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MODELLING OF A HOT-WALL CONDENSER USED IN DOMESTIC REFRIGERATORS

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Abstract. *This work presents a mathematical model of hot-wall condensers, commonly used in domestic refrigerators. The model predicts the heat transfer characteristics of the condenser and the geometric and operating parameters affecting the condenser tube length and capacity. The condenser tube of this model is divided into elemental units consisting of control volume, which contains the refrigeration fluid, the wall of the tube, and a bi-dimensional plate in contact with the atmospheric air. The heat transfer characteristics of the condensers are analyzed by considering the conduction heat transfer between the tube and the wall. Some correlations for inner heat transfer coefficient and pressure drop are analyzed depending on temperature, fluid phase, and the fluid in the tube, also taking into consideration the ambient temperature variation in heat transfer coefficient. The results obtained are compared with experimental data available in the literature. The hypothesis shows how the parameters of the condenser can influence the heat transfer of this equipment. The model presented here can be used for the design and analysis of hot-wall condensers in the future.*

Keywords: ***Hot-wall condenser, refrigeration, two-phase flow.***

1. INTRODUCTION

In recent years a major concern facing the scientific community is related to the rational use of natural resources. The effect of halogenated refrigerants, CFCs, and other gases on the stratospheric ozone layer, the need for clean energy sources (non polluting) and improved performance of power stations and cooling systems have motivated extensive research and large investments. These studies focus especially on solutions to reduce production costs and energy consumption.

In terms of refrigeration, such studies focus on refrigerant fluids that are less harmful to the environment and on analyzing the behavior of refrigeration system components in order to improve their energy efficiency and reduce production costs.

Some of these components such as compressors, heat exchangers, have been extensively studied, namely condensers and evaporators and expansion devices, particularly capillary tubes, used in small refrigeration systems (up to 10 kW). Well designed heat exchangers not only improve system performance, but also reduce the space occupied and the amount of manufacturing material, for a given cooling capacity.

There are currently two basic configurations for constructing condensers that operate in refrigeration cycles of domestic refrigerators. The first is the condenser outside the cabinet, in direct contact with the environment, and the second is the internal condenser, whose heat exchange does not take place directly with the environment. The first configuration is found in most refrigerators manufactured in Europe and South America and Asia, while the second configuration is commonly found in refrigerators produced in North America.

The most commonly used model to manufacture external condensers is the wire-and-tube type. These condensers comprise a single steel or copper pipe, to form the serpentine arrangement of a multi-pass tube. A cylindrical cluster of wires, serving as fins, is symmetrically welded to the outer surface on both sides along the normal direction of the tube, due to the higher thermal resistance of air in relation to the coolant.

The refrigerant fluid flows inside the tube as it undergoes phase change and the air flows externally, exchanging heat by radiation and natural convection with the outer surface of the tube and with the cluster of fins. Figure 1 shows that the refrigerant, before entering the condenser, flows over a region of the tube with no fins, known as the entry region of the condenser. The refrigerant fluid in this region is in superheated vapor state and also transfers heat to the air. Along the condenser, the refrigerant reaches the saturation state and the convective condensation process occurs afterwards, until the refrigerant exits the condenser, in the saturated state or subcooled liquid state.

The wire-and-tube condensers have been used in refrigeration systems and have also been widely studied since the 1950s. Some major difficulties have been encountered in studies on this type of condenser, namely satisfactory results concerning the heat transfer between its outer surface, the tube and wires, and the ambient air, which can often occur by natural convection or by forced air convection.

Placing the refrigerator near the wall, which often confines the flow, decreases the thermodynamic efficiency of refrigerators with internal condensers and also with external condensers. The current trend in the industry is to produce more compact refrigerators to take up as little space as possible and strive to combine this trend with maintenance or improved efficiency of such appliances.

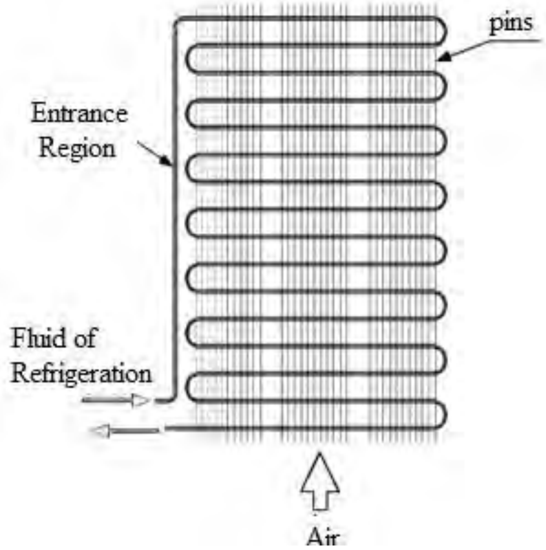

Figure 1 – Hot-and-wire condenser scheme.

One of the solutions found was to connect the condenser tubes and the evaporator to a uniform contact surface with the environment, for instance a plate, enlarging the heat exchange area while reducing the physical space.

Therefore, the most common model used to manufacture internal condensers was developed, the hot-wall. These condensers are formed in the same way as the wire-and-tube type, by a single steel or copper tube, arranged in a serpentine multi-pass shape. The serpentine is welded to the inner surface of the lining plate of the refrigerator's side walls, usually consisting of carbon steel, responsible for the heat transfer with the environment. Figure 2(b) schematically shows the condenser located inside the cabinet between the insulating foam and steel plate.

