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CHARACTERISATION OF A PLATE HEAT EXCHANGER WORKING AS CONDENSER. COMPARATIVE ANALYSIS

J.R. García-Cascales[1*], J. Villalba-Reguera[1], F. Illán-Gómez[1], F.J.S. Velasco[1], R.A. Otón-Martínez[1], J.P. Delgado-Marín[1]

1: Thermal and Energy Systems Modelling Research Group, Universidad Politécnica de Cartagena.
C/ Dr. Fleming s/n 30202, Cartagena, España.
e-mail: jr.garcia@upct.es

Abstract: *This paper presents a method to evaluate the heating capacity of a plate heat exchanger (phe) working as condenser. As in previous works analysing phe working as evaporators, the proposed methodology relies on a discretisation of the heat exchanger in cells, in such a way that the outlet temperature is assumed, and the heat transferred is calculated by integrating along the heat exchanger by means of a 1D approach which allows us to determine the outlet variables iteratively based on a heat transfer area calculation.*

The input variables of the model are pressure, enthalpy, and flow rate of both fluids at the inlet of the heat exchanger.

The study presents the method, pays attention to the heat transfer coefficient correlations utilised, and is used to comparatively determine the heat transferred by the refrigerant to the secondary fluid. Experimental results are used for validation and to analyse the goodness of the different correlations tested. Some conclusions are finally drawn.

Keywords: Condensation, plate heat exchanger, heat exchanger modelling.

1. INTRODUCTION

When it comes to analysing the behaviour of a heat exchanger, different options are available, namely lumped parameter methods which rely on the LMTD or NTU-ε [1] or methodologies which consider the discretisation of the heat exchanger into cells, in which the conservation equations are solved and later extended to the whole heat exchanger. The complexity of this last approach depends on the type of analysis considered (1, 2, or 3 dimensional) and the number of cells in which the system is discretised. In the existing literature, different examples may be encountered, of special significance are the already pioneer works carried out in [2,3] for different types of heat exchangers. These models sometimes are very time-consuming when they are integrated inside models which aim to analyse a complex installation comprised of multiple elements such us a heat pump or an organic Rankine cycle.

Having this in mind, the authors recently proposed a simple method to analyse a plate heat exchanger working as evaporator which relies on a discretisation of the heat exchanger in cells, in such a way that the outlet temperature is assumed and the heat transferred is calculated by integrating the conservation equations along the heat exchanger by means of a 1D approach which allows determining the outlet variables iteratively based on a heat transfer area calculation [4,5]). The number of cells considered, the time consumption, and the accuracy reached permitted to claim that the method is good enough for the purpose it was conceived. Thus, the objective of this work is to extend this method to the analysis of a phe working as a condenser so, this may be used to dynamically model a heat pump which will also need the consideration of the models of the rest of the heat pump components (expansion device, evaporator, internal heat exchanger, and compressor). As it is known, this sort of models are very sensitive to the correlations used to evaluate the heat transfer coefficient and pressure drop. In this paper, special attention is also given to them, so, after presenting the methodology, the correlations analysed in this study are briefly described and the results obtained are compared to some experimental results. This allows evaluating the goodness of the model and the correlations used. After that, some conclusions close this work.

2. MODEL DESCRIPTION

Let us consider a plate heat exchanger as that in Figure 1 working as condenser. Three possible zones may be encountered, a de-superheating, a condensation, and a subcooling zone. The model considers the discretisation of each zone of the heat exchanger into *n* cells, so there will be 3*n*+1 nodes as represented in Figure 1.

