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ABSTRACT

Fluted tube-in-tube condensers are key components in advanced energy efficient water heating heat pumps. To optimise the design of heat pumps, there exists a need for a computer design tool that incorporates all the essential features of these heat exchangers. This paper describes a detailed model to simulate fluted tube refrigerant-to-water condensers. The model allows the surface area to be divided into any number of sections for which all the refrigerant and water properties can be evaluated. This allows for the extension of the model to simulate heat exchangers for cycles employing zeotropic refrigerant mixtures. The model is based on the effectiveness-NTU method and incorporates appropriate refrigerant-side heat transfer coefficients for the superheated, two-phase and sub-cooled regions given the detailed geometry of the heat exchanger. The model furthermore incorporates a simplified approach to simulate the influence of refrigerant charge that negates the need for a complicated void fraction model. The model is validated with the aid of results from independent tests on a commercial fluted tube heat exchanger.

INTRODUCTION

The geometry of a counter-flow fluted tube-in-tube heat exchanger is shown schematically in Figure 1. The condensing refrigerant flows in the annulus while the water being heated flows inside the inner fluted tube. Fluted tube condensers such as these are able to produce high heat transfer coefficients by enhancing the flow conditions on both sides of the inner tube wall. The water heat transfer coefficient is improved by micro-circulation of the water without a significant increase in the pressure drop. Heat transfer in the annulus is improved by two phenomena. In the first instance the condensate is drawn towards the corners of the channels by surface tension, clearing the remaining surface in contact with the hot gas. Secondly, the liquid phase experiences some micro-circulation, particularly towards the outlet side, leading to the replacement of cold liquid with hotter liquid next to the surface. This however leads to an increase in the pressure drop.

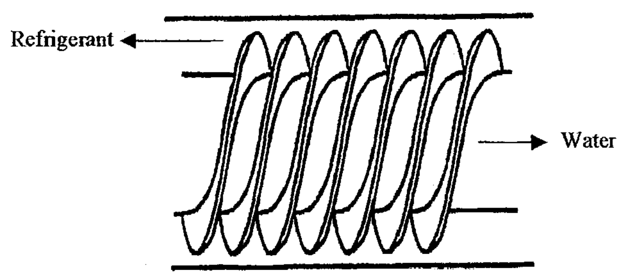

Figure 1 Geometry of fluted tube condenser.

In the condenser a counter-flow arrangement is used to obtain the maximum heat transfer. This also makes it possible to achieve outlet water temperatures above the condensing temperature. This is illustrated in Figure 2 where the inlet cold water is at T_{w1} and the hot outlet water is at T_{w4}. Note that T_{w4} is higher than the refrigerant condensing temperature, which varies between T_{r2} and T_{r3}. This paper describes a detailed simulation model for fluted tube water heating condensers that forms part of an integrated heat pump simulation and design program.

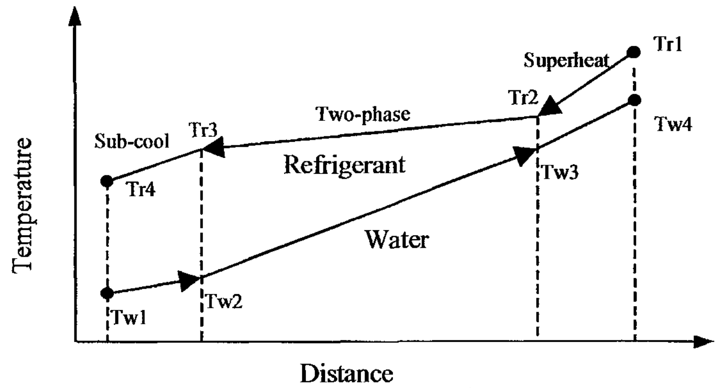

Figure 2 Temperature distribution in condenser.

