Dynamic Modeling and Validation of a Triple-Evaporator Domestic Refrigerator/Freezer with R-600a

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ABSTRACT

As domestic refrigerator/freezer designs become increasingly more complex and the technology implemented advance, dynamic modeling of the entire system becomes more important for predicting behavior and reducing energy consumption. In this work, a triple-evaporator domestic refrigerator/freezer with a variable speed compressor and R-600a as the working fluid is investigated with the goals of identifying an improved cycle architecture and control algorithm. A detailed dynamic model has been developed to predict the performance of the triple-evaporator refrigeration/freezer. The model uses detailed component models (*e.g.*, heat exchangers, compartment dynamic loads, compressor, among others) and implements various control schemes. The refrigerator/freezer has been fully instrumented and tested inside a psychrometric chamber at the Ray W. Herrick Laboratories at Purdue University. The experimental results have been used to assess the performance of the unit, conduct steady-state and transient model validation. The validated model is used to identify potential cycle improvements.

1. INTRODUCTION

Domestic refrigerator/freezer systems are essential appliances in modern households, but contribute to a significant amount of energy consumption around the world. In fact, approximately 6% of all energy consumed worldwide is attributed to domestic refrigerators according to the study done by Caglayan et al. (2022). High energy demand can be accounted not only by the large number of units running but also the low thermodynamic efficiency as mentioned by Hermes et al. in 2008. Improvements to efficiency and performance of the domestic refrigerator can reduce energy consumption on a global scale.

To study the performance of a refrigerator/freezer system, it is important to create an accurate model that will predict the performance of the unit, since experimental testing is typically time-consuming and costly, as mentioned in Borges et al. (2015). Negaro et al. (2011) concluded that steady-state mathematical models are not suitable for predicting the performance of refrigerators due to the fact that energy consumption is highly affected by the compressor run time for an on-off controlled compressor as well as the power consumption while in operation. This is still true for a variable-speed compressor unit with multiple evaporators circuits controlled by a multi-way valve. During the operation of such a system, the multi-way valve will be controlled to switch between circuits to satisfy the cooling demand. A dynamic model that can capture the unit's transient behavior should be used to fully study the performance of such a system.

In recent years, the demand for more versatile and flexible domestic refrigerators has prompted manufacturers to develop and commercialize products that include a third convertible cabinet with flexible temperature set point alongside the traditional freezer and refrigerator cabinets. For some products, this means a dedicated third evaporator in the system for the convertible cabinet. The third evaporator adds an extra layer of complexity with respect to the typical dual-evaporator system which in turn requires more complicated control algorithms to optimally control the unit to meet the cooling demand of the system. Different evaporators need to be activated and deactivated based on the cabinet temperatures and their set points. The switching between all the evaporators causes transient responses in the system that impact the performance of the unit. To study the energy consumption of such a system, a dynamic model that can capture this transient behavior and incorporate the controls of the system needs to be developed in order to reliably predict its performance.

In this work, a dynamic model was developed for a triple-evaporator domestic refrigerator/freezer system using R-600a as the working fluid in order to study its performance and explore improvement options. The system model was developed using the Modelica language in the Dymola environment to enable modeling of dynamic system and predict its transient responses. The dynamic model was based on detailed component models for the heat exchangers, compressor, expansion valves and cabinets. The system was tested experimentally at the Ray W. Herrick Laboratories at Purdue University. The results of the model were validated against experimental data and further used to explore performance improvement options.

2. SYSTEM MODELING

2.1 System Overview

The domestic refrigerator/freezer system in this study is a triple-evaporator free-standing model. This unit features three cabinets, each with its own evaporator. This system has a convertible cabinet along with the traditional refrigerator and freezer cabinets. It has a total net unit capacity of 591 liters and the vapor compression cycle uses a R-600a variable-speed hermetic reciprocating compressor.

The schematic of the system is shown in Figure 1. As seen in the figure, the system has a total of three evaporators, the refrigerator cabinet (RC) evaporator, the convertible cabinet (CC) evaporator and the freezer cabinet (FC) evaporator, respectively. The three evaporators are connected together and working in a bypass circuit cycle. This means that while the freezer evaporator will always run, the refrigerator and convertible evaporator are bypassed when not be in operation. The circuits are illustrated in Figure 2. There are a total of three circuits that can be run in the unit as seen in the figure. The four-way valve in this system will change the direction of the flow and allows only one circuit to run at one time.

