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AN INSIGHT INTO THE APPLICATION OF THE NTU-ε APPROACH FOR MODELLING VAPOUR-COMPRESSION LIQUID CHILLERS

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ABSTRACT

This paper presents a steady-state model that is useful for predicting the performance of centrifugal liquid chillers over a wide range of operating conditions. The model employs an elemental NTU-ε methodology to model both the shell-and-tube condenser and the flooded evaporator. The approach allows the change in heat transfer coefficients throughout the heat exchangers to be accounted for, thereby improving the accuracy of the simulation model. The model requires only those inputs that are readily available to the user (ie. condenser inlet water temperature and evaporator water outlet temperature). The outputs of the model include system performance variables such as the compressor electrical work input and the coefficient of performance (COP). The model is validated with data from a 450 kW open-drive centrifugal chiller where the agreement is found to be within ±10%.

NOMENCLATURE

A	Surface area	m^2
C_{min}	Minimum heat capacity	$kW.K^{-1}$
C_{max}	Maximum heat capacity	$kW.K^{-1}$
C_{pr}	Refrigerant specific heat	$kJ.kg^{-1}K^{-1}$
COP	Coefficient of performance	
h	Specific enthalpy	$kJ.kg^{-1}$
$\dot{m}_r$	Refrigerant mass flow rate	$kg.s^{-1}$
N	Number of rows in tube bundle	
NTU	Number of transfer units	
p	Pressure	kPa
p_{frac}	Pressure drop fraction	
$\dot{Q}$	Heat transfer rate	kW
R	Ratio of heat capacities	
T	Temperature	K
U	Overall heat transfer coefficient	$kW.m^{-2}K^{-1}$
ε	Heat exchanger effectiveness	

Subscripts

act	Actual
cond	Condenser
ci	Cold fluid inlet
co	Cold fluid outlet
desuper	Desuperheating component
hi	Hot fluid inlet
ho	Hot fluid outlet
sp	Single phase component
2	Exit state of compressor
3	Saturated vapour state in condenser
4	Exit state of condenser

INTRODUCTION

Vapour-compression liquid chillers are widely employed in both commercial and industrial applications to provide chilled water for air-conditioning purposes. It is well known in the HVAC industry that these machines consume large amounts of energy and that they often operate under part-load conditions. A recent review of the open literature [1] highlighted a distinct lack of research pertaining to the modelling of vapour-compression liquid chillers. Of the models that had been developed, the majority were "black-box" or empirically based steady-state models.

This paper presents a steady-state model specifically for centrifugal liquid chillers. The current model is an extension of a previously developed physical model [2] which utilised an NTU-ε approach through a "bundle average" method for the evaporator and the condenser. While the change in heat transfer coefficients with operating conditions were accounted for, their variation within the heat exchangers was neglected. Other existing models [1] also have this limitation in their approach to modelling the heat exchangers. Therefore an "elemental" methodology is employed in the current model to account for the variation in the heat transfer coefficients *throughout* the heat exchangers with operating conditions on both the shell-side and the tube-side. Component models for shell-and-tube heat exchangers utilising elemental approaches have been developed by Webb et al. [3] and Gabrielii and Vamling [4] but have yet to be employed in a complete cycle simulation. Pressure drop in both the heat exchangers (ie. over the tube bundles) is also accounted for due to the effect it has on the refrigerant saturation temperature and hence the heat transfer coefficients. This approach has important implications for all simulations as it is physically more realistic and can be used to model refrigerant mixtures. It may also be important for dynamic modelling where the refrigerant charge inventory is important.

DETAILS OF THE CHILLER

The model is based around an open-drive centrifugal chiller. The chiller was installed on the basis that it would be able to supply the cooling needs of the building on the hottest summer days. This however means that most of the time the chiller is operating at part load conditions and hence running very inefficiently. The chiller employs R-11 as the working fluid and prerotation vanes are used as a means of capacity control which allows the machine to operate down to a claimed 10% of the rated full load capacity (450 kW). These vanes modulate in response to the leaving chilled water temperature. Shut-down occurs when the water temperature leaving the evaporator is around 4°C, although design conditions are between 6-7°C. The design point for the condenser water inlet temperature is about 25°C. Shell-and-tube type heat exchangers are used for both the evaporator and condenser where the water flows through the tubes while the refrigerant boils or condenses on the outside of the tubes. The methodology for modelling various parts of the chillers is discussed in the following sections.