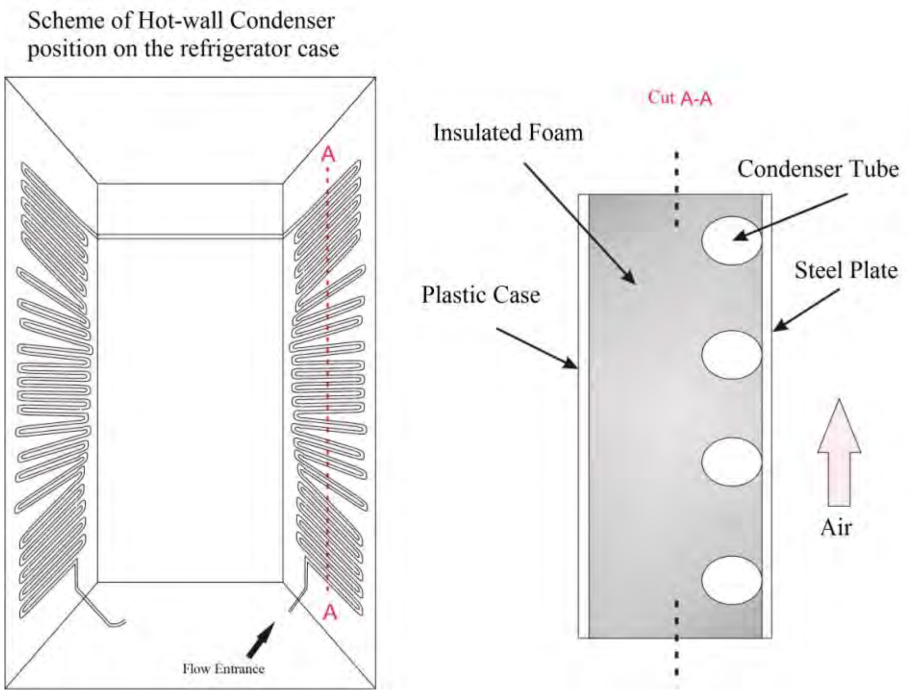

Figure 2 – Scheme of the hot-wall condenser.

Gupta and Gopal (2008) reported that the hot-wall condensers, besides displaying better aesthetics and dust protection in relation to the wire-and-tube condensers, also eliminates possible moisture condensing on the outer surface of the refrigerator. Therefore, many manufacturers have chosen to use this type of condenser.

Unlike the wire-and-tube condenser, the refrigerant fluid flows inside the tube as it changes phase and exchanges heat with the steel plate, which in turn transfers heat by natural convection and by radiation for the air to flow externally. Figure 2 shows one possible configuration for the hot-wall condenser in which the refrigerant, before entering the condenser, flows along a region of the tube which is not in direct contact with the side plates. The refrigerant then reaches the sides of the refrigerator and transfers heat to the environment through the plates, condensing along the length of the condenser. Along the condenser, the refrigerant reaches the saturation state and then the convective condensation process occurs until the refrigerant exits the condenser in the saturated or subcooled liquid state.

The convective condensation process within the pipe is quite complex, as there may be a variety of flow patterns. The vapor-liquid two-phase flow system established along the pipe depends on the average velocity and the properties of each phase. Although such properties do not generally vary during the convective condensation, the phase change causes an appreciable relative speed variation between the two phases, and the flow pattern can drastically change over the tube [Carey (1992)]. The flow patterns often observed in convective condensation are: round, undulated, slug and bubbles.

There are few studies that have investigated hot-wall condensers. Some researchers such as: Rebora and Tagliafico (1997), Bansal and Chin (2002), Gupta and Gopal (2008) and Zhang *et al.* (2010) developed computational models to analyze hot-wall condensers with different approaches to solve heat transfer and internal flow as well as other fundamental aspects.

Rebora and Tagliafico (1997) designed a horizontal refrigerator where the evaporator is located along the four internal walls and the condenser is positioned along the four external walls of the cabinet, separated only by a thermal insulator. A numerical modeling of the condenser was also elaborated to analyze the heat transfer behavior, compared with the experimental data obtained.

According to Rebora and Tagliafico (1998), the transient system requires from 3 to 5% of the heat capacity used throughout the refrigerator's period of use, permanently, thus, given the complexity of the transient equations and computational cost, the solution of the system's governing equations only for the permanent system is more advantageous and representative of the heat transfer problem as a whole.

A numerical model to analyze hot-wall condensers was developed by Bansal and Chin (2002), in which the two-phase flow of the refrigerant inside the tube was considered homogenous and the steel plate responsible for heat exchange with the environment was considered as a one-dimensional fin. The solution of the equations governing the internal flow and external heat transfer is iterative, solving the internal flow with the estimated heat transfer rate, and then calculating the parameters that will be used to determine the heat released in the fin. With the updated heat transfer rate value, the internal flow solution process to convergence restarts.

In the model developed by Gupta and Gopal (2008) the flow and heat transfer solution along the condenser is simultaneously obtained for each element of the condenser. All surfaces participating in the heat transfer, including the tube and the weld between the tube and plate are considered as overlapping fins, in other words, the circumference of the tube is regarded as a flat plate, with their respective contact regions with the heat insulator, steel plate and welding. The heat transfer of each simulation element is considered one-dimensional.

In their results analysis, Gupta and Gopal (2008) determined a higher pressure drop in the two-phase flow region and in that same region the heat transfer coefficient value has the highest values, when compared with the other flow regions.

In the work of Zhang *et al.* (2010), the heat transfer rate is modeled by considering that the plate in contact with the condenser tube is a fin, with its length similar to the relative spacing between the tubes. Zhang *et al.* (2010) show the results comparing different configurations to arrange the condenser tubes over the steel plate, showing the variations of the thermodynamic properties along the flow, and also the heat transfer capacity of each configuration.

This paper proposes a model for analyzing the performance of hot-wall condensers used in domestic refrigerators. This analysis encompasses modeling the flow of the refrigerant fluid inside the condenser tube, the heat transfer by conduction through the cabinet wall, and heat transfer by natural convection and radiation between the outer lining plate and the environment. The model is validated by comparing its results with experimental data available in the literature and/or obtained by other models.

2. MATHEMATICAL MODELLING

The proposed model is developed based on energy, mass and momentum conservation equations for the steady state regime. The variation of properties is only considered in the axial direction, which requires the use of constitutive relations to calculate the pressure drop due to friction and the heat transfer coefficient by convection. Figure 3 illustrates the condenser discussed in this work.

