INSTITUTO DE INGENIERÍA ENERGÉTICA
UNIVERSIDAD POLITÉCNICA DE VALENCIA

CPI, Edificio 8E, Cubo F, 5ª planta, Camino de Vera s/n, 46022 Valencia, Spain Tel. (34) 963877270, Fax (34) 963877272, Email: energeti@upvnet.upv.es

INSTITUTO DE INGENIERÍA ENERGÉTICA
(Institute for Energy Engineering)

Research Publications

Modelling of compact evaporators and condensers

J. M. Corberán[1], P. Fernández de Córdoba[2], S. Ortuño[1], V. Ferri[1], J. Gonzálvez[1], T. Setaro[3], G. Boccardi[3].
[1]*Applied Thermodynamics Dept., Univ. Politécnica de Valencia, Spain.*
[2]*Applied Mathematics Dept., Univ. Politécnica de Valencia, Spain.*
[3]*ENEA,Casaccia, Italy.*

Abstract

In this paper a model for compact evaporators and condensers is presented. The flow along a channel of a compact HE is considered to be 2-D and split into separated 1-D paths. Then, every 1-D flow path is discretised in as many elements as required. First, both fluid flow evolution along the heat exchanger are calculated through the integration of the 1-D conservation equations for single or two phase flow, assuming that wall temperatures are known. This calculation is performed through an explicit finite volume scheme, following the flow path. Once the temperature field for both fluids is calculated, then, the wall temperatures are obtained from the integration of the wall energy equation by means of an explicit scheme again. Then a new iteration starts till convergence throughout the HE is obtained. The presented method has proven to be very robust and very fast, being able to take into account local variation of properties and coefficients, and also include the calculation of longitudinal conduction.

1 Introduction

When evaporation or condensation take place in a Heat Exchanger (HE), a great variation in the properties, and in the heat transfer coefficient and friction factor, take place, 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 plate HE will be considered as example to illustrate the basis and capabilities of the model.

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.

The method presented in the paper is devoted to the thermo-hydraulic 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.

2 Governing equations

The governing equations for a single phase 1D steady flow along a channel of a PHE are:

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

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

$$A \cdot G \cdot \frac{d\left(i + u^2/2\right)}{dz} = \sum_{j=1}^{2} P_j h_j \left(T_{wj} - T\right) \quad (2)$$

where the fluid exchanges heat with the two surrounding plates.

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

$$G = \rho u = constant$$

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

$$AG\frac{\partial}{\partial z}\left[x\left(i_g + \frac{G^2x^2}{2\rho_g^2\alpha^2}\right) + (1-x)\left(i_f + \frac{G^2(1-x)^2}{2\rho_f^2(1-\alpha)^2}\right)\right] + AG\frac{\partial}{\partial z}\left(zgsin\theta\right)$$

$$-\sum_{j=1}^{2} P_j h_j \left(T_{wj} - T\right) = 0 \quad (4)$$

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.

---

* Constant cross section will be considered throughout the study.

At a fluid cell, the number of equations to be considered are two (energy and momentum), 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, so that the following set of state equations can be added to the system:

$$\begin{aligned} T &= T_{sat}(p) & \rho_v &= \rho_g(p) \\ i_v &= i_{v\_sat}(p_{sat} = p) = i_g(p) & \rho_l &= \rho_f(p) \\ i_l &= i_{l\_sat}(p_{sat} = p) = i_f(p) \end{aligned} \tag{5}$$

Now, only one equation is missing to close the system. Obviously some equations stating the relationship between the void fraction and the rest of the variables should be added into the balance

$$\alpha = \frac{1}{1 + S\dfrac{1-x}{x}\dfrac{\rho_g}{\rho_f}} \tag{6}$$

But this equation introduces a new variable, the slip ratio, therefore one more equation is still missing. The only way to close the problem is to consider some empirical relationship among those variables. A number of those correlations can be found in the literature, for instance: Wallis [4] and Premoli [5]. In this paper, the Chisholm [6] correlation is employed:

$$S = \left[1 - x \cdot \left(1 - \frac{\rho_l}{\rho_v}\right)\right]^{\frac{1}{2}} \tag{7}$$

Now the system is closed, i.e., nine equations combined with nine unknowns.

For the plates, 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.:

$$k\, t\, \nabla^2 T_w + \sum_{i=1,2} q_i = 0; \quad q_i = \frac{Q_i}{\Delta z_i \Delta y}; \quad Q_i = \int_0^{\Delta z_i} P_i\, h_i \left(T_w - T_i\right) dz_i \tag{8}$$

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

$$\sum_{i=1,2} Q_i = 0$$

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 plates 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 [1]. 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. 8). 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 and 2-phase flow.