HEAT EXCHANGER MODELLING

Both heat exchangers are modelled using an elemental NTU-ε method. The basic principle of this approach is to divide both the tube-side region (ie. water) and the shell-side region (ie. refrigerant) into elements to better predict the heat transfer. This requires the length of each tube to be divided into an arbitrary number of elements and that the tube bundle be divided into elements dictated by the number of tube rows in the bundle. Figure 1 shows a schematic of the methodology. As the water enters the heat exchanger it will either be cooled (as for the evaporator) or heated (as for the condenser). This change in temperature from the entry to the exit of the heat exchanger alters the temperature gradient between the refrigerant and water-sides and hence has a large effect on the heat transfer coefficients. By dividing the heat exchanger into elements, this effect on the heat transfer can be more accurately modelled. Also as the refrigerant enters the tube bank, pressure drops resulting from drag and momentum losses cause the local saturation temperature of the refrigerant to vary throughout the heat exchanger, hence affecting the heat transfer coefficients. Once again the row-by-row formulation allows these changes to be accounted for, increasing the accuracy of the simulation.

When modelling the heat exchangers the following assumptions were made:

1. The refrigerant entering both the evaporator and the condenser is evenly distributed over the length of the tube bundle with homogeneous properties.
2. The thermodynamic properties of both the water and the refrigerant remain constant within an element.
3. The change in saturation temperature of the refrigerant due to pressure drops is assumed to occur immediately after the exit from the element ie. this change in temperature is not considered to affect the heat transfer within the element itself.

The effectiveness of any heat exchanger is defined as the ratio of actual heat transfer that occurs in the heat exchanger to the maximum heat transfer that could be obtained in an infinitely long counterflow heat exchanger. Following Figure 1, this can be written as:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \qquad \text{where} \qquad \dot{Q}_{max} = C_{min}\left(T_{hi} - T_{ci}\right) \tag{1}$$

The actual heat transfer that would occur in the heat exchanger (or in any element of the heat exchanger) can be calculated using:

$$\dot{Q}_{act} = \varepsilon . C_{min}\left(T_{hi} - T_{ci}\right) \tag{2}$$

For a pure refrigerant it can be assumed that the coolant is the fluid with the minimum heat capacity as the refrigerant that is undergoing a phase change appears to have a very large heat capacity. Therefore the effectiveness was found from:

$$\varepsilon = 1 - e^{-NTU} \qquad \text{where} \qquad NTU = \frac{UA}{C_{\min}} \tag{3}$$

although a more correct formulation would be to use an equation for a crossflow condition which uses the fluid with the minimum heat capacity to calculate the actual heat transfer. This is especially the case for refrigerant mixtures where the refrigerant saturation varies significantly during the condensation and boiling processes. These equations were employed to find the total heat transfer by summing the heat transfer in all of the individual elements for both the evaporator and the condenser. As an example, the methodology for simulating the condenser (similar to the evaporator) will be explained. With a given inlet water temperature the heat transfer in the first pass is calculated starting with the sum of the elements in the first (top) tube row and continuing down the bundle for all of the rows in the first pass. As the refrigerant passes over each tube row, the pressure drop is calculated from a user specified fraction and apportioned evenly over each row as:

$$\Delta p = \frac{p_{frac} \cdot p_{cond}}{N} \tag{5}$$

and the new saturation temperature is calculated from property routines. The heat transfer in the second pass is then calculated in a similar fashion except that the inlet water temperature to each row is now the average exiting water temperature from all of the tube rows in the first pass. Appropriate correlations [5,6] are employed to find the heat transfer coefficients for the water and the boiling and condensing refrigerant. The overall heat transfer coefficient is then found and the number of transfer units (NTU) and the effectiveness is calculated from equation (3) using the tube surface area of the element.

The traditional problem when modelling the condenser using "bundle average" methods is how to account for the desuperheating of the incoming refrigerant. The elemental approach significantly simplifies this problem as it is possible to utilise single phase heat transfer correlations for the superheated refrigerant. The amount of desuperheating can be found from

$$\dot{Q}_{desuper} = \dot{m}_r\left(h_2 - h_3\right) \tag{6}$$

If the heat transfer calculated using the single phase heat transfer correlations is greater than $\dot{Q}_{desuper}$, the heat transfer in the single phase region could be neglected with the heat transfer being found using correlations for the condensing region. However, if the heat transfer is less than that required for condensation to occur, the refrigerant temperature is adjusted by

$$\Delta T = \frac{\dot{Q}_{sp}}{\dot{m}_r C_{pr}} \tag{7}$$

and the process is repeated on the next tube row.

