A Moving-Boundary Model of a Centrifugal Chiller System

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ABSTRACT

Dynamic models of liquid chillers are gaining importance in recent years as tools for evaluating feedback controllers and to study transient system performance in general. Such system models consist of component models for the heat-exchangers, the compressor and the expansion device. The modeling of the heat-exchangers is particularly significant as they constitute the largest thermal capacitances in the system and thus determine the system response. Existing models of liquid chillers use either highly simplified lumped parameter approximations for the heat-exchangers or highly detailed finite-volume formulations. A moving boundary formulation applied to both shell-and-tube construction heat-exchangers within a system model with a centrifugal chiller has not been found in the literature. This paper presents a complete system dynamic model of a centrifugal liquid chiller using this approach for both the evaporator and the condenser. The heat-exchangers are flooded types with refrigerant on the shell-side and water as the secondary fluid. The dynamics of the compressor and expansion valve are also included. Simulation results are presented for the system model during different load-change transients. The model is validated using data from a 90-ton single-stage, centrifugal chiller test stand with R134a as the refrigerant. The model is shown to predict system behavior accurately while running faster than real time on a contemporary desktop computer.

1. INTRODUCTION

Dynamic performance modeling of vapor compression systems has been of interest for over 25 years. An extensive literature survey of vapor compression system dynamic models (Bendapudi and Braun [2002a]) reveals that relatively few models exist for chiller systems, and even less for centrifugal systems. This paper documents the validation results of a centrifugal chiller system using the moving boundary heat-exchanger models described by Bendapudi et al [2004]. Load-change transients triggered by changes in entering water temperatures and set-points are presented. The steady-state and transient predictions are validated using data from a 90-ton single-stage centrifugal chiller test-stand.

2. TEST STAND DESCRIPTION

The test stand used for the validation of the model is a 90-ton water chiller charged with R134a. The refrigeration system consists of a centrifugal compressor with variable inlet guide-vanes for capacity control. The heat exchangers are of shell and tube construction, with two-water passes and one refrigerant pass. The expansion valve is a cascaded device consisting of a main valve driven by a pilot valve. The compressor is powered by an electric motor. The motor and transmission are cooled by the refrigerant through a bleed line tapped off of the liquid line, as shown in Fig. 1. This refrigerant returns to the evaporator inlet. The load on the chiller system is controlled by an arrangement of heat exchangers that simulate the building load, as shown in the Fig. 2. For clarity, the pumps and valves that control the water flow rates are not shown. The load on the system can be varied by altering the temperatures of the water entering the evaporator and condenser, i.e., varying the operating conditions of these peripheral heat exchangers. A detailed description of the test stand and instrumentation is provided in Comstock [1999].

3. SYSTEM MODEL

The system model consists of individual component dynamic models for the compressor, the heat-exchangers and the expansion valve, brought together with mutually consistent input-output information. Details of the compressor and valve model are presented in Bendapudi et al [2002b] and are only briefly described here.

The compressor is modeled in three parts – the controller, a quasi-steady grey-box model of the impeller and a black-box maximum capacity flow-rate model of the overall compressor assuming it to be operating with the inlet-guide vanes wide open. Using known boundary conditions of entering refrigerant pressure and enthalpy (or temperature or superheat) and discharge pressure, the maximum capacity flow-rate is computed. Based on the error difference between the actual and desired chilled water temperatures, the controller is used to output a correction factor that scales down the maximum capacity flow-rate to the actual flow-rate. With information of inlet refrigerant state, exit pressure and the flow-rate a quasi-steady state energy balance across the impeller is used to predict the exit enthalpy and power.

The expansion valve is modeled as a single thermo-static expansion device with a sensing bulb that has a constant thermal lag. The sensing bulb is charged with R500. From information of refrigerant state at the evaporator exit, the superheat is computed and determines the lift of the valve above the seat. Using the geometry of the valve seat, the flow-area is estimated and the flow-rate through the valve is determined using the orifice equation with a constant discharge coefficient. The valve modeled is different from what exists on the test-stand and thus results in different predictions of superheat that do not necessarily match the measurements. However, the valve model and hence the overall the system model, is physically, thermally and thermodynamically consistent and differences between model predictions of superheat and measurements are not considered to be of serious concern in terms of viability of the system model.

The heat-exchangers are modeled using the moving-boundary formulation outlined by Bendapudi et al [2004]. The evaporator is presumed to operate either with fully two-phase vapor, or with a two-phase evaporating and a superheated zone. The condenser is modeled to operate in three modes – fully superheated, superheated and condensing, superheated, condensing and sub-cooled. Both heat-exchangers are treated as concentric tubes-in-tube in counter-flow arrangement. The refrigerant flows through the outer tube and the water through the inner tubes.

