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A COMPARISON BETWEEN TWO NUMERICAL MODELS TO PREDICT THE THERMAL-FLUID BEHAVIOR OF WIRE-ON-TUBE CONDENSERS

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Abstract. *This paper presents two numerical models to predict the stationary thermal-fluid behavior of wire-on-tube condensers used in domestic refrigerators. The first one is a distributed model and it is based on a finite volume discretization of the fundamental equations of mass, momentum and energy conservation, giving detailed spatial variation of properties. The second one it is a zonal model, which gives a bulk description of properties, since the condenser is divided in three or two zones. For both models simplified assumptions are made: serpentine tubes curvature and viscous effects are neglected and the refrigerant flow and heat transfer are considered one-dimensional. Empirical correlations available on the literature were applied to predict the refrigerant and air heat transfer coefficients and friction factor on the refrigerant side. The numerical codes were implemented on EES®. The numerical results were compared with results in a work available in the literature and showed that although the zonal model is a simpler modeling approach, it describes the thermal-fluid behavior of the wire-on-tube condenser with reasonable results. Also, the distributed model (based on the finite volume method) describes a more accurate thermal-fluid behavior of the condenser, since it consists on more detailed formulation, with a higher computational cost. Those analyses were carried out aiming a simplified stationary simulation model for a domestic refrigerator, in which the condenser model should give good results and reasonable computational cost.*

Key-words: *wire-on-tube condensers, zonal model, distributed model, numerical simulation.*

1. INTRODUCTION

Wire-on-tube condensers are devices commonly used in domestic refrigerators. In the refrigeration system they are responsible to reject heat absorbed during compression process and refrigeration effect. Due to its importance on the system, experimental investigation and mathematical modelling of wire-on-tube condensers are important to better understand the phenomena involved, especially during condensation. Many mathematical models to describe the thermal- fluid behavior of those condensers are available in the literature, most of them presented in household refrigerators simulation models.

Xu (1996) developed a dynamic simulation model for a household refrigerator, in which the condenser model was based on the solution of mass, momentum and energy equation followed by a distributed formulation in space and implicit in time. Two-phase flow was modeled assuming a void fraction model proposed by Zivi (1964). The author considered empirical correlations to compute heat transfer coefficients, friction factors.

Klein (1998) developed a numerical model to simulation the thermal-fluid behavior of a domestic refrigerator. The author proposed a three zone model to investigate the wire-on-tube condenser, the resulting system of algebraic equations was solved. Empirical correlations for heat transfer coefficients and friction factor were considered. Results of global simulation of the refrigerator were compared with experimental data, showing well agreement.

Hermes (2000) developed numerical models to simulate all components of the same refrigerator analyzed by Klein (1998). For the wire-on-tube condenser, the numerical model was based on the solution of mass, momentum and energy equations using finite volume method, resulting on a spatial characterization of all variables. The author compared numerical results with Klein (1998) work, showing all agreements of numerical data. Later, Hermes et al. (2008) applied a simpler version (with pressure spatially constant) of this numerical model to simulate start-up and cycling transients of household refrigerators.

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Bendapudi et al. (2004) compared two formulations to dynamic simulate the behavior of heat exchangers: moving boundary and finite volume methods. The first one is usually used to overcome converge problems normally associated with distributed models, based on finite volume method.

Ameen et al. (2006) used finite element method to simulate the behavior of a wire-on-tube condenser from a retrofitted domestic refrigerator. A three-zone model was proposed and an analysis of the position where condensation occurs were done, as a function of refrigerant mass flow rate and ambient temperature. Numerical results were confronted with experimental data, also showing well agreement. Quadir et al. (2002) carried a similar study on wire-on-tube heat exchangers based on finite element modelling.

Gonçalves et al. (2009) used a three zone model for the wire-on-tube condenser coupled with models for other components of a domestic refrigerator to simulate the system under steady-state conditions. Empirical data obtained from experiments were used to reliable simulate the global behavior of the refrigerator. Results were compared against experimental data showing good results.

Despite more sophisticated from the mathematical side, distributed models carry several advantages against three zone model, since they can give spatial variation of variables and hence, a more detailed analysis of the thermal-fluid behavior of the refrigerant flow. However, the solution of the equations of a distributed model can result on non-convergence of the resulting system of equations, especially on the transition between saturated to subcooled regions, where thermal properties present higher gradients. The three zone model are generally applied when the main interest is the global behavior of the system, since it is simpler from both mathematical and computational views.

