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THERMAL-HYDRODYNAMIC NUMERICAL MODELING OF HOUSEHOLD VAPOR COMPRESSION REFRIGERATION SYSTEMS

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Abstract. *Efforts and solutions to improve the efficiency of refrigeration systems have been carried out along time, showing its importance, both on academy and industry. To evaluate the system performance, experimental tests are very usual. Several parameters can impact performance, such as heat exchangers (condenser and evaporator) and compressor efficiencies, cabinet insulation, compressor control strategies, and the choose of refrigerant and its amount. So, on the development phase of a project, these parameters should be correctly defined and, this way to adjust them, on each condition and each modification considered, another experiment is carried out, which is usually expensive and time demanding. This way, the modeling and simulation of such systems are very useful features to support the tests. On this work, a program on Python language was developed to perform a transient simulation of the system. Therefore, with the compressor rotation and the refrigerant charge as inputs, parameters as sub-cooling and superheating, mass flow on compressor and on capillary tube, mass of refrigerant on condenser and evaporator, pressures dynamics, surface temperatures and performance parameters, such as compressor consumption, refrigeration capacity and the coefficient of performance (COP) can be obtained. Besides, a simulation of a pull-down test was performed.*

Keywords: *Numerical Modeling, Refrigeration, Efficiency, Thermal-hydrodynamic model*

1. INTRODUCTION

According to recent works (PROCEL – Eletrobrás, 2007), the energy consumption of household refrigerators corresponds to about 27% of the brazilian domestic demand, totalizing approximately 8% of Brazil's total energy consumption. These data show how the improvement on these devices are important to the country. Normally, household refrigerators are composed by compressor, condenser, capillary tube and an internal heat exchanger with the suction line, evaporator and the cabinet, with one or two compartments. Several parameters influence the performance of the system, including, temperatures, pressures, refrigeration capacity and energy consumption. Among them, there are heat exchangers (condenser and evaporator) and compressor efficiencies, cabinet insulation, compressor control strategies and the choose of refrigerant and its amount. On the project of such systems, these parameters must be calculated and tested on the whole system operation, to find the combination that results on the best behavior. These tests are necessary due the complex interactions between components. Therefore, so many experiments are demanded, fact that takes considerable time and cost. This way a good feature to give support to experiments in a faster and reliable way is the modeling and simulation of such systems.

The models for vapor compression refrigeration systems simulation can be classified on the following categories: continuous or discrete; theoretical, empirical or balanced; steady-state or transient; lumped or distributed parameters. Continuous models are normally solved analytically and the equation problem has no discontinuities. The discrete ones are normally solved iteratively, commonly on time domain. Theoretical models are based on Physics principles with the evaluation of the heat transfer and transport phenomena and conservation equations. Empirical models consider equations and correlations obtained from experiments, which became the solution process more direct and fast, but works only on the operation conditions studied. The balanced models have as main structure, a theoretical model, however, some parameters are acquired experimentally, such as heat transfer coefficients, thermal conductances and capacities and efficiency maps. These models are also called “semi-empirical” (Rasmussen, 2012). Steady-state approaches provide the final system operation characteristics, after all the mass flow and heat transfer rates, temperatures and pressures are stable. Transient models can describe the system dynamic since the start to the stable condition. Finally, on lumped models, the calculations are based on average parameters, that represents the whole system component operation, for example superficial component temperatures. On the other hand, distributed models divide the component in several volumes and the governing equations are applied on each one, providing the parameters behavior along the component structure.