SOLUTION METHODOLOGY OF THE MODEL

The model arrives at a steady-state solution through an iterative solution process. The inputs to the model include the following variables:

1. The condenser inlet water temperature and condenser water mass flow rate.
2. The evaporator water outlet temperature and evaporator mass flow rate.
3. Degree of superheat at evaporator outlet.
4. Degree of subcooling at condenser outlet.
5. Compressor motor efficiency.
6. Compressor shell heat loss fraction (if desired).
7. Condenser and Evaporator pressure loss fractions (if desired).
8. The evaporator capacity.

The model guesses certain values of the refrigerant state (condenser refrigerant outlet state and evaporator refrigerant outlet state based on the given coolant temperatures) and proceeds to evaluate the complete thermodynamic cycle of the chiller. Knowing the pressure drop fraction in the condenser and evaporator allows the exit pressure from the compressor and inlet pressure to the evaporator to be found respectively. Empirical equations based on manufacturers and experimental data are then employed to calculate the isentropic efficiency of the compressor based on the pressure rise over the compressor (also a constant motor efficiency of 95% is assumed as is an arbitrary shell heat loss fraction based on the compressor work input). This allows the state of the refrigerant at the exit of the compressor to be found. The evaporator is then modelled with the calculated cooling capacity compared to the specified evaporator capacity. The evaporator refrigerant outlet temperature is then adjusted using both secant and bisection convergence techniques and the process repeated until the two values are within a specified tolerance. A similar process is used to model the condenser where the calculated capacity is compared with the actual condenser capacity given by

$$\dot{Q}_{cond} = \dot{m}_r\left(h_2 - h_4\right) \qquad (8)$$

The condenser refrigerant outlet temperature is then adjusted and the process is repeated until the system heat balance is satisfied at which time a steady-state solution has been found. The performance parameters such as the COP and condenser capacity are then calculated.

RESULTS

Due to the transient nature of the chilling system only a relatively small data set was obtained under steady-state conditions. Data for evaporator capacity, compressor electrical work input, and evaporator water inlet and outlet temperatures were collected. The condenser water inlet temperature was assumed to be at 25°C in all simulations. Figures 2-5 highlights the effect that the number of elements have on the accuracy of the simulation and the importance of accounting for the variation of the heat transfer coefficients. Figure 2 shows the effect of the number of elements on the average overall heat transfer coefficient for the evaporator. It was simulated for a cooling capacity of about 230 kW, an evaporator water outlet temperature of 281.5 K, and a refrigerant pressure drop of 10% of the evaporator pressure. The average overall heat transfer coefficient varied by about 10% depending on the number of elements used in the simulation although this would vary depending on the boiling correlation employed. Figure 3 shows the variation of the overall heat transfer coefficient with position along the tubes for the first five rows of a condenser modelled with smooth tubes. It can be seen that the effect of condensate inundation was large and should not be neglected. It has be shown [5] that enhanced tubes employed in modern heat exchangers also have relatively large reduction in heat transfer coefficients due to inundation effects. Figure 4 shows the effect of the number of elements in both the condenser and evaporator on the COP for an evaporator capacity of 290 kW and an evaporator outlet water temperature of 282.3 K. It can be seen that the simulation results more closely approximated the actual values as the number of elements were increased. At around 150 elements the simulation differed from the actual COP by 10% and as the number of elements approaches 2500 the accuracy was increased to about 4.9%. It does however become a case of diminishing returns with the extra elements and computing time producing little extra accuracy. It can be seen in Figures 5 and 6 that the model predicted the majority of values for COP and compressor electrical work input to within 10% for the chiller. While some degree of scatter was seen in the data, this was probably due to the fact that the condenser water inlet temperature (which plays a significant role in the performance) was taken be a constant 25°C when in fact it probably varied by ±2°C [7]. Also it appears that the model slightly underestimated the compressor electrical work input at low load conditions. This was probably due to the fact that there was little reliable data (under steady conditions) from which to correlate the compressor efficiency curve.

