SIMULATION OF A CROSS-FLOW PLATE FINNED TUBE CONDENSER USING A SIMPLIFIED APPROACH

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Abstract. *The present work presents a simplified computational procedure for simulating cross-flow plate finned tube condensers which found a wide application in condensing domestic refrigeration units, as well as in air conditioning equipments. The procedure follows some published works from literature, incorporating a few different computational aspects and considering a comparison of results from a more developed dedicated software, the COILDESIGNER software from CEEE (Radermacher et. al., 2005). The main aim is to show that a simplified procedure can offer satisfactory simulation results, which can be used for a first trial set of simulation data in detailed projects of vapor compression refrigeration systems.*
Keywords: *Tube-fin heat exchanger, simulation heat exchanger, condensers, model simplified, generation entropy number.*

1. INTRODUCTION

Heat exchangers are devices used to exchange heat between two or more fluids with a temperature difference. They are present in several industry sectors and they can be found in refrigerators, air conditioning systems, solar heating devices, industrial processes and aerospace applications, among other processes. The objective of the present study is to analyze the performance of a condenser using two different programs: COILDESIGNER software from CEEE (Radermacher et. al., 2005). and a simple computational program which solves a section type mathematical model.

Nowadays, there is a great variety of computational tools used to simulate heat exchangers. Mello (2000) states that the computational models can be used in several kinds of analysis, nevertheless, the obtained results will always be limited by the type of the mathematical model used. In general, these models can be divided in two basic categories, with peculiar advantages and disadvantages: the models by sections or regions and the finite volumes models. The models by sections have shorter calculus methodologies and produce faster results. The models that discretize the heat exchanger using the finite volumes technique, are more sophisticated, requiring more processing time and produce more reliable results. A classic model by sections was developed by Rich (1976) to the design of coils. The procedure is based on the division of the heat exchanger into two parts: a superheating region and an evaporation region. In both regions it is assumed that the surface is wet, since the refrigeration devices work in this condition. Results showed that, in spite of the low complexity, the model presents satisfactory results for coil design. Oskarsson et. al. (1990-a,b) developed three models to the design of coils. The first model used the finite volume. The second model uses the section method aforementioned. The third model is based on the ε-NUT relations, and is dedicated to simulate the performance of heat pumps. Domanski (1991) presented a model for the simulation of coils and domestic air-conditioning devices. The model uses the tube-by-tube method, in which each tube is separately analyzed, considering the non-uniformity of the airflow over the circuit. Bensafi et. al. (1997) proposed a model for the design of flat plate finned tube heat exchangers to evaporators, condensers and heat exchangers with refrigerant flowing in one phase. The model considers the conditions of dry and wet surface and use the 3D finite volumes technique. The results are initially satisfactory; nevertheless, due to the lack of reliable and detailed experimental data, the model was not completely validated. Vardhan and Dhar (1998) developed a model of numerical simulation to air conditioning systems. The model discretizes the heat exchanger in several nodes and the fluid properties depend upon the surface condition (wet or dry). The model allows the test of several types of circuit for the refrigerant. The model was developed based on the cold water flow, but ir can be easily adapted, if two-phase correlations are included. A simplified model was developed by Elmahdy and Mitalas (1977) for the simulation of air conditioning systems, considering the refrigerant as cold water.

2. MATHEMATICAL MODEL

In order to solve the proposed heat transfer problem, the ε-NTU (effectiveness – number of transfer units) method will be used. The cross flow heat exchanger is divided in three regions: the superheating (sh), saturation (sat) and sub-cooling (sc) regions. These regions correspond to the thermodynamic states in which the refrigerant is found. The total heat transfer rate $\dot{q}$ is the sum of the heat transfer rates in each region:

$$\dot{q} = \dot{q}_{sh} + \dot{q}_{sat} + \dot{q}_{sc} \tag{1}$$

The actual heat transfer rate in each section can be determined considering the energy balance in a control volume involving only the refrigerant:

$$\dot{q} = \dot{m}_R (i_{R,in} - i_{R,out}) \tag{2}$$

where $\dot{m}_R$ is the refrigerant mass flow rate, $i_{R,in}$ and $i_{R,out}$ correspond to the specific enthalpies of the refrigerant in the inlet and outlet of each region, respectively. The effectiveness of the heat exchanger ε is given by:

$$\varepsilon = \frac{\dot{q}}{\dot{q}_{max}} \tag{3}$$

In Eq. (3) $\dot{q}_{max}$ is the maximum heat that could be transferred if the exchanger were a counterflow exchanger with infinite area.