The main objective of this paper is to analyze two formulations used to model heat exchangers: the distributed model, based on finite volume method and three-zone model, in which the heat exchangers is divided in two or three regions, depending on the condition of the refrigerant. In this paper, those formulations will be applied to model a wire-on-tube condenser, typically used in household refrigerators.

2. MATHEMATICAL MODEL

This section presents the numerical formulation used to build the distributed model and the three zone model. This wire-tube condenser considered in this paper was obtained from Hermes (2000) and it is showed below:

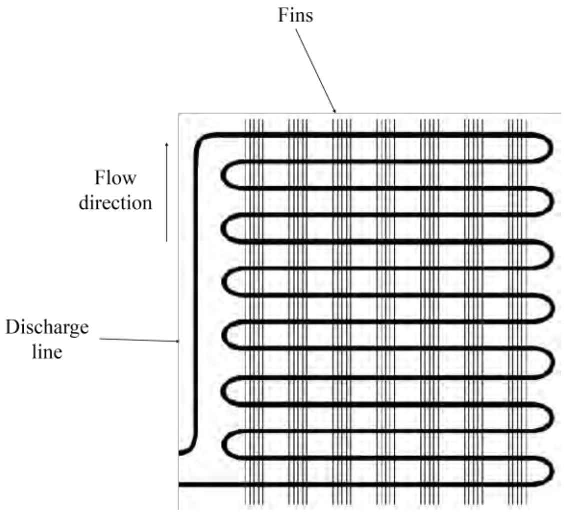

Figure 1. Wire-on-tube condenser

Some assumptions were made to simplify the mathematical modelling, including:

- Flow is assumed to be one-dimensional, fully developed and steady state
- R134a is considered to behave as a Newtonian fluid
- Heat transfer is one-dimensional and heat conduction is neglected
- Potential energy and viscous dissipation are neglected
- Condenser tube was modeled as a horizontal tube with constant diameter
- Fins were considered as uniformly spatially distributed
- Heat transfer coefficients and friction factor were calculated using empirical correlations available in the literature
- Air-side heat transfer is governed by natural convection and air temperature is assumed to be uniform

2.1 Distributed model

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In the distributed model the condenser tube is divided into a number of control volumes and the governing equations are applied to each of them. The mass velocity of the refrigerant, G, can expressed in terms of the product of the density, ρ, and the average velocity, u, according to Eq. (1):

$$G = \rho u \tag{1}$$

The momentum equation obtained represents the momentum balance over the control volume (Fig. 2), according to Eq. (2):

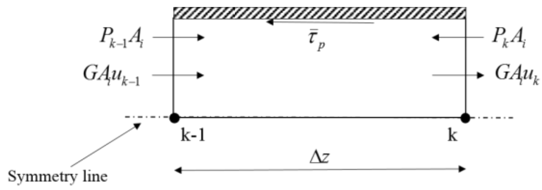

Figure 2. Momentum equation over a general control volume

$$P_{k-1}A_i + GA_iu_{k-1} - \bar{\tau}_p \pi D_i - P_k A_i + GA_iu_k = 0 \tag{2}$$

The wall shear stress is expressed according to Darcy friction factor and the dynamic pressure, according to Eq. (3) (WHITE, 1986):

$$\bar{\tau}_p = \frac{f}{4}\frac{Gu}{2} \tag{3}$$

To compute the friction factor was considered the correlation proposed by Churchill (1977):

$$f = 8\left[\left(\frac{8}{\text{Re}}\right)^{12} + \frac{1}{\left(\left(2{,}457\ln\left[\left(\frac{7}{\text{Re}}\right)^{0{,}9} + 0{,}27\frac{\varepsilon}{D_i}\right]\right)^{-16} + \left(\frac{37530}{\text{Re}}\right)^{16}\right)^{1{,}5}}\right]^{1/2} \tag{4}$$

The Reynolds number can be expressed by:

$$\text{Re} = \frac{GD_i}{\mu} \tag{5}$$

For two-phase flow, the absolute viscosity is computed for the liquid phase. The energy equation is obtained considering the energy balances inside and outside the condenser control volume, according to Eq. (6) and Eq. (7):

