Dynamic Simulation of Liquid Chillers

I. N. Grace
Brunel University

S. A. Tassou
Brunel University

DYNAMIC SIMULATION OF LIQUID CHILLERS

I N Grace, S A Tassou

Department of Mechanical Engineering
Brunel University
Uxbridge
Middlesex UB8 3PH
United Kingdom
Tel.: +(44) 1895 274 000 Ext. 2718
Fax: +(44) 1895 256 392
E-mail address: mesring@brunel.ac.uk

ABSTRACT

The impact of refrigeration systems on the environment can be reduced by: a) the use of alternative refrigerants which are less harmful to the environment and b) the optimisation of systems and control strategies to deliver increased levels of energy efficiency. Mathematical modelling offers the opportunity to test the performance of systems under different operating conditions and with alternative refrigerants. Dynamic models allow comparison of both transient and steady-state behaviour and this is of particular importance for liquid chillers since these systems can operate under transient conditions for long periods.

This paper covers the development of a general dynamic model for the simulation of liquid chillers. The reasons for mathematical modelling are discussed and the paper includes brief descriptions of the system component models including a semi-hermetic reciprocating compressor and thermostatic expansion valve as well as a shell-and-tube evaporator and condenser. The integration of the component models into the overall solution for a vapour compression refrigeration system is illustrated. The paper demonstrates the application of the model to predict the steady-state and dynamic performance of water-to-water chillers under a number of different operating conditions. Predicted results are compared to experimental data from a liquid chiller test rig to demonstrate the accuracy of the model.

NOMENCLATURE

A	area
c	specific heat capacity
H	heat transfer coefficient
h	enthalpy
m	mass
$\dot{m}$	mass flow rate
$\dot{Q}$	heat transfer rate
R	thermal resistance
T	temperature
t	time
u	velocity
V	volume
$\dot{W}$	work transfer rate, power
x	length

Greek symbol

ρ	density

Subscripts

a	ambient
ev	evaporator outlet vapour
ew	evaporator outlet wall
hx	heat exchanger wall
pw	phial wall
r	refrigerant
s	secondary fluid, coolant

INTRODUCTION

The impact of refrigeration and air-conditioning systems on the environment can be reduced by the use of new refrigerants which are less harmful to the atmosphere and the optimisation of systems and control strategies to deliver increased levels of energy efficiency. Theoretical assessment of alternative refrigerants and system optimisation can be carried out using a validated mathematical model. Dynamic models allow examination of both steady-state and transient performance and this is of particular importance as the operation of many refrigeration systems and chillers is transient in nature (Browne and Bansal, 1998).

A large number of mathematical models have been developed for vapour compression refrigeration and heat pump systems although the bulk of these have been steady-state treatments (Browne and Bansal, 1998). Literature detailing liquid chiller models is more limited. This paper presents the development of a dynamic model for the simulation of a liquid chiller and compares the predicted results to experimental data.

MATHEMATICAL MODEL

Heat Exchanger Models

The modelling technique used for the two heat exchangers was based on the fundamental equations describing the laws of conservation for physical systems. The principle of the conservation of mass can be written,

$$\frac{\partial \rho}{\partial t}+\frac{\partial \rho u}{\partial x}=0 \tag{1}$$

where the expression represents unsteady, one-dimensional mass conservation for a compressible fluid. The principle of the conservation of energy for a compressible fluid in one dimension can be given by,

$$\frac{\partial \rho h}{\partial t}+\frac{\partial \rho u h}{\partial x}+\dot{Q}=0 \tag{2}$$

The refrigerant, heat exchanger wall and coolant zones were divided into a series of discrete control volumes and the conservation equations applied to each volume. Note that, for the coolant, the assumption of incompressibility eliminates the need for a mass flow or density equation and, similarly, there is no requirement for a density equation for the solid wall.