Jakobsen (1995) developed a transient semi-empirical model to simulate a household refrigerator with one compartment. It was a discrete model based on lumped parameters. Porkhial et al (2005) presented a work focused on the wire-and-tube condenser of a household refrigerator description with R12. They developed a distributed model for the refrigerant and wall temperature, mass inventory and quality of refrigerant. Hermes (2006) proposed a transient model to simulate a frost-free refrigerator with two compartments, reciprocating compressor, wire-and-tube condenser with natural convection, capillary tube and internal heat exchanger, finned tube evaporator with forced convection and R134a as refrigerant. This model is predominantly theoretical with some parameters fitted through experiments. The heat exchangers models are distributed and the effectiveness method was used for the internal heat exchanger. Hermes and Melo (2008) presented a transient semi-empirical modeling for a frost-free refrigerator (with the same configuration of the one on the previous work). The heat exchangers models are distributed, based on the work of Rossi and Braun (1999) for an unitary air conditioner. This is a discrete model with variable time step. Borges et al. (2011) developed a quasi-steady semi-empirical model to simulate the steady-state of a frost-free household refrigerator. This lumped model is based on the works of Hermes and Melo (2008) and Hermes (2009). Guzella (2013) proposed three approaches: a steady-state semi-empirical modeling with the heat exchangers described with average parameters on each region (sub-cooled, two-phase and superheated), another with the same conditions and a distributed description of heat exchangers and a transient semi-empirical model with a distributed approach for condenser and evaporator. The refrigerator studied was presented by Klein (1998), with one compartment, reciprocating compressor, wire-and-tube condenser with natural convection, capillary tube and internal heat exchanger, roll-bond plate evaporator and refrigerant R134a. Zsembinszki et al. (2017) developed a steady-state semi-empirical model. This modeling considers average parameters on each region for the heat exchangers. The refrigerator tested was a walk-in freezer with one compartment, reciprocating compressor and finned tube condenser with forced ventilation (condensing unit), an electronic expansion valve, finned tube evaporator with forced convection and R404A as refrigerant.

The modeling developed and implemented is discrete, semi-empirical and transient, with lumped parameters, based on the work of Jakobsen (1995) and can provide a transient simulation with results as sub-cooling and superheating, mass flow rate on compressor and capillary tube, mass of refrigerant on condenser and evaporator, pressure dynamics, surface temperatures and performance parameters such as compressor consumption, refrigeration capacity and the coefficient of performance (COP). On this work there is a comparison between the results obtained with the simulations and the ones presented on Jakobsen (1995) for a refrigerator operating under an on/off logic and the simulation of a pull-down test, which is a very common proceeding on industry and researches in terms of design and performance evaluation. This test can provide results for the time to reach a certain temperature, sub-cooling and superheating behavior, allow an analysis to adequate the refrigerant charge and contribute to the development of control strategies for variable speed compressors.

2. METHODOLOGY

The modeling was carried out with a code developed on Python programming language, with the library of thermodynamic properties CoolProp, and consists on the analysis of control volumes around the system components. The refrigerator studied and the data about it, including geometric parameters, thermal conductance and capacities were according to the one presented by Jakobsen (1995). The refrigerator is controlled by an electronic thermostat, has one compartment of 325 L, reciprocating compressor, wire-and-tube condenser, capillary tube with an adiabatic portion and other portion on the heat exchanger with the suction line, roll-bond evaporator and the refrigerant is R134a, with approximately 80g. The ambient temperature was 25ºC and the compressor rotation, 2850 rpm.

2.1 Numerical description

The mathematical model is solved iteratively, in time steps, until the operation time simulated reaches the desired time preset and has an equation system with the parameters for each component, where they are interdependent. This equation system calculation is performed using the Newton-Raphson Method and the ordinary differential equations, present on the First Law of Thermodynamics application and on pressures calculation, with the 4th order Runge-Kutta Method, both present on other Python libraries.

Figure 1 shows the algorithm of the program implemented on Python.

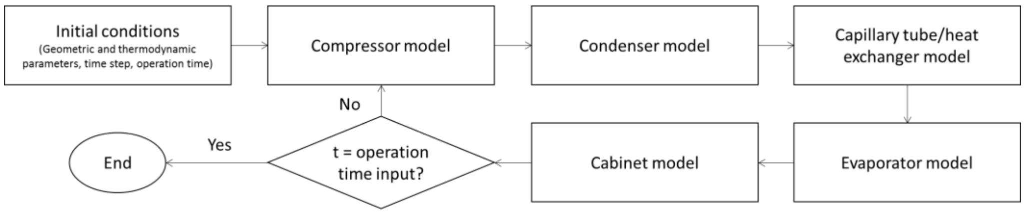

Figure 1. Algorithm of the program developed.

