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NUMERICAL SIMULATION OF THE PERFORMANCE OF A HOUSEHOLD REFRIGERATOR DURING START-UP AND CYCLING OPERATIONS

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Abstract. *This paper presents a numerical model developed on GT-SUITE® to describe the thermal-fluid behavior of a single-compartment household refrigerator during start-up and cyclic operations. A domestic refrigerator from a study available in the literature was considered and the geometrical and performance parameters of the components were applied as input data. Empirical correlations for refrigerant side and air-side heat transfer correlations, as well as friction factors were considered. The model is capable of simulating the start-up operation based on refrigerant charge and ambient temperature. The simulation model was used to reproduce pull-down tests for different ambient temperatures, namely 21 °C, 32°C and 43°C, and for cyclic operations simulated by acting over the thermostatic valve model. For pull-down test simulations, transient variations of several variables were analyzed. During cyclic operation, the model can predict on-off behavior of heat exchangers and compressor, including such parameters difficult to measure experimentally, as refrigerant migration in the system. Although no experimental data were available for comparison, the model provided good qualitative predictions of the refrigeration system as well as the behavior of the air inside the refrigerator compartment during both pull-down and cyclic operations.*

Keywords: *household refrigerator, GT-SUITE®, numerical simulation*

1. INTRODUCTION

Domestic refrigerators are units characterized by low cooling capacities (50-250W) and low quantities of charge (20-200g) when compared to air conditioning systems. A hermetically sealed compressor, a condenser, an evaporator and a capillary tube as expansion device, which is placed in contact with the compressor suction line, forming an additional heat exchanger, compose domestic refrigerators.

According to Hermes (2000), their performance can be evaluated by standardized experiments on a controlled environmental chamber, by using components characteristic curves coupled with simple calculations or by using a simulation model. The first provides most reliable information about the behaviour of the equipment, however is time consuming (more than 12 hours to be performed) and it is expensive. The second is applied only to obtain general understanding about the equipment performance. Finally, the use of numerical models to investigate the thermal-fluid behaviour of domestic refrigerators are increasingly used, since they provide reliable information in a short time, especially due to increasing capability of digital computers.

Several numerical models were developed on the last years: Melo *et al*. (1988) proposed one of the first domestic refrigerators simulation model, applied to a two-compartment domestic refrigerator operating with CFC-12. The proposed model was applied to reproduce pull-down tests and experimental data were used for validation and good agreement between results were observed.

Jakobsen (1995) presented two numerical models to predict the transient behaviour of a household refrigerator, the first one disregarding the refrigerant distribution on the system and the second considering their effect over the system. Lumped models were applied to heat exchangers (condenser and evaporator), as well as for the cabinet model. Compressor model was based on isentropic compression, corrected by empirical correlations. Capillary tube model was also modelled using to empirical correlations. Results were compared against experimental data showing poor agreement, especially during start-up. The author also analyzed the influence of control strategies over the performance coefficient of the refrigerator. Additionally, Chen and Li (1991), Xu and Clodic (1996), Li and Alleyne (2010), Tulapurkar and Kandelwal (2010), Lin *et al*. (2011) and Tagliafico *et al*. (2012) also published similar models for household refrigerators simulations.

More recently, Hermes and Melo (2008) presented a numerical methodology to perform transient simulations of household refrigerators during start-up and cycling transients. The methodology was applied to a frost-free 440-l top-mount refrigerator. Cycling transient operation was controlled considering the freezer temperature, while a thermo-mechanical damper was used to control fresh food compartment temperature. Experimental data were used to calibrate the compressor model and to validate the proposed model. Results of system energy consumption agreed within 10% of experimental data while air temperatures presented a maximum deviation of ± 1%.

Using a semi-empirical quasi-steady approach, Borges *et al*. (2011) developed a first-principles simulation model to predict the cycling behaviour of household refrigerators and therefore predict their energy consumption. Numerical results were confronted against experimental data showed a maximum deviation of ± 2%. Additionally, a sensitivity analysis was also carried out to identify potential method for energy savings. Gonçalves *et al*. (2004) also developed a semi-empirical model dedicated to project domestic refrigerators operation under steady state conditions.

