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NUMERICAL MODELING OF A DOMESTIC VAPOR COMPRESSION REFRIGERATION SYSTEM AND DETERMINATION OF REFRIGERANT CHARGE

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Abstract. The increasing on domestic refrigeration systems efficiency is an important theme of researches. This is achieved through optimizations in system components. Improvements can be obtained by advances on compressor efficiency, insulation and heat exchangers heat transfer coefficient, changing common capacity control strategy from compressor on/off cycling to a variable capacity control strategy or finding the ideal refrigerant charge for the system on the chosen characteristics of operation. This last subject often is a challenge for engineers and industries of this area. Usually, standard tests are made based on trial and error strategy to define this parameter, however, these tests demand long periods of time and bring another source of costs. So, on this work, a modeling of a domestic refrigeration system (freezer) with a mass inventory of refrigerant, in order to determine the optimal charge of fluid, is presented.

Keywords: Numerical and Mathematical Modeling, Refrigeration, Efficiency, Refrigerant charge or mass inventory

1. INTRODUCTION

Domestic refrigerators are widely used and are responsible for approximately 8% of the total energy consumption in Brazil (Empresa de Pesquisa Energética, 2015; Eletrobrás, 2007). This situation is aggravated due the low thermodynamic efficiency of these products. Therefore, the increasing on the refrigerators efficiency is a great way to improve the energy use in the country. The majority of these systems are composed by the following components: compressor, condenser, capillary tube, evaporator and cabinet, and operate under an on/off cycle, in other words, the desired range of temperatures for internal air of the cabinet is preset and, when this parameter reaches the inferior limit, the system is shuttled-off, analogously, when the air reaches the superior limit the system is turned-on. The optimization of these systems can be obtained changing some parameters of each component: advances on compressor efficiency, insulation and heat exchangers heat transfer coefficients, changing the common capacity control strategy from compressor on/off cycling to a variable speed capacity control strategy or fitting the best charge of refrigerant for the application conditions set. The refrigerant charge influence both the condensing and evaporating pressures, through the excess or the scarcity of fluid on the heat exchangers (Boeng, J., 2012).

The behavior of a refrigeration system by mechanical vapor compression is affected by each component performance, so that to know the system responses, many standard tests are made on controlled environments. Based on this, the present work shows the modeling of a domestic refrigeration system focused on finding the right refrigerant charge for such given operation conditions.

2. METHODOLOGY

It was made a program on the programming language Python, using the thermodynamic library CoolProp. The model was based on the thesis of Jakobsen, A. (1995) and is composed basically by the application of the First Law of Thermodynamics on each system component, doing energy balances. The product modeled is a freezer with one compartment of 824 L, an alternative compressor, finned tube condenser with ventilation, capillary tube and coil

evaporator with R-290 (propane) as refrigerant. Tests were made (pull-down and cycling) with this refrigerator to find the thermal conductances and thermal capacities of the system components and to compare the results to the program ones.

2.1 Modeling

The refrigeration cycle that represents this system is showed on Fig. 1. The point 1 is the compressor inlet, point 2 is the compressor outlet and condenser inlet, point 3 is condenser outlet and capillary tube inlet, point 4 is the capillary tube outlet and evaporator inlet and the point 5 is the evaporator outlet.
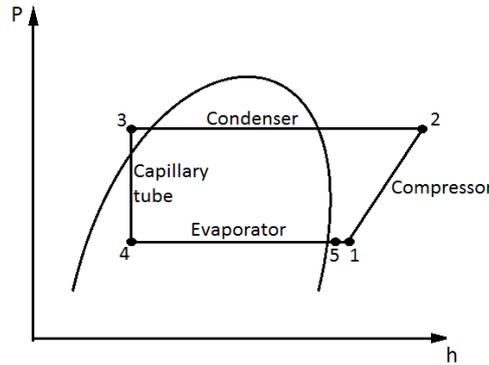

Figure 1. P-h diagram that represents the system operation.

The mathematical model was based on the following hypothesis:

- Hypothesis for the components: disregard: delays on transport, pressure loss and refrigerant accumulation on connector tubes; pressure loss on condenser and evaporator; spatial variation of temperature at condenser, evaporator and compressor surfaces; spatial variation of temperature inside the compartments of cabinet; changes on the amount of refrigerant dissolved on oil; dehumidification of air; snow and ice formation on evaporator surface; and door openings, air infiltration and the impact of goods cooling inside the cabinet on the thermal load calculating.

- Hypothesis 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; and negligible field forces.

Each component model provides information to the next one, in order to enable iterations. Following, the model of compressor is presented. Temperatures 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 mass flow provided by compressor is calculated as follows:

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

Where η_v is the compressor volumetric efficiency, calculated from the compressor curves from manufacturer, which are polynomial expressions that provide the cooling capacity, power consumption, mass flow rate and current, as function of the condensing and evaporating temperatures; ρ_1 is the density, at compressor inlet, V_s is the compressor displacement, and N_{rpm} is the compressor rotation speed, in rpm. The discharge temperature, T_2, is calculated using the housing temperature, T_{com}, based on the results of the tests.