As the refrigerant in the condenser inlet is in the superheated steam state and can become subcooled liquid, the flow along the tube, including a part of the condenser's inlet region, is divided into three regions: single-phase superheated steam, two-phase vapor-liquid and single-phase subcooled liquid. Figure 4 schematically shows the flow configuration

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along the condenser. Two different models for heat transfer between the condenser and the environment are addressed: (i) considering the direct contact hot-plate fluid; (ii) using a two-dimensional conducting modeling for the hot-wall.

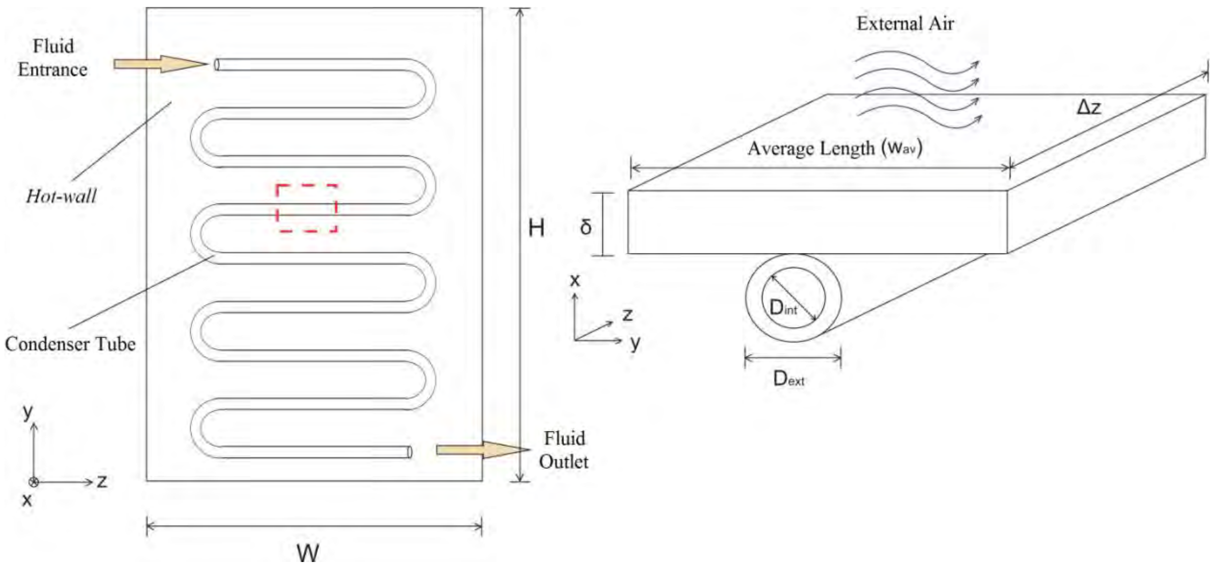

Figure 3 – Scheme of the hot-wall condenser modeling.

The hypotheses used for modeling the condenser are: (a) the condenser is viewed as a straight horizontal tube of constant diameter, that is, the curvature effects of the serpentine are disregarded; (b) the serpentine spacings are considered uniform; (c) The refrigerant's flow and heat transfer are considered one-dimensional; (c) The refrigerant is considered as a Newtonian fluid, oil free; (d) The mechanical balance is also regarded, meaning that the pressure is uniform at any cross-section of the tube and surface tension effects are disregarded; (e) the following are disregarded: the axial heat conduction in the fluid, the viscous energy dissipation, the potential energy variation in the flow along the condenser and the flow pulsation, characteristic of refrigerators operating with positive displacement machines; (f) the equation for the radiative heat transfer rate between the hot plate and the external environment is linearized; (g) the thermophysical properties of the material of the tube wall and hot plate are assumed constant; (h) the two-phase flow over the condenser is considered homogenous, that is, the flow is mathematically treated as a pseudo single-phase flow, whose properties are obtained considering the designation and properties of each individual phase. Consequently, both phases have the same velocity, pressures and temperatures in any cross-section along the tube.

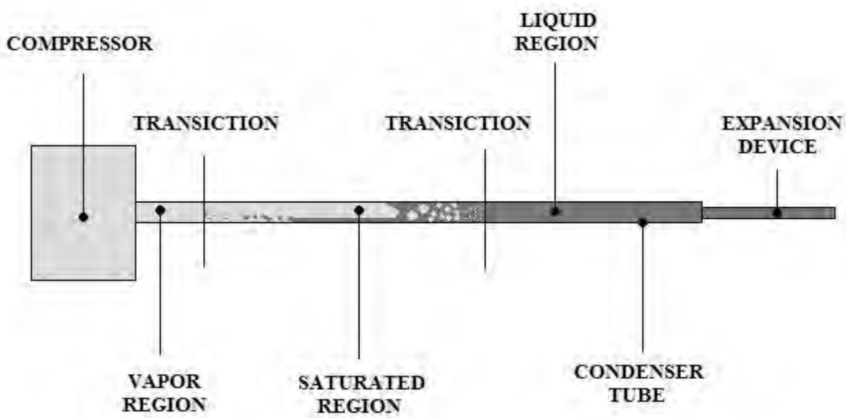

Figure 4 - Layout of the refrigerant flow along the condenser.