THEORETICAL BACKGROUND

Fluted Tube Geometry

Figure 3 shows the parameters describing the heat exchanger geometry. The outer diameter of the annulus is designated as D_o and is simply equal to the inside diameter of the outer tube.

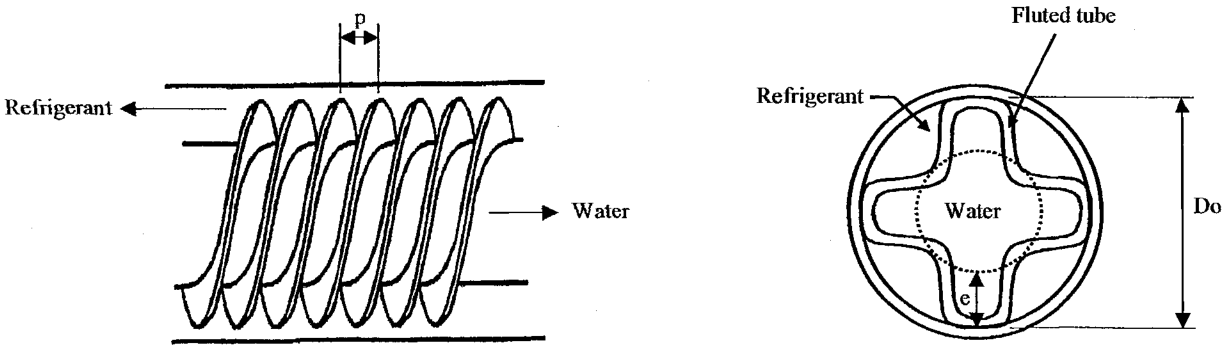

Figure 3 Parameters describing the fluted tube geometry.

Due to the complex shape of the inner fluted tube, the concept of "volume based diameters" is used rather than the conventional diameters associated with a circular cross-section. The volume based diameter is defined as the equivalent diameter of a circular cross-section that will result in the same enclosed volume inside the tube. The volume based inside diameter of the fluted tube (D_{vi}) is therefore calculated as

$$D_{vi} = \sqrt{\frac{4Vol}{\pi L}} \tag{1}$$

with *Vol* the volume enclosed inside the fluted tube and *L* the length of the fluted tube. The volume based outside diameter of the fluted tube (D_{vo}) is calculated as

$$D_{vo} = D_{vi} + 2t \tag{2}$$

with t the wall thickness of the tube. The cross-sectional flow areas of the annulus and the fluted tube can therefore be calculated directly from D_o, D_{vo} and D_{vi}.

The other important geometric parameters are the flute depth (e) and the flute pitch (p) which are defined in Figure 3. The flute depth and pitch are non-dimensionalised to obtain e^* and p^* respectively by dividing with D_{vi}. The helix angle (θ) is also required and is calculated as follows:

$$\theta = \arctan\left(\frac{\pi D_{vo}}{Np}\right) \tag{3}$$

with N the number of flute starts at any cross-section. The helix angle is also non-dimensionalised to θ^* by dividing by $\pi/2$.

Pressure Drop

Three different pressure drop regions exist namely (i) single-phase pressure drop for the water flowing inside the inner fluted tube, (ii) two-phase refrigerant pressure drop in the annulus as well as (iii) single-phase refrigerant pressure drop for the superheated refrigerant gas and sub-cooled refrigerant liquid in the annulus.

Inner fluted tube: Arnold and Christensen et al. (1993) have conducted extensive empirical work on flow inside fluted tubes and have developed correlations for the Darcy-Weisbach friction factor (see for instance Shames, 1992). The friction factor (f) is dependent on the Reynolds number (Re_w) and geometry as follows:

$\mathrm{Re}_w <= 1500$

$$f = 0.554\left(\frac{64.0}{\mathrm{Re}_w - 45.0}\right)\left(e^*\right)^{0.384}\left(p^*\right)^{-1.454+2.083e^*}\left(\theta^*\right)^{-2.426} \tag{4}$$

$\mathrm{Re}_w > 1500$

$$f = 1.209\left(\mathrm{Re}_w\right)^{-0.261}\left(e^*\right)^{1.26-0.05.p^*}\left(p^*\right)^{-1.66+2.033.e^*}\left(\theta^*\right)^{-2.699+3.67.e^*} \tag{5}$$

Re_w is based on the volumetric inside diameter (D_{vi}) and average water velocity.