The heat rejected through the compressor housing, Q_{com}, and the power that the compressor demands are:

$$\dot{Q}_{com} = UA_{com}(T_{com} - T_{ext}) \tag{2}$$

$$\dot{W}_{com} = \dot{m}_{com} \frac{(h_{2s} - h_1)}{\eta_s} \tag{3}$$

where UA_{com} is the thermal conductance of the compressor; T_{com} and T_{ext} are the compressor housing and the external temperature around compressor and condenser (this temperature is different from the ambient temperature, T_{amb}, because this components are inside a compartment, being ventilated by a fan, so, the temperature there is higher); h_1 is the compressor inlet enthalpy; h_{2s} is the enthalpy at compressor outlet considering isentropic compression and η_s is the isentropic efficiency, calculated with the same proceedings of the other compressor efficiency calculation.

The energy balance consider the power entering the control volume around the compressor through the suction mass flow and the external power supply and the power going out through the discharge mass flow and the heat lost through the compressor housing:

$$MC_{com} \frac{dT_{com}}{dt} = \dot{W} - \dot{Q}_{com} - \dot{m}_{com}(h_2 - h_1) \tag{4}$$

Being MC_{com} the compressor thermal capacity.

The next step is the modeling of the condenser. The condensing pressure is established as the saturation pressure at the condensing temperature T_{cond}. The degree of subcooling, ΔT_{sc}, is an input of the program, and is defined as the difference between the condensing temperature and the outlet temperature of the condenser, T_3. So, the heat transferred through the condenser is:

$$\dot{Q}_c = UA_c (T_{wc} - T_{ext}) \tag{5}$$

Being UA_c the thermal conductance of the condenser and T_{wc}, the condenser wall temperature that is approximated to the condensing temperature. With the First Law of Thermodynamics we obtain:

$$MC_c \frac{dT_{wc}}{dt} = \dot{m}_{com}(h_2 - h_3) - \dot{Q}_c \tag{6}$$

Where MC_c is the thermal capacity of condenser.

The capillary tube is considered adiabatic and, then, the expansion process is isenthalpic, so:

$$h_3 = h_4 \tag{7}$$

The evaporator model is very similar to the condenser one. The evaporating pressure is the saturation pressure at the evaporating temperature T_{evap}, which is approximated to the evaporator wall temperature T_{we}. The degree of superheat ΔT_{sh}, is an input analogously to the degree of subcooling. The outlet temperature of evaporator, T_5, is calculated summing the evaporating temperature with ΔT_{sh}. Finally, the inlet temperature of the compressor is obtained using T_5, based on the results of the tests. The heat transferred through the evaporator is:

$$\dot{Q}_e = UA_e (T_r - T_{we}) \tag{8}$$

Where, UA_e is the thermal conductance of the evaporator and T_r, the temperature of the air inside the cabinet. With the energy balance on the evaporator T_{we} can be calculated:

$$MC_e \frac{dT_{we}}{dt} = \dot{Q}_e - \dot{m}_{com}(h_5 - h_4) \tag{9}$$

Where MC_e is the thermal capacity of the evaporator.

Finally, there is the model of the cabinet. The thermal load $\dot{Q}_r$ (in other words, the heat that enters the cabinet) is:

$$\dot{Q}_r = UA_r(T_{amb} - T_r) \tag{10}$$

Being, UA_r the thermal conductance of the cabinet. Applying the First Law it is obtain the Eq. (11):

$$MC_r \frac{dT_r}{dt} = \dot{Q}_r - \dot{Q}_e \tag{11}$$

Where MC_r is the thermal capacity of the cabinet.

2.2 Mass inventory

To find the charge of refrigerant that provides the desired performance of the system, a mass inventory in steady state was made, so that with compressor rotation and degree of subcooling and superheat as inputs, the program returns the refrigerant mass inventory.

The great majority of refrigerant is in three components: compressor, condenser and evaporator. On this approach, the heat exchangers are divided in three parts: subcooled, two-phase, and superheated regions.

The volumes of each region must be calculated. For this purpose, the relation between these volumes and the total volume of the heat exchanger was approximated to the relation between the heat transferred on each part and the total heat transferred. For the condenser there is:

$$\dot{Q}_{c,2ph,s} = \dot{m}_{com,s}(h_{c,sat,g} - h_{c,sat,l}) \tag{12}$$

$$\dot{Q}_{c,sh,s} = \dot{m}_{com,s}(h_2 - h_{c,sat,g}) \tag{13}$$

$$\dot{Q}_{c,sc,s} = \dot{m}_{com,s}(h_{c,sat,l} - h_3) \tag{14}$$

Where, $\dot{Q}_{c,2ph,s}$, $\dot{Q}_{c,sh,s}$ and $\dot{Q}_{c,sc,s}$ are the heat transferred in the two-phase, superheated and subcooled regions, $\dot{m}_{com,s}$ is the mass flow rate and $h_{c,sat,g}$ and $h_{c,sat,l}$ are the saturated vapor and saturated liquid enthalpies at the condensing temperature. All variables are evaluated in steady state.

