MODELLING OF TUBE AND FIN COILS WORKING AS EVAPORATOR OR CONDENSER

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ABSTRACT

In this paper a model for tube and fin coils working as evaporators or condensers is presented, including dehumidification process of humid air. A comparison between calculated and measured results for the outdoor-coil of an Air-to Water 20 kW Heat Pump is included. The flow through the coil is considered to be steady. The refrigerant flow inside the tubes is considered to be 1D. The basis of the numerical scheme is to de-couple the calculation of the fluid flows from each other, by assuming that the tube temperature field is known at the fluid iteration. Then, temperature and pressure fields for both fluid flows are found from the integration of the conservation equations. This calculation is performed through an explicit finite difference discretisation technique, following the flow path. Once the temperature field for both fluids are calculated, then the conservation equation for the tubes, including longitudinal conduction, are integrated through an implicit scheme involving the temperature of the wall cell. Then, a new iteration starts till convergence throughout the HE is obtained.

1. INTRODUCTION

When evaporation or condensation takes place in a Heat Exchanger (HE), a great variation in the properties, and in the heat transfer coefficient and friction factor occurs, rendering general rating methods quite inaccurate. The discretisation of the HE becomes then necessary, and the use of an efficient numerical scheme for two-phase flow, able to account for the local variation of every parameter, becomes the key to obtain realistic predictions.

The model presented in this paper has been developed to be able to be applied to any kind of compact HE and flow arrangement. However, for space reasons, in the following, a fin and tube HE will be considered to illustrate the basis and capabilities of the model, especially for the calculation of the dehumidification process taking place at the air side of evaporators.

A HE is basically formed by two fluids that exchange heat throughout a series of wall pieces, formed in this case by a combination of tube and fins.

The method presented in the paper is devoted to the thermohydraulic analysis of HEs, more specifically, to the calculation of the fluids evolution throughout the HE for given values of mass flow rate and, inlet temperature and pressure of both fluid flows.

Many previous works have been carried out concerning this topic (see [1], for instance. However, most of them are only valid for fixed flow arrangements and do not allow the use of local values for properties and coefficients.

2. GOVERNING EQUATIONS

2.1 Refrigerant side

In the case of an evaporator or a condenser, a 2-phase flow with phase change occurs. A steady 2-phase flow following an annular pattern is considered to occur inside the pipes. Therefore, the separated fluid model will be considered, for which the governing equations are:

$$G = \rho u = constant \tag{1}$$

$$-\frac{dp}{dz} = \frac{2f \cdot G^2(1-x)^2}{D_h \rho_f}\Phi_f^2 + G^2\frac{d}{dz}\left(\frac{x^2}{\rho_g \alpha} + \frac{(1-x)^2}{\rho_f(1-\alpha)}\right) + (\alpha\rho_g + (1-\alpha)\rho_f)g sin\theta \tag{2}$$

$$AG\frac{\partial}{\partial z}\left[x\left(i_g + \frac{G^2 x^2}{2\rho_g^2\alpha^2}\right) + (1-x)\left(i_f + \frac{G^2(1-x)^2}{2\rho_f^2(1-\alpha)^2}\right)\right] + AG\frac{\partial}{\partial z}(zg sin\theta) = Ph(T_w - T) \tag{3}$$

The continuity equation states the conservation of the mass flow rate and the mass velocity all along a fluid path. Its value is known from the inlet conditions, so that G will be considered as a known constant in the following analysis.

At a fluid cell, the number of equations to be considered are two (energy, Eq.(3), and momentum, Eq.(2)), while the unknown variables in the equations are eight: x, α, ρ_l, ρ_v, i_l, i_v, p and T. The number of equations is clearly lower than the number of unknowns and hence some further relationships among these variables must be stated. First, the assumption of thermodynamic equilibrium is adopted, and then, one more equation is required, relating the void fraction with the rest of variables. See [2] for details.