The effectiveness in each region is calculated using the ε-NTU relations presented in Kays and London (1984) available in EES software. In the superheating and sub-cooling regions, where the fluid is not in a phase-change condition, Eq. (4) is used considering both fluids unmixed. In the saturation region, where the refrigerant condenses, Eq. (5) is used:

$$\varepsilon = 1 - \exp\left[\left(\frac{1}{C_r}\right)(NTU)^{0.22}\left\{\exp\left[-C_r(NTU)^{0.78}\right] - 1\right\}\right] \tag{4}$$

$$\varepsilon = 1 - \exp(-NTU) \tag{5}$$

where the capacity rate ratio C_r is given by:

$$C_r = \frac{C_{min}}{C_{max}} \tag{6}$$

In Eq. (6) C_{min} and C_{max} represent the smaller and the bigger of the hot and cold streams capacity rates, respectively. The NTU is determined based on the conductance UA and on the minimum capacity ratio, C_{min} :

$$NUT = \frac{UA}{C_{min}} \tag{7}$$

The conductance in each region is calculated from the total thermal resistance of the heat exchanger in that region, R_{total}. The conductive thermal resistance through the tubes can be neglected, since the tubes are thin and made of cupper. The conductance is obtained by:

$$UA = \frac{1}{R_{total}} = \frac{1}{R_{in} + R_{out}} \tag{8}$$

In Eq. (8) R_{in} is the convection resistance between the refrigerant and the inner surface of the tube, and R_{out} represents the resistance between the air and the surface of the plate fins and the outer tube surface due to the external convection and the conduction resistance of fins.

The thermal resistances from Eq. (8) are computed from the following equations:

$$R_{in} = \frac{1}{\bar{h}_{in} \pi D_{in} L_{t,\sec}} \tag{9}$$

$$R_{out} = \frac{1}{\eta_o \bar{h}_{out} A_{s,out} F_{\sec}} \tag{10}$$

In the previous equations (Eqs. 8 – 9) $\bar{\mathrm{h}}_{\mathrm{in}}$ and $\bar{\mathrm{h}}_{\mathrm{out}}$ stand for the average internal and external convective heat transfer coefficients, respectively, $L_{t,sec}$ means the total length of the tube of a particular region, $F_{\sec}$ means the fraction of the total length tube, η_o represents the overall external surface efficiency, and $A_{s,out}$ is the total external heat transfer area. This parameter is the sum of the total surface area of the fins, $A_{s,fin}$, and the un-finned tube wall surface, $A_{s,unfin}$, respectively. Considering the adopted heat exchanger geometry the following expressions are used for computing these geometrical variables,

$$\mathrm{A_{s,out}} = \mathrm{A_{s,fin}} + \mathrm{A_{s,unfin}} \tag{11}$$

$$A_{s,fin} = 2\frac{L_{tube}}{p_{fin}}\left(HW - N_{col} N_{row} \frac{\pi D_{out}^2}{4}\right) \tag{12}$$

$$A_{s,unfin} = \pi D_{out} L_{tube}\left(1 - \frac{th_{fin}}{p_{fin}}\right) \tag{13}$$

In Eq. (12), H and W are height and width of the front face of heat exchanger, the th_{fin} and p_{fin} are the fin thickness and pitch, respectively, L_{tube} is tube length, D_{out} is the external diameter, N is the tubes number where the subscript *col* corresponds to the number of columns and *row* corresponds to the number of rows. The overall surface efficiency is calculated as:

$$\eta_o = 1 - \frac{A_{s,fin}}{A_{s,out}}\left(1 - \eta_{fin}\right) \tag{14}$$

In Eq. (14) η_{fin} represents the fin efficiency. This parameter is computed from a solution to an annular type of constant cross-section fin, using EES internal fin efficiency function library. In order to apply this procedure it is computed and effective fin radius, $r_{fin,eff}$, which is calculated so that the fictitious annular fins have the same areas as the plane fins.