Although numerical models are increasingly being developed to perform numerical simulations, the use of a commercial software is still not common to perform household refrigerator simulations. Hence, the main aim of the present investigation is to present a simulation model developed in the GT-SUITE® platform that can be used for simulating the thermal-hydraulic behaviour of a vapour compression refrigeration system used in domestic refrigerators. The simulation model was constructed using models available in the software and considering geometrical and some experimental parameters from the refrigeration system in Hermes (2000).

2. NUMERICAL MODEL

The household refrigerator considered in this paper is composed of a wire-on-tube condenser, a roll-bond evaporator, a hermetic compressor and a concentric capillary tube-suction line heat exchanger. Condenser and evaporator interactions with ambient and refrigerated air are governed by natural convection. Geometrical parameters of this domestic refrigerator can be seen in Hermes (2000).

The simulation model developed in GT-SUITE® is based on one-dimensional Navier-Stokes and energy equations applied to each component of the refrigeration system considering an implicit formulation in time. It will be shown below a description of each component model.

2.1 Compressor model

In this paper, a simplified compressor model was considered, in which the volumetric and global efficiencies were specified and assumed as constant values. Compressor parameters such as displacement and nominal speed, given by V_D and n_{rpm} respectively, are used.

The mass flow rate displaced by the compressor is calculated as follows, based on the refrigerant density ρ_{inlet} in compressor suction line:

$$\dot{m}_{comp} = \frac{\rho_{inlet} V_D n_{rpm} \eta_{vol}}{60} \tag{1}$$

The electrical power consumption $\dot{W}_{comp}$ can be calculated assuming isentropic compression:

$$\dot{W}_{comp} = \dot{m}_{comp} \left(\frac{h_{outlet} - h_{inlet}}{\eta_g} \right) \tag{2}$$

As mentioned before, volumetric and global efficiencies, η_{vol} and η_g, were considered constant and equal to 85% and 43%, respectively. This is a limitation of the compressor model applied to simulate the refrigeration system.

2.2 Formulation for the heat exchangers: condenser, evaporator and capillary tube-suction line models

The following equations were obtained from GT-SUITE (2012). This formulation is applied to each equipment of the system, excluding the compressor whose model was presented before. The differential equations are solved numerically for each equipment of the system, which are divided into several control volumes and the equations are integrated implicitly in time.

The differential form of mass conservation is given by:

$$\frac{dm}{dt} = \sum_{contours} \dot{m} \tag{3}$$

For each sub-volume of the system, differential momentum equation is given by:

$$\frac{d\dot{m}}{dt} = \frac{A_i dp + \sum_{contours} (\dot{m}u) - 4C_f \frac{\rho u|u|}{2} \frac{A_i dz}{D_i} - C_p \frac{\rho u|u|}{2} A_i}{dz} \tag{4}$$

Where A_i is the cross sectional tube area, dp is the pressure differential across the control volume, u is the area-averaged refrigerant velocity, C_f is the friction coefficient, dz is the control volume length, C_p is the pressure coefficient and ρ is the area-averaged mixture density.

For single phase condition C_f is calculated using correlations depending on the Reynolds number:

$$C_f = \begin{cases} \dfrac{16}{\mathrm{Re}_{D_i}}, \mathrm{Re}_{D_i} < 2000 \\ \dfrac{1}{4}\left[4.781 - \dfrac{(A-4.781)^2}{B - 2A + 4.781}\right]^{-2}, \mathrm{Re}_{D_i} > 4000 \end{cases} \tag{5}$$

The constants A and B are given by the following equations:

$$A = -2.0 \log_{10}\left(\frac{\delta}{3.7D_i} + \frac{12}{\mathrm{Re}_{D_i}}\right) \tag{6}$$

$$B = -2.0 \log_{10}\left(\frac{\delta}{3.7D_i} + \frac{2.51A}{\mathrm{Re}_{D_i}}\right) \tag{7}$$