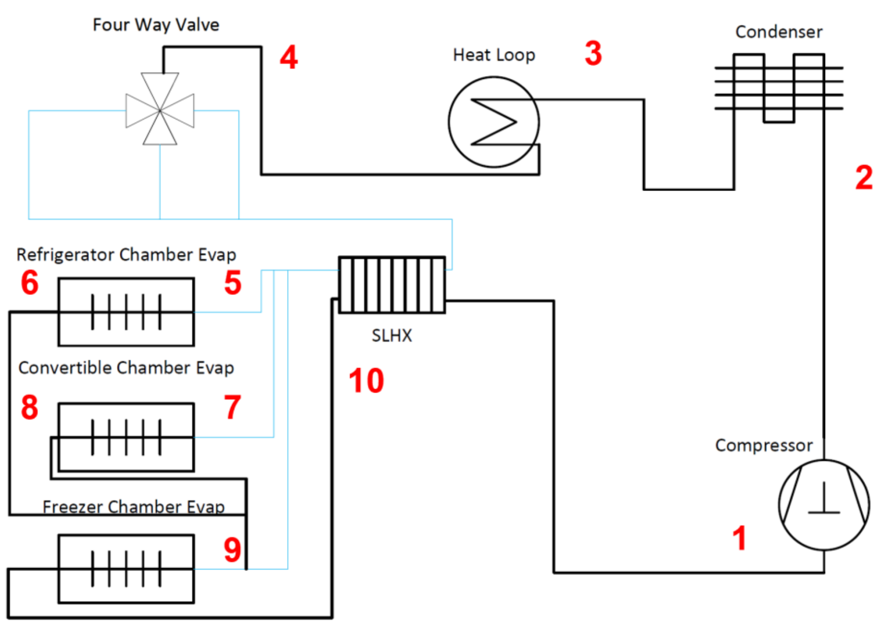


Figure 1 Schematic of the refrigerator/freezer system

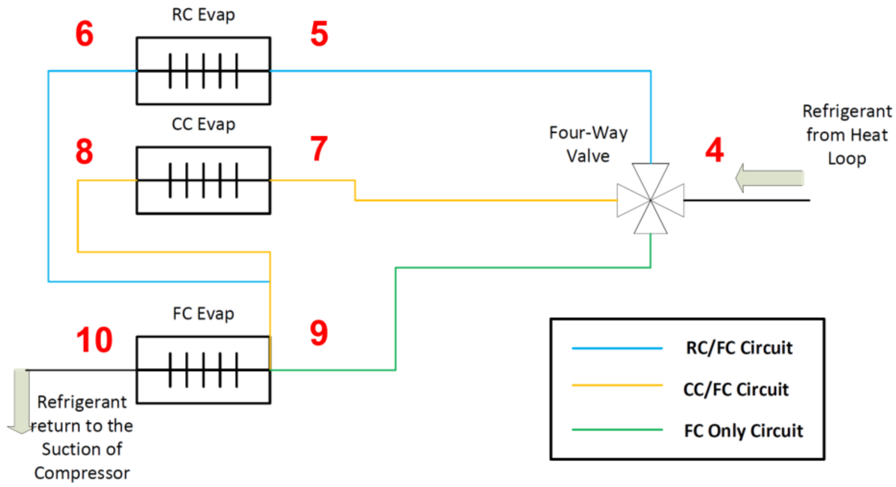


Figure 2 Illustration of the bypass circuit cycle of unit

2.2 Model Overview

The dynamic model developed in this study was created using the Modelica language in the Dymola environment utilizing TIL suite (2020). This modeling approach allows the implementation of the controls in the development of the model as well as obtaining transient results for performance analysis. The version of TIL used in this study is 3.10.0. The schematic of triple-evaporator refrigerator model in Dymola is shown in Figure 3.