2.2 Air side

For the air, the governing equations are those stated for the mass, energy and momentum conservation. In the case of the differential surface shown in figure 1, the following approximated equations are stated. See [3]:

$$-\dot{m}_a di = dQ - \dot{m}_a \cdot dW \cdot i_{f,wat}$$
$$dQ = \left[h_c\left(T - T_{wat}\right) + h_D\left(W - W_{s,wat}\right)\left(i_{g,T} - i_{f,wat}\right)\right]Pdz \tag{4}$$
$$-\dot{m}_a dW = h_D Pdz\left(W - W_{s,wat}\right) \tag{5}$$
$$\frac{dp}{dz} = -\frac{d\left(\rho u^2\right)}{dz} - f\frac{1}{2D_h}\rho u^2 \tag{6}$$

where $i_{s,wat}$ is the enthalpy of the saturated air at the water surface temperature.

The approach followed to treat the dehumidification process is the one proposed by Threlkeld [3].

Figure 1 shows an air cell in the more general case in which dehumidification of humid air takes place when the humid air is in contact with a cold surface. A water film is formed over the surface. There is a limit boundary layer of air next to the water surface. The hypothesis that the air in contact with the water film is saturated at the temperature of the water surface, T_{wat}, is assumed.

Using the relationship $Le = h_{c,0}/h_{D,0}C_{p,a}$, Eq.(4) can be written as (See [3]):

$$dQ = \frac{h_c Pdz}{C_{p,a}}\left[\left(i - i_{s,wat}\right) + \frac{\left(W - W_{s,wat}\right)\left(i_{g,T} - i_{f,wat} - 1061Le\right)}{Le}\right] \tag{7}$$

The second term in Eq.(7) is negligible compared with the first term, so that the heat transferred can be approximately calculated as follows:

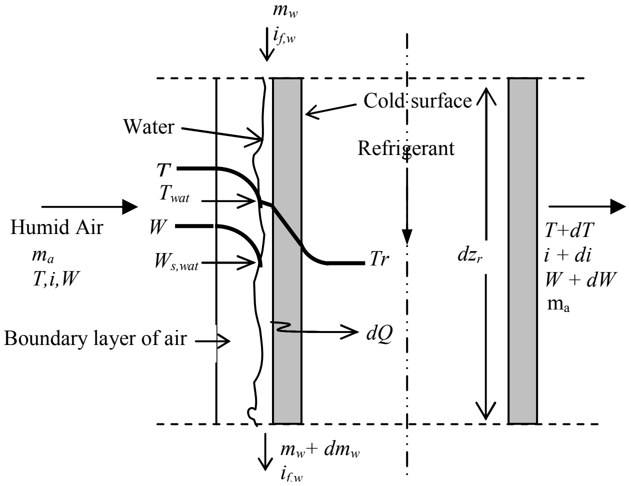

Figure 1. Schematic cooling and dehumidification of humid air

$$dQ = \frac{h_c}{C_{p,a}}\left(i - i_{s,wat}\right)Pdz \tag{8}$$

Eq.(8) allows an analysis much easier than Eq.(7). An additional equation is also normally used to simplify the description of the process, i. e. the assumption that the enthalpy of the saturated air is a linear function of the temperature: $i_{sat} = a + bT_{sat}$, where a and b are some coefficients which must be calculated from suitable plots [3]. Finally, in order to take into account the conduction across the water film, Eq.(8) is now modified [3], leading to:

$$dQ = \frac{h_w}{b_w}\left(i - i_{s,w}\right)Pdz \tag{9}$$

where $i_{s,w}$ is the enthalpy of the saturated air at the wall temperature, and:

$$h_w = \frac{1}{C_{p,a}/\left(b_w h_C\right) + y_w/k_w} \tag{10}$$

where k_w is the water thermal conductivity and y_w is the thickness the water film. Equation (9) shows that the heat transfer process is governed by the difference between the enthalpy of the humid air and the enthalpy of the saturated air at wall temperature as a driving potential.