For the transition condition of the flow regime, an interpolation between the expressions previously presented is performed. For two-phase flow, the correlation proposed by Friedel (1979), based on the two-phase multiplier, is applied:

$$\Phi^2 = E + \frac{3.24F\kappa}{Fr_h^{0.045} We_L^{0.035}} \tag{8}$$

The parameters E, F and κ are computed based on vapor and liquid properties and the vapor quality of the refrigerant, x:

$$E = (1-x)^2 + x^2 \frac{\rho_l}{\rho_g}\left(\frac{\mathrm{Re}_g}{\mathrm{Re}_l}\right)^{0.9} \tag{9}$$

$$F = x^{0.78}(1-x)^{0.224} \tag{10}$$

$$\kappa = \left(\frac{\rho_l}{\rho_g}\right)^{0.91}\left(\frac{\mu_g}{\mu_l}\right)^{0.19}\left(1 - \frac{\mu_g}{\mu_l}\right)^{0.7} \tag{11}$$

The Froude number Fr_h and Weber number We_L are calculated for the two-phase flow, which is modeled as homogeneous:

$$We_L = \frac{G^2 D_i}{\sigma \rho_h} \tag{12}$$

$$Fr_h = \frac{G^2}{g D_i \rho_h^2} \tag{13}$$

The pressure coefficient C_p is computed based on inlet and outlet pressure and inlet density and velocity of the control volume:

$$C_p = \frac{p_{inlet} - p_{outlet}}{\frac{1}{2}\rho_{inlet} u_{inlet}^2} \tag{14}$$

The energy equation for the internal flow is expressed as:

$$\frac{d(\rho h V)}{dt} = \sum_{contours} (\dot{m} h) + V\frac{dp}{dt} - \hbar_i A_{s,i}(T - T_w) \tag{15}$$

Empirical correlations were applied to predict heat transfer coefficients on the refrigerant side and air side. For the condenser and evaporator models, single phase heat transfer coefficient was calculated by Dittus and Boelter (1930):

$$\hbar_i = 0.023\frac{k}{D_i}\mathrm{Re}^{0.8}\,\mathrm{Pr}^n \tag{16}$$

Where n is taken as 0.3 for the condenser model and 0.4 for the evaporator model. During condensation, two-phase heat transfer coefficient is calculated according to Dobson and Chato (1998):

$$\hbar_i = 0.023\frac{k}{D_i}\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.3}\left(1 + 2.22 X_{tt}^{-0.87}\right) \tag{17}$$

The Lockhart-Martinelli X_{tt} parameter is given by

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1} \tag{18}$$

During evaporation, two-phase heat transfer coefficient is calculated according to Klimenko (1988), given by:

$$\hbar_i = 0.087\,\mathrm{Re}_{\mathrm{mod}}^{0.6}\,\mathrm{Pr}_l^{1/6}\left(\frac{\rho_g}{\rho_l}\right)^{0.2}\left(\frac{k_w}{k_l}\right)^{0.9}\frac{k_l}{b} \tag{19}$$

The modified Reynolds number Re_{mod} is expressed according to:

$$\mathrm{Re}_{\mathrm{mod}} = \frac{u_l}{\nu_l}\frac{\rho}{\rho_l}\left[1 + x\left(\frac{\rho_l}{\rho_g} - 1\right)\right]\sqrt{\frac{\sigma}{g(\rho_l - \rho_g)}} \tag{20}$$

The air side heat transfer coefficient on the condenser was calculated according to Tanda and Tagliafico (1998) correlation, given by:

$$\hbar_e = 0.66\frac{k}{H_c}\left(Ra_{H_c}\frac{H_c}{D_e}\right)^{0.25}\left[1 - \left(1 - 0.45\left(\frac{D_e}{H_c}\right)^{0.25}\right)e^{\frac{-s_f}{Z}}\right] \tag{21}$$

The parameters s_f and Z are showed below:

$$s_f = \frac{\theta_f - D_f}{D_f} \tag{22}$$

$$Z = \left(\frac{28.2}{H_c}\right)^{0.4}\left(\frac{s_f^{0.9}}{s_t}\right) + \left(\frac{28.2}{H_c}\right)\left(\frac{264}{T_w - T_{amb}}\right) s_f^{-1.5} s_t^{-0.5} \tag{23}$$

