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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 EVAP-COND software from NIST (Domanski, 2006). 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: *Shell-Tube exchanger heat, synergy of temperature fields and velocity, multivariable numerical optimization, the total cost.*

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: the EVAP-COND software, developed by NIST (Domanski, 2006) 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:

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

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

In the previous equations (Eqs. 8 – 9) $\bar{h}_{in}$ and $\bar{h}_{out}$ 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, 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,

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

$$A_{s,fin} = 2\frac{L}{p_{fin}}\left(HW - N_{tubes} N_{row} \frac{\pi D_{ext}^2}{4}\right) \tag{12}$$

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

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 Gray and Webb (1986) correlations for plane fins.

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

The Colburn coefficient, *J*, is computed as a function of number of tube rows, N_{row}, and the Colburn coefficient for four or more rows, J_4, using the following equations

$$J = J_4 F_{webb} \tag{17}$$

$$F_{webb} = 0.991\left(2.24\,\mathrm{Re}_{\min}^{-0.092}\left(\frac{N_{row}}{4}\right)^{-0.031}\right)^{0.607(4-N_{row})} \tag{18}$$

$$J_4 = 0.14\,\mathrm{Re}_{\min}^{-0.328}\left[\frac{S_v}{S_h}\right]^{-0.502}\left[\frac{p_{fin}-th_{fin}}{D_{ext}}\right]^{0.0312} \tag{19}$$

The Reynolds number, Re_{min}, is calculated as a function of the mass flux based on the minimum free flow area, G_c, and air dynamic viscosity, μ_{air}, by the next equation

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

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

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

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.

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

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

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

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 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.

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

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

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

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):

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

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

$$f_a = 0.0267\,\mathrm{Re}_c^{F1}\left[\frac{S_v}{S_h}\right]^{F2}\left[\frac{p_{fin}}{D_h}\right]^{F3} \tag{41}$$

being the parameters F1, F2, and F3 computed by the following relations:

$$F1 = 0.764 + 0.739\frac{S_v}{S_h} + 0.177\frac{p_{fin}}{D_h} - \frac{0.00758}{N_{t,row}} \quad (42)$$

$$F2 = -15.689 + \frac{64.012}{\ln(\mathrm{Re}_{\min})} \quad (43)$$

$$F3 = 1.696 - \frac{15.695}{\ln(\mathrm{Re}_{\min})} \quad (44)$$

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:

$$N_s = \frac{\dot{S}_{ger}}{(\dot{m}c_p)_{\min}} \quad (45)$$

3. COMPUTATIONAL METHODOLOGY

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 $\mathrm{L}_{\mathrm{tube}}$

$$F = \frac{L}{L_{tube}} \quad (46)$$

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. (48). At this point the F parameter is recalculated into the EES software.

$$UA_{sh} = \frac{1}{R_{sh}} \quad (47)$$

$$UA_{sh} = C_{\min,sh} NTU_{sh} \quad (48)$$

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. 49) and the ε-NTU method (Eq. 50).

$$UA_{sat} = \frac{1}{R_{sat}} \quad (49)$$

$$UA_{sat} = C_{\min,sat} NTU_{sat} \quad (50)$$

Sub-cooling Region

The F factor for the sub-cooling region is determined by Eq. (51)

$$F_{sc} = 1 - F_{sat} - F_{sh} \tag{51}$$

$$UA_{sc} = \frac{1}{R_{sc}} \tag{52}$$

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

$$NTU_{sc} = \frac{UA_{sc}}{C_{\min,sc}} \tag{53}$$

4. RESULTS AND DISCUSSION

In order to evaluate the method described above, two different condensers were considered, TC1 and TC2. In both cases are used plate finned circular tubes as described in Table 1. The dry air enters the condenser at 20°C, with a volumetric flow rate of 0.2 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 0.028 kg/s, pressure of 1 MPa and temperature of 60°C.

Table 1. Geometric configuration

Geometry	TC1	TC2
Tubes arrangement	Staggered	Staggered
Number of tube rows	2	2
Number of tubes in row	10	10
Inner diameter [mm]	10.21	17.17
Fin thickness [mm]	0.3302	0.4064
Fin pitch [mm]	3.175	3.277
Horizontal pitch [mm]	22	38.1
Vertical pitch [mm]	25.4	44.45

In Table 2 are presented refrigerant outlet temperature and pressure loss and the total heat exchanged in both condensers, using the EVAP-COND software and the developed model.