2.2 Modeling

Figure 2 shows the schematic refrigeration cycle representing the system. The state 1 is the compressor inlet, after the heat exchanger between capillary tube and suction line, state 2 is the compressor outlet and condenser inlet, state 3, the condenser outlet and capillary tube inlet, state 3i is the point where the adiabatic potion of capillary tube ends and the heat exchanger with suction line starts, 4 is the capillary tube outlet and evaporator inlet and 5 is the evaporator outlet.

To simplify the modeling some assumptions were adopted:

- For the components, disregard the following phenomena: delays on transport, pressure loss and refrigerant accumulation on connector tubes; pressure loss on condenser and evaporator; spatial variation on temperature at condenser, evaporator and compressor surfaces; spatial variation of temperature inside the cabinet; changes on the amount of refrigerant dissolved on oil.
- For the mathematical description: control volumes around the system components have only one inlet and one outlet; kinetic and potential energy variations are considered null on control volumes and their borders; Thermodynamic and transport properties are uniform on control volumes; negligible field forces.

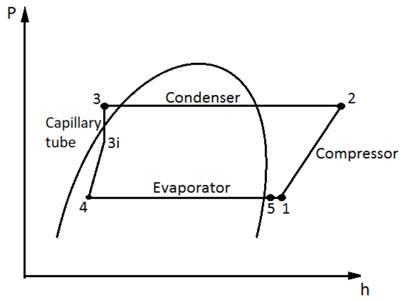

Figure 2. Diagram that represents the refrigeration cycle.

On the simulations there were no door openings and goods inside the cabinet.

The model consists on an energy balance in a control volume constructed around each component of the refrigeration system. From each component model is possible to obtain data that are used as inputs to the next component. The tempeartures are expressed in °C; pressure in Pa; density in kg/m³; volume in m³; mass flow in kg/s; enthalpies in J/kg; and heat transfers and power consumption in W.

The compressor model has as inputs the state of refrigerant on the suction, condensing and evaporating pressures (P_{cond} and P_{evap}), rotation and geometric parameters, besides the thermal conductance and capacity and ambient temperature.

The compressor mass flow rate ($\dot{m}_{com}$) is determined as follows:

$$\dot{m}_{com} = \eta_v \rho_1 V_s \frac{N_{rpm}}{60} \tag{1}$$

Where η_v is the volumetric efficiency, ρ_1, the density on compressor inlet, V_s = 3.13(10^{-6}), the compressor displacement and N_{rpm}, the rotation in rpm.

For the discharge temperature (T_2) it was considered a polytropic process, as shown on Eq. (2).

$$T_2 = \left(\frac{T_1 + T_{com}}{2}\right)\left(\frac{P_{cond}}{P_{evap}}\right)^{\frac{n-1}{n}} \tag{2}$$

Being T_1 and T_{com}, the compressor inlet and the housing temperatures and n, the polytropic exponent defined as 1.09 (Jakobsen, 1995).

The heat transfer through the housing ($\dot{Q}_{com}$) and the electric power consumption of compressor ($\dot{W}$) are:

$$\dot{Q}_{com} = UA_{com}\left(T_{com} - T_{amb}\right) \tag{3}$$

$$\dot{W} = \frac{\dot{m}_{com}\left(h_{2s} - h_1\right)}{\eta_g} \tag{4}$$

Where UA_{com} is the thermal conductance of compressor, T_{amb}, the ambient temperature, h_{2s} the enthalpy on discharge if the compression was isentropic, h_1, the enthalpy on compression inlet and η_g is the global efficiency.

The compressor efficiencies, according to Jakobsen (1995), were obtained through experimental data and structured in polynomials in function of the condensing and evaporating pressures (in bar), as shown below.

$$\eta_v = 0.542 - 0.08954\frac{P_{cond}}{P_{evap}} + 0.27523\sqrt{\frac{P_{cond}}{P_{evap}}} \tag{5}$$

$$\eta_g = -1.479 - 1.4916\,P_{evap} + 3.2409\sqrt{P_{evap}} + 0.08142\left(P_{evap}\right)^2 \tag{6}$$

Finally, for the compressor housing surface temperature calculation, the First Law of Thermodynamics is applied on a control volume around this component.

$$C_{com}\frac{dT_{com}}{dt} = \dot{W} - \dot{Q}_{com} - \dot{m}_{com}\left(h_2 - h_1\right) \tag{7}$$

The condenser model has as inputs the state of refrigerant on compressor discharge, the mass flow on compressor and on capillary tube ($\dot{m}_{cap}$), the total mass of refrigerant (M_{tot}), ambient temperature, geometric parameters, as volume and areas, thermal conductance and capacity.