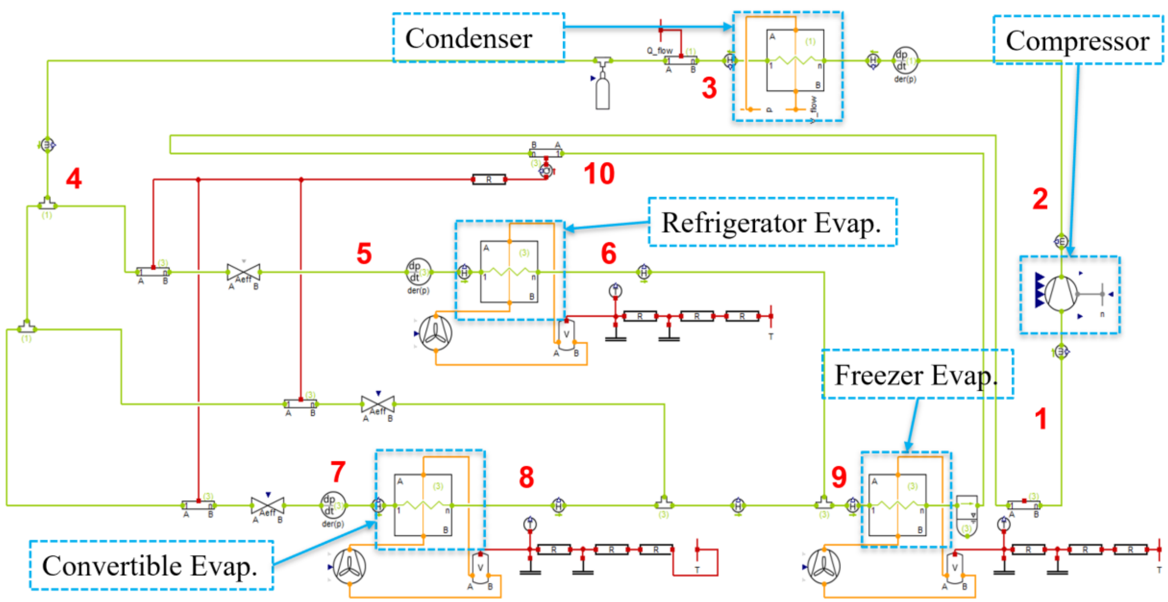


Figure 3 Triple-evaporator refrigerator/freezer system dynamic model in Dymola

2.3 Component Modeling

This section details the modeling approaches for each cycle component including a variable-speed compressor, fin-and-tube heat exchangers for both condenser and evaporators, diabatic capillary, a suction line heat exchanger and the cabinet wall insulation thermal model.

2.3.1 Compressor and Heat Exchanger Modeling

The compressor used in the system is a variable-speed hermetic reciprocating compressor with R-600a as the working fluid. The displacement of the compressor is 11.44 cm^3 and its operating frequency ranges from 40 to 143 Hz. A performance map of the compressor was provided by the manufacturer, and both the power consumption ($\dot{W}_{in}$) and mass flow rate ($\dot{m}_{map}$) of the compressor model are computed using a 10-coefficient map as mentioned in AHRI Standard 540 (2020) given the evaporation and condensation temperature of the system. Performance maps of the compressor at two frequencies were determined and used to linearly interpolate the compressor performance across the range of frequency, as detailed by Shelly et al. (2020).

Both the condenser and evaporators are fin-and-tube type heat exchangers. To model the fin-and-tube heat exchangers, the dimensions and details of the heat exchangers such as the fin thickness, fin frequency, number of passes among others are entered into a pre-defined fin-and-tube model within the TIL suite along with a choice of heat transfer and pressure drop correlations. This heat exchanger model is based on a discretization HEM, Homogenous Equilibrium Model. The heat transfer and pressure drop correlations used in the heat exchangers are summarized in Table 1. The air flow rate across the fin-and-tube heat exchanger is given by the fan datasheet. The fans used in the system are variable speed with a maximum volume flow rate of 0.029 m^3/s for the evaporator fans and 0.047 m^3/s for the condenser fans.