CONCLUSIONS

In this paper a new steady-state model for centrifugal liquid chillers has been presented. The model is based on physical laws and heat transfer coefficients that are uniquely applied using the NTU-ε methodology. The chiller can be simulated over a wide range of conditions and operating capacities which allows the part-load performance (the dominant operating characteristic in most chiller installations) to be studied. The model predicts the electrical work input to the compressor, and the coefficient of performance (COP) to within ±10% for the majority of operating conditions for an open-drive centrifugal chiller assuming a condenser water inlet temperature of 25°C. The model also demonstrated that the COP of centrifugal chillers increases with increasing load of the system. Also with slight modifications, the model lends itself to predicting the performance of refrigerant mixtures in vapour-compression liquid chillers as well as "system" simulation incorporating pump and fan work to allow optimum performance to be found.

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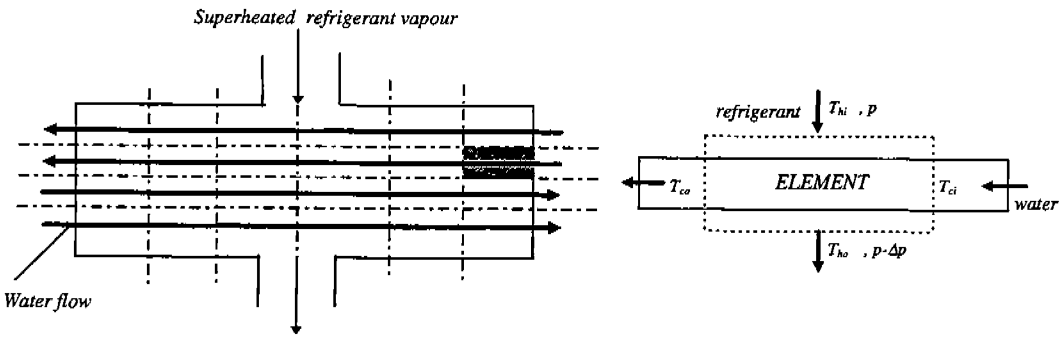

Figure 1: Schematic of the model formulation for the shell-and-tube condenser.

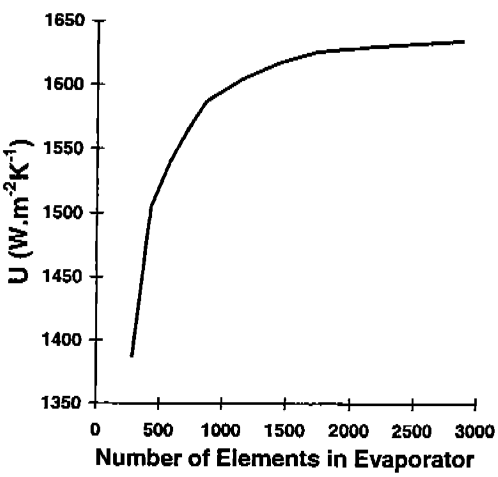

Figure 2: Plot showing number of elements vs. Overall heat transfer coefficient for the evaporator.

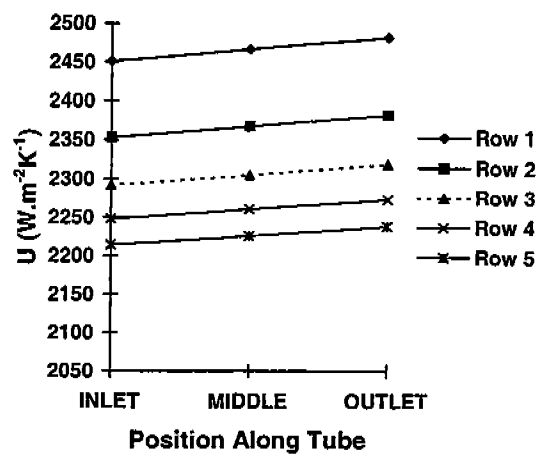

Figure 3: Plot showing the change in Overall heat transfer coefficient along the tube for the top five rows in the condenser.