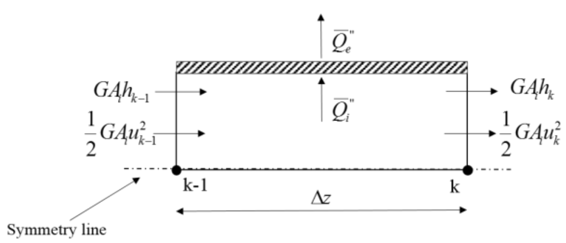

Figure 3. Energy equation over a general control volume

$$GA_i h_{k-1} + \frac{1}{2} GA_i u_{k-1}^2 - GA_i h_k - \frac{1}{2} GA_i u_k^2 - \overline{Q}_i'' \pi D_i \Delta z = 0 \tag{6}$$

$$\overline{Q}_i'' \pi D_i \Delta z = \overline{Q}_e'' \pi D_e \Delta z \tag{7}$$

The internal and external heat fluxes are given by Newton's cooling law. Hence, the internal and external heat fluxes can be given by Eq. (8) and Eq. (9):

$$\overline{Q}_i'' = \hbar_{\text{int}} \left(T - T_{wall} \right) \tag{8}$$

$$\overline{Q}_e'' = \hbar_{eq} \left(T_{wall} - T_{amb} \right) \tag{9}$$

The heat transfer coefficient on the refrigerant side is calculated by Dittus and Boelter (1930) correlation for single phase flow, Eq. (9), and by Shao and Granryd (1995) correlation for two-phase flow, Eq. (10):

$$\hbar_{\text{int}} = 0{,}023 \frac{k}{D_i} \text{Re}^{0,8} \, \text{Pr}^{0,3} \tag{9}$$

$$\hbar_{\text{int}} = \begin{cases} \dfrac{k_l}{D_i} \left\{ 0{,}084 \, \text{Pr}_l^{1/3} \left[\dfrac{h_v - h_l}{cp_l \left(T_r - T_c \right)} \right]^{1/6} \text{Re}_v^{0,67} \right\}, \text{Re}_v > 24000 \\ \dfrac{k_l}{D_i} \left\{ 015{,}9 \, \text{Pr}_l^{1/3} \left[\dfrac{h_v - h_l}{cp_l \left(T_r - T_c \right)} \right]^{1/6} \text{Re}_v^{0,15} \right\}, \text{Re}_v < 24000 \end{cases} \tag{10}$$

Where Reynolds number, Re_v, is given by Eq. (11):

$$\text{Re}_v = \frac{x G D_i}{\mu_l} \left(\frac{\rho_l}{\rho_v} \right)^{0,5} \tag{11}$$

The air-side heat transfer coefficient was calculated by Lefreve e Ede (1956) correlation for the discharge line and by Tanda and Tagliafico (1997) correlation for the finned region (see Fig. 1), given by Eq. (12) and Eq. (13), respectively:

$$\hbar_{ext} = \frac{k_{ar}}{L_{des}} \left\{ \frac{4}{35} \frac{\left(272 + 315 \, \text{Pr}_{ar} \right) L_{des}}{\left(64 + 63 \, \text{Pr}_{ar} \right) D_e} + \frac{4}{3} \left[\frac{7 Ra_{L_{des}} \, \text{Pr}_{ar}}{5 \left(20 + 21 \, \text{Pr}_{ar} \right)} \right]^{1/4} \right\} \tag{12}$$

$$\hbar_{ext} = 0{,}66 \frac{k}{H_c} \left(Ra_{H_c} \frac{H_c}{D_e} \right)^{0,25} \left[1 - \left(1 - 0{,}45 \left(\frac{D_e}{H_c} \right)^{0,25} \right) e^{-s_w / Z} \right] \tag{13}$$

Where the geometrical parameters are showed below:

$$s_w = \frac{\delta_w - D_w}{D_w} \tag{14}$$

$$Z = \left(\frac{28{,}2}{H_c} \right)^{0,4} \left(\frac{s_w^{\;0,9}}{s_t} \right) + \left(\frac{28{,}2}{H_c} \right)^{0,8} \left(\frac{264}{T_{wall} - T_{amb}} \right)^{0,5} s_w^{-1,5} s_t^{-0,5} \tag{15}$$