The pressure on condenser, when the system is on, is the saturation pressure on condensing temperature, and the mass of refrigerant on condenser is calculated as follows:

$$\frac{dM_c}{dt} = \dot{m}_{com} - \dot{m}_{cap} \tag{8}$$

Considering a linear variation of the quality on the condenser outlet (x_3) in function of M_c and the parameters M_{vc}, M_{vlc} and M_{lc}, which are the mass on condenser to establish x_3 equals 1, x_3 equals 0 and the quality on the condenser inlet equals 0, respectively, the quality on condenser outlet is calculated according to Eq. (9).

$$x_3 = \frac{M_{vlc} - M_c}{M_{vlc} - M_{vc}} \tag{9}$$

It was deducted an expression for the condenser outlet temperature (T_3) if $x_3 < 0$, presented in Eq. (10). If $0 \le x_3 \le 1$, the outlet temperature will be equals to the condensing one.

$$T_3 = T_{amb} + \left(T_{cond} - T_{amb}\right)\exp\left(-\frac{U_{sc}A_{sc}}{\dot{m}_{cap}c_{pl}}\right) \tag{10}$$

Where T_{cond} is the condensing temperature, U_{sc} and A_{sc}, the corrected value for the condenser overall heat transfer coefficient and the surface area of the region containing sub-cooled refrigerant and, c_{pl}, the specific heat of the liquid. The sub-cooled area is determined analogously to x_3, where it is considered linear with the mass on condenser. Therefore, A_{sc} is:

$$A_{sc} = A_c\left(\frac{M_c - M_{vlc}}{M_{lc} - M_{vlc}}\right) \tag{11}$$

Where A_c = 1.55 m^2 is the total surface area of condenser. The heat transfer through the condenser wall is:

$$\dot{Q}_c = UA_c\left(T_{wc} - T_{amb}\right) \tag{12}$$

Finally, with the First Law, the condenser wall temperature is calculated.

$$C_c\frac{dT_{wc}}{dt} = \alpha_c A_{ci}\left(T_{cond} - T_{wc}\right) - \dot{Q}_c \tag{13}$$

And the new condensing temperature, $T_{cond,n}$, for the next time step is:

$$T_{cond,n} = T_{wc} + \frac{\dot{m}_{com}\left(h_2 - h_3\right)}{\alpha_c A_{ci}} \tag{14}$$

Where α_c is the average internal heat transfer coefficient (calculated through Gnielinski (1976) correlation and with modified Tandon (1985) correlation, according to Shao and Granyrd (1995), for condensation), A_{ci} = 0.08 m^2, the internal surface area and T_{wc}, the surface temperature of condenser.

When the system is turned off, the pressure is strongly dependent of the fluid dynamic and, for both condenser and evaporator, if there is two-phase fluid, the pressure is:

$$\frac{dP}{dt} = \frac{\dfrac{dU}{dt} - f_2\dfrac{dM}{dt}}{M\dfrac{df_2}{dP} + V\dfrac{df_1}{dP}} \tag{15}$$

Being U, the refrigerant total internal energy, and f_1 and f_2 expressions for R134a evolving the saturation values of specific internal energy and volume fit through a polynomial in function of the pressure, according to Jakobsen (1995), M and V, the mass of refrigerant and the internal volume of the heat exchanger (V_c = 9.14(10^{-5}), the condenser volume and V_e = 7.63(10^{-5}), the evaporator volume), respectively.

If there is only gas, applying the perfect gas law with the compressibility factor (Z) to correct, the expression on Eq. (16) for the pressure can be obtained.

$$\frac{dP}{dt} = \frac{ZR\dfrac{dU}{dt}}{c_v V} \tag{16}$$

Where R is the gas constant, c_v, the specific heat at constant volume and V, the volume of the heat exchanger.

The capillary tube model has as inputs the state of refrigerant on condenser outlet, the suction and evaporator outlet temperatures, conductance on the heat exchanger and geometric parameters as tube diameter and length.