Annulus: The annulus can effectively be simulated as a number of helical coils in parallel. A so-called 'friction enhancement ratio' is employed in both the single and two-phase regions to determine the pressure drop in the helical coils. The friction enhancement ratio (r_f) is defined as the ratio of the effective friction factors when comparing helical and straight tubes namely:

$$r_f = \frac{f_{helical}}{f_{straight}}. \tag{6}$$

For the calculation of the friction enhancement ratio, $f_{straight}$ is based on the standard single-phase correlation by Swamee and Jain (1976) for straight tubes, while $f_{helical}$ is calculated from the single-phase correlation put forward by Das (1993) for helical coils namely:

$$f_{helical} = 4\left[0.079\,\mathrm{Re}_v^{\,-0.25} + 0.075\left(\frac{D_{ho}}{d_{coil}}\right)^{0.5} + 17.5782\,\mathrm{Re}_v^{\,-0.3137}\left(\frac{D_{ho}}{d_{coil}}\right)^{0.3621}\left(\frac{e}{D_{ho}}\right)^{0.6885}\right]. \tag{7}$$

In the correlation above D_{ho} is given by the difference between the inside diameter of the outer tube (D_o) and the outside volumetric diameter (D_{vo}). d_{coil} is given by $D_{ho}/\sin\theta$. This single-phase enhancement ratio is assumed to also apply to the two-phase region. Therefore, once it has been determined, the pressure drop in the helical coils for both the single and two-phase regions can be calculated by obtaining the pressure drop in straight tubes ($\Delta p_{straight}$) and then multiplying it by r_f.

The two-phase pressure drop for straight tubes is dependent on the refrigerant quality and is based on the correlation by Traviss et al. (1973) which states that:

$$\Delta p_{straight} = 0.09\,\mathrm{Re}_v^{-0.2}\, x^{1.8}\left[1+2.85\left(\mu_x^{-0.1}\left(\frac{1-x}{x}\right)^{0.9}\rho_x^{0.5}\right)^{0.523}\right]^2 \frac{G^2}{\rho_v D_{ho}} \tag{8}$$

with G the mass flux. The refrigerant properties i.e. quality (x), density (ρ), and viscosity (μ) are determined at the average refrigerant temperature in the tube section being analysed. Re_v is based on the vapour properties of the refrigerant at the average temperature.

In the simulation model allowance was made for an additional enhancement factor (e_f) to account for the differences between standard helical coils and the actual fluted tube annulus. The total pressure drop in the annulus ($\Delta p_{annulus}$) was therefore calculated as:

$$\Delta p_{annulus} = e_f r_f \Delta p_{straight}\,. \tag{9}$$

The value of the e_f was determined by comparing simulated results with measured data for a commercial fluted tube heat exchanger and will be addressed later on in the paper.

Heat Transfer Coefficients

Three different heat transfer modes are applicable namely (i) single-phase heat transfer for the water flowing inside the fluted tube, (ii) single-phase heat transfer for the superheated refrigerant gas and sub-cooled refrigerant liquid in the annulus as well as (iii) two-phase condensing heat and mass transfer in the annulus.