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$$s_t = \frac{\delta_t - D_e}{D_e} \tag{16}$$

Radiation heat transfer was taken into account and the heat transfer coefficient was incorporated in the external coefficient by a linearization of its radiative heat flux term. Hence, the total external heat transfer coefficient can be expressed by Eq. (17):

$$\hbar_{eq} = \hbar_{ext} + \varepsilon\sigma\left(T_{wall}^2 + T_{amb}^2\right)\left(T_{wall} + T_{amb}\right) \tag{17}$$

For the finned region, an equivalent external diameter was considered to take account of the presence of fins:

$$D_{eq} = D_e + \frac{D_w L_w N_w}{L_{cnd} - L_{dis}} \tag{18}$$

Two-phase flow was modeled using Hughmark (1962) void fraction model. The author proposed that the two-phase flow can be presented as a mixture of liquid and vapor bubbles with concentration of bubbles maximum in the center of the tube and decreasing monotonically in the radial direction. The void fraction is obtained by the implicit solution of Eqs. (19), (20) and (21), as shown in Hermes (2000):

$$\alpha = K_H \left[1 + \left(\frac{1-x}{x}\right)\frac{\rho_v}{\rho_l}\right]^{-1} \tag{19}$$

$$Z_H = \left[\frac{G D_i}{\mu_l + \alpha(\mu_v - \mu_l)}\right]^{1/6} \left[\frac{1}{g D_i}\left(\frac{Gx}{\rho_v\left\{\left[1+\left(\frac{1-x}{x}\right)\frac{\rho_v}{\rho_l}\right]^{-1}\right\}\left\{1-\left[1+\left(\frac{1-x}{x}\right)\frac{\rho_v}{\rho_l}\right]^{-1}\right\}}\right)^2\right]^{1/8} \tag{20}$$

$$K_H = \frac{-1.3740408 + 1.436221.Z_H}{1 + 1.5816751.Z_H - 0.00098720926.Z_H^2} \tag{21}$$

Additionally, for two-phase flow, in Eq. (1), density and average velocity are computed for the mixture.

2.2 Three-zone model

The three-zone model or zonal model considers that the condenser can be divided into zones, based on the condition of the refrigerant: superheated, saturated and subcooled (see Fig. 4). Governing equations are solved for each region giving the thermal-fluid behavior of the condenser.

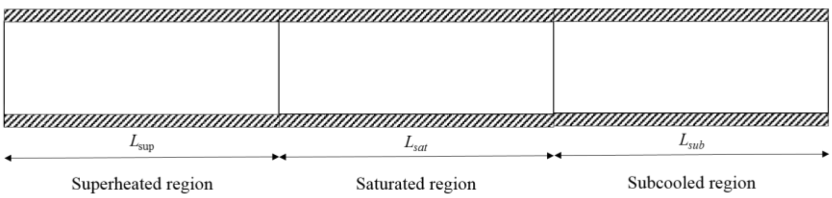

Figure 4. Three zone model

Similar equations were developed to build this numerical model. For each region, the heat transfer rate can be expressed in terms of the heat transfer coefficient, surface area and temperature difference. Eqs. (22) and (23) are used for the superheated region:

$$\dot{Q}_{sup} = \hbar_{eq} \pi D_e L_{sup}\left(\overline{T}_{wall,sup} - T_{amb}\right) \tag{22}$$

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$$\dot{Q}_{\sup} = \hbar_{\text{int}} \pi D_i L_{\sup} \left(\bar{T}_{ref} - \bar{T}_{wall,\sup} \right) \tag{23}$$

For the saturated region, internal heat transfer rate can be expressed as variation of refrigerant enthalpies, according to Eq. (24):

$$\dot{Q}_{sat} = \dot{m}\left(h_v - h_l\right) \tag{24}$$

For the subcooled region, similar expressions to the superheated region were developed:

$$\dot{Q}_{sub} = \hbar_{eq} \pi D_e L_{sub} \left(\bar{T}_{wall,sub} - T_{amb} \right) \tag{25}$$

$$\dot{Q}_{sub} = \hbar_{\text{int}} \pi D_i L_{sub} \left(\bar{T}_{ref} - \bar{T}_{wall,sub} \right) \tag{26}$$

Additionally, since the subcooled region may not necessarily exist, the calculation must consider this possibility. All properties were averaged in each zone and the resultant system of algebraic equations were solved until the convergence was reached for properties and length of thermal zones.