Table 1 Summary of heat transfer and pressure drop correlations used in heat exchanger model

	Heat Transfer Correlation	Pressure Drop Correlation
Fin-and-tube heat exchanger, R-600a	Evaporation: Shah et al. (1976), Chen et al. (1966) Condensation: Shah et al. (1979) 1-Phase: Gnielinski et al. (1976)	Konakov et al. (1946)
Fin-and-tube heat exchanger, Air	Haaf et al. (1988)	Haaf et al. (1988)

2.3.3 Diabatic Capillary Tube and Suction Line Heat Exchanger

The domestic refrigerator/freezer uses capillary tubes at the beginning of each circuit as the expansion devices. The capillary tubes are considered to diabatic in this case since they serve as counter-flow heat exchangers with the compressor suction line providing adequate superheat. In turn, the refrigerant in the capillary tube will further subcool in the process. Due to the nature of the Dymola environment and limitations of the TIL suite used at the time of this study, a single capillary tube model that can handle both the heat exchange and pressure drop was difficult to develop and was not included in this study. Instead, the capillary tube model is divided into two separate components: an orifice valve model responsible for producing the pressure drop and a tube model responsible for the exchange of heat. The heat transfer correlation used for the tube model is the same as in the fin-and-tube heat exchangers listed in Table 1. The pressure drop in the tube has been neglected.

In the refrigerator unit, the capillary tube is brazed to the suction line in a lateral configuration. The heat transfer between the capillary tube and the suction line is modeled with a thermal resistance as seen in Figure 1. The thermal resistance is calculated using the length of the capillary tube and its diameter. It is also adjusted later during the validation of the model.

The length of the capillary tube in the unit for each circuit was given by the manufacturer. The opening of the orifice valve is determined using the pressure drop that would occur given the length of the capillary tube at the design condition. In addition, the orifice valve opening is tuned during the validation process.

2.3.4 Cabinet Wall Model

The cabinet walls of the refrigerator/freezer system contribute to the thermal load entering the cabinet and affect the temperature of the compartment. For these reasons, it is important to accurately capture the dynamics of this interior space. In this study, the cabinet walls have been modeled with an RC equivalent circuit, as described by Belic et al. (2016). A schematic of the RC wall model is illustrated in Figure 5. The thermal resistance associated with the air on both sides of the cabinet walls is calculated using natural convection based on the dimensions of the system's cabinets. It was calculated to be 0.032 and 0.034 K/W for the refrigerator cabinet and freezer/convertible cabinet, respectively. The thermal resistance of the wall is calculated using the properties of the vacuum panel and foam insulation provided by the manufacturer. The wall thermal resistances are calculated to be 3.74 K/W, 4.13 K/W and 19.2 K/W for the refrigerator, freezer and convertible cabinets, respectively. There are two thermal capacitances in this model. The thermal capacitance C1 considers the energy that can be stored in the cabinet wall. The thermal capacitance C2 models the behavior of the air inside the cabinet and calculated using the properties of air and dimensions of the cabinet. The thermal capacitance of the air inside the cabinet, C2, is calculated to be 460.8 J/cm^{3}K for the refrigerator cabinet and 123.6 J/cm^{3}K for the freezer and convertible cabinets. The thermal resistances and capacitances of all three cabinets were initially calculated and then adjusted to better match the air temperature inside the cabinet with the experimental data obtained.

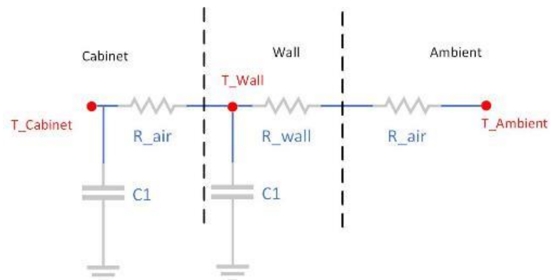


Figure 4 Equivalent RC model of cabinet walls

2.3.5 Miscellaneous Components

To successfully simulate the temperatures inside the cabinets, a finite volume of air that serves as the cabinet model was connected to air-side calculations of the evaporators. The volumes of the cabinets are 112 liters for both freezer and convertible cabinets and 355 liters for the refrigerator cabinet. The air-side calculations for the condenser are connected to air-side heat sink associated with ambient temperature within a household.

There is also a "heat loop" that is a long liquid line wrapped around the doors of the cabinets to prevent condensation forming on the exterior surface of the gasket and gasket/cabinet interface. The heat loop is modeled as a tube at the outlet of the condenser with a fixed heat loss of 10 W to the ambient as suggested by the manufacturer.