However, for the numerical scheme in the wall temperature calculation (section 5), it is more convenient to use a temperature difference as the driving potential. The total heat transferred consists of two parts: the sensible part and the latent part.

$$dQ = \left[h_C\left(T - T_w\right) + h_D i_{fg}\left(W - W_{s,w}\right)\right]Pdz$$

The above expression can be casted as a fuction of the temperature difference in the following way:

$$dQ = h_C\left[1 + \frac{i_{fg}}{C_{p,a}}\frac{W - W_{s,w}}{T - T_w}\right]\left(T - T_w\right)Pdz = h_{eq}\left(T - T_w\right)Pdz \tag{11}$$

where $h_{eq} > h_C$ and the term into brackets represents the enhancement factor of h_C due to the condensate.

The value for h_C comes from semiempirical correlations. In the present paper, the one by Chi Chuan Wang [4] & [5] will be used, for both latent and sensible processes.

2.3 Walls

For the walls, the equation to be written is the balance of the heat exchanged with the surrounding fluids and the heat transferred by longitudinal conduction along the wall, i.e.:

$$k\,e\,\nabla^2 T_w + \sum_{i=1,2} q_i = 0; \tag{12}$$

where the heat flux is the heat over the total heat transfer area. Normally, the effect of the longitudinal conduction is negligible so that the wall energy equation becomes the balance equation for the heat exchanged between fluids

$$Q_a = Q_r \tag{13}$$

The boundary conditions are given by the conditions of the fluids at the entrance of the heat exchanger.

3. GLOBAL SOLUTION STRATEGY

The global solution method employed is called SEWTLE (for Semi Explicit method for Wall Temperature Linked Equations) and is outlined in [6]. Basically, this method is based on an iterative solution procedure. First a guess is made about the wall temperature distribution, then the governing equations for the fluid flows are solved in an explicit manner, getting the outlet conditions at any fluid cell, from the values at the inlet of the HE and the assumed values of the wall temperature field. Once the solution of the fluid properties are got at any fluid cell, then the wall temperature at every wall cell is estimated from the balance of the heat transferred across it (Eq.(12)). This procedure is repeated until convergence is reached. The numerical scheme developed for the calculation of the temperature at every wall cell is also explicit, so that the global strategy consists in an iterative series of explicit calculation steps. The method is of application to any flow arrangement and geometrical configuration, and offers excellent computational speed. Moreover, it can be used, as it is the case of the present paper, for combined single, 2-phase flow and air.

In two-phase flow, the energy (Eq. (3)) and momentum (Eq. (2)) equations are coupled by the pressure through the influence of the it on the temperature. Fortunately the dependence is weak due to the usual small pressure drop inside the heat exchanger. Since all the variables mainly depend on the pressure, the momentum equation may be integrated first. Then, once the pressure at the outlet of the fluid cell is known, the energy equation can be integrated, leading to the evaluation of the enthalpy at the outlet, and of the quality and the rest of variables. The discretisation of the governing equations for 2-phase flow is briefly described in the following.

In the airside, the process in which this fluid is involved is found at the inlet of the air cell, in which the inlet temperature of the air is compared with the wall temperature and with the dew point temperature. The energy equation is integrated first, calculating the outlet enthalpy, and then the outlet temperature. The latent heat is calculated as the difference between the total heat and the sensible heat. From the latent heat, the outlet humidity is calculated. The momentum equation is then integrated, getting the outlet pressure of the air. Finally, the outlet properties of the air are calculated.

4. DISCRETISATION OF THE GOVERNING EQUATIONS

4.1 Refrigerant side

A simple, but effective discretisation has been adopted for the momentum equation. The friction and the gravity terms have been approached by the arithmetic average value of the corresponding function multiplied by the increment in distance along the channel. The acceleration term is integrated as the difference between the outlet and the inlet values.

$$p_o = p_i - \left[\overline{\frac{2fG^2(1-x)^2}{D_h\rho_f}\Phi^2{}_f}\right]'' \Delta z + G\left[CV_o - CV_i\right]^* + \overline{\left(\alpha\rho_g + (1-\alpha)\rho_f\right)}'' g\, sin\theta\,\Delta z \tag{14}$$