2.1 Governing Equations

Considering the above assumptions, the mass, energy and momentum conservation equations for the refrigerant are respectively given by:

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$$\frac{dG}{dz}=0 \tag{1}$$

$$\frac{d(Gu)}{dz}=-\frac{dp}{dz}-f\frac{Gu}{8}\frac{P_i}{A} \tag{2}$$

$$\frac{d(Gi_o)}{dz}+H_i\frac{P_i}{A_i}(T_r-T_c)=0 \tag{3}$$

where $G = (\rho u)$ is the refrigerant mass flow [kg/m²s], ρ is the specific mass [kg/m³], u is the average velocity of the refrigerant in a given cross section of the tube [m/s], z is the coordinate along the tube [m], p is the flow pressure [Pa], P_i is the internal perimeter of the tube [m], f is the Darcy friction factor, A_c is the cross-sectional area [m^2], H_i is the internal heat transfer coefficient by convection [W/m²K], T_r is the refrigerant temperature [K], T_c is the temperature of the condenser tube wall [K], and i_o is the specific enthalpy of the refrigerant stagnation [J/Kg],

Two approaches are proposed to analyze the heat transfer between the condenser and the environment:

(i) **Direct contact model.** In this case the thermal resistance of the tube wall is disregarded, that is, it is assumed that the fluid is in direct contact with the hot plate. Thus, the temperature distribution on the hot plate is obtained by the equation,

$$\frac{\partial^2 T_p}{\partial z^2}+H_i\frac{P_i}{k_P A_c}(T_r-T_p)-(H_e+H_{rad})\frac{w_{av}}{k_P A_c}(T_p-T_a)=0 \tag{4}$$

where k_p is the thermal conductivity of the hot plate material [W/mK], w_{av} is the average width of the hot plate section relative to the control volume considered [m], H_e is the heat transfer coefficient by convection between the hot plate and the external environment [W/m²K], $A_c=w_{av}\delta$ is the cross-sectional area of the hot plate [m^2], δ is the thickness of the hot plate [m], T_p is the temperature of the hot plate [K], T_a is the temperature of the external environment [K] and H_{rad} is the heat transfer coefficient by radiation [W/m²K].

(ii) **Two-dimensional plate model**. In this approach, it is assumed that heat is transferred from the refrigerant to the condenser tube wall and then to the hot plate in the axial direction, where the heat transfer is modeled two-dimensionally. The heat conduction in the tube wall is analyzed in a distributed manner in the axial direction and overall in the radial direction, and in the hot plate the heat conduction is analyzed two-dimensionally.

Thus, the temperature distributions along the tube wall and the hot plate are obtained by the equations.

$$\frac{\partial^2 T_c}{\partial z^2}+\frac{P_i H_i}{k_c A_{cc}}(T_r-T_c)-\frac{P_e k_c}{k_c A_{cc} dy}(T_c-T_p)=0 \tag{5}$$

$$\frac{\partial^2 T_p}{\partial z^2}+\frac{\partial^2 T_p}{\partial y^2}+\frac{k_c(T_c-T_p)}{k_p\delta dy}-\frac{(T_p-T_a)[H_{rad}+H_e]}{k_p\delta}=0 \tag{6}$$

where k_c is the thermal conductivity of the tube wall material [W/mK], $A_{cc} = [\pi(d_e^2 - d_i^2)/4]$ is the area of the pipe annulus [m^2], d_e and d_i, are respectively, the external and internal diameters of the tube [m] and P_e [m] is the external perimeter of the tube [m].

Thus, the proposed model consists of equations (1) to (3), which must be resolved, respectively, to calculate the distributions of the mean flow velocity, u, pressure, p, and stagnation enthalpy, i_o, and equations (4) or (5) and (6), depending on the model chosen for the heat transfer between the condenser and the environment, to calculate the temperature of the tube wall, T_c and the hot plate T_p.

In addition to these equations the following correlations are used: (i) Churchill (1977) to calculate the single-phase friction factor, (ii) Lin *et al.* (1991) to calculate the two-phase friction factor, (iii) Gnielinski (1976) to calculate the heat transfer coefficient by internal convection in the single-phase region; (iv) Mueller (1997) to calculate the heat transfer coefficient by internal convection in the two-phase region; (v) Cicchitti *et al.* (1960) to calculate the two-phase viscosity, and (vi) Churchill and Chu (1975) to calculate the heat transfer coefficient by convection between the hot wall and the environment. To determine the thermophysical properties of air and the refrigerant the REFPROP module 8.0 was used, Lemmon *et al.* (2007), which allows to analyze various refrigerants and their mixtures.

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2.2 Boundary Conditions

To analyze the flow along the condenser requires solving the governing equations in the superheated steam region, in the two-phase region and in the compressed liquid region. In Figure 5, the line between points 1 to 4 schematically illustrates a common flow situation along the condenser. The sections located between points 1-2, 2-3 and 3-4 correspond, respectively, to the superheated steam region, to the two-phase region and to the subcooled liquid region.

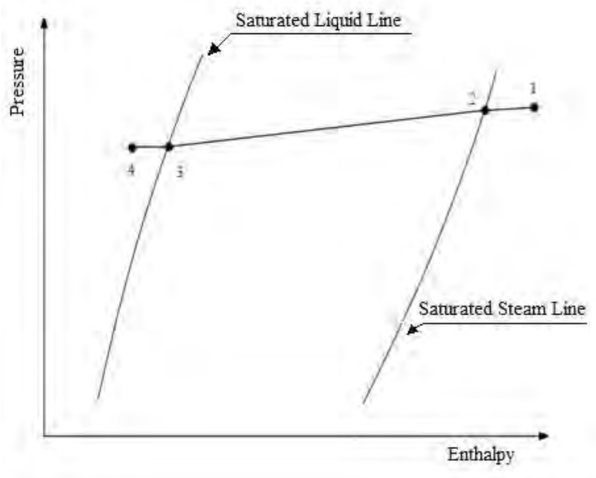

Figure 5. Schematic *p-h* diagram, representing the thermodynamic states of the refrigerant fluid along the condenser.

The refrigerant fluid that leaves the compressor discharge line as superheated steam enters the condenser in the thermodynamic state represented by point 1 in Figure 5 and begins to transfer heat to the environment, which decreases its temperature. The pressure of the refrigerant in the flow also decreases, due to viscous effects, until reaching the saturation pressure of the fluid in question, point 2 of Figure 5. From this point the changing phase process begins until the saturated liquid state is achieved, point 3 of Figure 5. Then, the refrigerant in the subcooled liquid state continues to transfer heat to the environment until it reaches the outlet of the condenser, point 4 of Figure 5.