The conservation equations can then be integrated with respect to time and distance over each control volume to produce the discretized form of each equation (Patankar, 1980, MacArthur and Grald, 1987). For the control volume illustrated in Figure 1, the discretized implicit form of the refrigerant mass conservation equation becomes,

$$[(\rho A)_P^{t+\Delta t}-(\rho A)_P^{t}]\Delta x+[\dot{m}_w^{t+\Delta t}-\dot{m}_e^{t+\Delta t}]\Delta t=0 \tag{3}$$

For the conservation of energy, the discretized equation for the refrigerant is as follows,

$$[(\rho A h)_P^{t+\Delta t}-(\rho A h)_P^{t}]\Delta x+[(\dot{m}h)_w^{t+\Delta t}-(\dot{m}h)_e^{t+\Delta t}]\Delta t+[(HA)(T_r-T_{hx})_P^{t+\Delta t}]\Delta t=0 \tag{4}$$

The upwind scheme can be used to approximate the enthalpy terms at the control volume interfaces by values at the volume centres. For the coolant, the discretized equation is written,

$$[(\rho AcT)_P^{t+\Delta t} - (\rho AcT)_P^t]\Delta x + [(\dot{m}cT)_e^{t+\Delta t} - (\dot{m}cT)_w^{t+\Delta t}]\Delta t + [(HA)(T_s - T_{hx})_P^{t+\Delta t}]\Delta t = 0 \quad (5)$$

The conservation of energy equation for the heat exchanger wall is given by,

$$[(\rho AcT)_P^{t+\Delta t} - (\rho AcT)_P^t]\Delta x + [(HA)(T_{hx} - T_s)_P^{t+\Delta t}]\Delta t + [(HA)(T_{hx} - T_r)_P^{t+\Delta t}]\Delta t = 0 \quad (6)$$

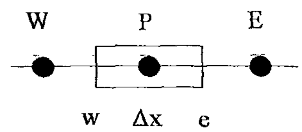

Figure 1 Control volume

An iterative approach is used to determine the heat exchanger pressure by comparing the outlet mass flow rate to a boundary condition. The solution process begins with an estimated heat exchanger inlet pressure. The refrigerant, coolant and wall equations are then solved for each control volume using an iterative process. Once the equations have converged, the refrigerant outlet mass flow rate is determined and compared to the target value provided as a boundary condition. The inlet pressure is modified and the process repeated until the two values converge to within a prescribed tolerance.

The simulated refrigeration system used shell-and-tube heat exchangers for both the evaporator and condenser, with refrigerant flowing on the shell-side in the condenser and on the tube-side in the evaporator. For the condenser, the refrigerant was assumed to flow vertically through the cooling tubes whilst the coolant flows horizontally through the coolant tubes. In the evaporator, the shell contains segmental baffles which introduce a crossflow pattern to the coolant flow path.

Expansion Valve Model

The simulation used an externally equalised thermostatic expansion valve (TEV) as the refrigerant flow control device. The remote phial is the key component in the operation of the TEV and this was modelled in detail. The zones modelled are shown in Figure 2.

The temperature of each zone was modelled using the first law of thermodynamics. For the evaporator outlet wall, the following equation was developed for the temperature, given the temperature at the previous time step and using the simple Euler method,

$$T_{ew}^{t+\Delta t} = T_{ew}^t + \frac{(HA)_{ev}(T_{ev} - T_{ew}) + (HA)_a(T_a - T_{ew}) - [(T_{ew} - T_{pw})/R]}{(mc)_{ew}}\Delta t \quad (7)$$

Similar equations were developed for the phial wall and the enthalpy of the phial refrigerant charge. The temperature and vapour pressure of the refrigerant charge can be found from the enthalpy and the refrigerant density in the remote phial.