Where θ_f is the fin spacing. The parameter s_t is related to tube spacing and external tube diameter, as shown:

$$s_t = \frac{\theta_t - D_e}{D_e} \tag{24}$$

Finally, for the evaporator mode, air-side heat transfer coefficient was calculated according to the correlation proposed by Churchil and Chu (1975), based on the evaporator height H_{evap}:

$$\hbar_e = \frac{k}{H_{evap}}\left\{0.825 + \frac{0.387 Ra_{H_{evap}}}{\left[1 + \left(\frac{0.492}{\mathrm{Pr}}\right)^{9/16}\right]^{8/27}}\right\}^2 \tag{25}$$

The capillary tube-suction line is modeled employing the same formulation previously presented. It is assumed that heat transfer mechanism of the compressor suction line and ambient air is natural convection. A constant value of 10 W/m2-K for the heat transfer coefficient is applied. The thermal cabinet is modeled using an available model in the *software*, which allows the automatic coupling with the evaporator.

In GT-SUITE®, two-phase flow is modeled as homogeneous and a void fraction correlation is applied to estimate the average density during two-phase condition. The homogeneous model is a simplified model that considers the two phases as a homogeneous mixture flowing with same velocity (Rice, 1987):

$$\varepsilon = \left[1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_l}\right)\right]^{-1} \tag{26}$$

3. RESULTS AND DISCUSSION

In this section it will be presented results of transient simulations (pull-down tests) for three different ambient temperatures, namely 21 °C, 32°C and 43°C, and also the cycling behavior of the refrigerator by the acting of the thermostatic valve for ambient temperature of 32°C. For all simulations, a refrigerant charge of 80g was considered.

3.1 Pull-down test simulation

Initially the numerical model developed in GT-SUITE® was applied to simulate pull-down tests. Fig. 1 shows the air temperature variation during the pull-down tests for three ambient temperature, namely 21°C, 32°C and 43°C.

In Fig. 1 it is observed that the air temperature decreases slowly due to the high thermal inertia of the air inside the cabinet. As expected, the air temperature establishes in a higher value as higher the ambient temperature value. Also, as higher is the ambient temperature, more time is required for the refrigeration system to reach steady state condition.

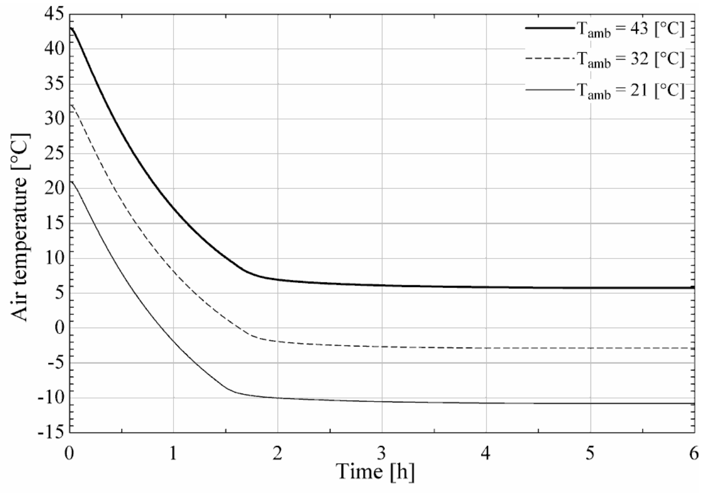

Figure 1. Air temperature variation during pull-down tests

Fig. 2 shows the compressor power consumption during pull-down tests for three ambient temperatures considered in this paper. It can be noticed that power consumption increases with ambient temperature, since the cabinet thermal load increases. For all cases it can be noticed a power peak when the system turns on, related to the mechanical inertia of the compressor. It should be pointed out that the compressor model applied on this paper is simple, since both volumetric and global efficiencies are constant and independent of the ambient temperature, which is not quite close to the actual performance of the compressors applied in refrigeration systems.