The individual component models are combined into a system model. The system states are integrated forward in time, given an initial condition in the form of system pressures, exit enthalpies and temperatures of the tube material and water in the different phase zones, and inputs of evaporator and condenser water temperatures and flow-rates and a chilled water set point temperature. The compressor equations are solved first and the flow-rate and exit enthalpy are determined from current values of system pressures, evaporator exit condition and chilled water temperature error. This is followed by the valve equations from which the valve flow rate is determined, again from current condenser and evaporator exit conditions. Knowing the compressor and valve flow-rates and compressor exit enthalpy, the condenser equations are integrated forward in time by 1 sec, using externally specified water flow-rate and entering temperature. The evaporator equations are similarly integrated forward in time by 1 sec using the valve flow rate and compressor flow rates. The enthalpy of refrigerant entering the evaporator is corrected by adding the heat loss from the motor and transmission. When both the heat exchangers are solved, a new set of system pressures and exit enthalpies are obtained, which can then be used to re-compute the compressor and valve conditions. In this manner, the cycle simulation is repeated. The external boundary conditions are updated every 10s, consistent with the sampling rate of measurements. The heat exchanger equations are integrated forward in time using an explicit Euler method with a one-step correction (Press et al [2000]) and a fixed time step size. This method ensures that the integration algorithm is second order accurate in time and yet avoids the iterative computations of a fully implicit scheme. All other differential equations in the system are integrated using an explicit Euler method with fixed time steps. The compressor model is solved using a Secant method search for the exit enthalpy.

4. VALIDATION RESULTS AND DISCUSSION

The system model was validated using data from a 90-ton centrifugal chiller test set-up. The chiller operates with R134a and uses water as the coolant. The model was initialized to an equilibrium condition corresponding to a full-load steady state. By driving the system model with boundary conditions updated every 10s, system is driven through a sequence of 27 load changes triggered by different combinations of changes in evaporator and condenser return water temperatures and chilled water set-point temperatures (refer Comstock [1999] for the experimental data collection). Simulation results of the model compared with the measurements are shown below for the 27 steady-state conditions as well as for a load change caused by a 2.78^{o}C drop in the chilled water set-point temperature, a 12.64^{o}C increase in evaporator water return temperature and a 1.48^{o}C increase in condenser water return temperature.

Figs. 3-6 show the models ability to capture the correct steady-state conditions accurately over a wide range of conditions. The controller on the test-stand restricts the motor power from exceeding a pre-set limit at high capacities. This feature in the model prevents the compressor from being able to deliver the throughput necessary to match the high capacities. This is the reason for the under-prediction in evaporator capacities at the high end. Figs.7-10 present the predicted system pressures and capacities and the measured responses during a transient triggered by simultaneous changes in chilled water set-point as well as evaporator and condenser return water temperatures. The chilled water temperature is seen to quickly follow the changed set-point. Although the evaporator pressure predicted is quite close to the measurements, it is not as well predicted as the condenser pressure. This is believed to be a consequence of approximating the expansion device model as a single thermostatic device to represent the cascaded pilot-driven main valve arrangement that exists on the system. The unaccounted non-linearities in the original arrangement and the approximated valve dimensions used in the valve model result in a different measured evaporator pressure response. A consequence of this is that the predicted superheat (not shown) also differ from the measurement. Also observed is a variation in some steady-state motor power predictions (Fig. 5) particularly during low and high load operation. One possible reason for this is the linear assumption made in the control action. This approach is reasonable in the mid-capacity range, but at extreme loads, the flow-rate through the compressor depends non-linearly on the inlet-guide vane position. The transient response however, is independent of this and is seen to match the measured power well (Fig. 9). Fig. 10 shows the predicted and measured sub-cooling versus time. During model execution, the total refrigerant charge required some adjustment (reduction) from the value used in the test stand during the measurements. This was necessitated by the fact that the liquid line, which contains a significant quantity of liquid refrigerant, was not included in the model. In addition, the assumption of homogenous two-phase conditions in the heat exchangers results in an under-prediction of refrigerant. The charge used in the system model was adjusted until the predicted sub-cooling matched the measurements.

The model was implemented in C++ and executed on a 1.8GHz processor computer in a Microsoft Windows environment. Execution speeds were found to be consistently close to half real time throughout the simulation that spanned about 14 real hours of data. The construction of the model allows for further improvements in speed, such as parallelizing, with no loss in accuracy.

5. CONCLUSIONS

A dynamic model of a vapor compression centrifugal liquid chiller using moving-boundary formulations for the shell-and-tube evaporator and condenser is validated using available data from an experimental test stand. The model is found to predict the chiller performance well by capturing the correct steady-states as well as load-change transients. The system model is fast and sufficiently accurate for transient studies.