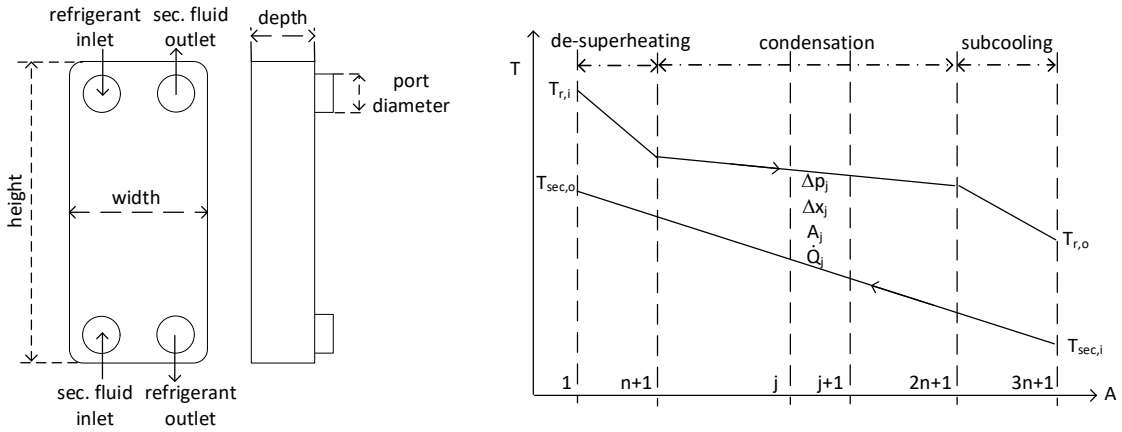

Figure 1. Heat exchanger geometry (left) and temperature distribution along the heat exchanger (right).

The inputs of the model are mass flow rate, inlet temperature (or enthalpy) and inlet pressure of both refrigerant and secondary fluid (water in this case). Initially, an arbitrary area (or length as it is 1D) is given to each of the three heat exchanger zones. They are divided into 3*n* cells with the same initial length $L_n = height/3n$, all variables of interest will be evaluated at the 3*n*+1 nodes. For the de-superheating zone, as a first guess, the outlet temperature of the last node is assumed to be the saturation one at the inlet pressure, $T_{r,n+1} = T_{sat}(p_{r,j})$. The same temperature difference is assumed in each cell so $\Delta T_{desuperheating,j} = (T_{r,j} - T_{sat}(p_{r,j}))/n$. The two-phase zone is divided into *n* cells with the same quality difference $\Delta x_j = 1/n$. Finally, the subcooling zone is divided into *n* cells with the same $\Delta T_j = (T_{r,2n+1} - T_{r,3n+1})/n$ in which $T_{r,2n+1}$is the saturation temperature at node 2*n*+1 pressure and $T_{r,3n+1}$is the outlet temperature of node 3*n*+1 defined as the average

value between the inlet temperature of the refrigerant and that of secondary fluid one $T_{r,3n+1} = (T_{r,j} + T_{sec,j})/2$,. Let us define two temperatures which will be useful in the methodology: They are the maximum temperature at which the refrigerant might leave the heat exchanger and the other is the minimum temperature at which it would leave the heat exchanger. After, the pressure of the refrigerant at each node is determined sequentially as refrigerant pressure and enthalpy are known at the inlet. This requires the use of an appropriate correlation for the pressure drop estimation. So, pressure at node *j*+1 is $p_{r,j+1} = p_{r,j} - \Delta p_r$. Now, the heat exchanged in each element may be determined by means of $\dot{Q}_j = \dot{m}_r(h_{r,j} - h_{r,j+1})$. The same procedure is applied to the nodes in the secondary fluid side by considering that $\dot{m}_r(h_{r,j} - h_{r,j+1}) = \dot{m}_{sec}(h_{sec,j} - h_{sec,j+1})$, the enthalpy at node *j* can be determined by means of $h_{sec,j} = h_{sec,j+1} + \dot{m}_r(h_{r,j} - h_{r,j+1})/\dot{m}_{sec}$. It worthwhile to recall that the value of temperature, enthalpy, and pressure of the secondary fluid at the inlet are inputs of the model and correspond to the node 3*n*+1. The evaluation of the pressure drop in the secondary fluid side is also done sequentially. Once all variables has been calculated, the heat transfer area associated to each cell is determined considering that $\dot{Q}_j = U_j A_j LMTD_j$.The global heat transfer coefficient U_j is calculated by using appropriate correlations for its evaluation so $U_j = \left(\frac{1}{h_{r,j}} + \frac{k_j}{e} + \frac{1}{h_{sec,j}}\right)^{-1}$. Once the heat transfer area is calculated, it is compared to the actual one, and $T_{r,o}$ is recalculated as follows