A suction accumulator of 50 ml is added at the freezer evaporator outlet to reflect the actual structure of the system. This small accumulator is used when the system runs in the FC only circuit. Since only one evaporator is in operation in this circuit compared to two evaporators running together in the other circuits, there will be extra refrigerant in the system that needs to be stored in the suction accumulator to ensure vapor at the suction inlet.

2.4 System Controls

Given the complexity of the refrigerator/freezer system with three evaporators and three different circuits, the control of the system is important to predict the performance of the unit. To capture the unit's controls in the dynamic model, experimental results were studied to understand the basic logic of the system when it is stabilized around set point. The control logic for maintaining cabinet temperatures and switching between circuits during stablized operation that was implemented in this study is as shown in Figure 6. Figure 6 refers to the three circuits in the system as CC/FC, RC/FC and FC only circuits as in Figure 2. This control logic is a simplified version compared to the actual control algorithm implemented in the actual system. Since this study focuses on the operation of the unit when the cabinet temperature is stabilized around the set point, the control logic of the unit during pulldown and ice making is not included.

This control logic compares the temperature in each cabinet to its set point and identifies if a cabinet needs cooling. Then the system will send refrigerant down the corresponding circuit to provide cooling to that cabinet. Since the freezer evaporator will always be running, the freezer evaporator fan is controlled separately with respect to the freezer cabinet temperature. The fan will stop running if the temperature is already at the set point to avoid over cooling the freezer.

To implement this control logic in the Dymola model, the orifice valve model used in the capillary tube model will open and close based on the circuit it will run. A four-way valve model is not available in the current version of the TIL suite. The orifice valve provides a good option to simulate three circuits running separately and each one with its own expansion device. Each evaporator circuit has one orifice valve model and only one among the three will open at one time during operation to allow only one circuit to run.

2.5 Modeling Parameters

In this study, the ambient temperature of the model was set to be 32 °C. The cabinets temperatures were 5 °C for refrigerator cabinet, -23 °C for the freezer cabinet and -22 °C for the convertible cabinet. The cabinet temperatures in all three cabinets follows around the set point within ±1 °C. Cabinet temperature and power consumption results are shown in detail in later sections.

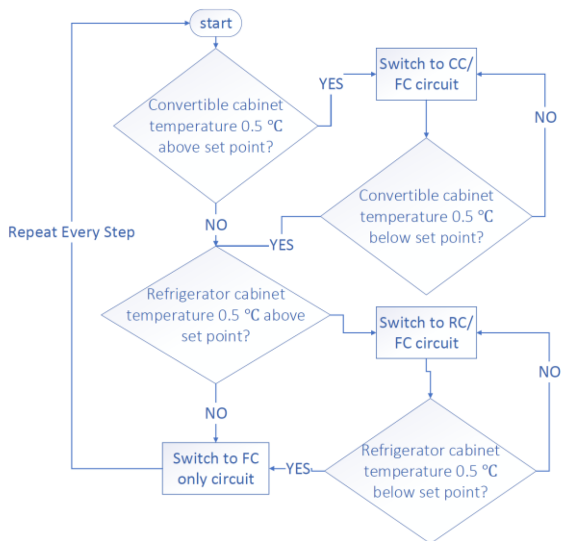


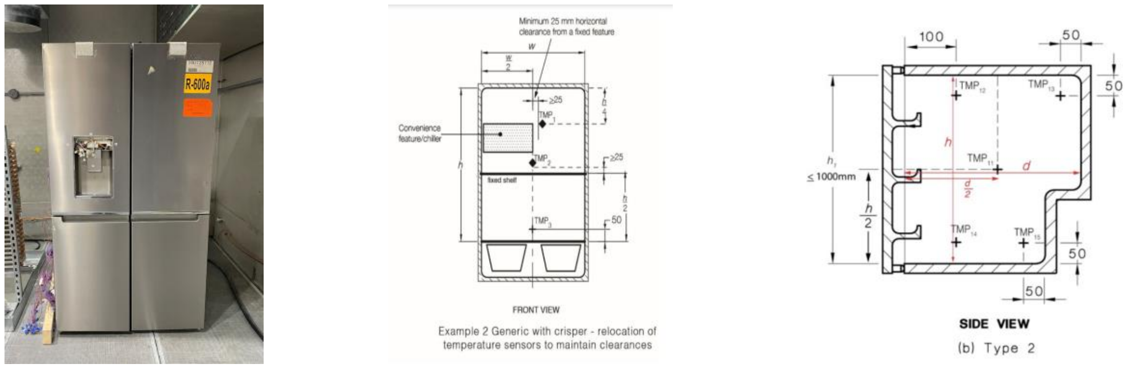
Figure 5 Flow chart of the control logic implemented in dynamic model