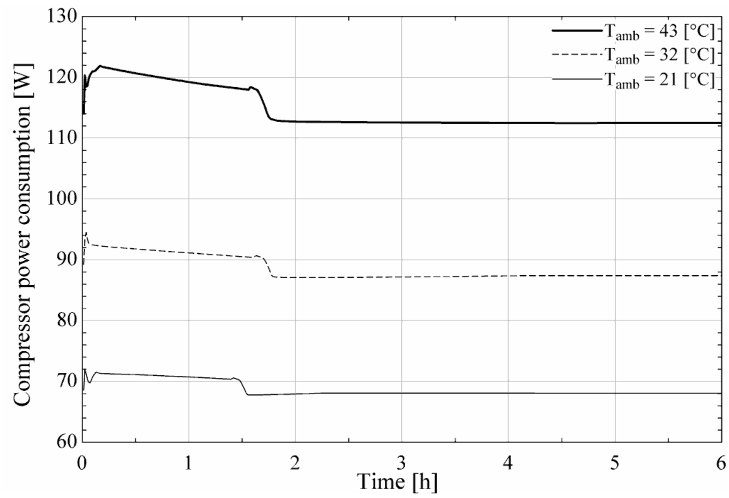

Figure 2. Compressor power consumption during pull-down tests

Additionally, it can be noticed two regions. The first one is transient and lasts approximately 1.5h and the other is stationary, where the system reaches steady state.

During pull-down test for ambient temperature of 32°C, it is shown in Fig. 3 the refrigeration capacity and the thermal load of the domestic refrigeration. It can be noticed that the refrigeration capacity during transient operation is higher, since on this period the air temperature inside the cabinet is higher. It can be observed that steady state is reached, meaning that the refrigeration capacity reaches the thermal load, approximately 2 hours after of the compressor start-up.

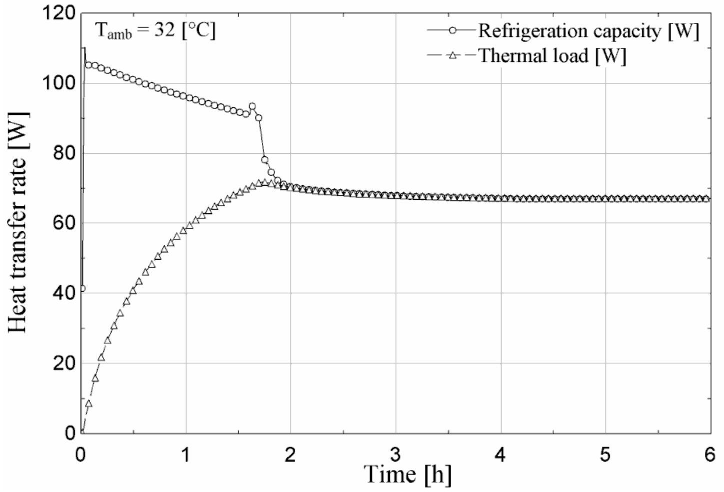

Figure 3. Refrigeration capacity and thermal load variation during pull-down tests

Fig. 4 shows the transient variation of COP (coefficient of performance) of the domestic refrigerator considered in this paper for ambient temperature of 32 °C. Since, as shown before, the refrigeration capacity of the system is higher during transient behavior the COP is higher during transient operation, exhibiting a behavior close to the refrigeration capacity presented before. For ambient temperatures of 21°C and 43°C similar results were observed.

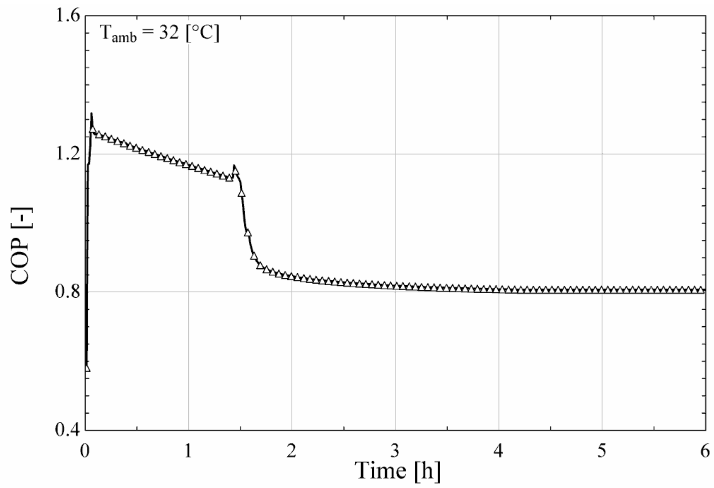

Figure 4. COP variation during pull-down tests

3.2 Cycling operation

The performance of the domestic refrigerator operation during cycling operation is analyzed by acting over the thermostatic valve. A model for this device was also developed in GT-SUITE® and the considered set-point temperatures were 5°C and 7°C, to shut off and turn on the compressor, respectively. Results presented is this section are concerned to an ambient temperature of 32°C. For other ambient temperature values results were similar.