Once the outlet pressure has been found, then the thermophysical properties of the fluid are calculated, and also the temperature. The energy equation can be discretised in a similar way, and used to provide the outlet enthalpy.

$$i_o = i_i + \frac{h}{\dot{m}}\left(T_w - \frac{T_i + T_o}{2}\right)P \cdot \Delta z - g \cdot sin\theta \cdot \Delta z \tag{15}$$

The discretisation of the heat transferred to the walls assumes a piecewise variation of the fluid temperature. See [6] for a detailed description**.**

On the other hand, the enthalpy at the outlet i_o can be written as:

$$i_o = x_o \cdot \left(i_{go} + \frac{G^2 x_o^2}{2\rho_o^2\alpha_o^2}\right) + (1 - x_o)\cdot\left(i_{fo} + \frac{G^2(1-x_o)^2}{2\rho_o^2(1-\alpha)^2}\right) \tag{16}$$

As can be seen from this equation, the enthalpy has a polynomial dependence on the quality (order 3). However, the contribution to the enthalpy of the kinetic energy term (G^2 term) is almost negligible, so that the equation can be explicitly solved for the outlet quality. With this updated value for the outlet quality, the outlet void fraction is now calculated from empirical correlations, and the calculation of the next fluid cell is started.

Notice that the described system of equations has been discretised in such a way that the calculation of the outlet conditions at every fluid cell is completely explicit, and that the calculation starts from the inlet section of the HE and progresses along every fluid path. However, the iterative nature of the global strategy allows for the adequate evaluation of the thermophysical properties of the fluids, and of the friction factor and the heat transfer coefficient at every cell.

4.2 Air side

Equation (9) is integrated by assuming that the enthalpy of the saturated air at the tube wall temperature is constant along the air cell. With this assumption, the evolution of the enthalpy of the air becomes exponential. The same treatment is applicable to the temperature evolution, which comes from the integration of the sensible heat equation. This equation represents the variation in temperature of the humid air due to the heat transferred to the walls without

* $\phi_o'' = \phi_i + (\phi_o - \phi_i)^*$; $\bar{\phi}'' = \phi_i + \frac{(\phi_o - \phi_i)^*}{2}$

where "*" means evaluated at previous iteration dehumidification. The solution for enthalpy and temperature becomes:

$$i_o = i_i e^{-\frac{h_w P \Delta z}{\dot{m}_a b_w}} + i_{s,w}\left(1 - e^{-\frac{h_w P \Delta z}{\dot{m}_a b_w}}\right) \tag{17}$$

$$T_o = T_i e^{-\frac{h_C P \Delta z}{\dot{m}_a C_{p,a}}} + T_w\left(1 - e^{-\frac{h_C P \Delta z}{\dot{m}_a C_{p,a}}}\right) \tag{18}$$

The outlet value for the humidity can be found from the latent heat value, which can be determined from the difference between the total and the sensible heat.

$$\begin{gathered} Q_l = Q - Q_s = \dot{m}_a (i_i - i_o) - \dot{m}_a C_{p,a}(T_i - T_o) \\ W_o = W_i - \frac{Q_l}{\dot{m}_a i_{fg}} \end{gathered} \tag{19}$$

After having found the value for the outlet temperature, and assuming, as actually occurs, that the pressure drop for the air is very small, the physical properties for the air at the outlet can be calculated.

Then, from the integration of the momentum equation the outlet pressure can be calculated:

$$\begin{gathered} p_o = p_i - \Delta p_{fric} \cdot \Delta z \\ \Delta p_{fric} = \frac{f}{4D_h}\frac{\dot{m}_a^2}{A^2}\left[\frac{(1+W_o)^2}{\rho_o^2} + \frac{(1+W_i)^2}{\rho_i^2}\right] \end{gathered} \tag{20}$$

where the acceleration term has been neglected.

5. CALCULATION OF THE TUBE WALL TEMPERATURE

Once the fluid temperature and the air temperature are known at every cell, the wall temperature can be determined at every wall cell, and the global iterative procedure repeated until the convergence is reached.