The specific mass and temperature of the refrigerant in the compressed liquid and superheated steam regions are calculated using the pressure and specific enthalpy values of the refrigerant, calculated by equations (2) and (3) in the form: $\rho_r=\rho_r(p,i)$ and $T_r\ (p,i)$. In the two-phase region, the specific mass and temperature of the refrigerant are calculated similarly, keeping in mind that in this region $\rho=[\rho_l+\ \alpha(\rho_v-\rho_l)]$, where ρ_l is the specific mass of the saturated liquid, ρ_v id the specific mass of saturated steam and α is the void fraction, the ratio between the cross-sectional area of the tube occupied by the vapor phase and the total cross sectional area.

The conditions defining the beginning of the two-phase flow regions, 2-3, and the subcooled liquid flow, 3-4, must be known. The governing equations of the flow along the condenser, Equations (1) to (3), are first order partial differential equations, therefore to be solved they require only one boundary condition.

At the condenser inlet the mass flow, pressure and temperature of the refrigerant are known. With the temperature and pressure of the refrigerant, the other thermodynamic properties at the inlet are determined. Furthermore, it is considered that at the condenser inlet, the tube wall is at the same temperature as the refrigerant. Thus, the conditions at the inlet of the condenser are:

$$z = z_1 = 0 \qquad \begin{cases} G = G_1 \quad p = p_1 & T_r = T_c = T_1 \\ \rho_r = \rho_r(p_1, T_1) & i = i(p_1, T_1) \end{cases} \tag{7}$$

For the direct contact model the hypothesis used is that there is a null derived condition for the wall temperature at the condenser outlet. For the two-dimensional plate model, it is assumed that all sides of the hot plate are isolated.

The beginning of the two-phase region is identified by comparing the calculated pressure with the saturation pressure relative to the temperature of the refrigerant. If the flow pressure is equal to or less than the saturation pressure for the local flow temperature, it is assumed that it reached the flow saturation region. The end of the two-phase region is identified by the name of the mixture, meaning that the beginning of the flow region of subcooled liquid corresponds to the location along the tube where $x\ =\ 0,$ and the thermodynamic properties of that location are those relative to the saturated liquid state.

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3. SOLUTION METHODOLOGY

The numerical solution of the differential equation system, both along the condenser tube and the hot plate is obtained by using the Finite Volume method. Therefore, the domain is divided into *m* control volumes with nodes located in the inlet and outlet sections of these volumes (see Fig. 6), and the governing equations are integrated over each control volume. The average values of the source terms of these equations, $\bar{S}^{\phi}$, where ϕ represents a generic dependent variable per unit mass, are approximated by the arithmetic average of the respective values in the inlet and outlet sections of the control volume,

$$\bar{S}^{\phi} = \frac{1}{\Delta z}\int_{k-1}^{k} S^{\phi} dz = \frac{S^{\phi,k} + S^{\phi,k-1}}{2} \tag{8}$$

where the lower *k* value represents the nodal points along the computational mesh.

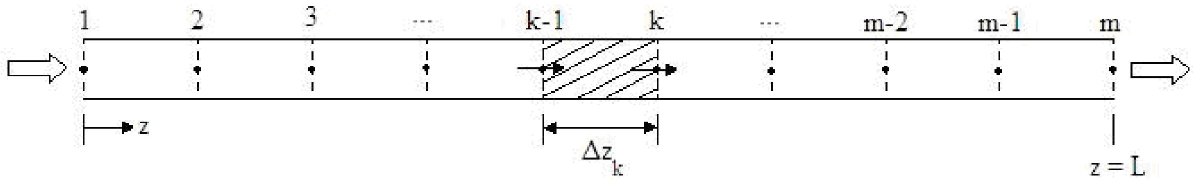

Figure 6 – Control volumes along the one-dimensional domain discretized.

This procedure enables to obtain an algebraic equations system that is resolved point to point along the condenser. The algorithm to solve this equations system is:

1. Provide the condenser's geometrical data and known properties at the condenser's inlet tube. Provide an estimated temperature range of the hot plate.
2. Calculate the other flow properties of the refrigerant at the condenser inlet using the known properties (k=1);
3. Increment k=k+1;
4. Calculate the average flow velocity, *u*, Eq.(1);
5. Calculate the pressure, *p*, by Eq. (2);
6. Calculate the stagnation enthalpy, i_o , by Eq. (3);
7. If the model adopted is the Direct Contact, proceed to Step 8. If the model adopted is the two-dimensional plate, calculate the temperature of the condenser tube wall, T_c , by Eq. (5);
8. Update the thermodynamic and thermophysical properties, and the empirical parameters of the flow;
9. Return to step 4 until the convergence of the variables;
10. Return to step 3 until the last volume control;
11. If the model adopted is Direct Contact, calculate the temperature of the hot wall, T_p, by Eq. (4) and proceed to step 12. If the model adopted is the two-dimensional plate, calculate the temperature field of the hot plate by Eq (6)
12. Compare the values of T_p calculated in step 11 with those estimated in step 2. If the convergence criterion has been achieved, terminate the algorithm solution, otherwise, return to step 2.

5. RESULTS

The results obtained are compared with experimental and computational data available in the literature. However, there are few experimental studies found in the literature on hot wall condensers for domestic refrigerators. Therefore, the comparison of results was restricted to the thermal load of the condenser and the pressure drop across the condenser of a specific static-type domestic refrigerator, operating with the refrigerant fluid HFC-134a or R134a under different operating conditions. An analysis of the influence of mass flow, ambient temperature and title on the performance of the condenser is performed in order to emphasize one of the model's potentiality.

The condenser studied is the same one used by Bansal and Chin (2002), and Table 1 shows its geometric dimensions and characteristics. The refrigerant fluid used in the tests is R134a, whose thermophysical properties are calculated using the REFPROP 8.0 module, Lemmon *et al.* (2007). The material of the condenser and the hot plate is steel, whose thermophysical properties are also shown in Table 1.