To compute the heat transfer coefficients and friction factor, the same empirical correlations showed before were considered. Additionally, pressure drop on each region is also accounted for:

$$\Delta P = \frac{f}{2\rho} \frac{L}{D_i} G^2 \tag{27}$$

3. NUMERICAL METHODOLOGY

Numerical procedures were implemented on EES® (Klein, 2004) to solve the governing equations for each control volume of the condenser for the distributed model and for each region for three zone model. Both procedures were implemented by an estimation-correction algorithm, until the convergence can be achieved for all variable involved.

Additionally, for the distributed model, were considered second order approximations for heat transfer and shear stress, in order to minimize high variations of thermal and friction effects on the transition of single phase to two-phase flow. Those approximations are suggested by Hermes (2000) and Hermes (2006).

Figure 5 shows the algorithms implemented for both numerical procedures, where on the left side it is shown the distributed model and on the right side the three-zone model:

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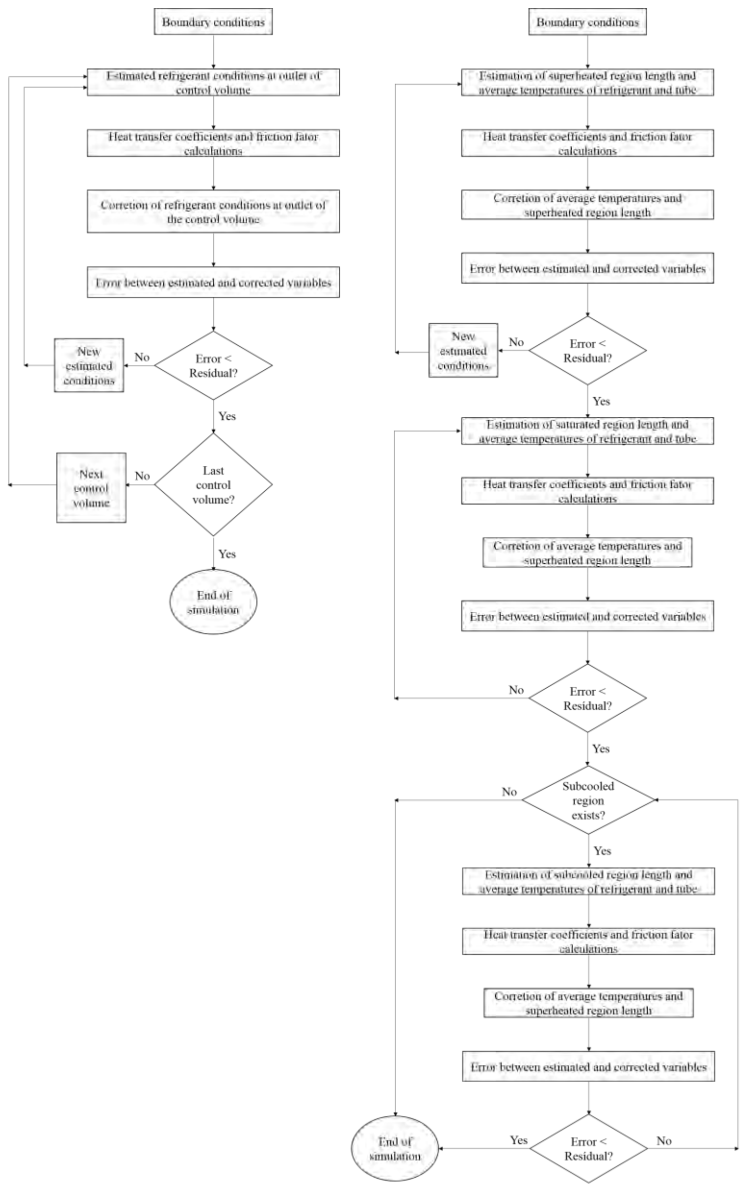

Figure 5. Algorithms for both numerical models

Due to non-convergence of the numerical solution on the distributed model, probably related to higher gradient of properties during the transition to single phase to two phase conditions, a sub-relaxation factor of 0.90 was applied during correction of estimated variables. No convergence problems were observed for the three zone model.