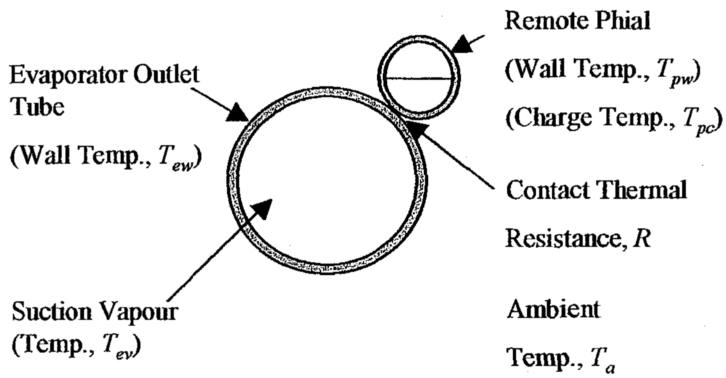

Figure 2 Remote phial zones

The orifice area was found using a force balance across the pressure diaphragm. At equilibrium, the force exerted by the remote phial must be equal to the combined forces of the evaporator outlet pressure and the superheat spring. The resulting spindle position is translated into an orifice area by a constant which represents the relationship between the spindle and the orifice area. The mass flow rate through the orifice area is then calculated using the Bernoulli equation for flow through an orifice plate.

Compressor Model

A dynamic model of a semi-hermetic reciprocating compressor was developed. The model was based on the application of the first law of thermodynamics to a number of discrete regions of the compressor. The regions are then modelled as lumped parameter volumes in which spatial variations are neglected.

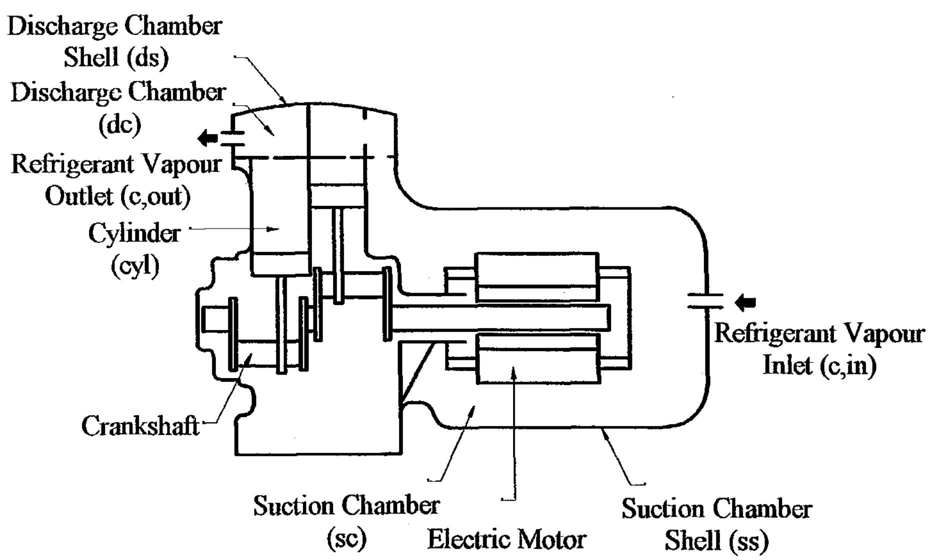

Figure 3 Compressor zones

For an open system, the first law for unsteady flow with a uniform state may be approximated by,

$$(\rho V)\ \frac{\partial h}{\partial t} = \dot{Q} - \dot{W} + \sum (\dot{m}h)_{in} - \sum (\dot{m}h)_{out} \tag{8}$$

where the control volume density is assumed to be constant, the work associated with the time rate of change of pressure is neglected and the energy is expressed in terms of enthalpy. This equation is used to determine the suction and discharge chamber enthalpy.

The compression process is modelled by the isentropic compression of refrigerant vapour from the inlet pressure to the outlet pressure. The resulting enthalpy increase is then modified by the isentropic efficiency to account for irreversibilities such as friction, pressure losses across the valve and cylinder wall heat transfer. The compressor mass flow rate is determined by the compressor speed and the cylinder displacement, modified by the volumetric efficiency. Temperatures are also calculated for the suction and discharge chamber shells and the cylinder wall.