$$A_{s,fin} = 2\frac{L_{tube}}{p_{fin}}\pi\left(r_{fin,eff}^2 - \frac{D_{ext}^2}{4}\right) \tag{15}$$

Using an EES function library the fin effectiveness is thus computed as a function of the fin thickness, fin pitch, external tube radius, overall external heat transfer coefficient and the fin wall conductivity, k_{fin}, respectively. The fins are made from cooper.

The average external convective heat transfer coefficient, $\bar{h}_{out}$, for a dry air it is computed using the Kim et. al. (1999)

$$\bar{h}_{out} = \frac{J G_c c_{air}}{\mathrm{Pr}_{air}^{3/2}} \tag{16}$$

The Colburn coefficient, J, is computed as a function of number of tube rows, N_{row}, using the following equations

$$J_{kim} = j_3 1{,}043 \left[Re_{dc}^{-0,14} \left[\frac{s_v}{s_h}\right]^{-0,564} \left[\frac{p_{aleta}}{D_h}\right]^{-0,123} \left[\frac{s_v}{D_h}\right]^{1,17} \right]^{3-N_{tubos,lin}} \tag{17}$$

$$J_3 = 0{,}163 Re_{dc}^{-0,369} \left[\frac{s_v}{s_h}\right]^{0,106} \left[\frac{p_{aleta}}{D_h}\right]^{0,0138} \left[\frac{s_v}{D_h}\right]^{0,13} \tag{18}$$

The Reynolds number, Re_{Dh}, is calculated as a function of hidraulic diameter and the terms S_v and S_h are the vertical and horizontal spacing adjacent between tubes, respectively. The mass flux, G_c, is based on the minimum free flow area shown below:

$$G_c = \frac{\dot{m}_{air}}{A_{\min}} \tag{19}$$

$$A_{\min} = \left[\left(\frac{H}{S_v} - 1 \right) C + S_v - D_{out} - (S_v - D_{out}) N_f th_{fin} \right] \tag{20}$$

$$\mathrm{Re}_{\min} = \frac{G_c D_{ext}}{\mu_{air}} \tag{21}$$

In the heat exchanger considered in this analysis, the tubes are disposed in a staggered arrangement. In Eq. (20) C is the lower distance between the tubes and N_f represents the number of fins per unit length.

The average internal convective heat transfer coefficient, $\overline{h}_{in}$, in each heat exchanger region is computed using an internal library implemented in the EES software. In the regions where the refrigerant is single-phase, are used library procedures that identify whether the flow is laminar or turbulent and which correlation should be used for both the average friction factor and average heat transfer coefficient or Nusselt number, respectively. The following equations are solved in the present case for refrigerant single-phase regions.

Convective heat transfer coefficient in the laminar flow regime (Shah and London, 1978):

$$Nu = 3.66 + \frac{\left[0.049 + \left(\frac{0.020}{\mathrm{Pr}} \right) \right] Gz^{1.12}}{\left[1 + 0.065 Gz^{0.7} \right]} \tag{22}$$

where the Graetz, Gz, and Reynolds, Re, numbers are computed as

$$Gz = \frac{D_{in} \mathrm{Re} \mathrm{Pr}}{L_{t,\mathrm{sec}}} \tag{23}$$

$$\mathrm{Re} = \frac{4 \dot{m}_R}{D_{in} \mu_R} \tag{24}$$

In Eq. (24) μ_R represents the refrigerant dynamic viscosity.