On the adiabatic portion of the tube, there is an isenthalpic expansion, so, the enthalpy on the outlet of this part is equals to the entering one ($h_{3i} = h_3$). This way, the enthalpy of refrigerant entering the heat exchanger with the suction line is determined. The pressure at this point is calculated using the factor $f_{\Delta P}$ (evaluated experimentally), which is the fraction of pressure loss occurring on the adiabatic portion. Therefore:

$$P_{3i} = P_{cond} - f_{\Delta P}\left(P_{cond} - P_{evap}\right) \tag{17}$$

To determine the temperature of fluid entering the heat exchanger (T_{3i}), a polynomial was obtained for the temperature on saturated liquid curve of R134a in function of the enthalpy, presented on Eq. (18).

$$T_{sat,liq}(h) = -7.10^{-10}h^2 + 0.001h + 102.91 \tag{18}$$

Another parameter necessary to the calculation is the saturation temperature at P_{3i}, $T_{3i,sat}$. This way,

$$T_{3i} = T(P_{3i}, h_{3i}) \tag{19}$$

If $T_{3i,sat} > T_{sat,liq}$. Else,

$$T_{3i} = T_{3i,sat} \tag{20}$$

Assuming that this is a counterflow heat exchanger and that the temperature vary linearly from T_{3i} to T_5 (evaporator outlet temperature), the Eq. (21) is obtained for the compressor suction temperature (T_1).

$$T_1 = T_{3i} - \frac{(T_{3i} - T_5)}{\frac{UA'_{he} L_{he}}{\dot{m}_{com} c_{pv}}} \left[1 - \exp\left(- \frac{UA'_{he} L_{he}}{\dot{m}_{com} c_{pv}} \right) \right] \tag{21}$$

Where UA'_{he} is the thermal conductance of the heat exchanger in function of the length, L_{he}, the heat exchanger length and c_{pv}, the vapor specific heat (admitting that there is vapor on suction line).

The enthalpy on the capillary tube outlet (h_4) is calculated through an energy balance on the heat exchanger.

$$h_4 = h_{3i} + h_5 - h_1 \tag{22}$$

Being h_5, the enthalpy on evaporator outlet.

Through experimental data (Jakobsen, 1995), an expression was obtained for the capillary tube mass flow rate, Eq. (23), according to the proposal of Christensen (1993).

$$\dot{m}_{cap} = 0.003755 \sqrt{\frac{(P_{cond} - P_{evap})}{v_3}} + 0.030689 \Delta T_{sc} + 0.074415 \tag{23}$$

Where v_3 is the specific volume on condenser outlet and ΔT_{sc} is the degree of sub-cooling, equivalent to (T_{cond} - T_3).

The evaporator model is similar to the condenser one. The inputs are state of refrigerant on capillary tube outlet, mass flow on compressor and capillary tube, total mass of refrigerant, cabinet internal air temperature (T_r), geometric parameters as volume and areas, thermal conductance and capacity.

The pressure on evaporator, when the system is on, is the saturation pressure on evaporating temperature, and the mass of refrigerant on this heat exchanger is:

$$\frac{dM_e}{dt} = \dot{m}_{cap} - \dot{m}_{com} \tag{24}$$

Analogously to the condenser, defining M_{le} as the mass on evaporator causing quality (x_5) zero on the outlet, M_{vle}, the mass for $x_5 = 1$ and M_{ve}, the mass for the evaporator inlet being saturated vapor, this evaporator outlet quality is determined according to Eq. (25).

$$x_5 = \frac{M_{le} - M_e}{M_{le} - M_{vle}} \tag{25}$$

The outlet temperature, T_5, when $x_5 > 1$, is:

$$T_5 = T_r + (T_r - T_{amb}) \exp\left(- \frac{U_{sh} A_{sh}}{\dot{m}_{com} c_{pv}} \right) \tag{26}$$