System Model

The refrigeration system was modelled by linking the four component models to pass system conditions from one to another. The refrigerant state and the mass flow rate were used as input and output parameters to each component. The system model also handled user-specified information such as initial conditions and simulation time. The system performance was calculated in terms of cooling capacity, heat rejection, compressor power and COP and this data was exported to a file for analysis.

RESULTS

Steady-state Prediction

The system model was used to predict the steady-state performance of a water-to-water chiller. The results were compared to experimental measurements taken from a laboratory test rig. Figure 4 shows the experimental and predicted results for the system cooling capacity, where cooling capacity is calculated from a heat balance applied to the coolant. Experimental data points are shown together with trend lines for condenser coolant outlet temperatures between 30°C and 40°C. The model is able to predict the steady-state cooling capacity to within ±0.75kW across this range of data.

Both the simulation and the experiments show an increase in cooling capacity with evaporator coolant temperature. This is due to the increase in refrigerant vapour density brought about by an increase in evaporator temperature and leading to increased system mass flow rates. The model and experiments also show that the cooling capacity increases as the condenser water temperature is decreased. This is a result of the reduction in condenser refrigerant temperature and pressure leading to lower enthalpy at the condenser outlet and evaporator inlet and therefore an increase in the evaporator refrigerating effect.

Figure 5 shows the compressor power input against a range of evaporator and condenser coolant temperatures. The model is able to predict the experimental values to within ±0.5kW. Power consumption increases with evaporator temperature as the refrigerant density increases and the mass flow rate through the compressor is raised. The power requirement also increases with condenser temperature due to the increased pressure ratio and compressor discharge enthalpy.