Convective heat transfer coefficient in the turbulent flow regime (Gnielinski, 1978):

$$Nu = \frac{\left(\frac{f}{8}\right)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7\left(\mathrm{Pr}^{2/3}-1\right)\sqrt{\frac{f}{8}}} \tag{25}$$

In Eq. (25) the turbulent friction factor is calculated by the Petukhov (1970) correlation

$$f = \frac{1}{\left[0.79\ln(\mathrm{Re})-1.64\right]^2} \tag{26}$$

The average internal convective heat transfer coefficient, $\overline{h}_{in}$, in the two-phase region, where the refrigerant condensates is computed using the Dobson and Chato (1998) correlation, implemented in the EES software (Klein e Alvarado, 2010). The correlation is divided into wavy or annular depending on the mass flux and the modified Froude number. The mass flux is computed as:

$$G = \frac{4\dot{m}_R}{\pi D_{in}^2} \tag{27}$$

If the mass flux is greater than 500 kg/(m^2-s), the flow is assumed to be annular regardless the modified Froude number and the local heat transfer coefficient is computed as:

$$h_{in} = \frac{k_{l,sat}}{D_{in}} 0.023\,\mathrm{Re}_{D,l}^{0.8}\,\mathrm{Pr}_{l,sat}^{0.4}\left(1+\frac{2.22}{X_{tt}^{0.89}}\right) \tag{28}$$

In Eq. (28) X_{tt} represents the Lockhart Martinelli parameter and $\mathrm{Re}_{D,l}$ is the superficial liquid Reynolds number, that is computed assuming that the liquid flows alone in the tube. Both parameters are computed according to:

$$X_{tt} = \sqrt{\frac{\rho_{v,sat}}{\rho_{l,sat}}}\left(\frac{\mu_{l,sat}}{\mu_{v,sat}}\right)^{0.1}\left[\frac{(1-x)}{x}\right]^{0.9} \tag{29}$$

$$\mathrm{Re}_{D,l} = \frac{G D_{in}(1-x)}{\mu_{l,sat}} \tag{30}$$

In Eqs. (29) and (30) x represents the local vapor quality. In all the above relations the sub indices, *l*, *v*, and *sat*, stand for liquid, vapor and saturated respectively.

If the mass flux is less than 500 kg/(m^2-s), then the flow is either annular or wavy depending on the modified Froude number, which is computed by:

$$Fr_{\mathrm{mod}} = \frac{0.025\,\mathrm{Re}_{D,l}^{1.59}}{Ga^{0.5}}\left(\frac{1+1.09X_{tt}^{0.039}}{X_{tt}}\right)^{1.5} \quad for \quad \mathrm{Re}_{D,l} \le 1250 \tag{31}$$

$$Fr_{\mathrm{mod}} = \frac{1.26\,\mathrm{Re}_{D,l}^{1.04}}{Ga^{0.5}}\left(\frac{1+1.09X_{tt}^{0.039}}{X_{tt}}\right)^{1.5} \quad for \quad \mathrm{Re}_{D,l} > 1250 \tag{32}$$

In the above two expressions Ga is the Galileo number, defined as:

$$Ga = \frac{g\rho_{l,sat}\left(\rho_{l,sat}-\rho_{v,sat}\right)D_{in}^3}{\mu_{l,sat}^2} \tag{33}$$

If the modified Froude number is greater than 20, then the flow is assumed to be annular and the local heat transfer coefficient is computed according to Eq. (28). If the modified Froude number is less than 20, then the flow is assumed to be wavy and the Eq. (34) is used to compute the local heat transfer coefficient.

$$h_{in} = \frac{k_{l,sat}}{D_{in}}\left[\left(\frac{0.23}{1+1.11X_{tt}^{0.58}}\right)\left(\frac{GD_{in}}{\mu_{v,sat}}\right)^{0.12}\left(\frac{\Delta i_{vap}}{c_{l,sat}\left(T_{sat}-T_{s}\right)}\right)^{0.25} Ga^{0.25}\,\mathrm{Pr}_{l,sat}^{0.25} + ANu_{fc}\right] \tag{34}$$

The parameter A in Eq. (34) is related to the angle from the top of the tube to the liquid level:

$$A = \frac{\arccos(2\,vf-1)}{\pi} \tag{35}$$

where vf represents the void fraction computed by the Zivi (1964) correlation:

$$vf = \left[1+\frac{(1-x)}{x}\left(\frac{\rho_{v,sat}}{\rho_{l,sat}}\right)^{2/3}\right]^{-1} \tag{36}$$