$$T_{r,o} = \begin{cases} T_{max} = T_{r,3n+1} \; if \; A_{total} < A, \\ T_{min} = T_{r,3n+1} \; if \; A_{total} \geq A, \end{cases}$$

where $A_{total} = \sum_{j=1}^{3n+1} A_j$ is the surface of the heat exchanger calculated after each iteration and A is the actual surface. This is done iteratively until convergence is reached. The flow chart in Figure 2, summarise the procedure described above. As was mentioned, the pressure drop and the heat transfer coefficient play an important role in the model as they have a strong influence on the results. Therefore, some correlations have been studied along this work, particularly for this model, although there are several reviews of these correlations in the existing literature specially for plate heat exchangers. Let us mention the studies carried out in the context of the Swetle method or its extensions [6]. The correlations considered in this paper for the heat transfer coefficients are, for single phase in the secondary fluid case, and in the refrigerant de-superheating and subcooling zones the correlation of Bogaer and Bölcs [7]

$$h_{r,sp} = \frac{k}{d_h} a \mathrm{Re}^b \mathrm{Pr}^c \left(\frac{\mu}{\mu_w}\right)^n,$$

where *a* and *b* depend on the value of Re, $c = 0.333\exp(6.4/(\mathrm{Pr}+30))$ and $n = 0.3/(\mathrm{Re}+6)^{0.125}$. This equation was derived for water and oil. The interested reader may see more details on this model in their original work. For the two phase flow zone, four correlations are studied; Thonon correlation [8] in which the htc is given by the correlation

$$h_{r,tp} = 1564 \frac{k_f}{d_h} \mathrm{Re}_{eq}^{-0.76} h_{fo}$$

where $h_{fo} = 0.347 \mathrm{Re}_f^{0.653} \mathrm{Pr}_f^{0.33}$, $\mathrm{Re}_{eq} = \frac{G_{eq} d_h}{\mu_f}$, and $G_{eq} = G((1-x) + x\left(\frac{\rho_f}{\rho_g}\right)$.
the author carried out experimental tests with R410A.
Han et al. correlation [9],

$$h_{r,tp} = \frac{k_f}{d_h} \mathrm{Ge}_1 \mathrm{Re}_{eq}^{\mathrm{Ge}_2} \mathrm{Pr}_f^{1/3}$$

where $\mathrm{Ge}_1 = 11.22(pitch/d_h)^{-2.83}(\pi-\beta)^{-4.5}$ and $\mathrm{Ge}_2 = 0.35(pitch/d_h)^{0.23}(\pi-\beta)^{1.48}$. Re_{eq} is defined as is shown above. It was derived for R-22 and R-410A.
Yan et al. correlation [10],

$$h_{r,tp} = 4.118 \frac{k_f}{d_h} \mathrm{Re}_{eq}^{-0.4} \mathrm{Pr}_f^{1/3}$$

which was obtained by using R134a experiments. Finally, the Longo et al. correlation is considered [11]

$$h_{r,tp} = \begin{cases} \varphi c_3^{4/3} \left(\frac{k_f^3 \rho_f^3 g h_{fg}}{\mu_f L_j q}\right)^{1/3} \; if \; \mathrm{Re}_{eq} < 1600, \\ h_{sat} + F(h_{sph} + \frac{c_{pv} q}{h_{fg}} \; if \; \mathrm{Re}_{eq} \geq 1600, \end{cases}$$

where $\varphi = A/A_{projected}$, L_j is the length of the cell considered, q is the heat flux, and c_3= 0.943. The

expression of the coefficients h_{sat}, F, h_{sph} may be found in original paper. Let us remark that this correlation differentiates between gravity-controlled condensation and forced convective condensation depending on the value of Re_{eq} (< 1600 gravity controlled and convective otherwise). In this work, the authors studied refrigerants R134a. R410A, R236fa, R600a, R290, R1270, R1234yf, R1234ze.