Where U_{sh} and A_{sh}, are the corrected value for the evaporator overall heat transfer coefficient and the surface area of the region containing superheated refrigerant and. If $0 \leq x_5 \leq 1$, the outlet temperature is the evaporating one (T_{evap}). The degree of superheating is $\Delta T_{sh} = T_5 - T_{evap}$ and the superheated area is determined analogously to x_5,

$$A_{sh} = A_e\left(\frac{M_{vle} - M_e}{M_{vle} - M_{ve}}\right) \tag{27}$$

Where $A_e = 0.475$ m^2 is the total surface area of evaporator. The heat transfer through the evaporator wall is:

$$\dot{Q}_e = UA_e\left(T_r - T_{we}\right) \tag{28}$$

With the First Law, the evaporator wall temperature is calculated.

$$C_e\frac{dT_{we}}{dt} = \dot{Q}_e - \alpha_e A_{ei}\left(T_{we} - T_{evap}\right) \tag{29}$$

And the new evaporating temperature, $T_{evap,n}$, is:

$$T_{evap,n} = T_{we} + \frac{\dot{m}_{cap}\left(h_5 - h_4\right)}{\alpha_e A_{ei}} \tag{30}$$

Where α_e is the average internal heat transfer coefficient (calculated through Gnielinski (1976) correlation and with Liu and Winterton (1991) correlation for evaporation), $A_{ei} = 0.0475$ m^2, the internal surface area and T_{we}, the surface temperature of condenser.

Finally, for the cabinet, the inputs are the heat transfer on evaporator, thermal conductance and capacity and ambient temperature.

The heat transferred through the walls is:

$$\dot{Q}_r = UA_r\left(T_{amb} - T_r\right) \tag{31}$$

And, the temperature on the cabinet air is determined with the following equation.

$$C_r\frac{dT_r}{dt} = \dot{Q}_r - \dot{Q}_e \tag{32}$$

3. RESULTS

Table 1 shows a comparison between some important performance parameters on the simulation of the refrigerator under an on/off logic and the ones presented on the work of Jakobsen (1995). Note that the simulated results are coherent and close to the ones acquired on the reference. The behavior of pressures, sub-cooling, superheating, mass of refrigerant on the heat exchangers and mass flow rate on compressor and capillary tube are presented on the figures below. Note on Fig. 3 the pressures dynamics, where the effect of "pressure equalization" is described when the system turns off. Fig. 4 shows the evolution of sub-cooling and superheating. When the system turns on, the mass flow on compressor is higher and, as well as the pressure difference increases, the mass flow on capillary tube is enhanced until the stabilization, as shown on Fig. 5. As a consequence of the mass flow dynamics, as mentioned, the mass on condenser raises faster, and, on the other hand, the opposite occur to the mass on evaporator. When the mass flow on capillary tube reaches the level of the compressor one, an equilibrium is established. See Fig. 6.

Table 1. Comparison between performance parameters from the simulation and the ones from Jakobsen (1995).

	Average power [W]	Average cooling capacity [W]	COP
Simulation	33.53	45.88	1.37
Jakobsen (1995)	33.38	45.02	1.41

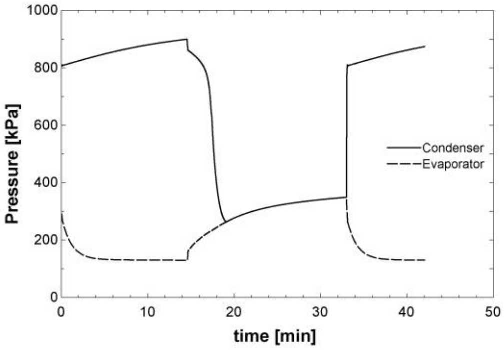

Figure 3. Simulated pressures on condenser and evaporator.

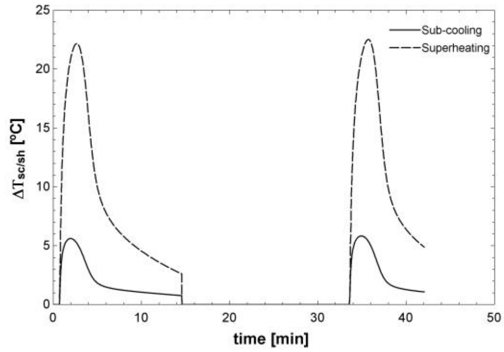

Figure 4. Simulated sub-cooling and superheating.

