation of refrigerant-lubricant mixtures (Eckels, 1991). Experimental data is presented for different oil concentrations (0, 1.2,2.5, and 5.4%) in both CFC-12 (naphthenic oil) and HFC-134a (PAG oil). Overall pressure drop and heat transfer coefficient data are presented graphically as a function of oil concentration for both refrigerants. A much more detailed set of data showing how heat transfer pressure drops vary with quality, mass flux, heat flux, and other parameters are presented by Wattelet, Chato, Jabardo, Panek, and Renie (Wattelet, 1991). The two-phase heat transfer tests are conducted for evaporation, with and without various oils. The refrigerants tested are CFC12 and HFC-134a with an inlet quality of 20% and saturation temperatures between 4.4 and 11.1°C. A similar study was conducted by Bonhomme, Chato, Hinde, and Mainland for condensation parameters Bonhomme, (1991).

Materials and Methods

MATERIALS

The material used is Refrigerator system operation of a frequency analyzer, Temperature monitor and Matlab/simulink for simulation.

METHODS

Determining Evaporator Heat Load (Q_{evp})

Vital to the estimation of evaporator parameters (e.g. UA_{evp} and air flow rate) is the reliability of the method used to calculate the evaporator heat load, Qevap. For this process, ftrst law energy balances are considered around two control volumes. One control volume is placed around the entire cabinet, excluding the compressor and condenser. The other control volume includes the refrigerant side of the evaporator as shown in figure 3.1. These two methods of determining the load are considered below.

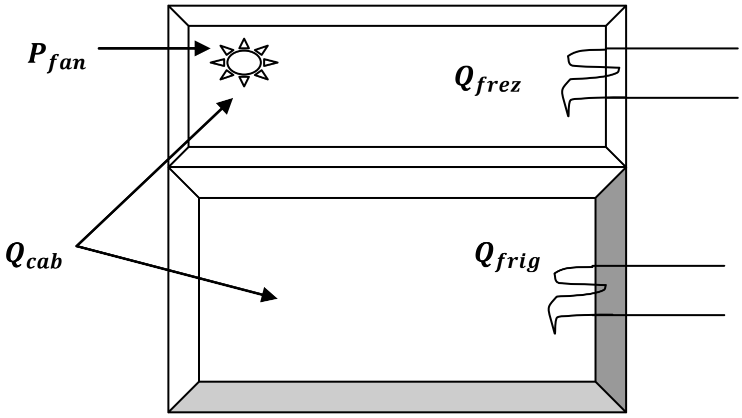

Figure 3.1: Heat Transfer Paths into Refrigerator and Freezer Cabinets

As described in, steady state operation is achieved through the use of variable heat loads in each of the freezer and refrigerator cabinets. Reverse heat leak information provides a measure of the heat transfer into the cabinets through the walls as a function of internal and external temperatures. The reverse heat leak procedure is documented in the M.S. thesis of a co-worker (Staley, 1992). Since the instrumentation yields a measure of the electrical power flowing into the cabinets via the heaters and evaporator fan, enough information is available to estimate the total energy flowing into the cabinets, all of which must be removed by the evaporator. Figure 3.1 shows the paths of heat transfer into the cabinet. An energy balance for this system is shown in Equation (3.1).

$$Qevap = Qcab + Qcrez + Qrrig + Pfan \tag{3.1}$$

Since three of the values are well known electrical inputs ($Qfrez, Qfrig, and\ Pfan$), the greatest uncertainty comes from the cabinet heat leak calculation ($Qcab$).

