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DEVELOPMENT OF A TRANSIENT SIMULATION MODEL OF A FREEZER PART II: COMPARISON OF EXPERIMENTAL DATA WITH MODEL

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Abstract

A dynamic and distributed parameter model was developed to analyze a freezer system. Basic conservation equations were used to develop a mathematical model of the freezer, which consisted of models of the condenser, the evaporator, the compressor model and the capillary. Based upon the freezer mathematical model, a computer program was developed to predict changes of pressure, temperature, flow rate, etc., with both time and space. Experiments were conducted on a 400 L capacity freezer to evaluate the accuracy of the simulation model. Results for the system startup pressures, power, and temperature were presented.

Experimental Setup

A schematic diagram of the experimental setup for these tests is shown in Figure 1. The freezer was located in a conditioned room (designated as the "experimental space" in Figure 1). Measurements were made on a 400 L capacity freezer to evaluate the accuracy of the simulation model. The ambient temperature outside the freezer was set at 25±0.5°C and relative humidity was controlled at less than 90%. The size of the freezer was 1318 mm x 592 mm x 830 mm. The running speed of the compressor was 2880 rpm. Transient startup conditions were at 25°C inside the freezer.

When the ambient conditions in the experimental space reached steady state, data were collected at twelve second intervals. During the test, the instantaneous power input into the refrigerator, pressure across the compressor and temperatures of the in-cabinet and out-cabinet were measured.

Measurement of Main Parameters

The tube wall temperature and the refrigerant temperature inside the tube were measured with calibrated copper-constantan thermocouples. Six to eight thermocouples were mounted in the evaporator and the condenser to obtain the temperature profiles with time and location of the refrigerant. The in-cabinet air and ambient temperatures also were measured by the thermocouples. The thermocouples were connected by a digital voltage meter which had a sensitivity of 0.1 μV. The sensitivity of the voltage and ampere meters was 0.3 μV and 0.5 μA, respectively. Pressures were measured at the inlet and outlet of the compressor. The instantaneous power to the refrigerator and daily energy consumption were recorded by independent digital watt and watt-hour meters whose accuracy was within ± 0.2 % of the reading.

Simulated and Tested Results

The whole process from start-up to shut-down was simulated on the 400 L freezer. Initial conditions in the simulation model were determined by measured data. The results were compared with the experimental data.

Figure 2 presents the predicted curve from the model that shows how the mass flow of the compressor and capillary tube varies with time during the start-up. When the compressor started, the mass flow rate reached its maximum value very quickly. Once the compressor began operation, the refrigerant was transferred from the low pressure to the high pressure side of the system. At the same time, the condenser pressure increased rapidly and the evaporator pressure decreased rapidly (Figures 3 and 4). With the increased pressure difference between the high and low pressure sides, the mass flow rate of the compressor increased while that in the capillary tube decreased gradually. The mass flow rate through the compressor equaled that through the capillary tube approximately 150 seconds after start-up and remained relatively constant after that time. The model appeared to track to the trends in condensing and evaporating pressures well during the first 150 seconds after startup.

Figures 5 and 6 show how the pressure of both the condenser and evaporator vary with time for 90 minutes after startup. Recall that the temperature in the freezer started at 25°C and dropped to -30°C during the course of the tests. This drop in temperature is reflected in the gradual drop in both the condensing and evaporationg pressures. The temperature drop made in the freezer was not a steady process and the working process of the cooling system continued to change from start-up to shut-down. Figure 5 also shows that initial condensing pressure increased rapidly for the first 600 seconds, then gradually decreased (Figure 5). The model appeared to provide good agreement with the data.

Figure 7 shows that the power input was not constant and also varied with time. At start-up, the cooling capacity and mass flow rate of the refrigerant were larger, and the pressure of the condenser and evaporator were higher because the temperature difference in the evaporator was larger. At the same time, the input power was also large. As the cooling process was carried out, the cooling capacity of the evaporator and the mass flow rate of the system refrigerant gradually decreased. It was noted in Figure 7 that the power input decreased with time.

Figure 8 shows the temperature distribution along the evaporator at a time of 3000 seconds. The model predicted evaporating temperatures that were between 1 and 3°C warmer than those that were measured.

