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J. M. Corberan
Universidad Politecnica de Valencia

P. Fernandez De Cordoba
Universidad Politecnica de Valencia

S. Ortuno
Universidad Politecnica de Valencia

V. Ferri
Universidad Politecnica de Valencia

P. Montes
Universidad Politecnica de Valencia

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J. M. Corberán[1], P. Fernández de Córdoba[2], S. Ortuño[1], V. Ferri[1], P. Montes[1]

[1]Applied Thermodynamics Dept., Univ. Polit. de Valencia, Spain.
Camino de Vera 14, ES 46022 VALENCIA, SPAIN.
Tel. 34 963877323, Fax. 34 963877329, email: corberan@upvnet.upv.es
[2]Applied Mathematics Dept., Univ. Polit. de Valencia, Spain.

ABSTRACT

In this paper a model for heat exchangers working as evaporator or condenser of a refrigeration system is presented, including a comparison between calculated and measured results for a plate heat exchanger and a tube & fins coil. In the paper, the main characteristics of the model and a comparison between experimental and calculated results is presented and discussed.

The flow inside the pipes or channels is considered to be one-dimensional, and discretized in as many elements as required. The basis of the numerical procedure is to de-couple the calculation of the fluid flows from each other. Then, both fluid flows evolution along the heat exchanger are calculated through the integration of the 1-D conservation equations. For the refrigerant, the two fluid (liquid & vapour) separated model under equilibrium is considered. Specific correlations for evaporation and condensation heat transfer in pipes and plate heat exchangers have been implemented into the model. Concerning the air, both sensible heat transfer and dehumidification are considered with appropriate correlations for the heat transfer coefficient and friction factor.

NOMENCLATURE

A	cross section area (m^2)
b_w	slope of i vs. T curve
C_p	specific heat (J/kg K)
D_h	hydraulic diameter (m)
e	wall thickness (m)
f	friction factor
g	gravity (m/s^2)
G	mass velocity (kg/s m^2)
h	heat transfer coefficient HTC (W/m^2 K)
h_C	sensible HTC in the air case(W/m^2 K)
h_D	Mass transfer coefficient (kg/m^2)
i	enthalpy (J/kg)
k	conductivity (W/m K)
m	mass flow rate (kg/s)
P	perimeter (m)
p	pressure (Pa)
q	heat flux (W/m^2)
Q	heat (W)
S	slip ratio
T	temperature (K)
u	velocity (m/s)
W	humidity (kg vap/kg dry air)
x	vapour quality
z	spatial co-ordinate (m)

Greek symbols

α	void fraction
Φ_f^2	2-phase fric. Multiplier
ρ	density (kg/m^3)
∇^2	Laplacian operator

subscripts

a	air
eq	equivalent
f	saturated liquid
g	saturated vapour
i	cell index
l	liquid
r	refrigerant
s	saturation
v	vapour
w	wall
wat	water

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 discretization 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. This paper deals with the application of the method to the modelling of a fin and tube HE and a brazed plate heat exchanger under both operation modes: evaporation and condensation.

A Fin and Tube Coil is basically formed by two fluids (refrigerant, in this case, inside the pipes and air throughout the fins) that exchange heat throughout a series of separating and heat transfer surfaces, formed in this case by a combination of tubes and fins.

A PHE is formed basically by a series of fluid channels which exchange heat throughout parallel separating walls. To allow the problem to be more general, more than two different fluids can exist, though the most common situation is to have just two fluids in one HE at the very same time. In this case, refrigerant and water.

The method presented in the paper is devoted to the thermohydraulic analysis of HE's, 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 devoted to this topic (see [1], for instance). However, most of the employed models are only valid for fixed flow arrangements and geometries, and do not allow the use of local values for properties and coefficients.

GOVERNING EQUATIONS

Single Phase Flow

The governing equations for a single phase 1D steady flow along a fluid cell are:

$$G = \rho u = Constant^{*};$$

$$\frac{dp}{dz} = -\frac{d(\rho u^2)}{dz} - f\frac{1}{2}\rho\frac{u^2}{D_h} - \frac{d(zg\rho)}{dz}* \tag{1}$$

$$A \cdot G \cdot \frac{d\left(i + {u^2}/{2}\right)}{dz} = Ph(T_w - T) \tag{2}$$

where the fluid exchanges heat with the surrounding walls at T_w (pipe wall for tubes and plates for PHE). These equations are the governing equations for the water flow in the BPHE, the refrigerant single phase regions (superheated vapour or subcooled liquid), and for the air flow when dehumidification does not take place.