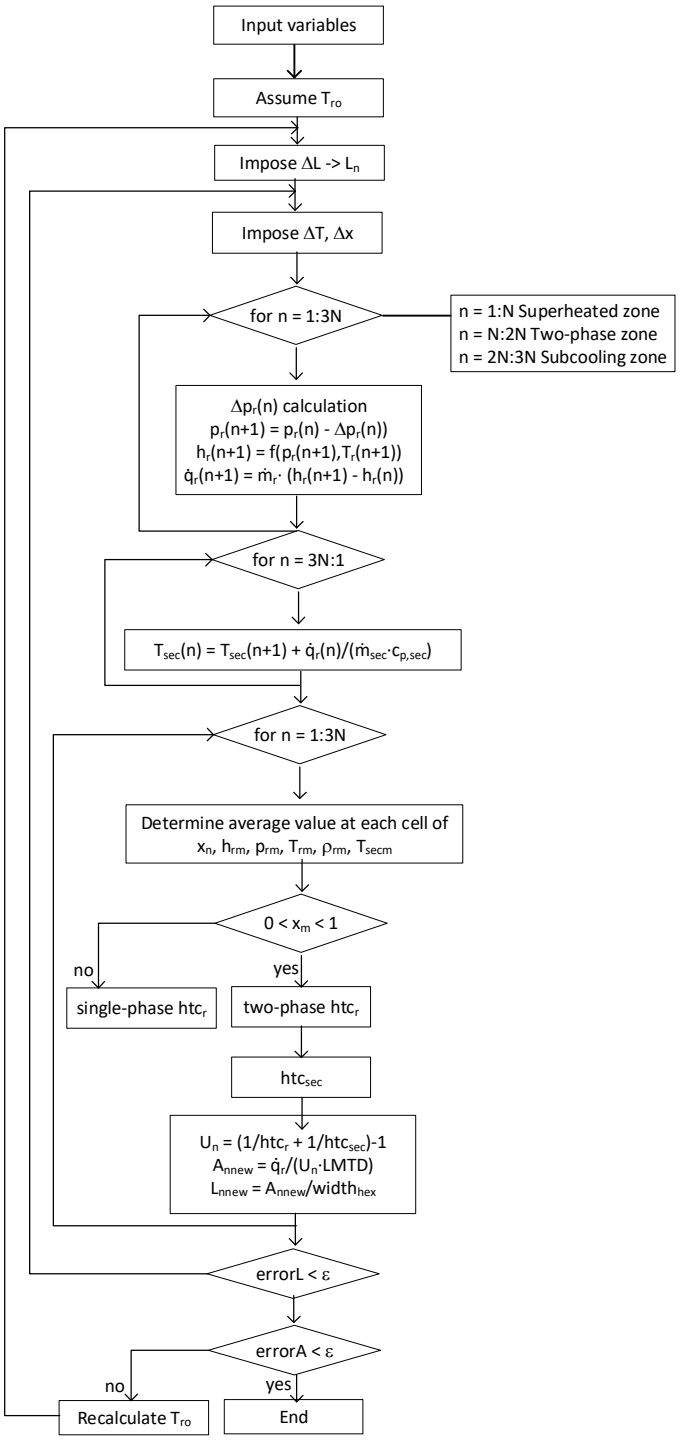

Figure 2. Flow chart.

Single-phase pressure drop is calculated by means of the correlation used by IMST-ART code [12], property of their developers. The two-phase flow pressure drop is calculated by summing the viscous friction, gravity, and acceleration components. The viscous friction component is evaluated by Chisholm correlation in which the pressure drop is given by $dp_{tp} = \phi_{tp} dp_f$, where the two-phase multiplier ϕ_{tp} is a function of the Martinelli parameter and dp_f is the viscous friction corresponding to the liquid phase flowing through the cell considered.