4. RESULTS AND DISCUSSION

Table shows main geometrical parameters of the wire-on-tube condenser. Numerical models were validated with numerical results obtained from Hermes (2000), who also proposed a numerical model for this condenser.

Table 1. Main geometrical parameters

Geometrical parameter	Value
External diameter	4.76 mm
Internal diameter	3.34 mm
Equivalent length of condenser tube	9 m
Number of fins	104
Fin diameter	1.5 mm
Fin length	855 mm

Tube spacing	56 mm

The numerical model developed by Hermes (2000) was based on the solution of mass, momentum and energy equations, followed by a distributed formulation on space and implicit formulation in time. The thermal conduction on the wall was also considered. The advantage of the implicit formulation in time is the possibility to run both steady and transient simulations.

Hermes (2000) used boundary conditions obtained from Klein (1998), who developed a steady state simulation model for the same domestic refrigerator. The boundary conditions for three cases are presented in Table 1:

Table 1. Boundary conditions for simulations

Boundary condition	Case 1	Case 2	Case 3
Ambient temperature (°C)	32	43	54
Refrigerant mass flow rate (kg/h)	1.46	1.78	2.21
Refrigerant inlet pressure (bar)	11.80	15.53	20.05
Refrigerant inlet temperature (°C)	74.9	89.7	102.6

Table 2 shows numerical results obtained by both models presented on this paper and the model developed by Hermes (2000). In the simulations using the distributed model were considered 300 control volumes and a residual target of 10^{-4}. This last parameter were also considered for simulations using three zone model.

Table 2. Validation of numerical models

Results	Numerical model	Case 1	Case 2	Case 3	Simulation time (s)
Refrigeration capacity (W)	Hermes (2000)	79.0	88.8	98.3	-
	Distributed model	79.8	88.7	95.4	129.8
	Three zone model	80.5	91.1	99.3	0.4
Outlet refrigerant temperature (°C)	Hermes (2000)	42.6	56.6	67.6	-
	Distributed model	40.9	56.2	66.9	131.6
	Three zone model	39.4	55.4	67.0	0.4
Outlet refrigerant condition*	Hermes (2000)	3 °C	3.3%	11.8%	-
	Distributed model	4.5 °C	3.0%	14.7%	155.5
	Three zone model	6.0 °C	0.9 °C	8.9%	0.4

*Quality (%) or degree of subcooling (°C)

From results showed in Table 2, it can be noticed that due to the similarity between the distributed model and Hermes (2000) model those results are closer than the three zone model, which tend to overestimate results for all cases. This probably occurred since this model is a simpler version of the distributed model. However it can be seen that this model provides results almost immediately and the distributed model presents a higher simulation time due to its solution complexity. All simulation were carried in an Intel® Core™ i7 CPU @ 2.20GHz with 8GB RAM computer.

The distributed model can be used to analyze spatial variations of a variety of variables, since the governing equations are solved for each control volume that represents a portion of the tube. The three-zone model can be used to analyze average parameters for each region. In Fig. 6 are shown the refrigerant and wall tube temperature profiles for both numerical models developed in this paper (distributed model on the left side, and three-zone model in the right side).

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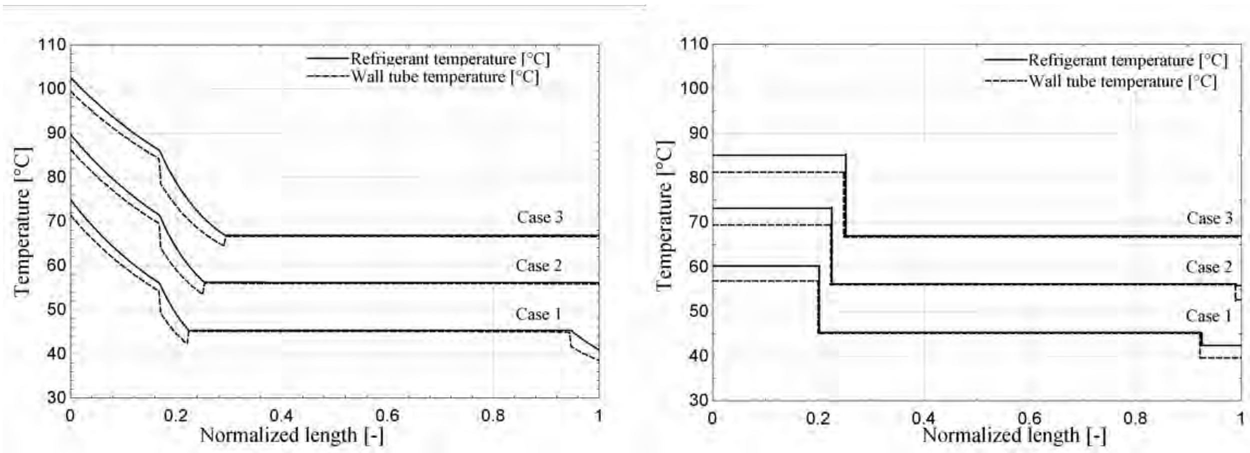