Fig. 5 show the air temperature inside the cabinet during cycling operation. It can be noticed that the first cycle is longer since the air temperature has to decrease from the ambient temperature to the defined set-point temperature. Once the air temperature reaches 5°C, the thermostatic valve shuts the compressor off and the air temperature increases until the upper limit, when the valve turns the compressor on and the cycle repeats indefinitely. It should be pointed that the behavior of a typical domestic refrigerator is always characterized by a cyclic transient operation.

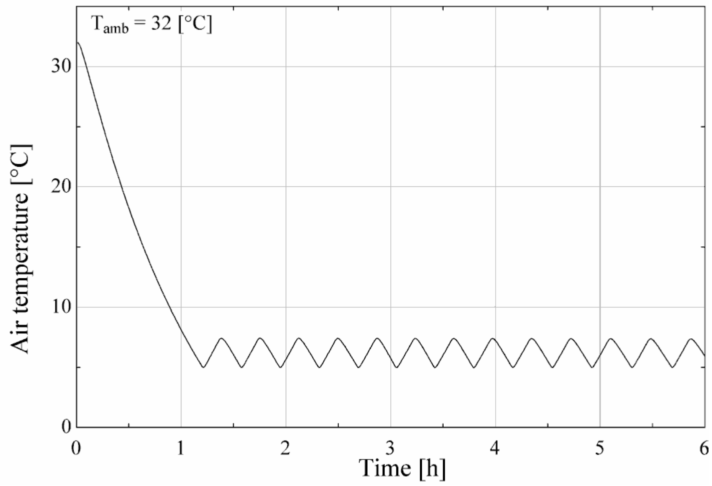

Figure 5. Air temperature variation during cycling operation

Fig. 6 shows the power consumption variation during on-off operation. Immediately when the compressor is turned on it is observed a pressure peak, related to the mechanical inertia of the compressor.

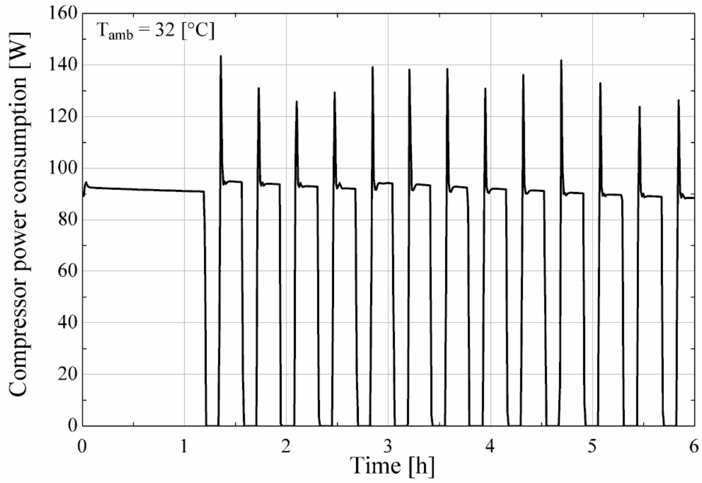

Figure 6. Compressor power consumption variation during cycling operation

Fig. 7 show the dynamic behavior of discharge and suction pressure during cycling operation. For simplicity, all variables were averaged over a normalized cycle. It can be noticed that during off-period, the discharge pressure decreases rapidly to the suction pressure value and when the upper set-point temperature is reached and the compressor is turned on. It is observed a high-pressure gradient on the discharge and a sudden decrease on suction pressure due to the compressor start-up.