In tests conducted by Bansal and Chin (2002), the refrigerant fluid has a superheating rate of 5 ^{o}C at the condenser inlet. The flow saturation temperature and the external environment temperature are provided, as well as the flow rates used in each case. Table 2 shows the operating conditions for the five tests conducted by Bansal and Chin (2001).

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Table 1 - Geometric characteristics and properties of the condenser tested (Bansal and Chin, 2001).

Geometric characteristics and properties	
Internal diameter of the tube [m]	0.004
External diameter of the tube [m]	0.00475
Single side panel area [m^2]	0.9786 (0.591 x 1.656 m)
Back panel area [m^2]	0.8736 (0.624 x 1.4 m)
Total tube length [m]	23.21
Internal heat transfer area (tubo) [m^2]	0.1990
External heat transfer area (plate) [m^2]	3.240
Thermal conductivity of steel [W/mK]	50
Plate thickness [m]	0.004
Absolute roughness [mm]	0.00058

Table 2. Operating conditions tested (Bansal and Chin, 2001).

Test Conditions	**Case 1**	**Case 2**	**Case 3**	**Case 4**
Ambient temperature [ºC]	20	25	20	25
Inlet temperature [ºC]	37	42	40	45
Inlet pressure [bar]	8.15	9.37	8.87	10.16
Flow range [g/s]	0.7 – 1.2			
Emissivity	0.92			

Figures 7(a) to 7(d) show for Cases 1 to 4 of Table 2, respectively, the comparisons between the heat capacities of the condenser calculated in this work, according to the Direct Contact model and the experimental data of Bansal and Chin (2001).

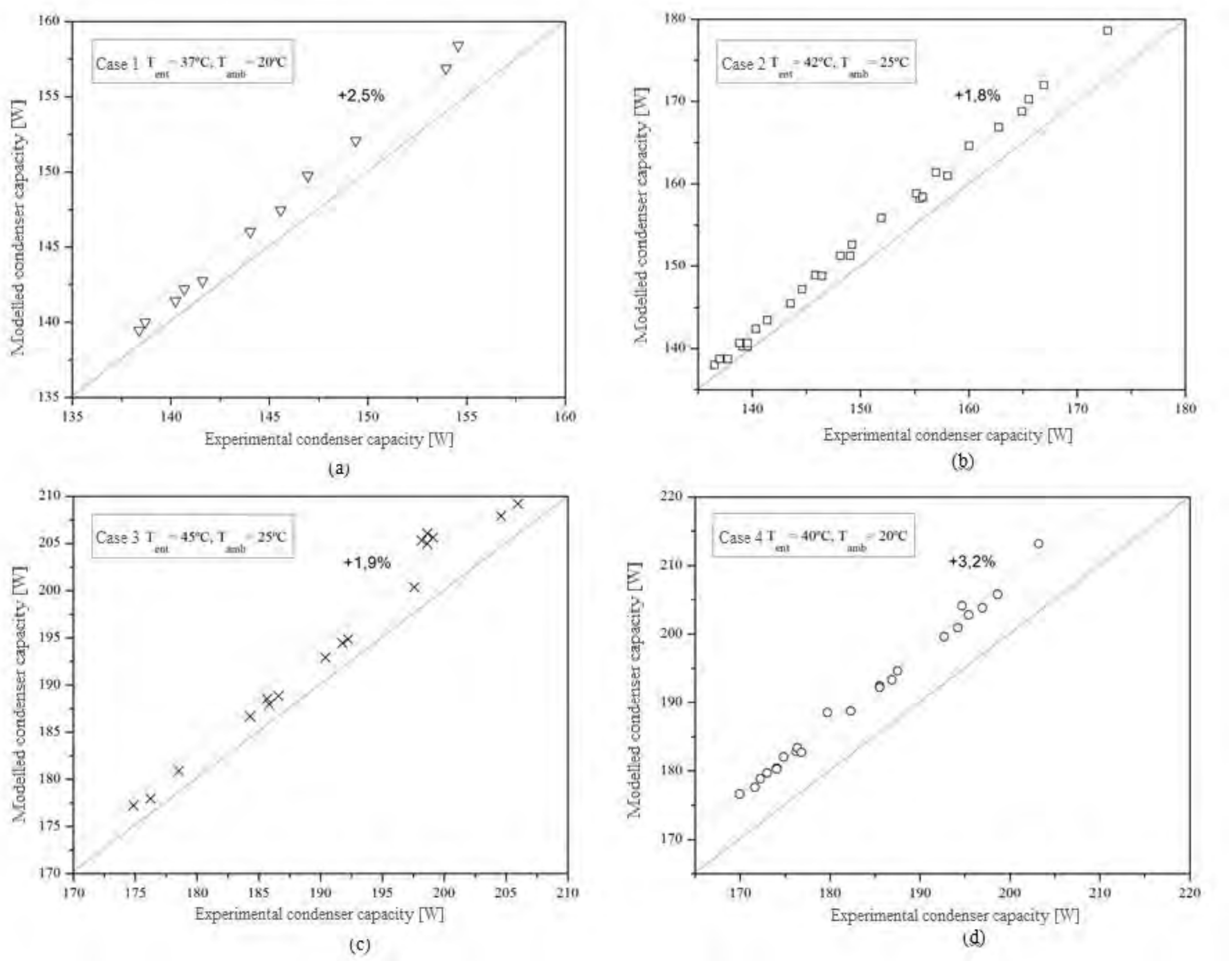

Figure 7 – Comparison between measured (Bansal and Chin, 2001) and predicted condenser capacity: (a) Case 1; (b) Case 2; (c) Case 3; (d) Case (4).

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The heat capacity of the condenser calculated in this study is in the 0 to 3.2% range in comparison to the experimental values, with the highest deviation for Case 3, Fig. 7(d), and the smallest deviation is of 1.8% for Case 2, Fig. 7(b). Also noted is the tendency of an increased difference between the calculated and measured results with the increasing heat capacity of the condenser, except for the comparisons in Case 4, Figure 7(d), with an approximately constant deviation throughout the analyzed range, 170 to 210 W.