The Nusselt number related to forced convection in the bottom pool, Nu_{fc}, is evaluated according to:

$$Nu_{fc} = 0.0195\,\mathrm{Re}_{D,l}^{0.8}\,\mathrm{Pr}_{l,sat}^{0.4}\sqrt{1.376+\frac{C_1}{X_{tt}^{C_2}}} \tag{37}$$

The parameters C_1 and C_2 are evaluated based on the Froude number:

$$\begin{aligned} &C_1 = 7.242 \;\; and \;\; C_2 = 1.655 && for \quad Fr > 0.7 \\ &C_1 = 4.172+5.48Fr-1.564Fr^2 \;\; and \;\; C_2 = 1.773-0.169Fr && for \quad Fr \le 0.7 \end{aligned} \tag{38}$$

The Froude number is computed as:

$$Fr = \frac{G^2}{\rho_{l,sat}^2\, g D_{in}} \tag{39}$$

The pressure loss of the air is determined by Kakaç (1998):

$$\Delta P = \frac{G_c^2}{2\rho_{air,in}}\left[f\,\frac{A_{tot}}{A_o}\frac{\rho_{air,in}}{\rho_{air,avg}} + \left(1+\left(\frac{A_o}{WH}\right)^2\right)\left(\frac{\rho_{air,in}}{\rho_{air,out}}-1\right) + K_c + K_e\frac{\rho_{air,in}}{\rho_{air,out}}\right] \tag{40}$$

where the friction factor f is based on the hydraulic diameter D_h and is computed by Kim et. al. (2000):

$$f_{kim} = f_f\,\frac{A_{s,aletas}}{A_{s,ext}} + f_t\left[1-\frac{A_{s,aletas}}{A_{s,ext}}\right]\left[1-\frac{e_{aletas}}{p_{aleta}}\right] \tag{41}$$

$$f_t = \frac{4}{\pi}\left[0.25+\frac{0.118}{\left[\frac{s_v}{D_{dc}}-1\right]^{1.08}}Re_{dc}^{-0.16}\right]\left[\frac{s_v}{D_{dc}}-1\right] \tag{42}$$

$$f_f = 1.455 Re_{dc}^{-0.656} \left[\frac{s_v}{s_h}\right]^{-0.347} \left[\frac{p_{aleta}}{D_{dc}}\right]^{-0.134} \left[\frac{s_v}{D_{dc}}\right]^{1.23} \tag{43}$$

In each section of the heat exchanger, the heat transfer rates depend on the heat transfer area. Therefore, a factor is defined in order to determine the exact size of the section. The *F* factor is defined as the ratio of the section length *L* and the total length of the heat exchanger L_{tube}

$$F = \frac{L}{L_{tube}} \tag{44}$$

Superheated Region

In this region, the F factor is determined by an iterative process. The first value is initially supposed and used to estimate the thermal resistances and conductance of the superheating region using Eqs. (8 – 10), as represented by Eq. (47). Then considering the energy balance and the effectiveness definition and relation with the NTU variable a new conductance of the superheating region is computed by Eq. (46). At this point the F parameter is recalculated into the EES software.

$$UA_{sh} = \frac{1}{R_{sh}} \tag{45}$$

$$UA_{sh} = C_{\min,sh} NTU_{sh} \tag{46}$$

Saturation Region

The same procedure described above is used on the saturation region. The conductance for the saturation region is defined as the product of the overall heat transfer coefficient and the heat transfer area. The F factor estimated for the saturation region is recalculated by an iterative process through the comparison between the conductances determined by the thermal resistance (Eq. 47) and the ε-NTU method (Eq. 48).