3. RESULTS

To analyse the goodness of the method, 52 experiments have been conducted, they consider three different refrigerants R134a, R513a, and R152a. These have been carried out in an air-to-water heat pump in which a base case was defined v_{air} = 2.5 m/s, $T_{air,i}$ = 15ºC, HR = 86.6%, $T_{water,i}$ = 30ºC and $T_{water,o}$ = 35ºC, around this, the following variables were varied; air velocity (1.5, 3.2 and 4.0 m/s), air inlet temperature (10, 20, 30ºC) and water inlet temperature 10, 20, 30, and 50ºC. The variables measured are inlet and outlet temperatures of both refrigerant and secondary fluid, pressure at the inlet, and pressure drop on both sides. These have been done by means of 1/10DIN 4-wire RTDs and high-accuracy pressure transmitters. Mass flow rate of refrigerant and volume flow rate of water have been also measured by means of a Coriolis flowmeter and an electromagnetic flowmeter respectively. Figure 3 shows condensation temperature for each of the correlations used to determine the two-phase htc and Figure 4 shows the secondary fluid outlet temperature and heating capacity. Lines with ±10% of deviation have been included in the figures to allow a better evaluation of each approach.

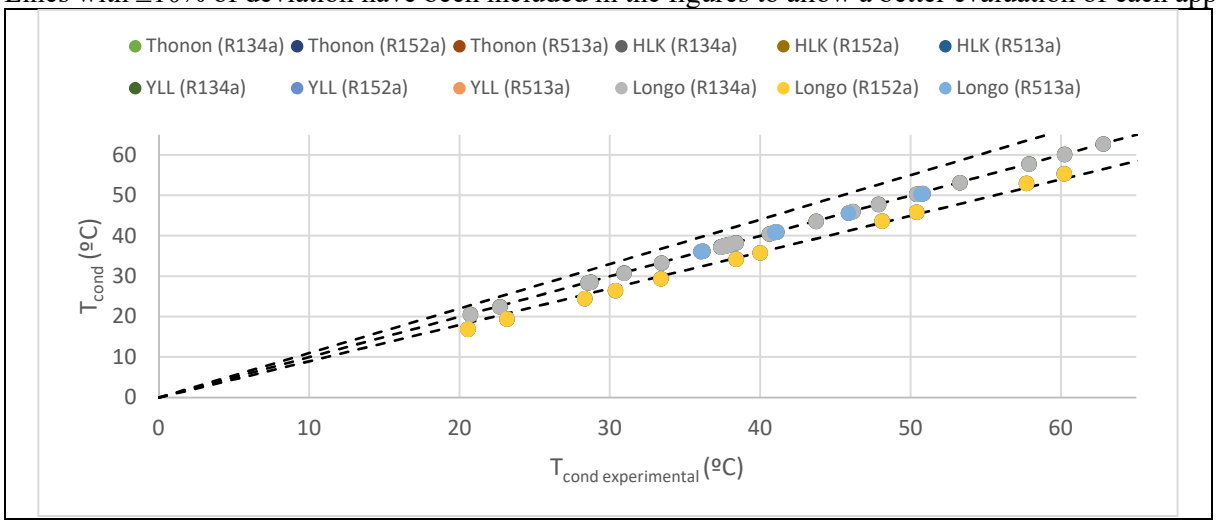

Figure 3. Condensation temperature.