Tabela 2. Simulation results comparison

Geometry	Refrigerants Parameters\Results	Evap-Cond	Simplified Model (EES)
TC1	Temperature (ºC)	32.1	31.4
	Pressure (kPa)	0.2000	0.2142
	Heat Transfer (kW)	0.552	0.554
TC2	Temperature (ºC)	39.4	30.5
	Pressure (kPa)	0	0.013
	Heat Transfer (kW)	0.514	0.558

It can be observed that, for the TC1 condenser, the results are very similar for the two computational programs. The refrigerant outlet temperature obtained by Evap-Cond software is 2.18% higher than the temperature of the simplified model. When the outlet pressure and the total heat exchanged are compared, the Evap-cond results are 7.1% and 0.36% lower than the simplified model, respectively.

For the TC2 condenser, the discrepancy between the results is greater, achieving up to 22.59% of the analyzed parameters. With the simplified model the outlet refrigerant thermodynamic state was a sub-cooled fluid, while with the EVAP-COND program the refrigerant exit thermodynamic state was two-phase. These differences could be caused by the different correlations which each model uses. In fact the simplified model does not considered the in-tube fluid single and two-phase pressure drops in the 180° tube bends. 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 mode details the model accuracy.

The next analysis is related to a parametric study of the entropy generation number behavior as a function of three geometric parameters: outer tube diameter, fin pitch and number of tube rows. The results are presented in Figures 1 to 3.

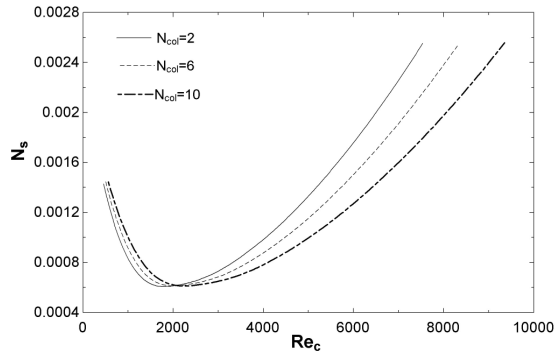

Figure 1: Entropy generation number versus Reynolds number for three different tube rows number

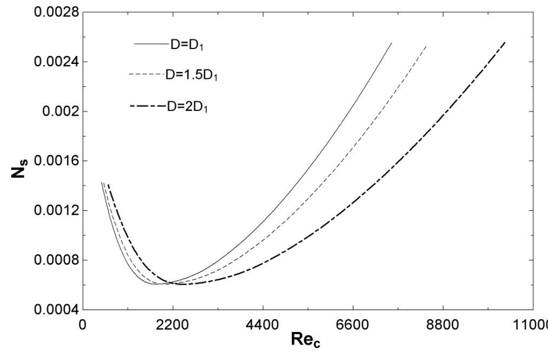

Figure 2: Entropy generation number versus Reynolds number for three outer tube diameters

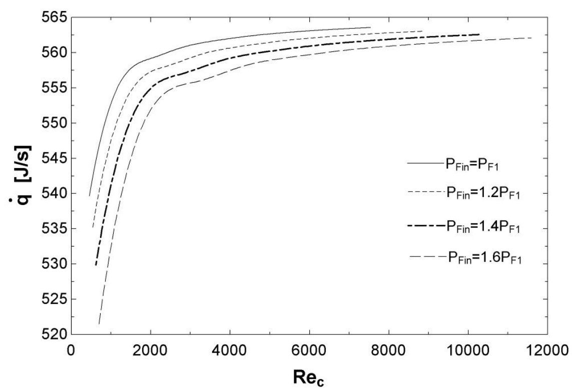

Figure 3: Heat transfer rate versus Reynolds number for four fin pitches

Figure 1 presents the behavior of the condenser when the number of tube rows is altered. It can be observed that the minimum entropy generation number remains almost constant (i.e. the Reynolds number is the same) when is increased the number of tube rows. For low Reynolds numbers, N_s, is influenced mainly by the heat exchanger temperature differences, while for high Reynolds numbers N_s depends strongly on the heat exchanger pressure losses. The same behavior is observed when the outer tube diameter is altered, as shown in Fig. 2.

Figure 3 shows the influence of the fin pitch on the heat transfer rate. An increase on the pitch causes a diminution on the heat exchanged in the whole Reynolds number interval. Nevertheless, for high Reynolds numbers this influence is not significant. For low Reynolds number values the diminution of the fin pitch produces a sharply increase of the external heat transfer convective coefficient, increasing in the same way the heat transfer rate. For high Reynolds number values the external air velocity is practically the same for different fin pitch and this last parameter does not influences in a great extent the heat transferred in the condenser.

5. CONCLUSIONS

This paper presents an analysis of two simulation models of cross-flow plate finned tube condensers: Evap-cond and a simplified model from EES. Two geometric configurations of the condenser were simulated and the results for outlet refrigerant temperature and pressure losses 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 model. 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 mode details the model accuracy.

A parametric analysis using an entropy generation number was also accomplished. The results show that for air-flow Reynolds number of 2000 up to 2200 the simulated condenser works at a minimum entropy generation condition.

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