Discussion

Generally speaking, reasonable agreements have been obtained between the predicted values and the experimental data measured in a specified test and so the reliability of the model was verified.

However, simulation results for the whole system deviated from actual data significantly at some points because there was some interaction of calculations from each of the individual component models. The theoretical values of condensing pressure were higher than experimental values for normal operating conditions. On the analysis of the condenser model, it was possibly because the calculated heat transfer coefficient was lower than the actual value. It resulted in a larger temperature difference between inside and outside the condenser and made the condenser pressure higher. In the analysis of component model interference, the calculated compressor discharge vapor was higher than the actual value. This led to an increasing condensing capacity and made the condensing pressure higher. In the analysis of the system model, the mass flow rate in the system could be higher than the actual value, which also resulted in the increasing condenser capacity and made condensing pressure higher. It was a complex process. It was difficult to explain the discrepancy between the predicted curve and measured data independently. Another possible reason for the differences could be experimental and calculation errors.

Conclusion

The whole process from start-up to shut-down was tested and simulated for a 400 L freezer that consisted of the compressor, the condenser, the evaporator, and the capillary. Good agreement between model predictions and experimental measurements was achieved. It is recommended that the model be tested with other equipment.

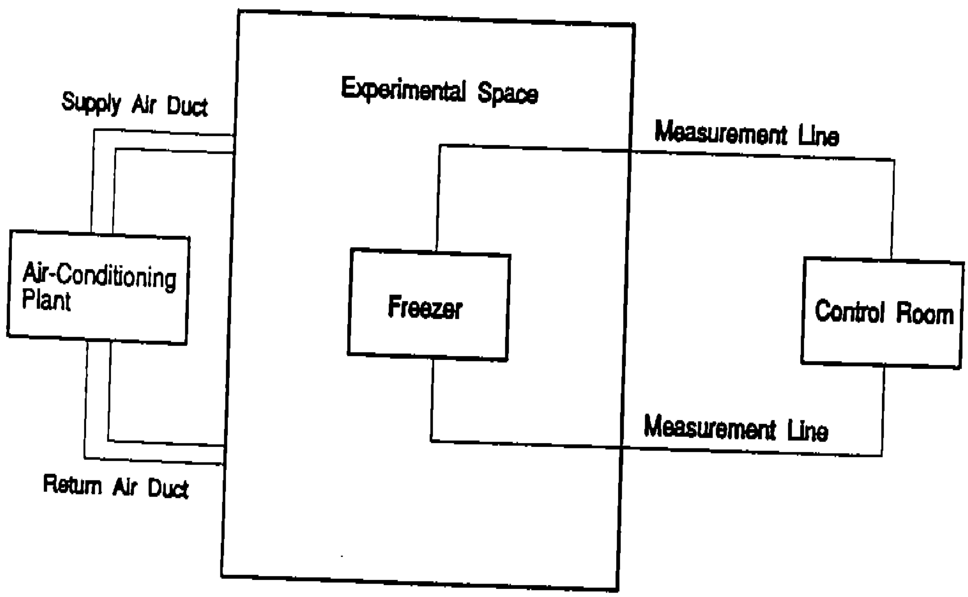

Figure 1 - Schematic view of the test setup and measurement system.