In single phase flow, the energy and momentum equations are uncoupled, leading to the possibility of first using the energy equation (2) and the equations of state to calculate the temperature solution at the outlet of the fluid cell and then using the momentum equation (1) to calculate the pressure and density variation. The employed discretisation for the energy equation and the numerical scheme (LFTV) is described in [1]. The momentum equation is discretised in a similar manner.

In two phase flow, the energy (3) and momentum (4) equations are coupled by the pressure, through the influence of the pressure on the temperature. Fortunately the dependency 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 described in the following.

4 Discretisation of the 2-phase flow governing equations

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.

$$p_o = p_i - \left[\frac{2 f G^2 (1-x)^2}{D_h \rho_f} \Phi^2{}_f \right]'' \Delta z + G\left[CV_o - CV_i \right]^* + \left(\alpha\rho_g + (1-\alpha)\rho_f \right)'' g \sin\theta \Delta z \qquad (9)$$

where * means evaluated at previous iteration, and CV states for:

$$CV = G\left[\frac{x^2}{\rho_g \alpha} + \frac{(1-x)^2}{\rho_f (1-\alpha)}\right] \tag{10}$$

Since the arithmetic averages in (9) require the knowledge of the outlet conditions, the employed values are updated values calculated from the present value at the inlet plus a variation of the property along the cell, by assuming that that variation is the same that the one happened in the previous iteration, i.e.

Outlet value: $\phi_o'' = \phi_i + (\phi_o - \phi_i)^*$ Averaged value: $\bar{\phi}'' = \phi_i + \frac{(\phi_o - \phi_i)^*}{2}$

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.

$$i_o = i_i + \sum_j \frac{h_j}{\dot{m}}\left(T_{wj} - \frac{T_i + T_o}{2}\right) P_{\cdot j}\, \Delta z - g \cdot \sin\theta \cdot \Delta z \tag{11}$$

The discretisation of the heat transferred to the walls assumes a piecewise variation of the fluid temperature. See [1] for a detailed description.
On the other hand, the enthalpy at the outlet i_o can be written as:

$$i_o = x_o \cdot \left(i_{go} + \frac{G^2 x_o^2}{2\rho_o^2 \alpha_o^2}\right) + (1 - x_o) \cdot \left(i_{fo} + \frac{G^2 (1-x_o)^2}{2\rho_o^2 (1-\alpha)^2}\right) \tag{12}$$

As can be seen from this equation, the enthalpy has a polynomial dependency 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. The non linearity induced by the kinetic energy term and the fact that the outlet void fraction is also an unknown is easily solved again by estimating those terms through their values at the inlet plus a correction, evaluated from previous iteration. Therefore, the outlet quality can be estimated by

$$x_o = \left[i_o - \left(i_{fo} + \frac{G^2 (1-x_o'')^2}{2\rho_{fo}^2 (1-\alpha_o'')^2}\right)\right]\left[i_{fgo} + \left(\frac{G^2 x_o''^2}{2\rho_{go}^2 \alpha_o''^2} - \frac{G^2 (1-x_o'')^2}{2\rho_{fo}^2 (1-\alpha_o'')^2}\right)\right]^{-1} \tag{13}$$

With this updated value for the outlet quality, the outlet void fraction is now calculated from (7) and (6), 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.

5 Wall temperature calculation

Once the fluid temperature is known at every fluid 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 wall temperature T_w the balance between the heat transferred to the two neighbour fluid cells 1 and 2 is stated as

$$Q_1 + Q_2 = 0 \tag{14}$$

where the heat transferred to fluid cell *i* has been discretised as

$$Q_i = h_i \cdot P_i \cdot \Delta z_i \cdot (T_w - \overline{T}_i) \tag{15}$$

In which $\overline{T}_i$ is the arithmetic average of the inlet and outlet fluid temperatures to be consistent with (11). Equation (14) leads to the following explicit expression for the wall temperature:

$$T_w = \frac{h_1 P_1}{h_1 P_1 + h_2 P_2} T_1 + \frac{h_2 P_2}{h_1 P_1 + h_2 P_2} \overline{T}_2 \tag{16}$$

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

6 Heat transfer coefficient and friction factor correlations

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. The most employed correlations for heat transfer coefficients and friction factor for annular flow in pipes were previously studied by the authors in [2]. However, for 2-phase flow in PHE, most of them 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. [3] leads to satisfactory predictions. That correlation is the one that the authors have used to produce the results shown in the present paper.