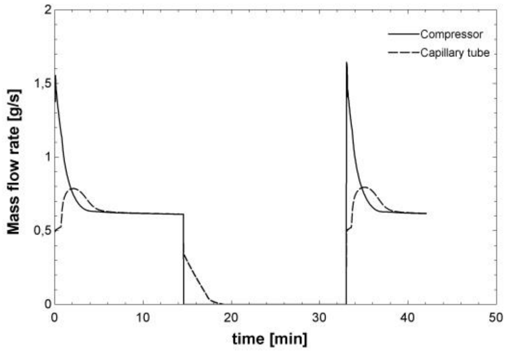

Figure 5. Simulated mass flow rate on compressor and capillary tube.

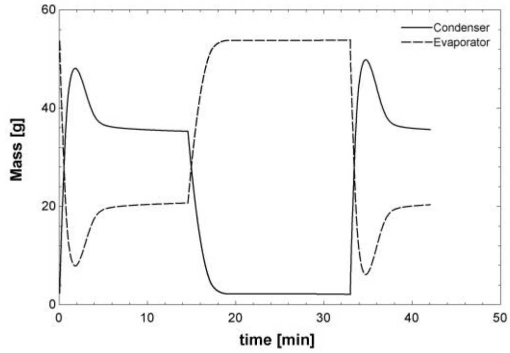

Figure 6. Simulated refrigerant mass on condenser and evaporator.

Following the simulation of a pull-down test is presented. On this test the refrigerator, on a climate chamber, starts after the thermal equilibrium with the ambient established and the thermostat is by-passed. Then, the system is left to operate until the steady state is established. The ambient temperature was 25ºC and the compressor rotation, 2850 rpm.

Figure 7 shows the surface temperature of each component as well as the cabinet inside air. The electric power consumption of compressor is presented on Fig. 8, where the peak on compressor start can be noticed. The pressures on condenser and evaporator and the mass flow rate on compressor and capillary tube are presented on Fig. 9 and Fig. 10. It is necessary more time (comparing to Fig. 5) to the mass flow rates reach the equalization, so, the condenser gets more filled and the evaporator emptier on this starting period, comparing to Fig. 6, as presented on Fig. 11. With less refrigerant mass on evaporator and higher temperature inside the cabinet, the superheating, on this initial functioning, reaches higher levels, according to Fig. 12. Finally, Fig. 13 brings the refrigeration cycle in steady state.

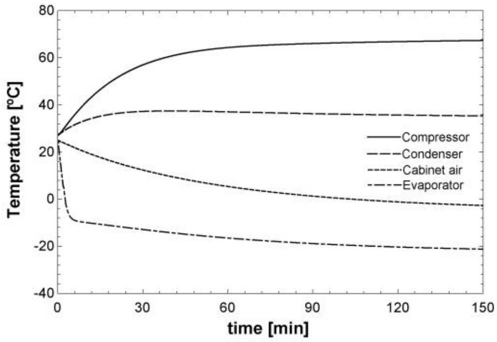

Figure 7. Simulated surface temperatures and cabinet air.

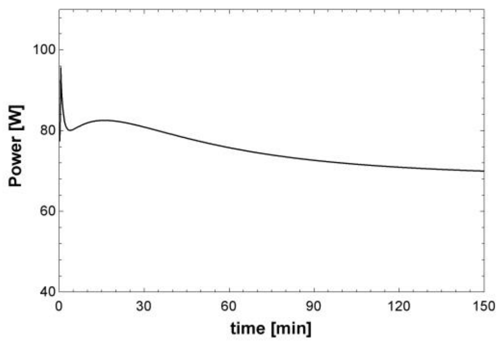

Figure 8. Simulated electric power consumption.

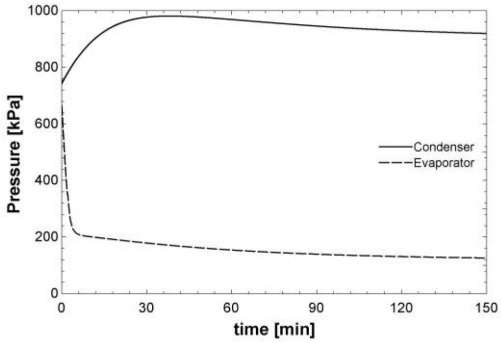

Figure 9. Simulated pressures on condenser and evaporator.