Inner fluted tube: The inside heat transfer coefficient is also based on the empirical work by Arnold and Christensen et al. (1993). The Nusselt number (Nu) is dependent on the Reynolds number (Re) as follows:

$Re <= 5000$

$$Nu = 0.014\,\mathrm{Re}^{0.842}\left(e^*\right)^{-0.067}\left(p^*\right)^{-0.293}\left(\theta^*\right)^{-0.705}\mathrm{Pr}^{0.4} \tag{10}$$

$Re > 5000$

$$Nu = 0.064\,\mathrm{Re}^{0.773}\left(e^*\right)^{-0.242}\left(p^*\right)^{-0.108}\left(\theta^*\right)^{0.599}\mathrm{Pr}^{0.4} \tag{11}$$

The Prandtl number (P_r) is determined by $P_r = \dfrac{cp\mu}{k}$ and the inside heat transfer coefficient by $h_i = \dfrac{Nuk}{D_{vi}}$.

Annulus: According to the Chilton-Colburn analogy (see for instance Incropera & DeWitt, 1996) the heat transfer enhancement ratio (r_h) for simple geometries should be approximately equal to the friction enhancement ratio. Therefore, the Nusselt number for the annulus ($Nu_{annulus}$) is calculated in terms of the Nusselt number for straight tubes ($Nu_{straight}$) as:

$$Nu_{annulus} = e_h r_h Nu_{straight} \quad with\ r_h = r_f\,. \tag{12}$$

Allowance is again made for an additional enhancement factor (e_h) to cater for differences in the helical coil and fluted tube annulus as well as other deviations from the assumptions originally made in the friction-heat transfer analogy. The value of e_h was determined by comparison between the simulated and measured results and will be addressed later. It was again assumed that the single-phase enhancement ratios also apply to the two-phase region.

The calculation of $Nu_{straight}$ for the single-phase regions is based on the standard Dittus-Boelter correlation (Incropera & DeWitt, 1996) namely:

$$Nu_{straight} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}\,. \tag{13}$$

The two-phase heat transfer coefficient is based on the technique used by Shah (1979) where the liquid heat transfer coefficient (h_{liq}) is first determined with the aid of the Dittus-Boelter equation. This is then adjusted for the average quality of the refrigerant in the tube section being analysed as follows:

$$h_{tp} = h_{liq}\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{pr^{0.38}}\right] \tag{14}$$

with pr equal to the ratio of the local static pressure to the critical pressure of the refrigerant.

SIMULATION MODEL

The inputs to the condenser model are the fluted tube geometry, the refrigerant type, the refrigerant mass flow rate, the refrigerant inlet enthalpy where it leaves the compressor as well as the water mass flow rate and inlet water temperature. The refrigerant inventory is not simulated based on a detailed void fraction model such as the one found in Orth (1995). The approach used is to fix the fraction of condenser volume that is filled with sub-cooled liquid refrigerant. The rationale for this is that due to the high density of the liquid, the volume of sub-cooled liquid in the system will remain virtually unchanged over a wide range of operating conditions. Our measurements have proven this to be a sufficiently accurate approach, especially given the deficiencies of current void fraction models. For this approach, either the degree of sub-cooling or the fraction of the volume containing sub-cooled liquid must therefore also be specified as input. The simulation model makes use of refrigerant property tables that can also be extended for zeotropic mixtures. The simulation model is based on the algorithm shown in Figure 4.