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. Therefore only the momentum and the energy equations must be integrated.

Two-Phase Refrigerant Flow

In the case of an evaporator or a condenser, a 2-phase flow with phase change occurs. In the case of the coil, a steady 2-phase annular or stratified flow takes place inside the pipes. A steady 2-phase flow similar to the annular pattern in tubes is considered to happen along the channels in between the plates in PHE. Therefore, the separated fluid model will be considered for both cases. Hence, the governing equations are:

$$G = \rho u = constant$$

$$-\frac{dp}{dz} = \frac{2f \cdot G^2(1-x)^2}{D_h \rho_f}\Phi_f^2 + G^2\frac{d}{dz}\left(\frac{x^2}{\rho_g \alpha} + \frac{(1-x)^2}{\rho_f(1-\alpha)}\right) + \left(\alpha\rho_g + (1-\alpha)\rho_f\right)g sin\theta \tag{3}$$

$$AG\frac{\partial}{\partial z}\left[x\left(i_g + \frac{G^2x^2}{2\rho_g^2\alpha^2}\right) + (1-x)\left(i_f + \frac{G^2(1-x)^2}{2\rho_f^2(1-\alpha)^2}\right)\right] + AG\frac{\partial}{\partial z}(zg sin\theta) = Ph(T_w - T) \tag{4}$$

* Constant cross section will be considered throughout the study.

The system becomes closed under the assumption of thermodynamic equilibrium when an empirical correlation for the void fraction is assumed. A number of those correlations can be found in the literature, for instance: Wallis [2] and Premoli [3]. In this paper, the Chisholm [4] correlation is employed:

$$S=\left[1-x\cdot\left(1-\frac{\rho_l}{\rho_v}\right)\right]^{\frac{1}{2}} \tag{5}$$

Air Flow Under Dehumidification

Concerning the airside under dehumidification, 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 [5]:

$$-\dot{m}_a di = dQ-\dot{m}_a\cdot dW\cdot i_{f,wat}$$

$$dQ=\left[h_C\left(T-T_{wat}\right)+h_D\left(W-W_{s,wat}\right)\left(i_{g,T}-i_{f,wat}\right)\right]Pdz \tag{6}$$

$$-\dot{m}_a dW=h_D Pdz\left(W-W_{s,wat}\right) \tag{7}$$

$$\frac{dp}{dz}=-\frac{d\left(\rho u^2\right)}{dz}-f\frac{1}{2D_h}\rho u^2 \tag{8}$$

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 [5].

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 $Le=h_C/h_D C_{p,a}$, Eq.(6) can be written as (See [5]):

$$dQ=\frac{h_C Pdz}{C_{p,a}}\left[\left(i-i_{s,wat}\right)+\frac{\left(W-W_{s,wat}\right)\left(i_{g,T}-i_{f,wat}-1061Le\right)}{Le}\right] \tag{9}$$

The second term in Eq.(9) is negligible compared with the first term, so that the heat transferred can be approximately calculated as follows:

$$dQ=\frac{h_C}{C_{p,a}}\left(i-i_{s,wat}\right)Pdz \tag{10}$$

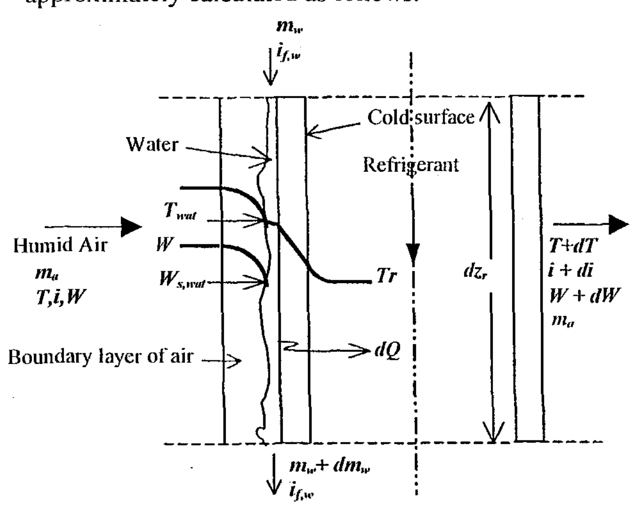

Figure 1. Schematic cooling and dehumidification of humid air

Eq.(10) allows an analysis much easier than Eq.(9). 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 fittings. See [5]. Finally, in order to take into account the conduction across the water film, Eq.(10) is now modified [5], leading to:

$$dQ=\frac{h_w}{b_w}\left(i-i_{s,w}\right)Pdz \tag{11}$$

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

$$h_w = \frac{1}{C_{p,a}/(b_w h_C) + y_w/k_w} \qquad (12)$$

where k_w is the water thermal conductivity and y_w is the thickness of the water film. Equation (11) 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, it is more convenient to use a temperature difference as driving potential. The total heat transferred consists of two parts: the sensible part and the latent part.

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

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

$$dQ = h_C\left[1 + \frac{i_{fg}}{C_{p,a}}\frac{W - W_{s,w}}{T - T_w}\right](T - T_w)Pdz = h_{eq}(T - T_w)Pdz \qquad (13)$$

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

The value for h_C comes from semiempirical correlations. In the present paper, the one by Chi Chuan Wang [6] & [7] will be used, for both latent and sensible processes.

Walls

Concerning the tube and plate 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.:

$$k\, e\, \nabla^2 T_w + \sum_{i=1,2} q_i = 0; \qquad (14)$$

where the heat flux is the total heat transferred over the projected heat transfer area.

In evaporators and condensers, the effect of the longitudinal conduction is negligible so that the wall energy equation becomes the balance equation for the heat exchanged between fluids.

$$\sum_{i=1,2} Q_i = 0 \qquad (15)$$

Boundary Conditions

The boundary conditions are given by the temperatures and pressures of the fluids at the entrance of the heat exchanger, and by the adiabatic condition for the outer walls of the heat exchanger.

GLOBAL SOLUTION STRATEGY

The global solution method employed is called SEWTLE (for Semi Explicit method for Wall Temperature Linked Equations) and is outlined in [8]. 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 obtained at any fluid cell, then the wall temperature at every wall cell is estimated from the balance of the heat transferred across it (Eq.(14)). 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 dehumidification.

In single phase flow, the momentum and the energy equations are uncoupled, and then, both of them can be integrated independently. From the energy equation, the outlet temperature can be figured out and then, from the momentum equation, the outlet pressure can be obtained. For a detailed treatment of the equations , the reader is submitted to [9].

In two-phase flow, the energy (Eq. (4)) and momentum (Eq. (3)) equations are coupled by the pressure through its influence on the temperature. Fortunately, under thermodynamic equilibrium, 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 must 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. A finite volume scheme is used for the discretization of the equations. See [8] and [9].

In the airside, the process in which this fluid is involved is found at the inlet of the air cell, where 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. See [10]

HEAT TRANSFER COEFFICIENT AND FRICTION FACTOR CORRELATIONS

For single-phase flow in pipes very well established correlations are known. The authors have used, for turbulent flow, the Gnielinski correlation for the HTC and the Petukhov correlation for the friction factor. For laminar flow the Blausisus equation has been used. For single phase flow in PHE some very well established correlations are published. The authors have used the one proposed in [11].

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 [6] & [7] have been used. These correlations cover both dry process and dehumidification.

The most employed correlations for heat transfer coefficients and friction factor for two-phase annular flow in pipes have been studied by the authors in [12]. 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.

An accurate evaluation of these parameters for two-phase flow in PHE is very difficult, and only a few specific references can be found in the open Literature. For two-phase flow in PHE, most of the correlations used for refrigerants inside pipes are not of application. The study of appropriate correlations for PHE is a complex matter and a detailed study is planned for the future. So far, the authors have found that for the case of evaporation, the strategy proposed by Thonon et al. [13] leads to satisfactory predictions. That correlation is the one that the authors have used to produce the results shown in the present paper. For condensation process, an Shah correlation [14] was used, but the parameters of the correlation had to be slightly adequate.