To find out the temperature of the wall T_w, Eq.(18) is solved, where:

$$Q_r = h_r \cdot P_r \cdot \Delta z_r \cdot \left(T_w - \overline{T}_r\right) \tag{21}$$

in which $\overline{T}_r$ is the arithmetic average of the inlet and outlet fluid temperatures.

In the case of the air, the integration of Eq.(11) considering the exponential profile for the temperature evolution given by Eq.(18), allows the heat transferred from the air to the wall to be stated as:

$$Q_a = h_{a,eq}(T_w) \cdot P_a \cdot \Delta z_a \cdot \left(T_{a,i} - T_w\right) \tag{22}$$

where $T_{a,i}$ is the inlet temperature of the air to be consistent with Eq.(13) and

$$\begin{gathered} h_{a,eq}(T_w) = h_C\left[1 + \frac{i_{fg}}{C_p}\frac{W - W_w}{T - T_w}\right]\frac{\left(1 - e^{-\varphi(T_w)}\right)}{\varphi(T_w)} \\ \varphi(T_w) = \frac{P_a \Delta z_a h_C(T_w)}{\dot{m}_a C_p} \end{gathered} \tag{23}$$

To find out the temperature of the wall that balances the heat transferred from the wall to both fluids, Eq.(13) will be written explicitly:

$$P_a \Delta z_a h_{a,eq}(T_w)\left(T_{a,i} - T_w\right) = P_r \Delta z_r h_r \left(T_w - \overline{T}_r\right) \tag{24}$$

As can be seen from Figure 2, the HTC for the air side is not a continuous function of T_w. Two well-separated parts can be observed in the plot. The left-hand side part shows the typical behaviour for HTC when dehumidification exists. On the other hand, when only sensible heat transfer takes part in the process, then the HTC is a constant value.

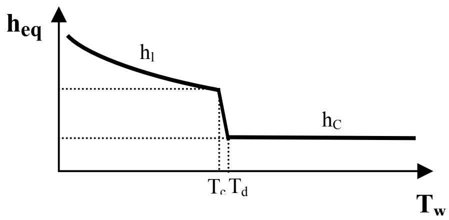

Figure 2. Variation of HTC with T_w

Due to the continuous nature of the physical problem of air cooling with condensation of water vapour, we expect the numerical scheme to be stable and the convergence to be reached. In order for the model to avoid the discontinuity in the HTC of the air, a third region has been introduced as it is shown in figure 2. This corresponds to an interpolation of the HTC between the dew point temperature and an arbitrary temperature called “critical temperature”. After some trials, an arbitrary value of 0.1 degrees has been used in the present study, allowing the solution of the mentioned numerical problem.

In case that the parameter h is not a function of the wall temperature for none of the fluids, as it is the case for the sensible heat (cooling without dehumidification or heating of humid air), then

$$h_{a,eq} = h_C \frac{\left(1 - e^{-\varphi}\right)}{\varphi}; \quad \varphi = \frac{P_a \Delta z_a h_C}{\dot{m}_a C_{p,a}}$$

and Eq.(16) leads to the following explicit expression for the temperature of the wall:

$$T_w = \frac{h_{a,eq} P_a}{h_{a,eq} P_a + h_r P_r} T_{a,i} + \frac{h_r P_r}{h_{a,eq} P_{a,eq} + h_r P_r} \overline{T}_r \tag{25}$$

Therefore, the calculation of the wall temperature at every cell is based on a very fast explicit formula.

On the other hand, in case that h_i for one of the two fluids is dependent of T_w, as it is the case for the air during cooling with dehumidification, then Eq.(24) is implicit, and an iterative method must be used everytime. In this case, the equation to be solved is:

$$T_w = \frac{1}{1+\beta(T_w)}\left[\beta(T_w)T_{a,i} + \overline{T}_{ref}\right] \quad (26)$$

$$\beta(T_w) = \frac{P_a \Delta z_a h_{a,eq}(T_w)}{A_r h_r}$$

6. HEAT TRANSFER COEFFICIENT AND FRICTION FACTOR CORRELATIONS

The most employed correlations for heat transfer coefficients and friction factor for annular flow in pipes have been studied by the authors in [7]. For the results shown in this paper, the VDI correlation was used for evaporation, while the Travis correlation was used for condensation. For the friction factor coefficient, the Chisholm correlation was used, both for condensation and evaporation.