For single phase flow in PHE some very well established correlations are published. The authors have used the one proposed in [7].

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 0.7 has been considered as the start 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.

7 Cells in which the saturation line is crossed by

An additional difficulty appears in the cells where the saturation line is crossed by, for instance, the cell in which the evaporation* will be completed and the cell is long enough to also produce a superheating of the vapour. A possible numerical option could be to consider the outlet conditions as saturated. The mistake introduced by this assumption could be small as the cell size becomes small enough. However, this strategy could lead to problems in convergence, since during the iterative calculation, a cell could be alternately considered as evaporation taking place inside, or as superheating taking place. This is a very undesired property for an iterative procedure. A way to overcome this difficulty is to recognise what part of the cell is dedicated to finish the evaporation process, and then the rest of the cell can be treated as a single phase superheating zone.

In general, in evaporators and condensers, four different cases could be found: i) From evaporation to superheating. ii) From desuperheating to condensation. iii) From condensation to subcooling and iv) from subcooled heating to evaporation. The transition from the two phase flow evaporating region to the superheated region is discussed in the following. In the other mentioned cases, a similar treatment is required.

Two different portions of the cell (end of the evaporation 2-phase 2ph, and superheating region 1-phase 1ph), of total length Δz must be identified. The heat transferred in both regions are:

$$Q_{2ph} = \dot{m} \cdot (1 - x_i) \cdot i_{fg} \qquad Q_{1ph} = \dot{m} \cdot C_{p,v} \cdot (T_o - T_{sat}) \tag{17}$$

Besides:

$$Q_{2ph} = \sum_j P_{2phj} \cdot h_{2phj} \cdot \Delta z_{2ph} \cdot (T_{wj} - T_{sat})$$

$$Q_{1ph} = \sum_j P_{1phj} \cdot h_{1phj} \cdot \Delta z_{1ph} \cdot \left(T_{wj} - \frac{T_{sat} + T_o}{2} \right) \tag{18}$$

If pressure losses are negligible, T_{sat} will be constant and coincident with T_i. By combining these expressions, the length of each portion of the cell Δz_{1ph} and Δz_{2ph} can be obtained:

$$\Delta z_{2ph} = \Delta z \cdot \frac{(1 - x_i) \cdot \dot{m} \cdot i_{fg}}{\sum_j p_{2ph,j} \cdot h_{2ph,j} \cdot (T_{wj} - T_{sat})} \qquad \Delta z_{1ph} = \Delta z - \Delta z_{2ph} \tag{19}$$

* In the text, the boiling process taking place inside the evaporators will be referred to as evaporation

Therefore, the fluid temperature at the outlet of the cell can be found by writing:

$$T_o = \frac{1}{\left(\dot{m} C_{p,v} + A\right)} \left[\dot{m} C_{p,v} T_{sat} - \dot{m}\left(1 - x_i\right) i_{fg} + B + C\right] \tag{20}$$

where

$$A = \tfrac{1}{2} \sum_j p_{1ph,j} h_{1ph,j} \Delta z_{1ph}; \quad B = \sum_j p_{1ph,j} h_{1ph,j} \Delta z_{1ph} \left(T_{wj} - \frac{T_{sat}}{2}\right);$$
$$C = \sum_j p_{2ph,j} h_{2ph,j} \Delta z_{2ph} \left(T_{wj} - T_{sat}\right)$$

Additionally, in order to make consistent the calculation of the heat transferred from the fluid to the wall, $Q_{2ph} + Q_{1ph}$, with the one employed to state the heat transferred across the wall (15), the heat transfer coefficient must be corrected in the following way.

$$\bar{h}_{cell,j} = \frac{\Delta z_{1ph} * h_{1ph,j} + \Delta z_{2ph} * h_{2ph,j}}{\Delta z} \tag{21}$$

8 Experimental results

An experimental test campaign has been carried out in order to study the performance of the BPHE of a 20 kW air-to-water reversible heat pump with two different refrigerants: R22 and propane. A comparison between the calculated and measured results for the BPHE working as evaporator 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). A schematic of the test loop is shown in Fig. 1.