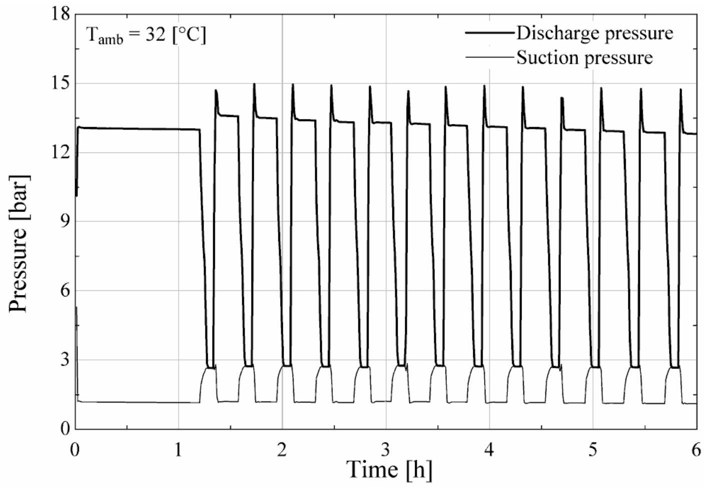

Figure 7. Compressor suction and discharge pressures variation during cycling operation

The numerical model can also predict the refrigerant mass migration during cycling operation, which is depicted in Fig. 8:

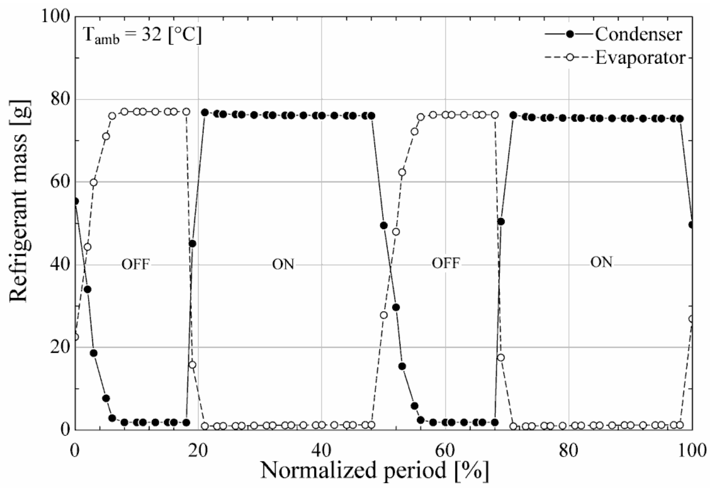

Figure 8. Refrigerant mass migration variation during cycling operation

During on-period, most of the refrigerant mass is located on the condenser, since the compressor is creates the a high pressure gradient and the capillary tube acts as a restriction. During off-period, the pressure gradient decreases rapidly and most of the refrigerant mass is located in the evaporator, since this equipment is at a lower temperature. The mass migration during on-off period is difficult to be measured experimentally, showing this to be an important feature of the model developed in GT-SUITE®. The analysis carried with the presented model considers that the compressor operates in steady state, which is not the actual operation of this equipment. The model disregards the refrigerant and oil interaction and the refrigerant mass withheld inside the compressor housing. This is a strong limitation of the presented model.

4. CONCLUSIONS

This paper presented a simulation model developed in GT-SUITE® to perform transient simulations of a 230L single compartment household refrigerator during start-up and cycling operations. Geometrical parameters from all equipment of the refrigeration system and refrigerant charge were set to build the simulation model. Empirical correlations for heat transfer coefficient and friction factor were also considered. The simulation model was used to reproduce pull-down tests for three different ambient temperatures, namely 21°C, 32°C and 43°C. Cycling operation was modeled by acting over the thermostatic valve and results were shown for an ambient temperature of 32°C. Results showed that the model could reproduce the thermal-fluid behavior of the domestic refrigerator in which several variables could be analyze, including refrigeration capacity and thermal load. During cycling operation the model was able to predict the cycling behavior of the system, including results difficulty to measure experimentally, such as refrigerant mass migration. Those results gave a qualitative understating of the global behavior of the system, since no experimental data were available for comparison and validation.

5. ACKNOWLEDGEMENTS

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