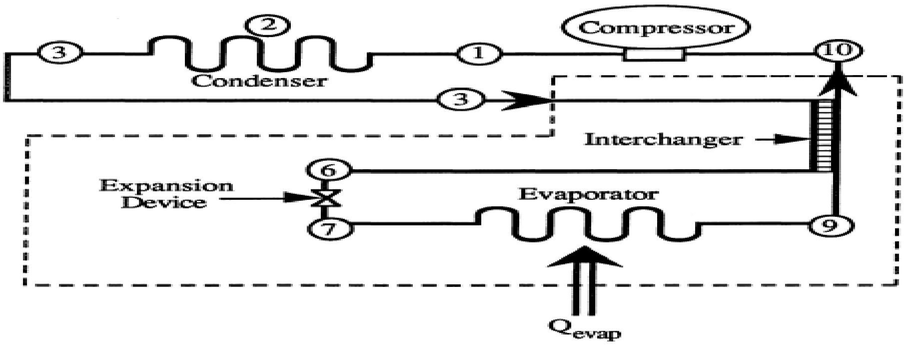

Figure 3.2: Refrigerant Side Energy Balance on the Evaporator

The other method of determining the evaporator heat load involves a refrigerant side energy balance. Ideally, the desired control volume would include only the evaporator, and the evaporator heat transfer would simply be the product of the refrigerant mass flow rate and the difference between the outlet and inlet enthalpies. This, however, is not possible since the enthalpy at the two-phase inlet is not uniquely determined by temperature and pressure, and cannot be determined from experimental data. By extending the control volume to include the expansion device and the capillary tube/suction line heat exchanger (interchanger), as shown in Figure 3.2, the refrigerant side energy balance is now possible. The control volume inlet enthalpy is taken at the sub-cooled outlet of the condenser and the control volume exit enthalpy is taken at the super-heated exit of the suction line. Further, by assuming that there is no heat transfer between the interchanger and the surroundings, it is possible to formulate Equation (3.2).

$$Qevap = mdot\,(qinlet - qoutlet) \qquad (3.2)$$

An examination of these two methods and a comparison of their respective results are presented in results. The evaporator heat load obtained from the cabinet load analysis is used for the following parameter estimations.

Air Side Energy Balance

With the evaporator heat load, $Qevap$, determined, a simple first law air side energy balance is conducted around the evaporator. If the air inlet and outlet temperatures are known, then the flow rate of air can easily be estimated for a given data set. The air side energy balance is represented in Equation (3.3).

Where

$$Qevap, air = mdotairCp, air\,(Tma - Tair, out) \qquad (3.3)$$

$$Tma = mixedairtemperatureatevaporatoroutlet$$

$$Tair, out = air\ temperature\ at\ evaporator\ outlet$$

$$mdotair = {}^{60\,Vdotevap}/_{v}\ (lb/lb)$$

$$v = tqespecificvolumeofair\,(ft^3/lb)$$

$$Vdotevap = tqevolumetricflowrateofair\,(ft^3/min)$$

$$cp, air = tpespecific\ teatofair$$

The specific heat (at the air outlet temperature, $T_{air,out}$is used rather than an enthalpy difference because the measured, well-mixed outlet air downstream of the fan is accurately known, and the evaporator air inlet, $Tair, in$, is not. The refrigerator and freezer air mixes only inches from the inlet, so the $T_{air,in}$ thermocouple reading was initially suspect. Therefore, an air split analysis is used to calculate this temperature as a function of the two return air temperatures from the cabinet, as defined by Equation (3.4).

$$Qma = fz\,qfreezer + (l - fz)q\,refrigerator \qquad (3.4)$$

Where q = entaply of the air, fz = mass ration of air from freezer

This energy conservation equation assumes adiabatic mixing. By assuming constant specific heat, an excellent assumption over the temperature range in question, Equation (4.4) simplifies to Equation (3.5).

$Tma = fzTfreezer + (l - f0\, T\, refrigerator$(3.5)

Where T= temperature of the air for fz= $massratioofairfromfreezer$

To be rigorous, Equation (3.4) is used in the following analysis. Multidimensional optimization is used to estimate both the air split fraction, fz, and the total volumetric flow rate over the evaporator, $Vdotevap$, simultaneously. The objective function is shown in Equation (3.6).

$$minimize \sum \left(Q_{evap} - Q_{evap\,,air}\right)^2 \qquad (3.6)$$

The volumetric air flow rate, $Vdotair$, of 46 cfm resulting from this procedure matched quite well with the manufacturer's estimate of 45-47 cfm (Elsom, 1991). The air split fraction, fz, of 84.7% is also in excellent agreement with the manufacturer’s estimate of 85% (Elsom, 1991). Using these values of fz and $Vdotair, Tair$, in is calculated for each run (from the cabinet return air temperatures) and compared to the air inlet temperature measured using the thermocouple array.

Design of the refrigerator temperature analyzer

Figure 3.3 models a basic refrigeration temperature analyzer that transfers heat between the refrigerant two-phase fluid and the environment moist air mixture. The compressor drives the R134a refrigerant through a condenser, a capillary tube, and an evaporator. An accumulator ensures that only vapor returns to the compressor.