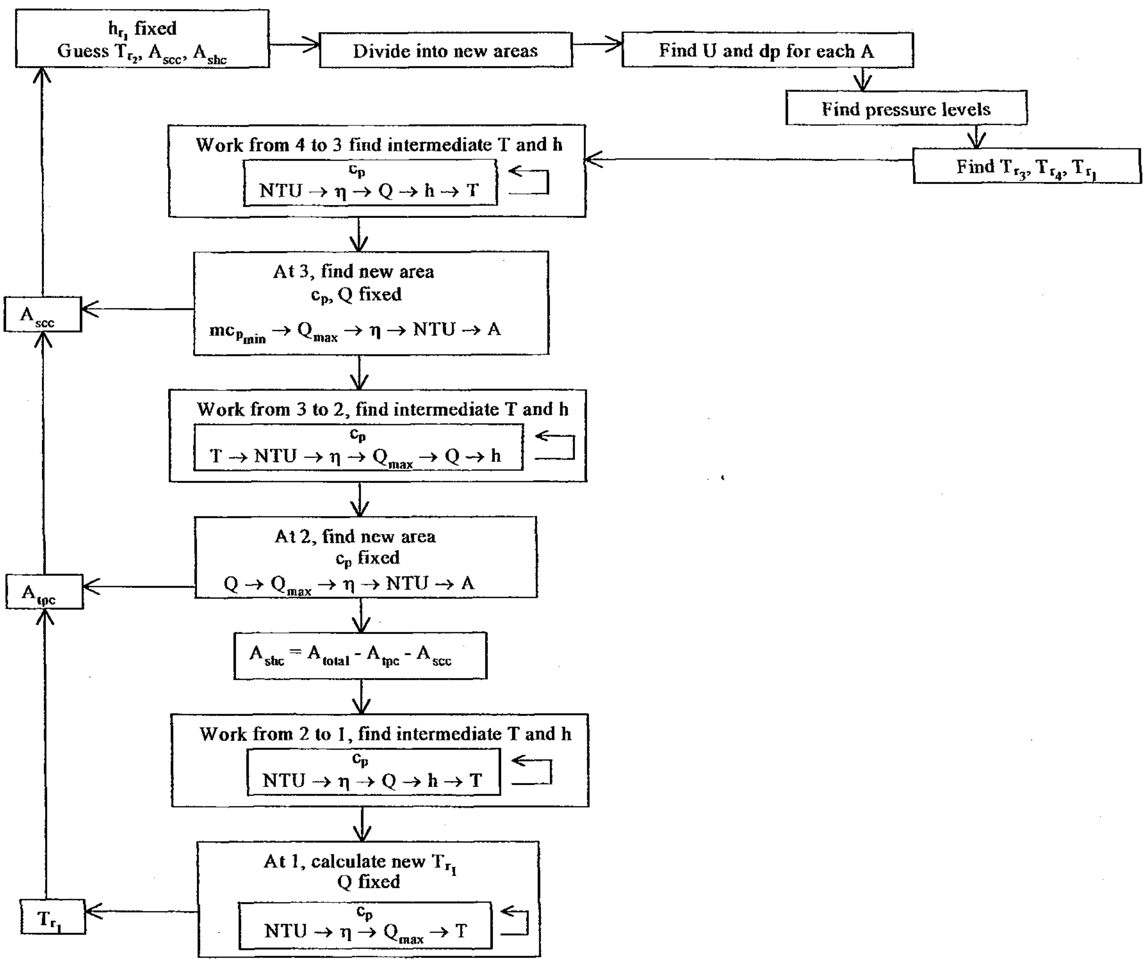

Figure 4 Flow diagram for the condenser simulation routine.

Initial guesses are made for the size of the heat exchanger sections associated with the superheated and sub-cooled regions as well as the condensing temperature (T_{r2}). The heat exchanger area is divided up into a specified number of finite increments each representing a section of tubing as shown in Figure 5. This allows for the simulation of a temperature glide in the two-phase region and therefore the extension of the model to simulate heat exchangers for cycles employing zeotropic refrigerant mixtures. The section areas are adjusted so that the phase interfaces coincide exactly with section interfaces. Given the geometry of each section, the heat transfer coefficients and pressure drops can now be determined using the correlations and enhancement factors described above. Once the pressure levels at each section interface are known, the inlet temperature, phase interface temperatures and outlet temperature can be obtained. Starting at the sub-cooled liquid side, each section is then solved iteratively to obtain a heat balance between the water and the refrigerant while the area of the section is adjusted to satisfy the effectiveness-NTU equation for counter-flow given the calculated heat transfer coefficient. Having worked through the whole condenser once, new superheat and sub-cooled areas as well as a new value for the condensing temperature are obtained. This whole procedure is then repeated until no further adjustments in areas or condensing temperature are required.