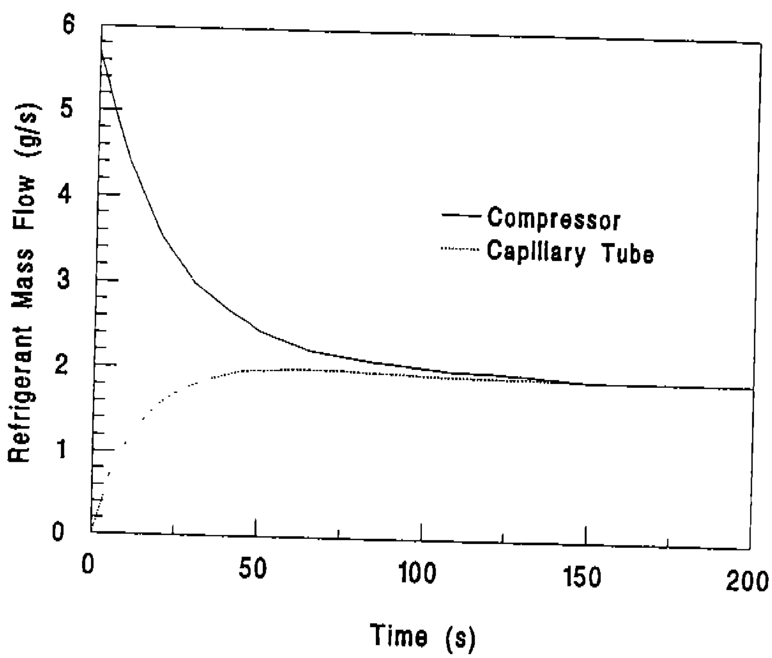

Figure 2 - Simulated startup behavior of mass flow rates in the compressor and capillary tube.

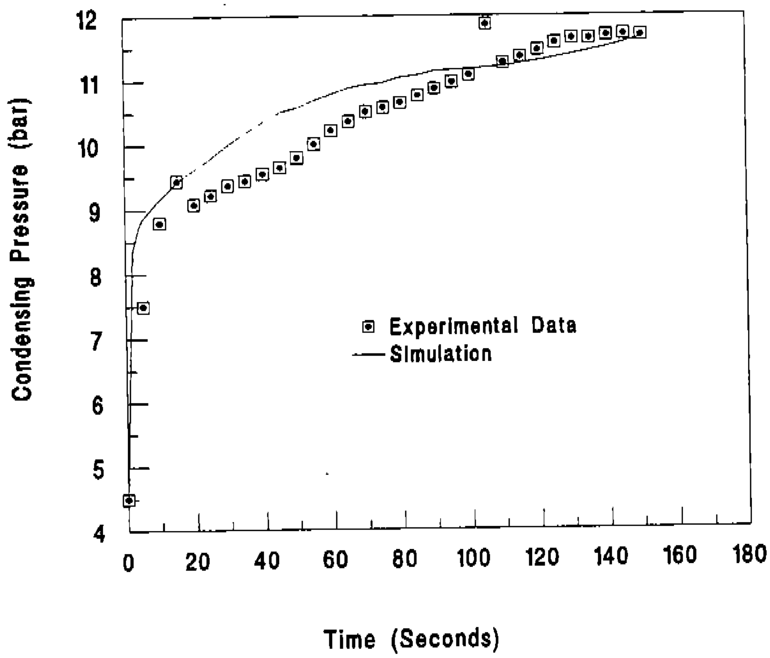

Figure 3 - Comparison of model and data for condensing pressure at startup.

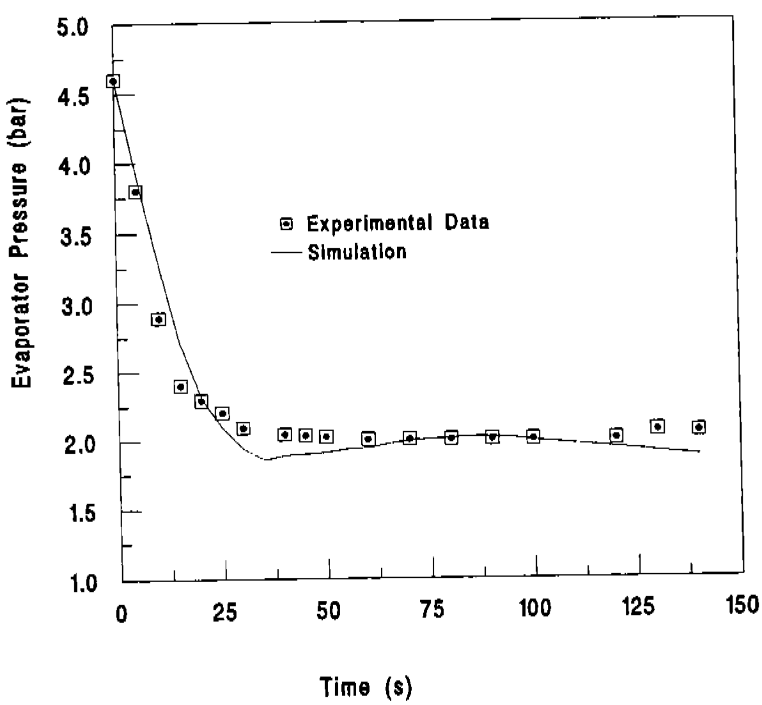

Figure 4 - Comparison of model and data for evaporating pressure at startup.