Two fans drive moist air flow over the condenser and the evaporator. The evaporator air flow is divided between the freezer compartment and the regular compartment. The controller turns the compressor on and off to keep the compartment air temperature at around 4 degree C.

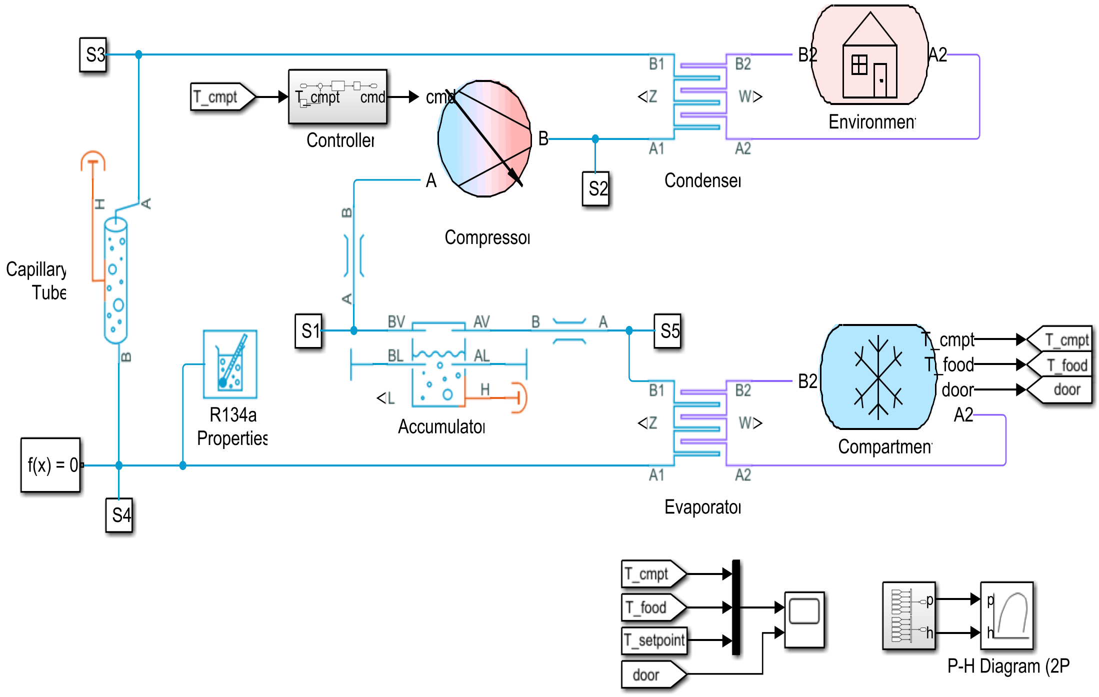

Figure 3.3: simulink model of refrigerator temperature analyzer

IV Result and discussions

4.1 Results

4.2 Simulation Results from Simscape Logging

This plot shows the rate of heat transfer between refrigerant and moist air in the condenser and evaporator as well as the rate of heat loss through the insulation of the compartment and freezer. It also shows the temperature of cold air and food in the compartment and freezer. At 11000 s, the compartment door is opened for 60 s, resulting in a spike in compartment temperature, table 4.1 shows the heat transfer rate and temperature respectively of the refrigerator system.

Table 4.1: heat transfer rate

HEAT FLOW RATW (W)	Condenser	evaporator	comparator	Freezer
	900	310	50	180
	200	183	50	43
	0	36	45	430
	210	178	110	100
	0	38	48	40
	200	180	100	80
	0	40	41	38