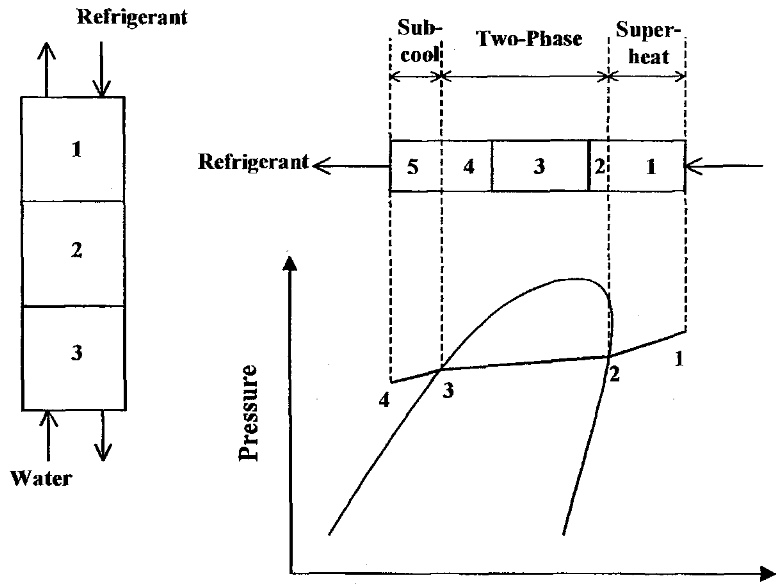

Figure 5 Pressure enthalpy diagram for the fluted tube condenser.

VERIFICATION

In order to obtain suitable values for the enhancement factors and to verify the simulation model, simulations were conducted for a commercial fluted tube heat exchanger for which results of independent tests were made available by the manufacturer. An optimisation routine was linked to the simulation model and used to obtain a single value for e_f and e_h resulting in the best fit between the empirical and simulated pressure drops and heat transfer rates for all the data sets. The resultant values of e_f and e_h are 2.052 and 0.685 respectively.

Figures 6 and 7 show the good comparison obtained between the simulated and measured results. The average difference between the simulated and measured pressure drops is 5.6 % and the maximum difference is 14.2 %. Figure 7 shows that the measurements covered a sufficiently wide range of heat transfer rates. The average difference between the simulated and measured heat transfer rates is 0.12 % and the maximum difference is 0.27 %.

The value of e_h = 0.685 implies that if the heat transfer coefficient for simple helical coils were used without the additional enhancement factor, the heat transfer rate would be over-predicted for the fluted tube annulus. This is probably due to the fact that part of the heat transfer area of the annulus (on the outside of the outer tube) is not in contact with the cooling medium as was the case in the tests conducted by Das from which the helical coil correlations were derived. This means that the heat transfer area of the analogous helical tube is effectively reduced.

The value of $e_f = 2.052$ implies that the pressure drop in the fluted tube annulus is much higher than that of a corresponding smooth helical tube. This is realistic since the manufacturing process is such that the annulus is no longer simply a smooth channel but rather contains many indentations and rougher areas along the flow path.

The results show that the approach followed here is valid although only a relatively small data set of measurements was used. A need therefore exists for extension of the validation and the development of suitable enhancement factor correlations that include the effect of the various fluted tube geometrical parameters.