3. EXPERIMENTAL VALIDATION

3.1 Experimental Setup

The refrigerator/freezer system was installed and tested at the Ray. W Herrick Laboratories at Purdue University. The refrigerator was set up inside a psychrometric chamber able to handle flammable refrigerants, as shown in Figure 7(a). The system has been instrumented with T-type thermocouples (±0.1 °C) both inside the cabinets and in the refrigeration circuits. A total of 23 thermocouples were used in the unit to conduct the experiments. In particular, 13 out of 23 thermocouples were located in the cabinets to measure the air temperature distribution, and the remaining thermocouples were placed on the refrigeration circuit. A power meter with ±1% error across its range was also installed to measure the power consumption of the unit during operation. The data acquisition device used intervals of 50 seconds to collect all the data during the experiment.

The experimental testing followed the energy testing procedures outlined in Standard EN 62552 (2019). The energy testing does not require any load inside the cabinets during testing. The ambient temperature was kept at 32 °C during testing. The thermocouples are instrumented based on the instructions given in this standard, as shown in Figure 7(b) and 7(c).



(a) **(b)** **(c)**

Figure 6 Experimental set up of the experimental unit: (a) Experimental unit; (b) Refrigerator cabinet thermocouple layout; (c) Freezer/convertible cabinet thermocouple layout

3.2 Model Validation

To validate the dynamic model, the temperature inside each cabinet as well as the power consumption of the unit must be studied to confirm the prediction of its dynamic behavior. The dynamic behavior that can be captured in the results of this model is a major advantage compared to the results in a steady-state model. Moreover, the dynamics associated with the compressor operation must also be captured to predict the energy consumption.

The comparison between measured and predicted values of the freezer cabinet temperature is shown in Figure 8. All the figures in the section are depicted in 50 second intervals as the experimental data is collected every 50 seconds. As seen in the figure, the model results can maintain the temperature of the freezer cabinet around the set point. The cabinet temperature oscillates around the set point in a range of around 1 °C. The average temperature of the freezer cabinet is -22.72 °C in the experimental result and -23.14 °C in the modeling result. There is a 0.42 °C difference between the average values.

The modeling result shows around 0.5°C more oscillation than the experimental result. This is due to the fact that the control logic implemented in the model prioritize the refrigerator and convertible cabinet before the freezer cabinet. In addition, the unit does not cycle off when the cooling demand is met. This means the unit continues to provide cooling in the freezer while the other two cabinets are satisfied, even if the freezer temperature is at the set point. This does not disrupt the performance of the unit since the components are designed precisely so that the freezer does not get too much cooling before the system needs to switch to another circuit in this situation. Since the model inevitably has deviation from the physical unit in components such as capillary tubes and heat exchangers, the freezer gets unnecessary cooling in the model and the temperature in the freezer will have a higher oscillation. In fact, the oscillation range of the convertible cabinet temperature in the model is set to be slightly smaller than that in the experiment as seen in Figure 10. This way the CC/FC circuit can be activated more frequently and the excess cooling in the freezer can be reduced.