$$UA_{sat} = \frac{1}{R_{sat}} \tag{47}$$

$$UA_{sat} = C_{\min,sat} NTU_{sat} \tag{48}$$

Sub-cooling Region

The F factor for the sub-cooling region is determined by Eq. (49)

$$F_{sc} = 1 - F_{sat} - F_{sh} \tag{49}$$

$$UA_{sc} = \frac{1}{R_{sc}} \tag{50}$$

The ε-NTU method defines the required conductance for the sub-cooling region as:

$$NTU_{sc} = \frac{UA_{sc}}{C_{\min,sc}} \tag{51}$$

The number of entropy generation is the ratio between the entropy generation and the minimum heat capacity rate, as proposed by Bejan (1996). The entropy generation depends on the pressure losses and the heat temperature difference:

$$N_s = \frac{\dot{S}_{ger}}{\left(\dot{m}c_p\right)_{\min}} \tag{52}$$

3. RESULTS AND DISCUSSION

In order to evaluate and validate to a certain extent the computational method described above, was performed a comparison of the numerical results with the results obtained with the COILDESIGNER software from CEEE (Radermacher et al., 2005). This comparison was performed simulating one compact heat exchanger with plate finned circular tubes denoted as TC1 and with the geometrical characteristics described in Table 1. The simulations were performed considering the following operational conditions. The dry air enters the condenser at 20°C, with a volumetric flow rate in a range of 0.1 up to 0.5 m^3/s and pressure of 101325 Pa. The refrigerant is R-134a entering into the inlet tube as superheated vapor with a mass flow rate of 0.0028 kg/s, pressure of 1 MPa and temperature of 60°C in all cases.

Table 1. Geometric configuration

Geometry	TC1
Tubes arrangement	Staggered
Number of tube rows	2
Number of tubes in row	10
Inner diameter [mm]	10.21
Fin thickness [mm]	0.3302
Fin pitch [mm]	3.175
Horizontal pitch [mm]	22
Vertical pitch [mm]	25.4

In Figures 1 to 5 is shown a comparison of the results obtained from the simplified model and those calculated with the COILDESIGENR software. The variables considered in the comparisons of Figures 1 – 5 are the refrigerant and air outlet pressures, the refrigerant and air outlet temperatures and the heat transfer rate, respectively. As can be seen the maximum deviation found are small, being about – 0.5 to +0.5 % for the outlet refrigerant and air temperatures and for the outlet refrigerant pressure. For the heat transfer rate were found maximum deviation of the order of -2 to +2%; and the air pressure drop was the parameter that shows the higher variations in the order of about 5%. Considering that these deviations are for the relative errors it is found that the simulation results obtained with the proposed simplified model are very good in comparison with the well-established COILDESIGNER code. In the case of the air pressure drop it should be commented that the maximum relative errors were for low air Reynolds numbers, for which the pressure drop was very small, about 25 Pa, approximately.

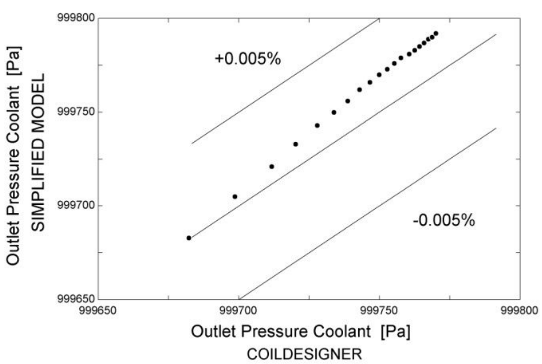

Figure 1: Deviation between the simplified model refrigerant outlet pressure values and COILDESIGNER values.

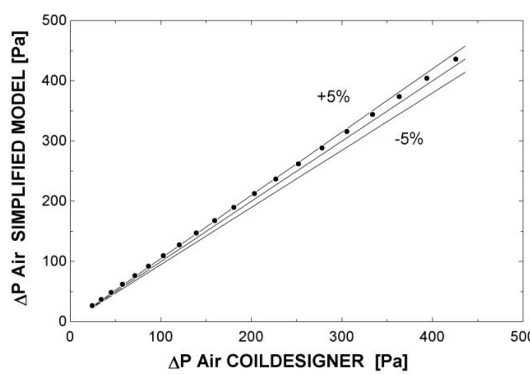

Figure 2: Deviation between the simplified model air pressure loss air values and COILDESIGNER values.