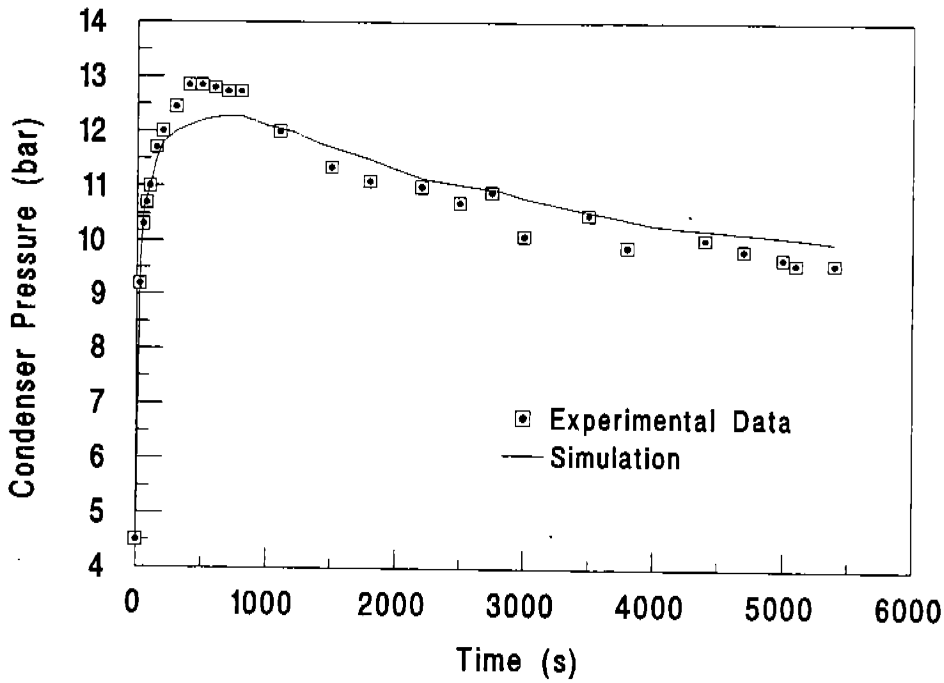

Figure 5 - Comparison of model and data for condenser pressure for long term operation.

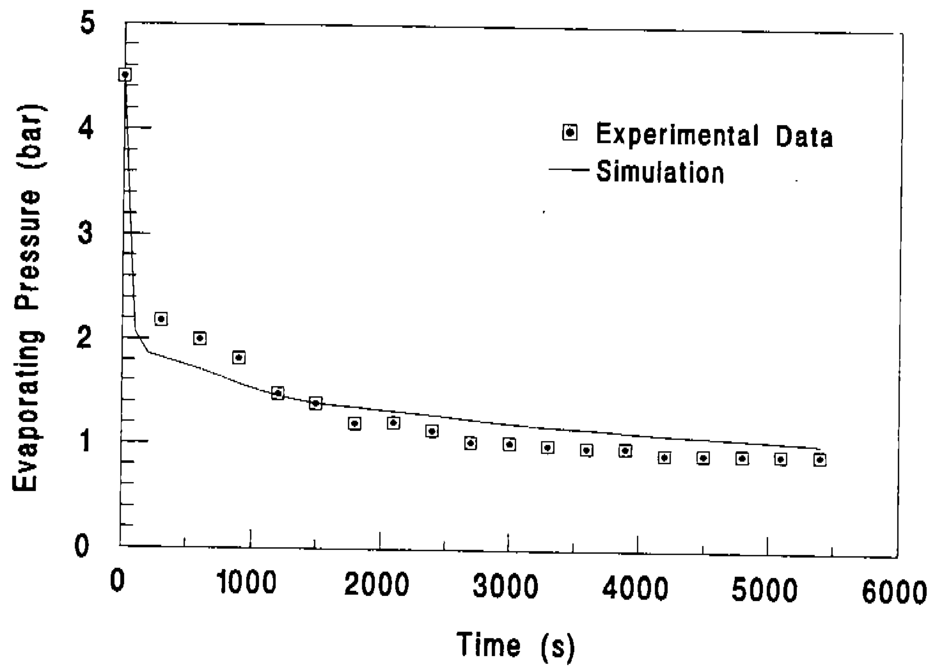

Figure 6 - Comparison of model and data for evaporating pressure for long term operation.