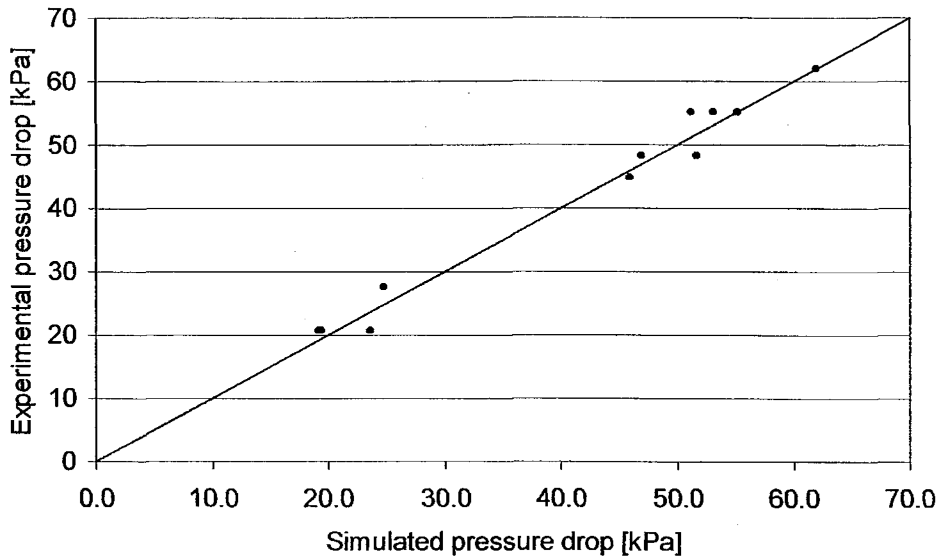

Figure 6 Simulated versus measured pressure drop.

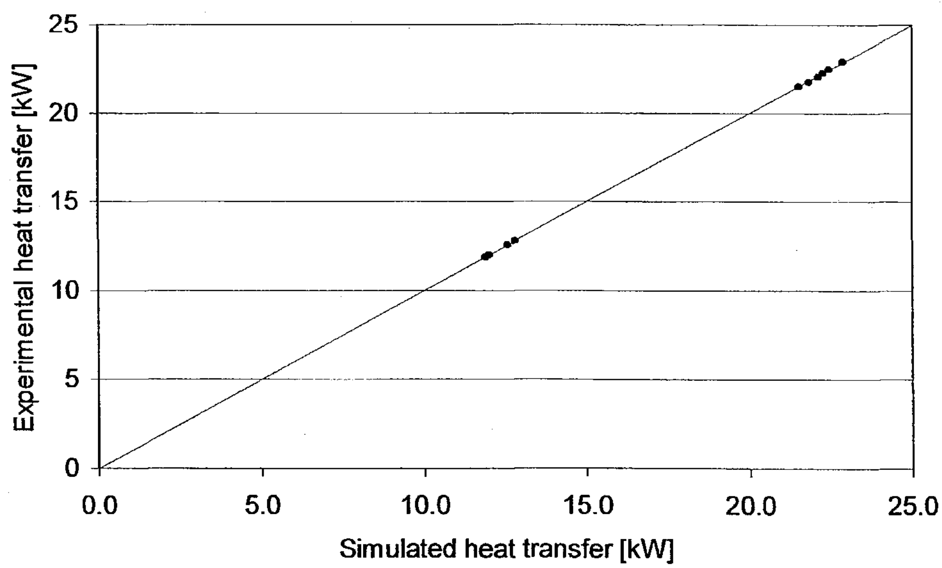

Figure 7 Simulated versus measured heat transfer rate.

CONCLUSIONS

This paper described a detailed model to simulate fluted tube refrigerant-to-water condensers. The model allows the surface area to be divided into any number of sections for which all the refrigerant and water properties can be evaluated. This allows for the extension of the model to simulate heat exchangers for cycles employing zeotropic refrigerant mixtures. The model is based on the effectiveness-NTU method and incorporates appropriate refrigerant-side heat transfer coefficients for the superheated, two-phase and sub-cooled regions given the detailed geometry of the heat exchanger. The model incorporates a simplified approach to simulate the influence of refrigerant charge that negates the need for a complicated void fraction model. The model was validated successfully with the aid of results from independent tests on a commercial fluted tube heat exchanger. A need was however identified for extension of the validation and the development of suitable enhancement factor correlations that include the effect of the various fluted tube geometrical parameters.

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