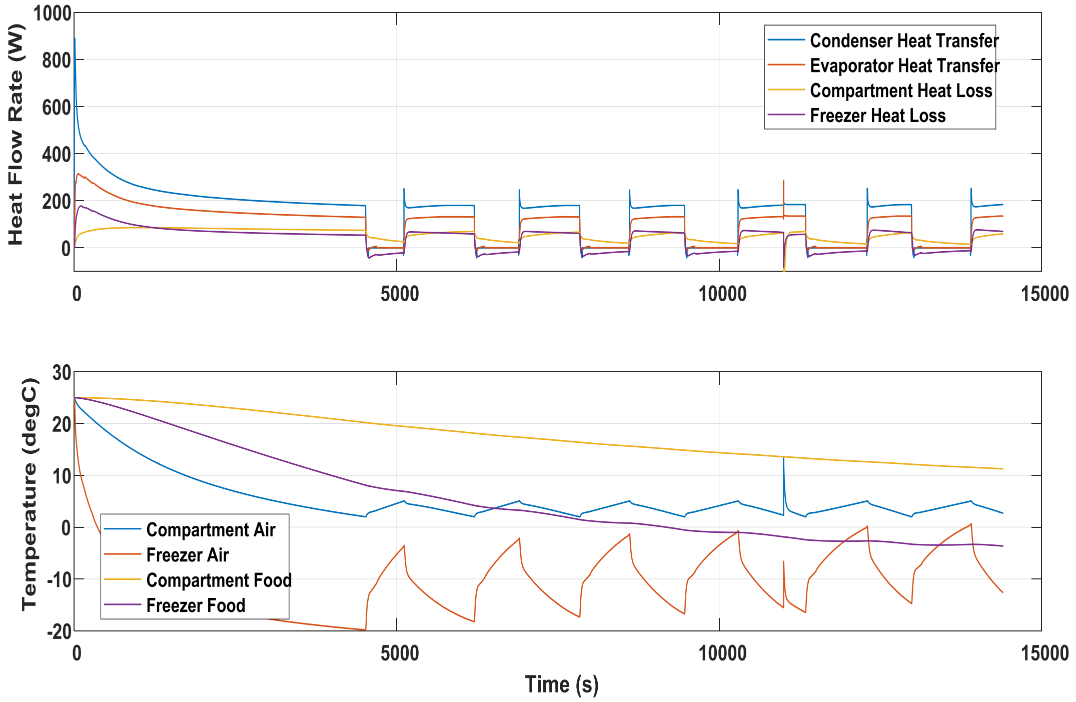

Figure 4.1: plots showing heat flow rate and temperature of refrigerator system

Figure 4.1 shows the temperature consumed by the freezer air and the cooling load of the compartment food system, which is the rate of heat transfer in the evaporator. The coefficient of performance is the ratio of the cooling load and the time variants to consumed.

Table 4.2: Temperature

Temperature deg	Condenser	evaporator	comparator	Freezer
	25	25	25	25
	8	-5	22	15
	8	-3	20	8
	7	-18	18	2
	6	0	15	-0.5
	7	0	12	-3
	8	0	40	-4

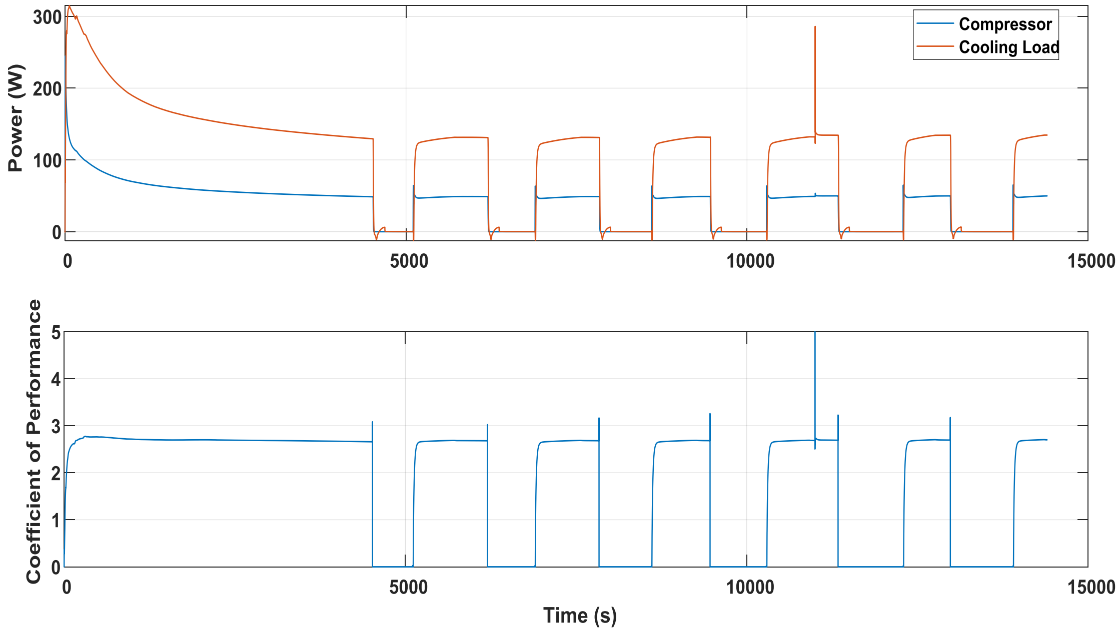

Figure 4.2: plot showing power consumption and coefficient of performance

Figure 4.2 shows the power consumed by the compressor and the cooling load of the refrigeration system, which is the rate of heat transfer in the evaporator. The coefficient of performance is the ratio of the cooling load and the power consumed.