Figure 6. Temperature profiles for both numerical models

As expected, the temperature profiles obtained by the distributed model considers spatial variations for each node of all control volumes. The three zone model presents uniform temperature profiles for each region of the condenser. The temperature profile given by the distributed model show a smooth temperature variation, included inflection points related to the end of the discharge line and beginning of refrigerant saturation. This last inflection can be explained by the fact that for heat transfer during two-phase flow, the refrigerant side heat transfer coefficient is higher than the one for heat transfer during single phase flow. Although this inflection can be explained this way, this is not probably the real behavior of the temperature profiles.

Table 3 shows a comparison between both numerical models regarding the length of each zone of the condenser and average temperatures of refrigerant and tube:

Table 3. Comparison of results between numerical models

Case	Numerical model	Superheated Region			Saturated Region			Subcooled Region		
		Length (m)	$\overline{T}_{ref}$ (°C)	$\overline{T}_{sup}$ (°C)	Length (m)	$\overline{T}_{ref}$ (°C)	$\overline{T}_{sup}$ (°C)	Length (m)	$\overline{T}_{ref}$ (°C)	$\overline{T}_{sup}$ (°C)
1	Distributed model	1.980	61.1	58.0	6.510	45.3	45.1	0.510	42.9	40.0
	Three zone model	1.813	60.3	56.9	6.492	45.4	45.2	0.695	42.4	39.6
2	Distributed model	2.250	74.1	70.9	6.750	56.2	56.0	0	-	-
	Three zone model	1.995	73.2	69.5	6.911	56.3	56.0	0.0934	55.9	52.7
3	Distributed model	2.610	85.7	82.4	6.390	66.9	66.6	0	-	-
	Three zone model	2.231	85.1	81.3	6.769	67.0	66.7	0	-	-

According to Table 3, it can be noticed slightly difference on calculation of regions lengths between the models presented in this paper. For the superheated region, the three zone model tends to predict a smaller length for all cases analyzed. For the saturated region, in cases 2 and 3 the three zone models predicted a longer length and in case 1 almost the same length was predicted by both models. Finally for the subcooled region, in case 2 the three zone model predicted existence of this region although no subcooled condition was predicted by the distributed model. In case 1, the distributed model predicted a smaller region than the three zone model. The behavior of both models were very similar on predicting the average temperature for the refrigerant and tube, as expected.

5. CONCLUSIONS

This paper presents a comparison between two numerical models to predict the thermal-fluid behavior of wire-on-tube condensers, typically applied on domestic refrigerators. The first model, called distributed model, was based on the discretization of the momentum and energy equations for each control volume of the condenser. This model was used to analyze spatial variations of parameters and properties of the refrigerant. The second model, called three zone model, was based on the division of the condenser on three regions and the governing equations were applied to each of them. Algorithm were implemented on EES® (Klein, 2004) based on estimation and correction of the variable involved. Numerical validation of results were done considering a work available on the literature, where a numerical model was developed. Results showed relative good agreement for both models. The distributed model is computationally more sophisticated, hence takes more time to fully characterize the thermal-fluid behavior, giving better results than the three zone. This last one gives results almost immediately and showed to be a useful tool to perform simulation, despite of its

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inherent mathematical simplicity. The above models can be considered a useful tool for a first-step thermal project of a desired wire-on-tube condenser.

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