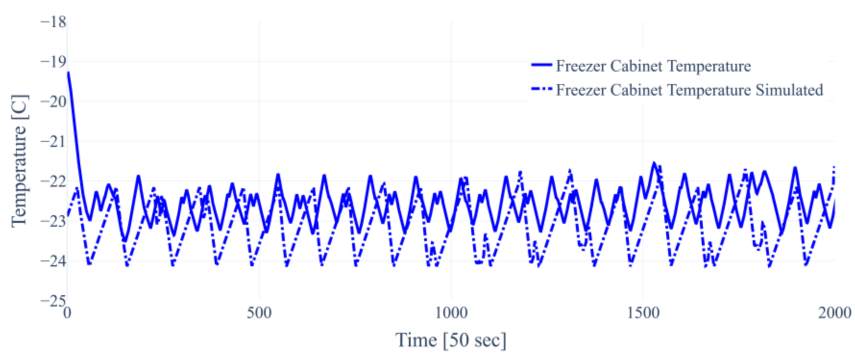


Figure 7 Freezer cabinet temperature comparison between model and experimental results

The convertible and refrigerator cabinet temperature comparison results are shown in Figure 9 and Figure 10. The modeling results of these two cabinets match well with the experimental results. The average refrigerator cabinet temperature in the experimental result is 4.54 °C and in the modeling result is 4.88 °C. Further, the dynamics of the refrigerator cabinet result is shown to be captured quite accurately in Figure 9. The average convertible cabinet temperature is -21.59 °C in the experimental results and -21.24 °C in the modeling result. The difference between the average temperature of experimental and modeling result is 0.34 °C for the refrigerator cabinet temperature and 0.35 °C for the convertible cabinet temperature. Figure 9 and 10 illustrate the ability of the model to accurately predict the dynamic behavior of the temperatures that includes the effects of both the cabinet wall and heat exchanger models.

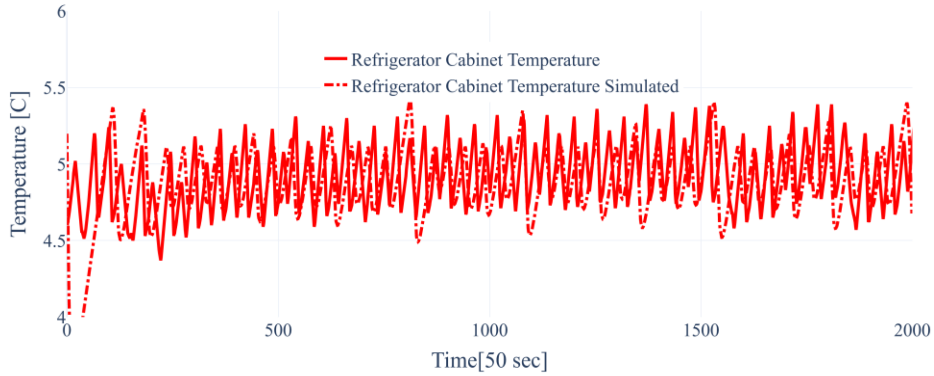


Figure 8 Refrigerator cabinet temperature comparison results

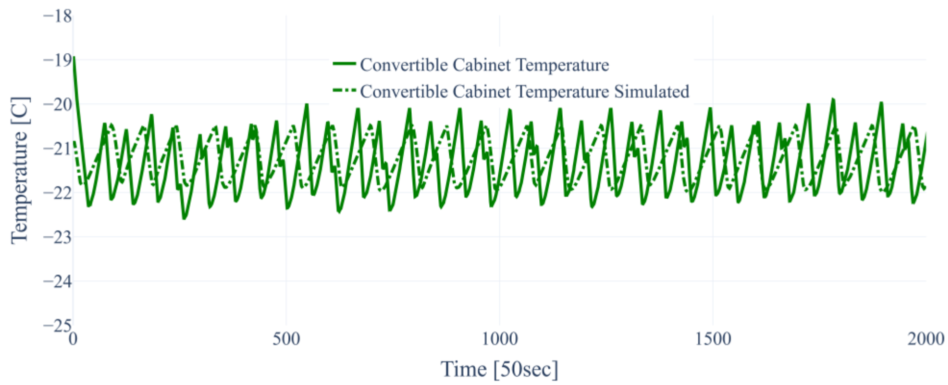


Figure 9 Convertible cabinet temperature comparison results

With the cabinet temperature of the modeling results validated with the experiments, the power consumption must be compared as well to ensure the accuracy of the compressor model and the refrigeration circuit in general. The comparison of the power consumption is shown in Figure 11. The average power consumption predicted by the model during the experimental run was 74.03 W, which only deviated by 2.03% from the measured average power consumption of 75.54 W.