Table 3 Pressure and Mass low Rate

Pressure and mass flow rate	condenser	evaporator	comparator	Freezer
	25	5	3	0.3.0
	8	-5		
	8	-4		
	7	-2	2	2
	6	0	1	-0.5
	7	0		-0.6
	8	0	0	-0.4

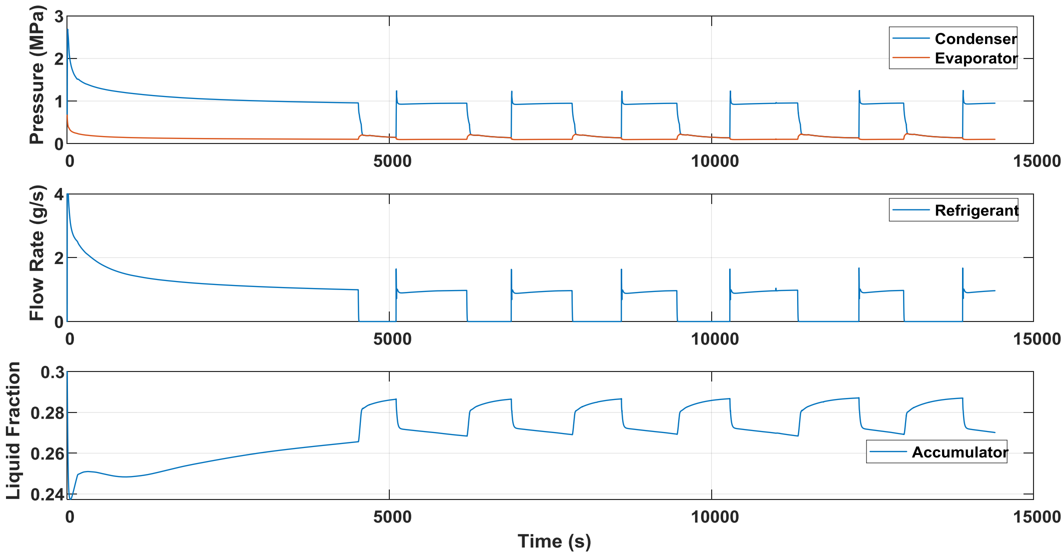

Figure 4.3: plot showing refrigerant pressure and mass flow rate

Figure 4.3 shows refrigerant pressure and mass flow rate. The high pressure line is at around 1 MPa and the lower pressure line is at around 0.1 MPa. The nominal refrigerant flow rate is 1 g/s. The plot also shows the liquid volume fraction in the accumulator.

Conclusions

One of the goals of this work was to develop a model that is better able to simulate refrigerator performance under off-design operating conditions. Our experimental results indicate that this goal can be meet fairly well through the use of relatively simple multi-zone heat exchanger models. By assigning a constant effective heat transfer coefficient (U) for each zone (sub cooled, two-phase, and superheated) and scaling the zones with the area (A), the model's ability to predict refrigerator performance under conditions where there is significant condenser sub cooling and evaporator superheat has been greatly improved. The confidence levels for our experimental estimates of overall heat transfer coefficients (U) of the heat exchangers are within 5%, the volumetric air flow rate over the heat exchangers (V dot) within 5%, the effectiveness of the capillary tube/suction line heat exchanger ($EinVwitEin$ 4%, and the effective heat transfer coefficient of the compressor (hbar) within 10%. The experimental data also confirm that the generic performance maps provided by compressor manufacturers may contain significant errors, especially with regard to refrigerant mass flow rate. These errors could be due to several factors. First, it is not uncommon to observe as much as ±5% variation in the performance of two compressors of the same model, Second, to generate the maps, the compressors are tested in a calorimeter where the compressor is exposed to a constant air temperature different from that seen under a refrigerator compartment. Moreover, variations of ±5% among compressor calorimeters have been observed. In our experimental setup, these effects are minimized, if not eliminated, by measuring mass flow directly or by calculating it from energy balances that are based on direct measurements of cabinet loads and refrigerant pressures and temperatures.

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