Figures 8(a) and 8(b) show the condenser's heat capacity distribution according to the mass flow of the refrigerant, respectively, for cases 1 and 2 (Table 2). These figures show the experimental data of Bansal and Chin (2002) compared with the results calculated in the model presented herein, when both approaches are used in terms of heat transfer between the hot plate and the environment: direct contact model and two-dimensional plate model. Figs. 8 (a) and 8 (b) also show the results calculated by Bansal and Chin (2002), adiabatic fin tip model, and the results calculated with the model presented herein with the adiabatic fin tip approach to calculate the adiabatic temperature of the hot plate.

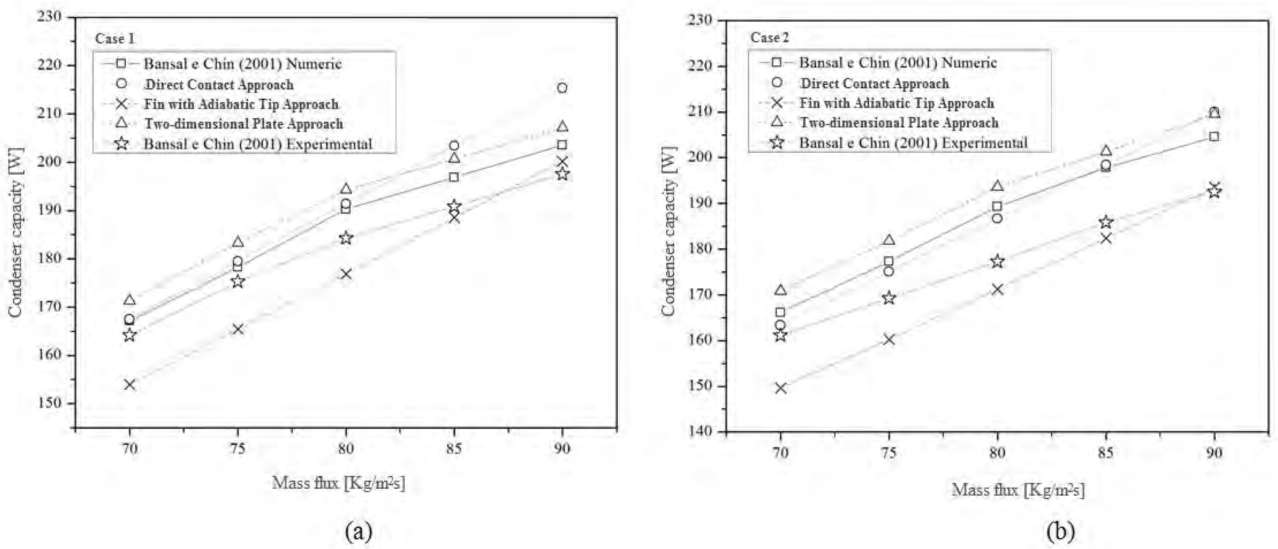


Figure 8 – Condenser capacity variations with mass flux: (a) Case 1; (b) Case 2.

Figures 8(a) and 8(b) show that the results calculated by the present model, both with the direct contact approach and with the two-dimensional plate, are overestimated compared with the experimental data of Bansal and Chin (2002). For Case 1, Figure 8 (a) shows a closer approximation between the results calculated by the two-dimensional plate modeling and the experimental data of Bansal and Chin (2002), exhibiting a behavior similar to the numerical results of Bansal and Chin (2002). For the direct contact model there is a closer approximation to the experimental data, particularly for Case 1, smaller mass flows and an increasing deviation between the measured and calculated heat capacities with increasing mass flow.

For the adiabatic fin tip approach, Figs. 8(a) and 8(b) show an opposite behavior, meaning that the results calculated by this model are underestimated compared to the experimental data of Bansal and Chin (2002). Furthermore, the deviations between the results calculated and the experimental data decrease with increasing mass flow. It is also seen in Figures 8(a) and 8(b) that the linear distribution behavior of the condenser's heat capacity according to mass flow obtained by the direct contact and the adiabatic fin tip models, differ from the results obtained by two-dimensional plate model, which followed the same tendency as the experimental data of Bansal and Chin (2002) with increasing mass flow.

Figures 9(a) and 9(b), respectively, show the flow pressure drop variations along the condenser tube according to the mass flow shown for Cases 1 and 2 (Table 2). In these figures the results calculated by Bansal and Chin (2002) are compared with the results calculated by the model presented herein, using the three heat transfer approaches between the hot plate and the environment: direct contact model, two-dimensional plate model and adiabatic fin tip model.

Figures 9(a) and 9(b) show that the present model underestimates the flow pressure drop compared with the model of Bansal and Chin (2002), independent of the approach used to transfer heat between the hot plate and the environment, except for the 70 kg/m^{2}s mass flow for the direct contact model. This may be related to the inclusion of the pressure reduction term due to the action of the gravitational field and also the effect of the curvature of the tube along the flow, according to Bansal and Chin (2002).

The linear tendency of a pressure drop increase in the mass flow was observed in both cases analyzed, Figs. 9(a) and 9(b), when the adiabatic fin tip model and the two-dimensional plate model were used. This tendency is also seen in the results calculated by the model of Bansal and Chin (2002). However, the direct contact model, besides not exhibiting this linear tendency, showed an oscillation with the increasing mass flow value, especially for Case 2, Figure 9(b).

The two-dimensional plate model allows obtaining the temperature field along the hot plate of the condenser in contact with the environment, where the condenser tube is located. Figure 10 schematically shows the position of the hot plate of a hot-wall condenser in a domestic refrigerator.