Since the system operation entails the switching of three circuits, the power consumption behavior will be dynamic as each circuit is different in terms of the length of capillary tube and pressure drop which affects the mass flow rate of the system. In Figure 11, the modeling results of the power consumption exhibits a very similar pattern compared to that of the experimental results. It also shows a close range of oscillation compared to the experimental results. This means that the dynamic effect of switching circuits in the system can be captured by this model. To quantify the performance of the model in capturing the system's dynamic behavior, the root-mean-squared deviation of the model results compared to the experimental results shown in Figure 11 is calculated to be 7.71 W. This means on average the modeling result has a 7.71 W deviation compared to the experimental result. This difference can be explained by the difference in the control algorithm implemented in this model and that in the actual unit. The difference can also be caused by use of orifice valves in the model to activate and deactivate the circuit and the actual unit uses a four-way valve system that can have a difference in dynamic behavior.

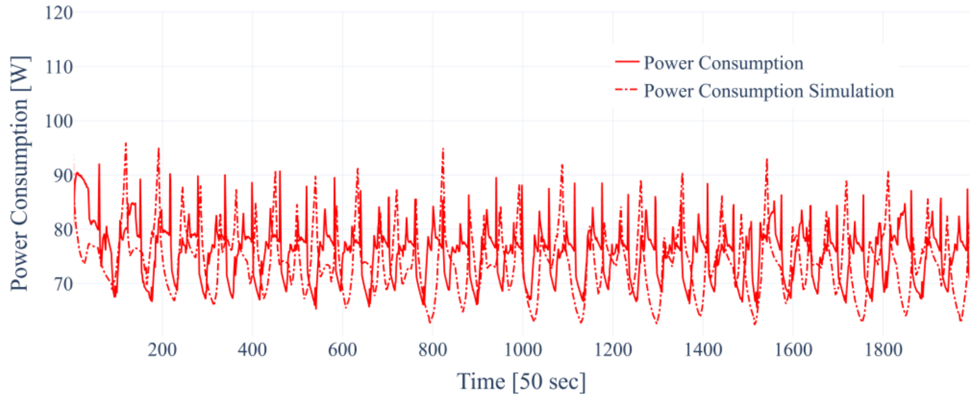


Figure 10 Power consumption comparison between modeling and experimental results

3.3 Performance Improvements

The validated dynamic model is useful for understanding the performance of the system and as a platform for exploring advanced cycle architectures for performance improvement. Utilizing this model, the performance of the system with a parallel circuit cycle can be explored with the same components. The parallel circuit cycle is such that each evaporator runs individually. Choi et al. (2018) investigated advances for domestic refrigerator/freezer system and mentioned that the parallel circuit cycle can have better performance when compared to a bypass circuit cycle such as the system in this study. However, Choi et al. also reported that a parallel circuit cycle requires a more complicated specific control algorithm to achieve the performance enhancement. With this dynamic model, the effects of different control algorithms on the unit performance can be studied.

The dynamic model can also serve as a good platform for considering more advanced cycle architecture options such as a multi-stage cycle with vapor injection. The impact of a cycle architecture change can be studied in detail and control algorithms needed for a more complex system can also be built and tested using this model.

4. CONCLUSION

In this study, a detailed dynamic model was developed for a triple-evaporator domestic refrigerator/freezer system with detailed component models and control algorithms to accurately predict the dynamic performance of the system. The results of the model were validated with experimental results of the unit in terms of both the average performance and the dynamic behavior. The model was found to accurately simulate the dynamic behavior of the actual unit at the correct level of temperature and power consumption. The differences between the average temperature in all three cabinets do not exceed 0.5 °C. The average power consumption comparison shows a 2.03% difference between the experimental and modeling results. The root-mean-squared deviation of the power consumption modeling results compared to the experimental result is 7.71 W.

Further work will include employing the dynamic model to explore and study different cycle architectures for performance improvement options. Utilizing the nature of the Dymola model, different configurations of the system such as a parallel circuit cycle can be implemented and studied. The parallel circuit cycle can potentially be modified to become a multi-pressure cycle where the pressure and temperature of the medium temperature cabinet evaporator can be separated from the lower temperature cabinet evaporators. Further, this multi-pressure cycle can be modified with additional enhancements such as vapor injected compression.

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