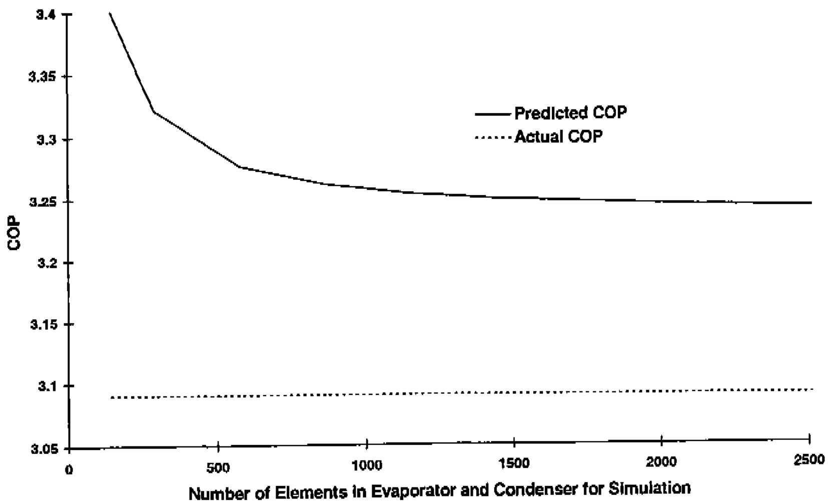

Figure 4: Predicted COP versus number of elements in condenser and evaporator simulation.

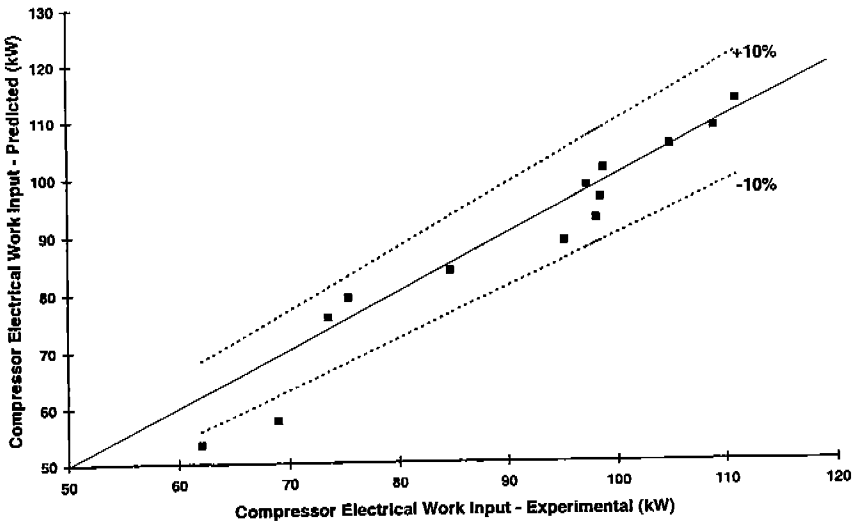

Figure 5: Actual compressor electrical work input versus predicted electrical work input for 450 kW centrifugal chiller.

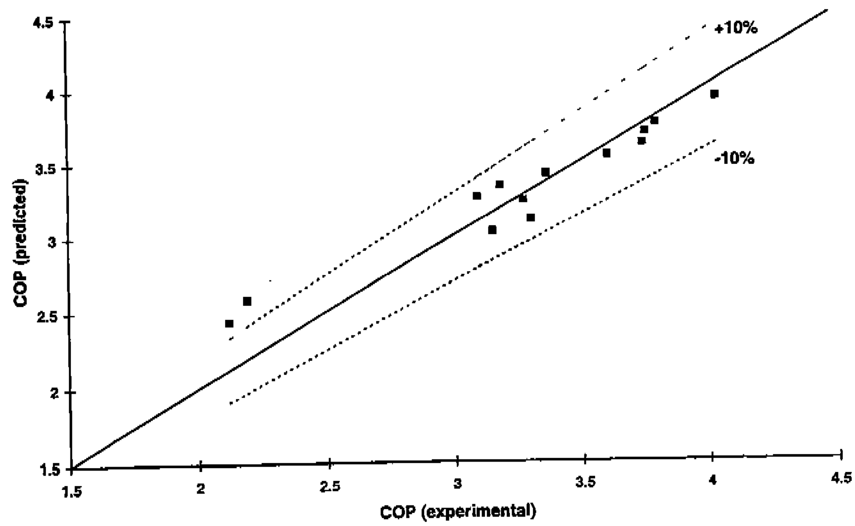

Figure 6: Actual COP versus predicted COP for 450 kW centrifugal chiller.