For single-phase flow in pipes some very well established correlations have been published. The authors have used, for turbulent flow, the Gnielinski correlation for the HTC and the Petukhov correlation for the friction factor. For laminar flow, Blausisus equation has been used.

In the description of the air processes, the correlations for the heat transfer coefficient and for the friction factor proposed by Chi Chuan Wang in [4] & [5] have been used. These correlations cover both dry process and dehumidification.

7. EXPERIMENTAL RESULTS

An experimental test campaign has been carried out in order to study the performance of the fin and tube coil of a 20 kW air-to-water reversible heat pump for two different refrigerants: R22 and propane. A comparison between the calculated and measured results for this coil working as evaporator and condenser is presented in the following.

The experimental tests have been carried out at the ENEA Research Centre of Casaccia with a facility named PROHPHETA (PROpane Heat Pump & Heat Exchangers Thermal-hydraulic Activity). The test loop consists mainly in three circuits: the air loop, the water loop and the refrigeration loop. The instrumentation installed allows the measurement of the temperature, mass flow rate and pressure in all the relevant points of the three loops. The studied coil was inserted in the air loop.

In Figures 3 to 6, the measured and calculated results are presented. The employed coil is an Alfa Laval coil, 1144 x 850 x 85 mm, with three tube rows, 36 tubes per row and 6 identical circuits. The fins used were corrugated, with a fin pitch of 2.9 mm made on aluminium. The tubes had an outer diameter of 12.7 mm and were on copper. Tests were performed with two different refrigerants: R22 and R290 (propane), following a test matrix which tried to reproduce typical operation conditions, varying the air temperature and the refrigerant inlet temperature and pressure.

The figures plotted below reproduce the results for both condenser (Figures 5 & 6) and evaporator (Figures 3 & 4) modes. As can be seen from the figures, the difference between the calculated and measured values in the heat transferred is typically lower than 6% for both refrigerants for the coil working as evaporator, and lower that 8% working as condenser. The straight lines plotted in the figures of the heat transferred represent a range of error of ± 5%. The error in the pressure drop for the coil as evaporator is lower than 20%, but the trend is well predicted (Figure 4). For the coil working as a condenser, a big discrepancy exists between measured and calculated points. The explanation for this may come from the inaccuracy of the measurements, since the absolute values of the pressure difference are small.

The air pressure drop through the coils could not be compared because of the lack of results concerning this magnitude.

A low number of iterations is required till convergence is reached. Typical values range from: 10-25 iterations. The model always behaves stable and convergence is always finally reached.