In figure 2, the HTC along one fluid channel of R290 for condensation is shown. In this plot three different regions can be seen. The first one corresponding to the superheated zone with low HTC. A second region with higher values of the film coefficient corresponding to the two-phase condensation region and finally, a small part dedicated to the subcooling of the refrigerant inside the fluid channel. In Fig. 3, an example of evaporation for R22 is shown. In this plot, a first region corresponding to high values of the HTC can be observed. Also the two-phase region with quality values greater than the dry-out ones is presented. Finally, a third region of the HTC near 200 W/m^2K (single phase region) can be distinguished in the plot.

Normally, correlations for 2-phase flow are not defined for the high quality region. Therefore, it is necessary to interpolate between an arbitrary point before saturation (typically x = 0.9) and x = 1 (saturated vapour). In the case of evaporators, it should be also taken into account the effect of the dry-out process which is expected to happen once the quality is high enough. For the comparison of results shown in this paper, a value of 0.8 has been considered as the starting point of the dry-out process, so that for higher qualities the heat transfer coefficient is interpolated from the value corresponding to that quality and the one corresponding to saturated vapour.

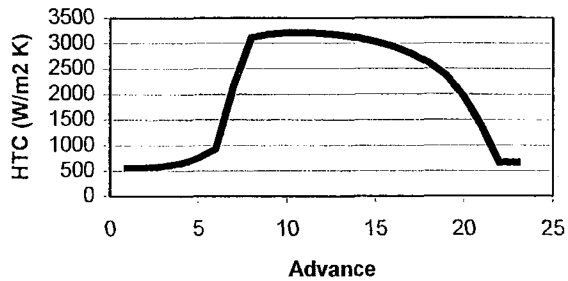

Figure 2. HTC for R290 in condensation

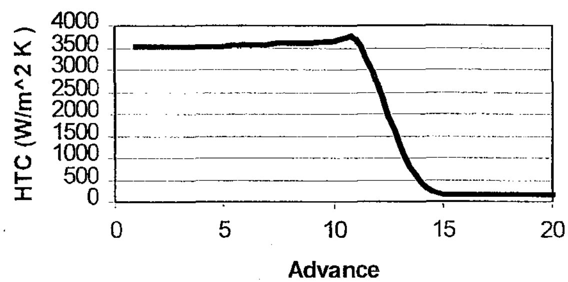

Figure 3. HTC for R22 in evaporation

EXPERIMENTAL RESULTS

The results from an experimental test campaign of a 20 kW air-to-water reversible heat pump carried out under a R&D project dealing with the optimisation of the heat pump for propane, was used to assess the agreement of the calculated results. A comparison between the calculated and measured results for the two BPHE and the coil of the heat pump, working as evaporator and as condenser is presented in the following.

In Figures 4 to 11, 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. The used BPHE is an Alfa Laval PHE, ref: CB52-HX with 46 plates made of aluminium. 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 HE working as condenser (BPHE: figures 6 & 7; coil: figures 10 & 11) and evaporator (BPHE: figures 4 & 5, coil: fig. 8 & 9) modes. As can be seen from the figures, the difference between the calculated and measured values in the heat transferred is lower than 6% for both refrigerants and both HE's. 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 BPHE working as evaporator (Fig. 5) is lower than 6%. For the PHE working as a condenser (Fig. 7), no conclusions can be drawn due to the non-reliability of the experimental data. The difference in pressure drop for the coil working as evaporator is lower than 20%, but the trend is well predicted. For the coil working as a condenser (Fig. 11), a big discrepancy exists again between measured and calculated results. The explanation for the disagreement found in Fig. 7 & 11 may come from the inaccuracy of the measurements, since the absolute values of the pressure difference are small. The air pressure drop through the coil could not be compared because of the lack of results concerning this magnitude. In the plots, the R290 points are those represented by a full diamond, and the R22 points are those represented by an empty triangle.