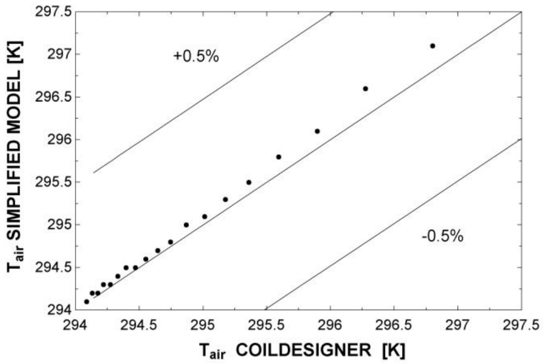

Figure 3: Deviation between the simplified model outlet air temperature values and COILDESIGNER results.

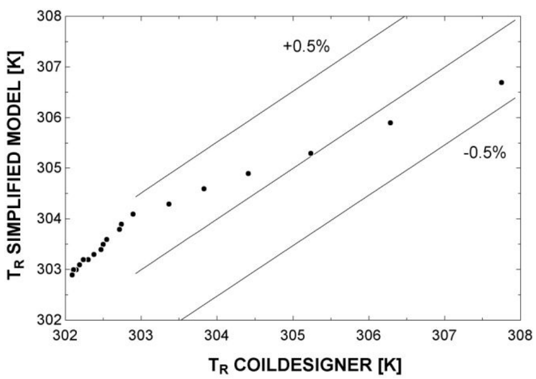

Figure 4: Deviation between the simplified model refrigerant outlet temperature values and COILDESIGNER results.

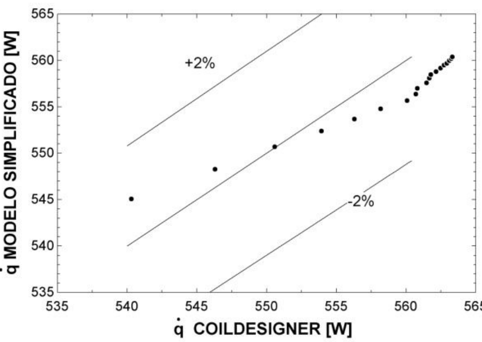

Figure 5: Deviation between the EES model heat transfer rate results and COILDESIGNER results.

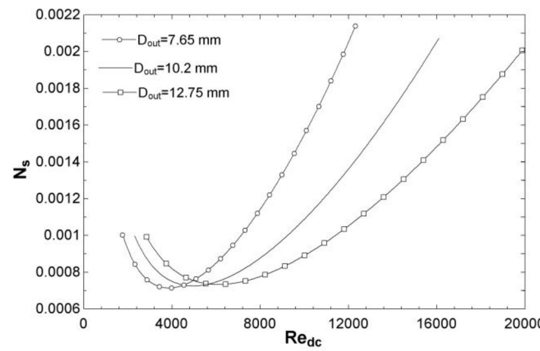

Figure 6: Entropy generation number versus Reynolds number for three outer tube diameters.

The next analysis is related to a parametric study of the entropy generation number variation (defined in Bejan 1996) as a function of three geometric parameters: outer tube diameter, fin pitch value and number of tube rows. The results are presented in Figures 6 to 8. In all these figures it is observed the same trend of the entropy generation number variation as a function of the air Reynolds number. As can be noted all the presented curves show a minimum value of the entropy generation number. This behavior is due to the combination of two irreversibility mechanisms. For small air Reynolds number the irreversibilities due to the temperature difference is the dominant mechanism. In this case the increase of the Reynolds number increases the heat transfer rate, leading to small temperature differences and to a less entropy generation. After reaching the minimum point, the irreversibility mechanism due to the pressure drop in both fluids starts to become the dominant effect leading to the augmentation of the entropy generation number for air Reynolds numbers above the minimum value.

Now considering the results displayed in Figure 6 regarding the external diameter variation influence it can be commented that this variable leads to two different trends. For the small air Reynolds numbers higher values of D_{out} produce higher irreversibilities, which can be caused mainly by the lesser convective heat transfer coefficients are consequently higher heat exchanger temperatures. When higher air Reynolds number values are considered the previous observed trend is inverted because smaller tube diameters leads to a higher pressure drops increasing the pressure drop irreversibility contribution to the overall irreversibilities, namely the entropy generation number.