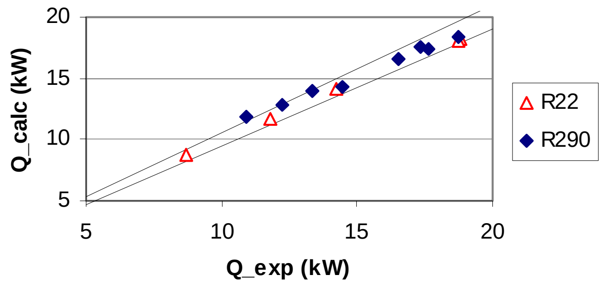

Fig. 3. Calculated vs. experimental capacity. Evaporator

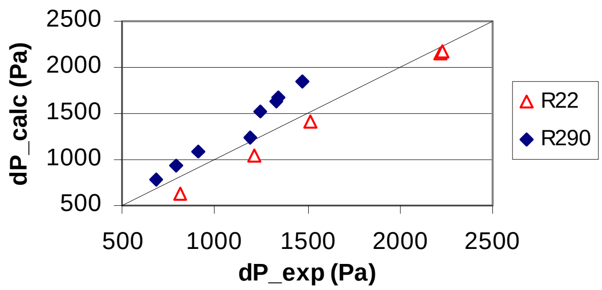

Fig. 4. Calculated vs. experimental pressure drop. Evaporator

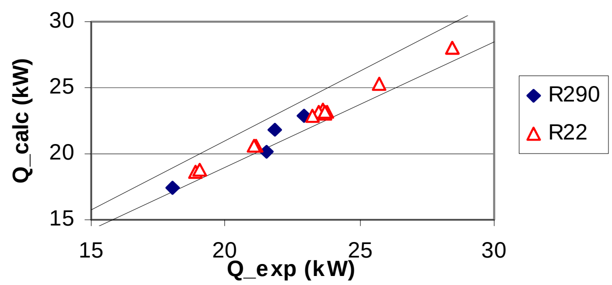

Fig. 5. Calculated vs. experimental capacity. Condenser

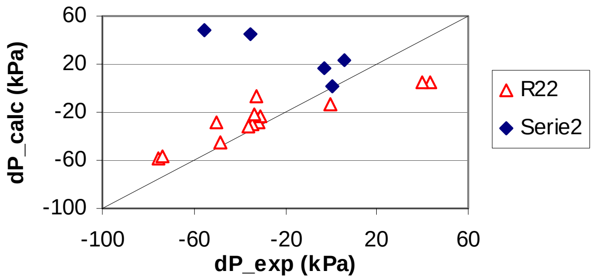

Fig. 6. Calculated vs. experimental pressure drop. Condenser

8. CONCLUSIONS

The following main conclusions can be drawn:

- A model for tube and fin coils has been presented where the local variation of properties, friction factor and heat transfer coefficient, are adequately taken into account.
- The model is of application to any flow arrangement and circuit geometry, and is able to also include the possible effect of longitudinal conduction.
- The solution strategy is iterative and consists of a series of successive explicit evaluations of the fluid temperatures and then of the wall temperatures. The method is called by the authors SEWTLE for Semi Explicit method for Wall Temperature Linked Equations. Typically, the number of required iterations ranges from 10 to 20.
- The model has been implemented with the equations governing both sensible and latent process for humid air, obtaining accurate results in the comparison with experimental measurements.
- A full comparison of results on a fin and tube HE working as an evaporator and as a condenser has been performed for two different refrigerants, showing a difference between measured and predicted values for the heat transferred typically lower than 6%.

9. ACKNOWLEDGEMENTS

This research has been financed in part by the European Comission within the Fourth Framework Programme of the rational use of energy of the Joule Programme. Thanks also to Alfa Laval Artec for having collaborated in this research.

10. NOMENCLATURE

A	cross section area (m^2)	α	void fraction
b_w	slope of i vs. T curve	Φ_f^2	2-phase fric. multiplier
C_p	specific heat (J/kg K)	ϕ, φ	generic variable
CV	$G\left[\frac{x^2}{\rho_g \alpha} + \frac{(1-x)^2}{\rho_f (1-\alpha)}\right]$	θ	angle with horizontal
D_h	hydraulic diameter (m)	ρ	density (kg/m^3)
e	wall thickness (m)	∇^2	Laplacian operator
f	friction factor	Δ	Increment
g	gravity (m/s^2)	**subscripts**	
G	mass velocity (kg/s m^2)	*a*	air
h	HTC (W/m^2 K)	*eq*	equivalent
i	enthalpy (J/kg)	*f*	saturated liquid
k	conductivity (W/m K)	*g*	saturated vapour
m	mass flow rate (kg/s)	*i*	inlet, cell index
P	perimeter (m)	*j*	cell index
p	pressure (Pa)	*l*	liquid
q	heat flux (W/m^2)	*o*	outlet, cell index
Q	heat (W)	*r*	refrigerant
S	slip ratio	*s*	saturation
T	temperature (K)	*v*	vapour
u	velocity (m/s)	*w*	wall
W	humidity (kg vap/kg dry air)	*wat*	water
x	vapour quality		
z, y	spatial co-ordinates (m)		

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