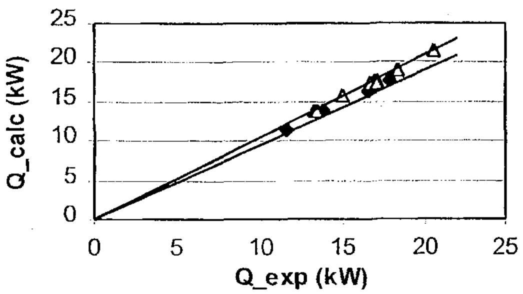

Fig.4. Capacity.BPHE. Evaporator

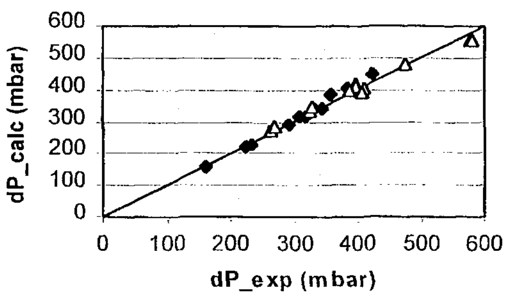

Fig.5. Pressure drop. BPHE. Evaporator

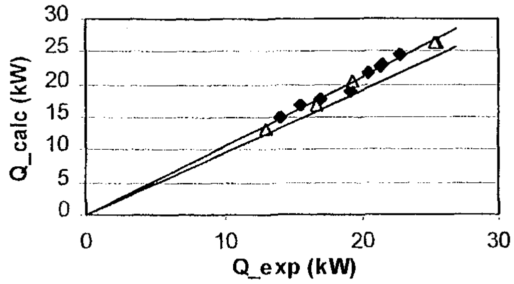

Fig.6 Capacity. BPHE. Condenser

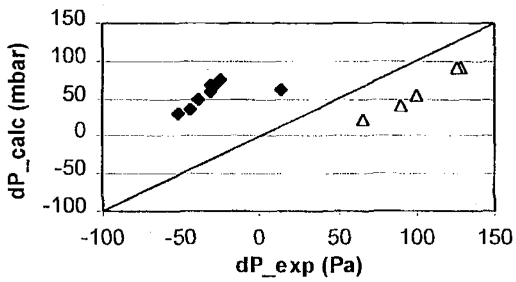

Fig.7 Pressure drop. BPHE. Condenser

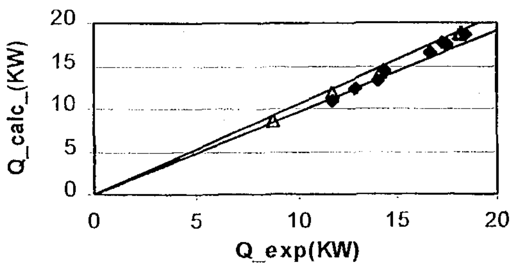

Fig. 8 Capacity. Coil. Evaporator

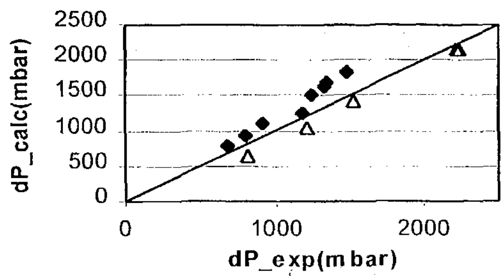

Fig. 9 Pressure drop. Coil. Evaporator

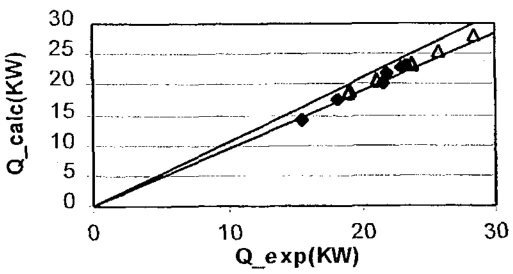

Fig. 10 Capacity. Coil. Condenser

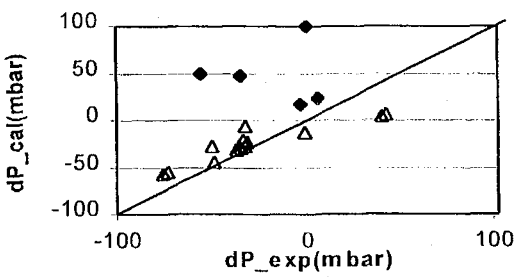

Fig. 11 Pressure drop. Coil. Condenser

CONCLUSIONS

The following main conclusions can be drawn:

- A model for compact HE 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, a low number of iterations are required till convergence is finally reached, and a low CPU calculation time is needed.
- The numerical scheme for the integration of the energy equation is based on the assumption of a piecewise profile for the fluid temperature.
- A full comparison of results on a fin and tube HE and a PHE, both 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%.

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