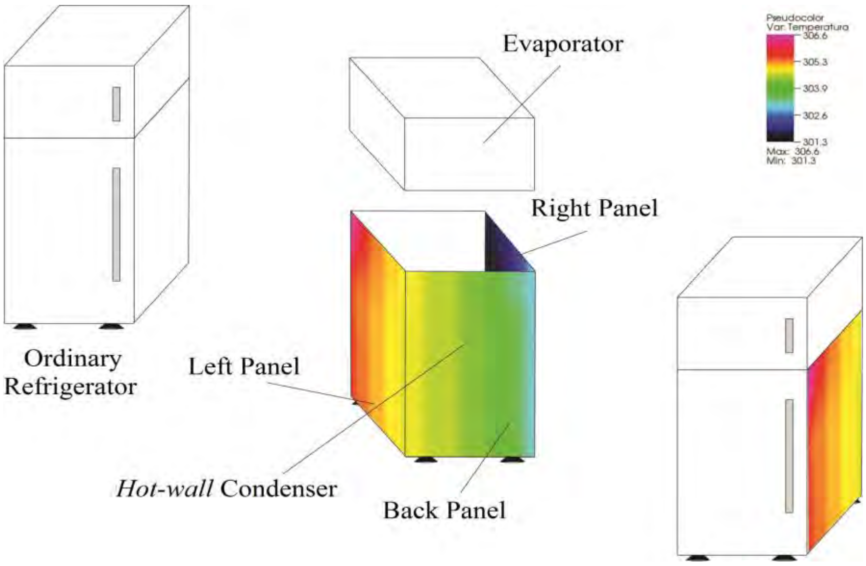
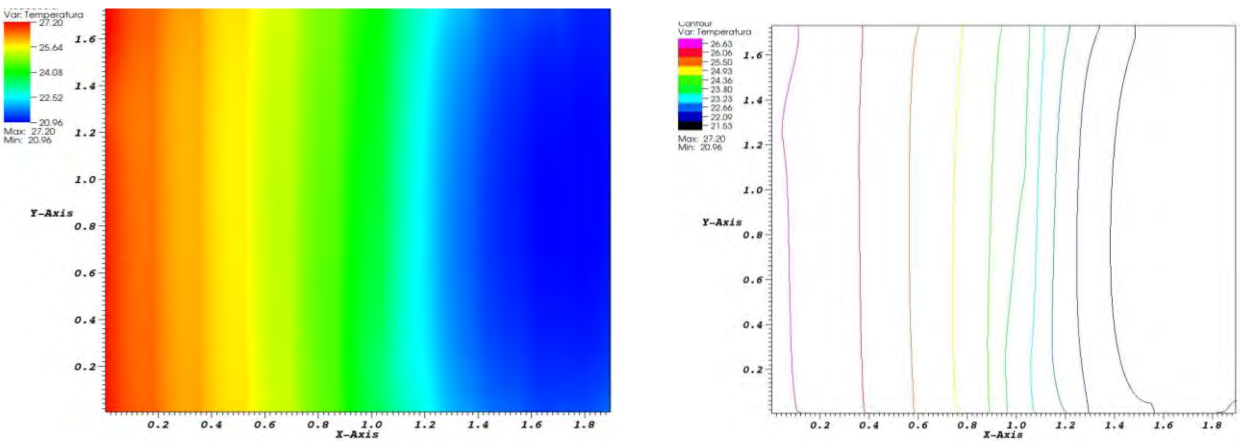


Figure 10 – Scheme positioning a hot-wall condenser in a domestic refrigerator.

Figure 11 shows the temperature field and isothermal lines on the hot plate for Case 1 (Table 2) for the mass flow rate of 1.2 g/s. It can be observed that the left side panel (0≤x≤0.591 m) shows the highest temperatures since the condenser inlet is located in that region where the fluid has the highest heat load. It is also seen that the lowest temperatures are in the right panel (1.215≤x≤1.806 m), where the condenser outlet is located.



(a) (b)

Figure 11 – (a) Distribution of temperature and (b) isothermal lines along the hot plate for Case 1, $\dot{m} = 1{,}2$ g/s .

With the analysis of the isothermal lines, Figure 11(b), it is observed that the temperature field is more stratified at the rear panel of the refrigerator (0.591≤x≤1.215 m), where the isothermal lines are grouped, and is almost homogeneous at the right panel (1.215≤x≤1.806 m), where the temperature distribution remains almost constant.

5. CONCLUSIONS

This work presented a numerical model to simulate flow and heat transfer, in a continuous regime, along a hot-wall condenser located on the side surfaces of a domestic refrigerator. The model allowed to calculate the distributions of temperature, specific mass, pressure, enthalpy of the refrigerant fluid, heat load, temperature of the condenser wall and temperature field in the plate.

The results obtained, in terms of heat capacity for the direct contact approach, used in the modeling of heat transfer between the hot plate and the environment, are in good agreement with the experimental data extracted from the

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literature. Considering all cases tested, the lowest deviation found between the experimental and numerical values was of 1.8% and the highest was of 3.2%.

With respect to the pressure drop along the condenser tube, there was a marked difference with respect to the calculated values found in the literature, especially in relation to the direct contact modeling, which showed significant variations in the results. This fact will be discussed in the future, as it can be related to the non-inclusion of the pressure reduction term according to the action of the gravitational field and also the effect of the tube curvature along the flow in the model presented herein.

The temperature field in the condenser hot plate, obtained by the two-dimensional plate model, showed almost the same configuration for the cases analyzed, where it was observed that the left panel is the one with the highest temperatures since it is in contact with the inlet region of the condenser. The right panel exhibits a nearly homogeneous temperature field, as it is in contact with the outlet region of the condenser tube, while the rear panel exhibited a very stratified temperature distribution, as it is in contact with the largest region of the tube where the two-phase flow occurs.

The results showed that the model can be used to analyze the performance of hot-wall condensers. However, the considerations presented indicate the need for a broader study of the phenomena and methods used in order to improve the model, hence a closer approximation between the calculated and experimental results of the performance parameters of hot-wall condensers, such as heat capacity and pressure drop, as well as the properties of the refrigerant fluid along the flow.

4. ACKNOWLEDGEMENTS

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1. 6. RESPONSIBILITY NOTICE