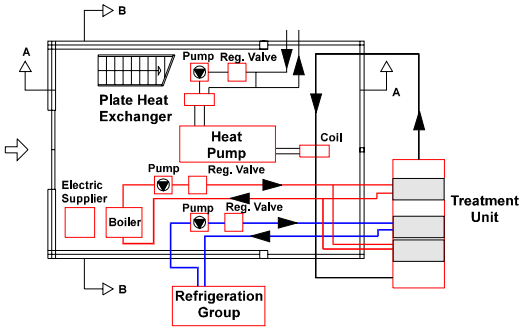

Fig.1 Schematic of the test loop PROPHETA

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 BPHE was inserted in the water loop.

In figures 2 and 3, the measured and calculated refrigerant pressure drop, and the cooling capacity, are presented. The employed BPHE is an Alfa Laval CB52-HX with 46 plates. Tests were performed with two different refrigerants: R22 and R290 (propane). The inlet water temperature was always kept to 12ºC and the outlet temperature varied depending on the refrigerant capacity. At nominal conditions the outlet temperature was 7ºC.

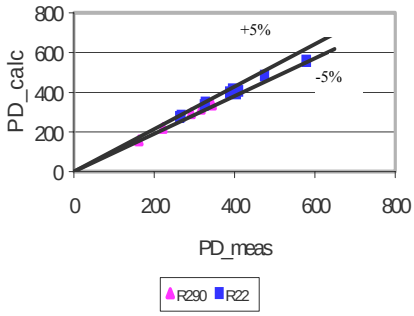

Figure 2. Calculated vs. Measured pressure drop

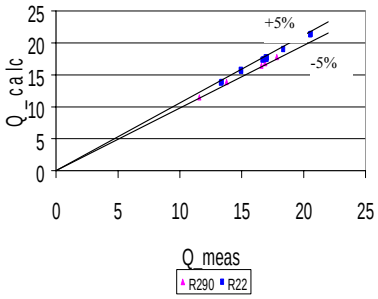

Figure 3 Calculated vs. Measured capacity

As it can be observed in both figures the difference between the predicted values and the measured ones is small.

A low number of iterations is required till convergence. Typical values range from: 10-12 iterations for evaporation in co-current flow, up to 15-20 iterations for condensation in countercurrent mode.

9 Conclusions

The following main conclusions can be drawn,

- A model for compact evaporators and condensers 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 geometry, and is able to also include the calculation 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 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 BPHE working as a evaporator has been performed for two different refrigerants, showing a difference between measured and predicted values typically lower than 5%.

Nomenclature

A	cross section area (m^2)
C_p	specific heat (J/kg K)
D_h	hydraulic diameter (m)
f	friction factor
g	gravity (m/s^2)
G	mass velocity ($kg/s\ m^2$)
h	heat transfer coefficient ($W/m^2\ K$)
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	wall thickness (m)
T	temperature (K)
u	velocity (m/s)
x	quality
z, y	spatial co-ordinates (m)

α	void fraction
Δ	increment
Φ_f^2	two phase friction multiplier
ϕ	generic variable
θ	angle with horizontal
ρ	density (kg/m^3)
∇^2	Laplacian operator

subscripts

f	saturated liquid
g	saturated vapour
i	inlet, cell index
j	cell index
l	liquid
o	outlet
sat	saturation
v	vapour
w	wall

References

[1] Corberán, J.M., Fernández de Córdoba, P. Gonzálvez, J.,Alias, F. Numerical Schemes for the Calculation of Heat Exchangers including Multistream, Local Evaluation of Properties and Heat Transfer Coefficients, and Longitudinal Conduction. (In preparation).

[2] Corberán, J.M., García, M. Modelling of plate finned tube evaporators and condensers working with R134a. *Int. Journal of Refrigeration.* Vol. 21, Nº 4, pp. 273-284, 1998.

[3] Thonon, B., Vidil, R. & Marvillet, C. Recent Research and Developments in Plate Heat Exchangers. *J. of Enhanced Heat Transfer.* Vol. 2,pp. 149-155, 1995.

[4] Wallis, G. B. *One dimensional Two-Phase flow*, Mc-Graw-Hill: New York, 1969.

[5] Premoli, A. Et al. A dimensionless correlation for determining the density of two-phase mixtures. *Termotecnica*, Vol. 25, pp.17-26, 1971.

[6] Chisholm, D. A theoretical basis for the Lockhart-Martinelli correlation for two-phase flow. *Int. J. of Heat and Mass Transfer*, Vol. 10, pp. 1767-78, 1972.

[7] Cooper A., Dennis Usher J., Friction Factor Correlations and Heat Transfer Correlations, 3.7.4 and 3.7.5 in *1998 Heat Exchanger Design Handbook*, Ed. Hewitt G.F., Begell House, 1998.