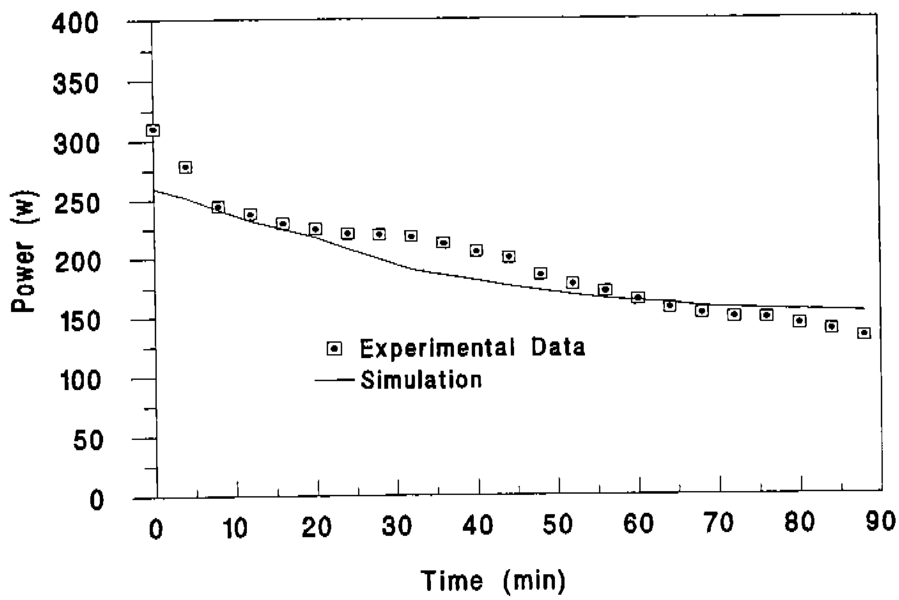

Figure 7 - Comparison of simulation and data for compressor power.

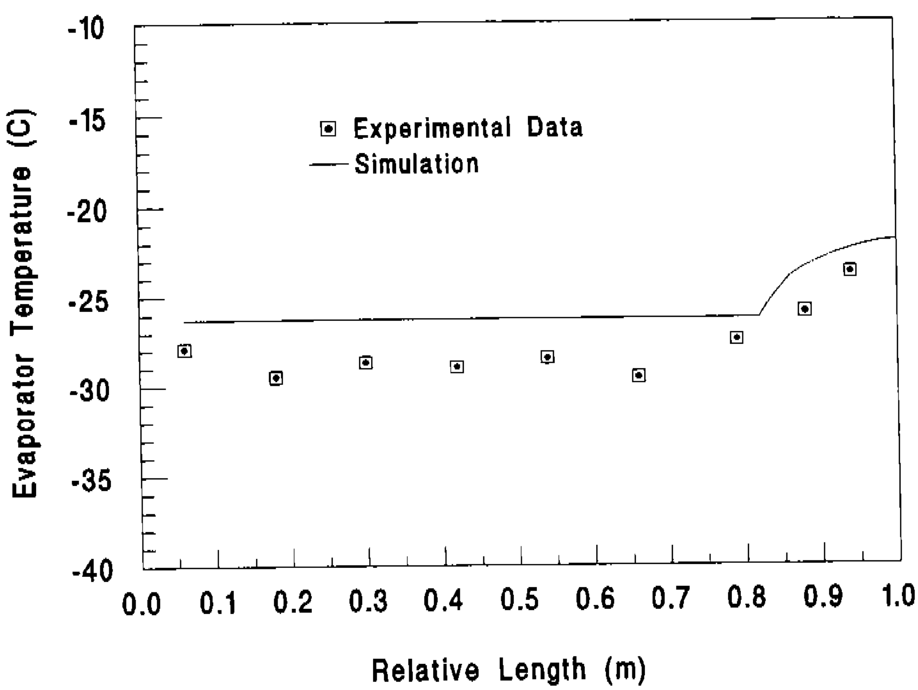

Figure 8 - Comparison of simulation and data for temperatures along the evaporator at time: 3000 seconds.