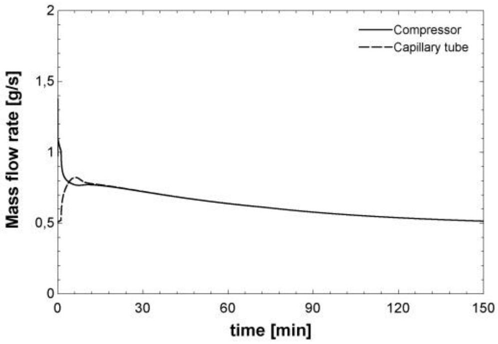

Figure 10. Simulated mass flow rate on compressor and capillary tube.

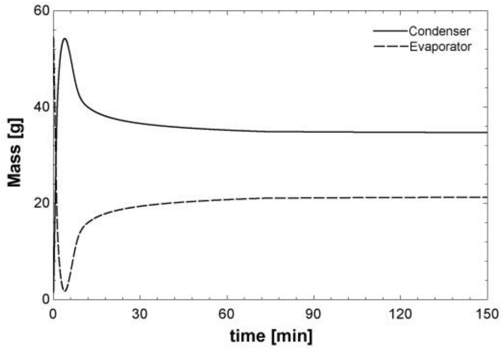

Figure 11. Simulated refrigerant mass on condenser and evaporator.

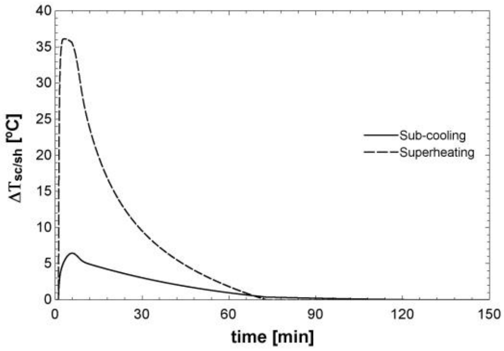

Figure 12. Simulated sub-cooling and superheating.

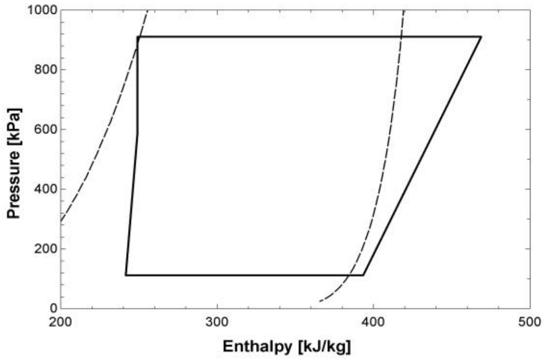

Figure 13. Diagram of the refrigeration cycle on steady state.

On this test the compressor power consumption was 68.72W, the refrigeration capacity, 67.59W and the COP, 0.98. All of them evaluated when steady state was established.

The electric power consumption, with a peak at the start, the behavior of the condenser average temperature, the refrigerant mass in it (considering the variations since the start due the mass flow rates variation on compressor and capillary tube), the condenser pressure and the COP value are coherent, according to Porkhial et al. (2005). In the work of Nunes et al. (2015), the same behavior on condensing and evaporating pressures, on sub-cooling and superheating (with the peaks on both close to the start and posteriorly a stabilization) and on the refrigerant mass dynamic accommodation on the heat exchangers can be noticed.

4. CONCLUSIONS

In the present work, a numerical simulation model, coupling the thermal and the hydrodynamic analysis and computing the performance of a household vapor compression refrigeration system is developed. The mathematical description was implemented on a program using Python language, with the library of thermodynamic properties CoolProp. The characterization data for the refrigerator used is presented on Jakobsen (1995) and the results of the simulation were compared to the ones on this work. The results are in good agreement with the literature, providing bases to conclude that the work is coherent. Besides, the simulation of a pull-down test returned a coherent behavior. As future works, the characterization of components on a common refrigerator from brazilian market, with two compartments and a numerical model to characterize and select capillary tubes for the operation conditions, based on Stoecker and Jones (1985) and Guzella et al. (2016) are being carried out.

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