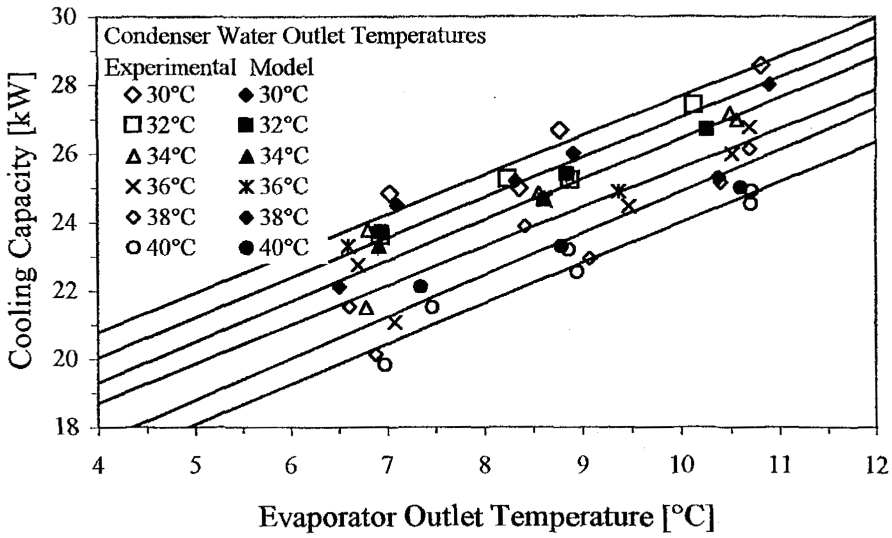

Figure 4 Steady-state cooling capacity

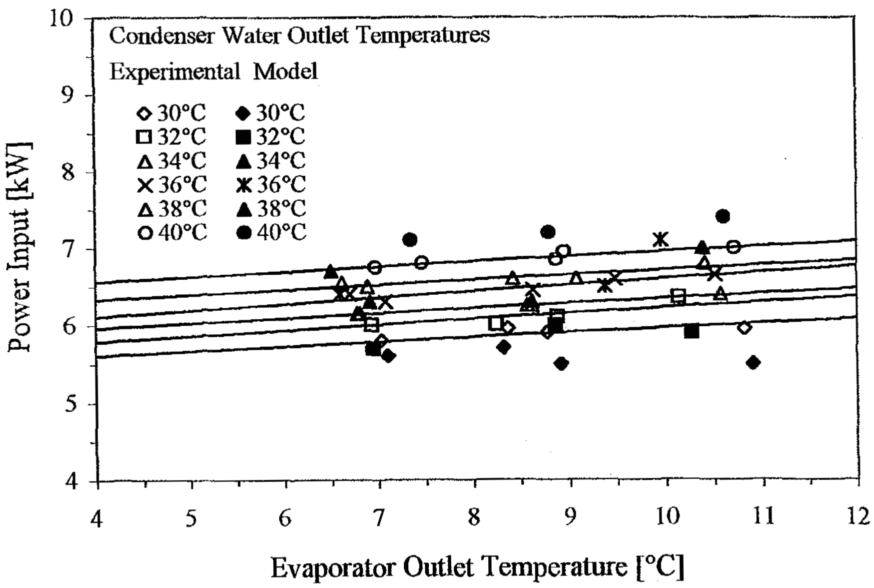

Figure 5 Steady-state power consumption

The experimental and predicted COP values for steady-state performance are shown in Figure 6. The model predicts the experimental values to within ±0.3. COP is shown to increase with evaporator temperature as the pressure ratio and compressor power consumption is reduced. The COP decreases with condenser temperature as the pressure ratio and power consumption is increased and the cooling capacity is decreased.

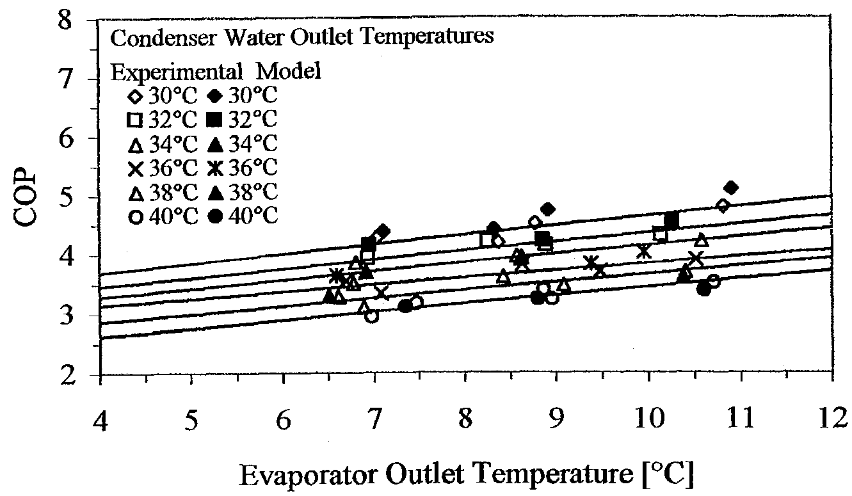

Figure 6 Steady-state COP

Start-up Prediction

The model was then used to simulate the dynamic start-up performance of a water-to-water chiller. Figure 7 shows the experimental and predicted pressure responses. It can be seen that the model tends to predict higher start-up condenser and evaporator pressures than those measured experimentally. The model also predicts a faster convergence to the steady-state than the experimental data.