In the case of Figures 7 and 8 it is seen that the results always show the same trends regarding the fin pitch and number of tubes influence on the entropy generation number. In these two cases a less fin pith and a higher number of tube rows leads to a higher irreversibilities because in the two cases the overall pressure drop is higher for all air Reynolds number values. Nevertheless, as above explained for small air Reynolds number the prominet irreversibility mechanism is constituted from the main temperature difference, leading in all case to a minimum values of the entropy generation number for a particular air Reynolds number. This behavior is less accentuated in Figure 8.

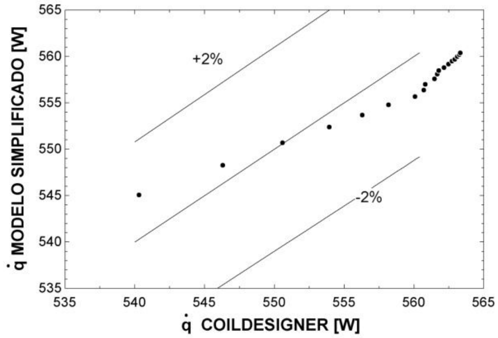

Figure 5: Deviation between the EES model heat transfer rate results and COILDESIGNER results.

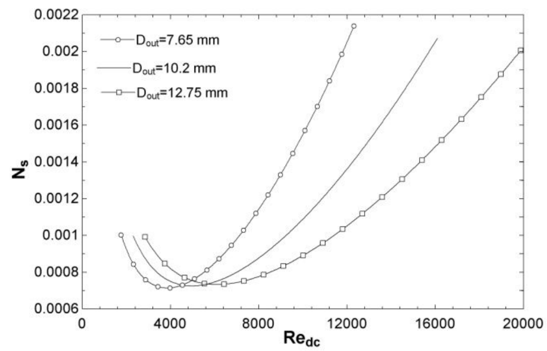

Figure 6: Entropy generation number versus Reynolds number for three outer tube diameters.

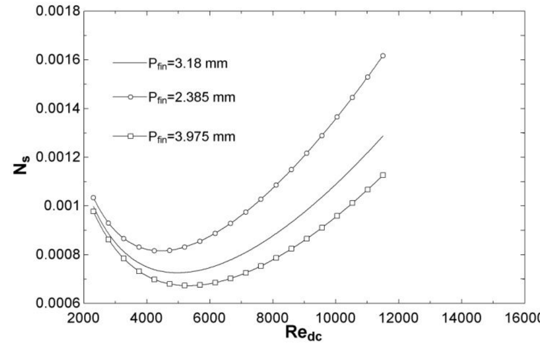

Figure 7: Entropy generation number versus Reynolds number for three different fin picht numbers.

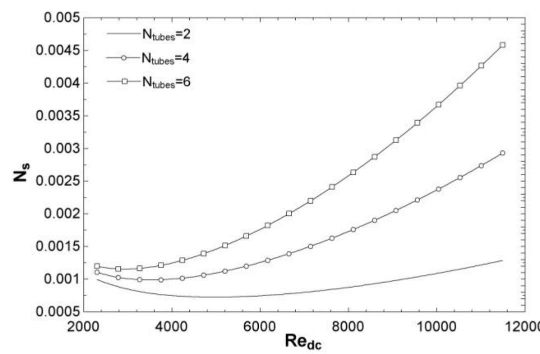

Figure 8: Entropy generation number versus Reynolds number for three different tube rows numbers.

5. CONCLUSIONS

This paper presents an analysis of two simulation models of cross-flow plate finned tube condensers: Coildesigner and a simplified model from EES. One geometric configuration of the condenser was simulated and the results for outlet refrigerant and air temperature and pressure and for the heat transfer rate were compared. It can be concluded that the utilization of the simplified model for the analysis of the performance of circular plate finned condensers provides a satisfactory results when compared to a more sophisticated software. Nevertheless, it is important to take into account the fact that the simplified model does not consider yet the in-tube fluid single and two-phase pressure drops in the 180° tube bends. However, in any case the simplified model seems to be a very practical tool for first trial calculations. Experimental data are needed in order to confirm in more details the model accuracy.

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