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FIGURES

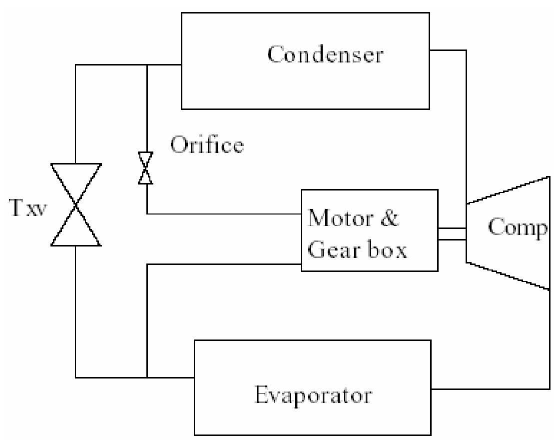

Fig. 1: Refrigerant flow-paths

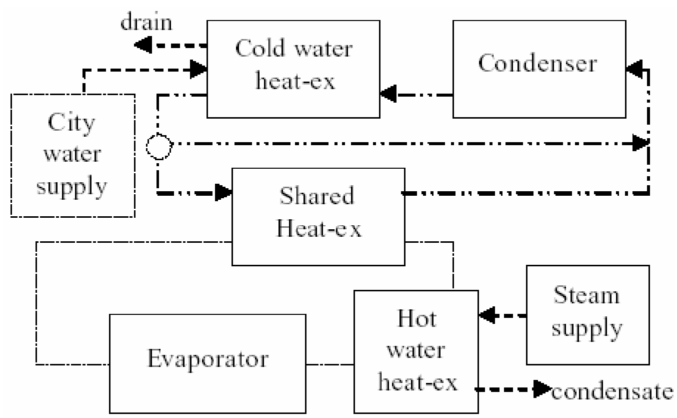

Fig. 2: Test-stand schematic

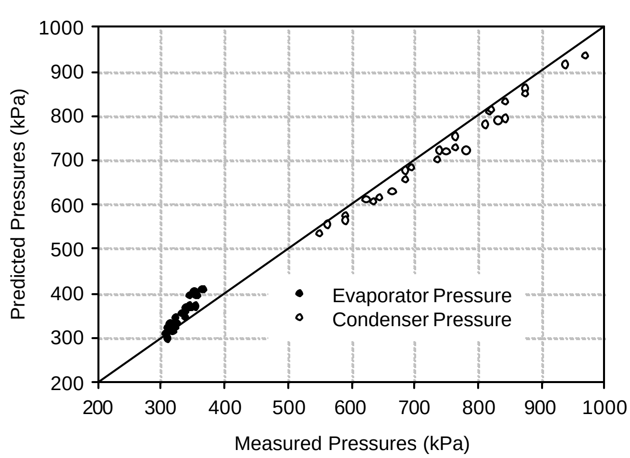

Fig. 3: Steady state pressures

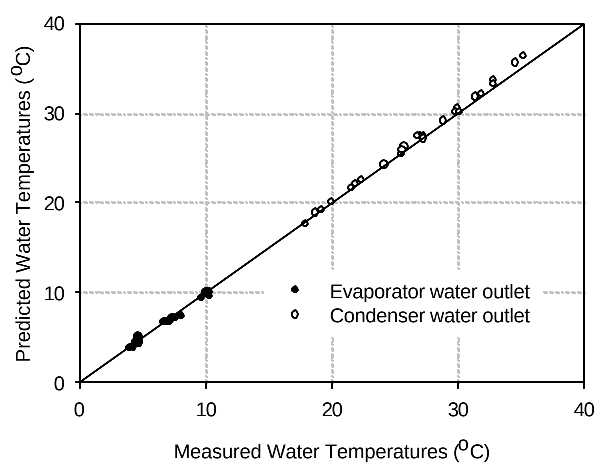

Fig. 4: Steady state leaving water temperatures

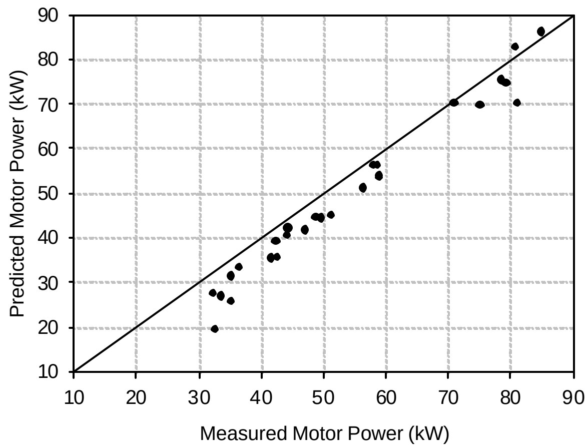