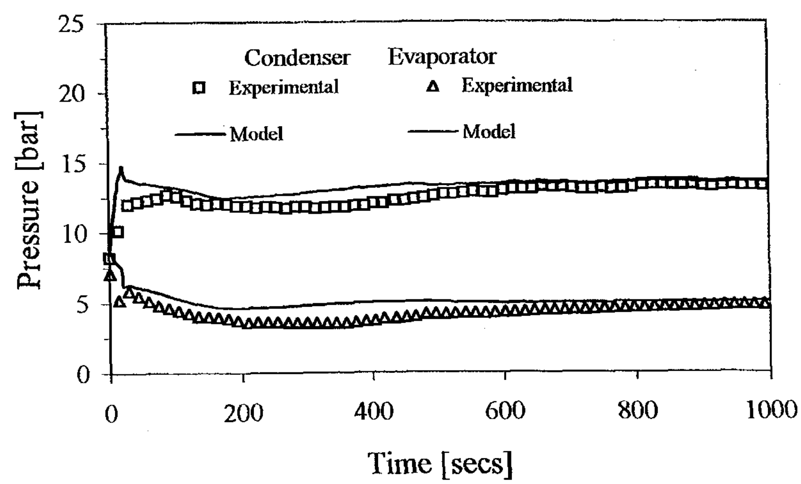

Figure 7 Start-up refrigerant pressures

The experimental and predicted coolant temperatures are shown in Figure 8. The simulated condenser outlet temperature is shown to follow the experimental data after predicting a peak at 20s in the same way as the condenser pressure. The predicted evaporator temperature is shown to model the experimental values closely. The system cooling capacity and heat rejection were calculated using heat balances applied to the coolant in each heat exchanger. The accuracy of the system performance prediction was therefore dependent upon the ability of the model to predict the coolant outlet temperatures.

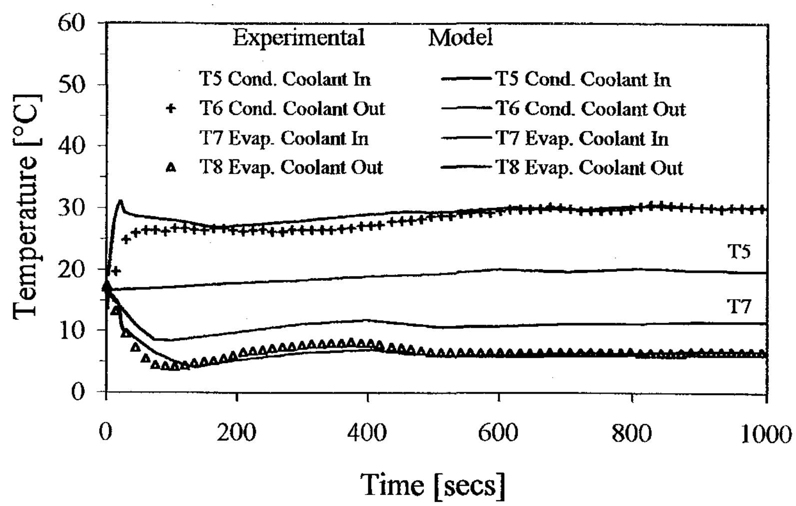

Figure 8 Start-up coolant temperatures

CONCLUSIONS

A detailed dynamic model of a liquid chiller has been developed. The evaporator and condenser are modelled using a distributed parameter technique to allow detailed investigation of the system behaviour. The compressor is modelled using a "lumped" parameter approach and the expansion valve is simulated by a simple orifice flow model and a detailed thermodynamic model of the remote phial. The heat transfer coefficients used are not specific to any one refrigerant allowing simulation with any refrigerant without modification. The component models are integrated by a central routine which allows the solution of each component model sequentially.

The model can be used for the simulation of both the steady-state and the dynamic performance of liquid chillers. The applications of the model include system optimisation, the development of new control strategies and the evaluation of alternative refrigerants. The long execution times required prevent the model's use in real-time applications. Future work includes the simplification of this model to increase execution speed and enable real-time applications such as on-line fault detection and diagnosis.

REFERENCES

Browne, M.W. and Bansal, P.K. 1998. *Challenges in modelling vapor-compression liquid chillers*, ASHRAE Transactions, 1998, Vol.104, Part 1A, No.4141, pp.474-486.

MacArthur, J.W. and Grald, E.W. 1987. *Prediction of cyclic heat pump performance with a fully distributed model and a comparison with experimental data*, ASHRAE Transactions, 1987, Vol.93, Part 2, No.NT-87-05-2, pp.1159-1178.

Patankar, S.V. 1980. *Numerical heat transfer and fluid flow*, Hemisphere Publishing Corporation, USA.