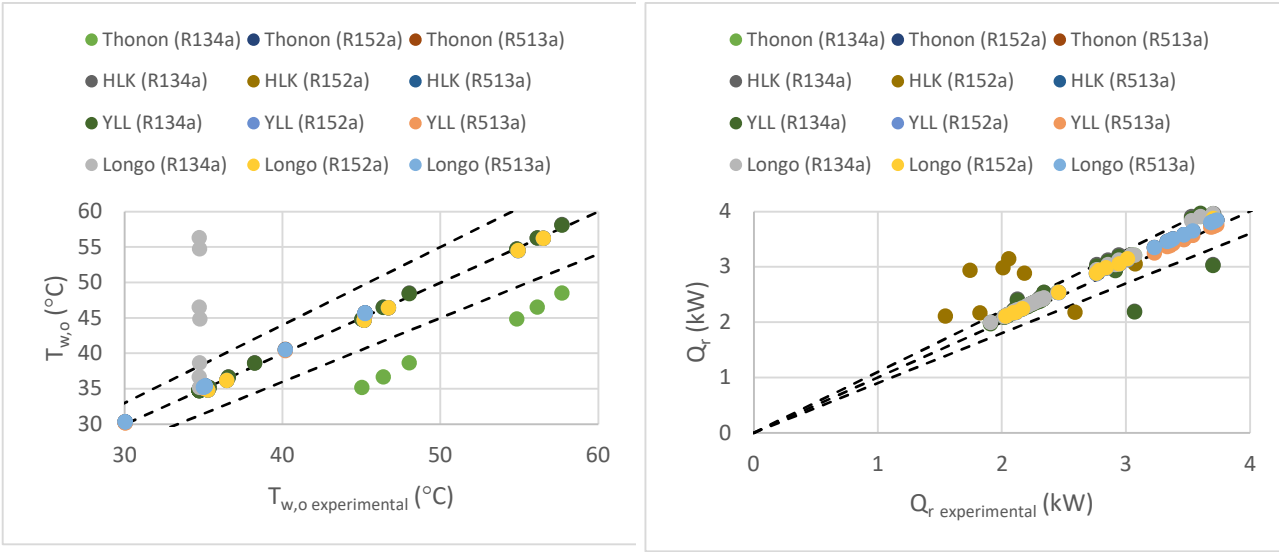

Figure 4. secondary fluid outlet temperature and heating capacity.

The mean absolute difference (MAD = $\sum|x_{measured} - x_{calculated}|/\#\text{measurements}$) and the mean absolute relative difference (MARD = $\sum|x_{measured} - x_{calculated}|/(x_{calculated}\#\text{measurements}) \times 100$) between the measured and the calculated values are gathered in Table 1. Despite the good agreement shown in the figures between the experimental and the numerical results, there is certain underestimation of the condensation temperature and an overestimation of the outlet temperature of the secondary fluid in the case of the Longo correlation for R152a and R134a tests respectively. Thonon correlation in the case of the R134a tests, underestimates the experimental data. As far as the heating capacity, Longo correlation overestimates the

R152a data and Yan et al. correlation underestimates the result corresponding to R134a. In general, the results obtained are very satisfactory and let us consider that the model will be suitable for its integration in a dynamic model.

Table 1. MAD and MARD corresponding to each model.

MAD/MARD	p_{ro}(kPa/%)	T_{seco}(ºC/%)	T_{cond}(ºC/%)	Q_{cond}(kW/%)
Thonon	2.83/0.30	0.21/0.61	1.05/3.05	0.13/4.50
HanLeeKim	2.72/0.29	0.21/0.61	1.05/3.04	0.13/4.50
YanLioLin	1.51/0.16	0.17/0.50	1.01/2.92	0.10/3.45
Longo	2.57/0.29	0.21/0.61	1.05/3.05	0.13/4.50

4. CONCLUSIONS

In this work, a model to characterise numerically a plate heat exchanger working as condenser has been presented. It relies on a 1D cell discretisation of the exchanger and a further sequential cell-by-cell analysis. Due to the influence of the htc and pressure drop in the results, a study has been carried out to analyse the goodness of the model comparing the numerical results to 52 experiments carried out for different refrigerants. The results obtained have proved to be interesting enough to be included in a more complex model for the dynamic simulation of a heat pump.

11. ACKNOWLEDGMENT

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