Fig. 5: Steady state motor power

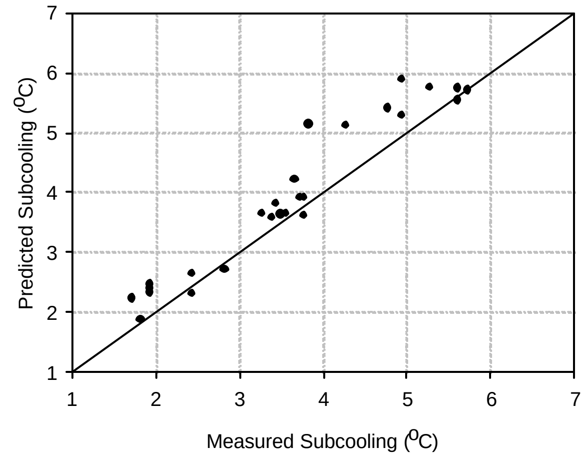

Fig. 6: Steady state sub-cooling

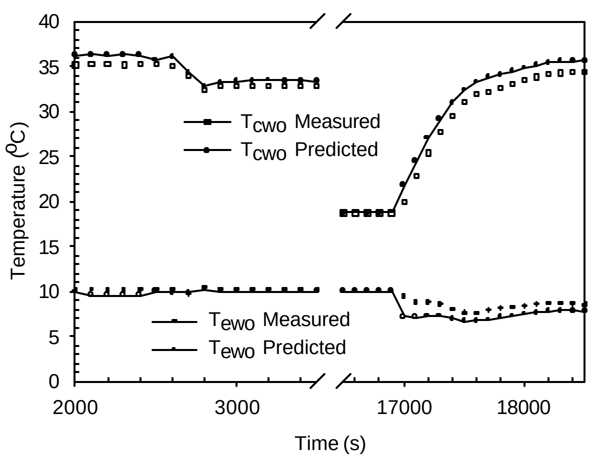

Fig. 7: Transient pressures variations

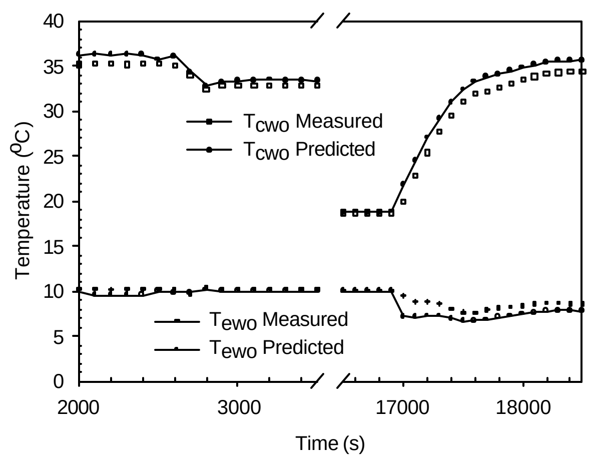

Fig. 8: Transient water temperatures

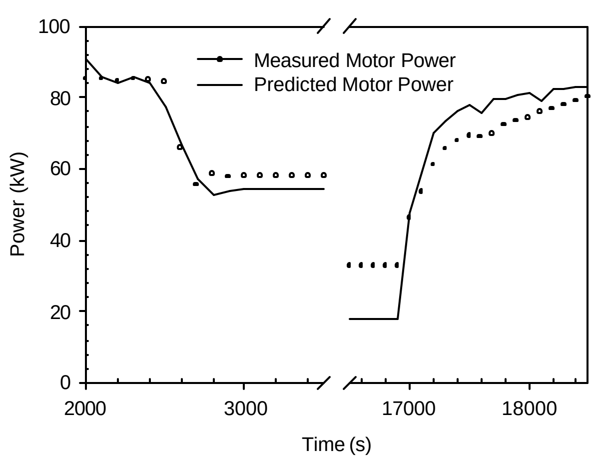

Fig. 9: Transient motor power

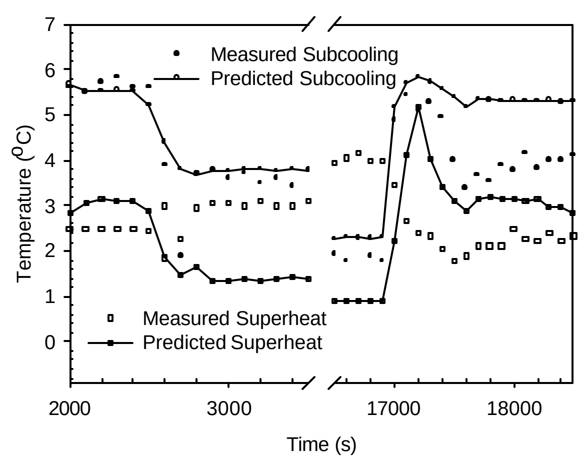

Fig. 10: Transient sub-cooling & superheat