The volumes of the regions are:

$$V_{c,2ph,s} = \frac{Q_{c,2ph,s}}{Q_{cond}} V_c \tag{15}$$

$$V_{c,sh,s} = \frac{Q_{c,sh,s}}{Q_{cond}} V_c \tag{16}$$

$$V_{c,sc,s} = \frac{Q_{c,sc,s}}{Q_{cond}} V_c \tag{17}$$

Where, $V_{c,2ph,s}$, $V_{c,sh,s}$ and $V_{c,sc,s}$ are the volumes of the two-phase, superheated and subcooled regions in steady state respectively. V_c is the total volume of condenser.

Analogously, the same processes are adopted for the evaporator:

$$\dot{Q}_{e,2ph,s} = \dot{m}_{com,s}(h_{e,sat,g} - h_4) \tag{18}$$

$$\dot{Q}_{e,sh,s} = \dot{m}_{com,s}(h_5 - h_{e,sat,g})$$ (19)

Where, $\dot{Q}_{e,2ph,s}$ and $\dot{Q}_{e,sh,s}$ are the heat transferred in the two-phase and superheated regions, and $h_{e,sat,g}$ is the saturated vapor enthalpy at the evaporating temperature.

$$V_{e,2ph,s} = \frac{Q_{e,2ph,s}}{Q_{evap}}V_e$$ (20)

$$V_{e,sh,s} = \frac{Q_{e,sh,s}}{Q_{evap}}V_e$$ (21)

Where, $V_{e,2ph,s}$ and $V_{e,sh,s}$ are the volumes of the two-phase and superheated regions in steady state respectively. V_e is the total volume of evaporator.

Then, in the single-phase regions, the mass of refrigerant on each heat exchanger is calculated as follows:

$$m_{c,sh,s} = V_{c,sh,s}\rho_{c,sh,s}$$ (22)

$$m_{c,sc,s} = V_{c,sc,s}\rho_{c,sc,s}$$ (23)

$$m_{e,sh,s} = V_{e,sh,s}\rho_{e,sh,s}$$ (24)

Where the pairs m and ρ are the mass and the density of the fluid on each single-phase state.

For the two-phase regions, the calculation includes a model for the void fraction of the flow. It was used the Wallis void fraction model:

$$\alpha = \left[1 + \left(\frac{1-x}{x}\right)^{0.72}\left(\frac{\rho_v}{\rho_l}\right)^{0.40}\left(\frac{\mu_l}{\mu_v}\right)^{0.08}\right]^{-1}$$ (25)

Being, α, the void fraction, x, the quality, ρ_v and ρ_l the density of saturated vapor and saturated liquid respectively, and μ_v, μ_l their viscosities.

Now, the masses of refrigerant in two-phase regions are:

$$m_{c,2ph,s} = V_{c,2ph,s}\left(\alpha\rho_{c,sat,g} + (1-\alpha)\rho_{c,sat,l}\right)$$ (26)

$$m_{e,2ph,s} = V_{e,2ph,s}\left(\alpha\rho_{e,sat,g} + (1-\alpha)\rho_{e,sat,l}\right)$$ (27)

Where the densities ρ are in saturated conditions for the condenser and evaporator.

For the compressor, there are two regions: high pressure (discharge line) and low pressure (housing and suction muffler). The calculation evolves only gas phase, so the masses are:

$$m_{com,h,s} = V_{com,h,s}\rho_{com,h,s}$$ (28)

$$m_{com,l,s} = V_{com,l,s}\rho_{com,l,s}$$ (29)

Where, m, V and ρ are analogous to the case of heat exchangers single-phase states. The h and the l in the sub-indices indicate high pressure and low pressure regions, respectively.

Finally, the total mass of refrigerant m_{tot} is:

$$m_{tot} = m_{c,sh,s} + m_{c,sc,s} + m_{e,sh,s} + m_{com,h,s} + m_{com,l,s} + m_{c,2ph,s} + m_{e,2ph,s} \tag{30}$$

3. RESULTS

The Tab. 1 shows the values obtained for the thermal conductances and capacities on a pull-down test at 32°C of ambient temperature.

Table 1. Thermal conductance and thermal capacity of each component.

Component	Thermal conductance (W/°C)	Thermal capacity (J/°C)
Compressor	7.50	15000.0
Condenser	38.88	62205.56
Evaporator	18.36	6975.31
Cabinet	3.56	53332.35

The inputs of the program, provided for the pull-down test are 4100 rpm as compressor rotation, 1.0°C as degree of subcooling and 0.5°C as degree of superheat. The Fig. 2. shows the temperatures of compressor housing, condenser wall, evaporator wall, cabinet air and the compressor power consumption, comparing these parameters obtained on the experimental test (dashed line) to the simulation ones (solid line).
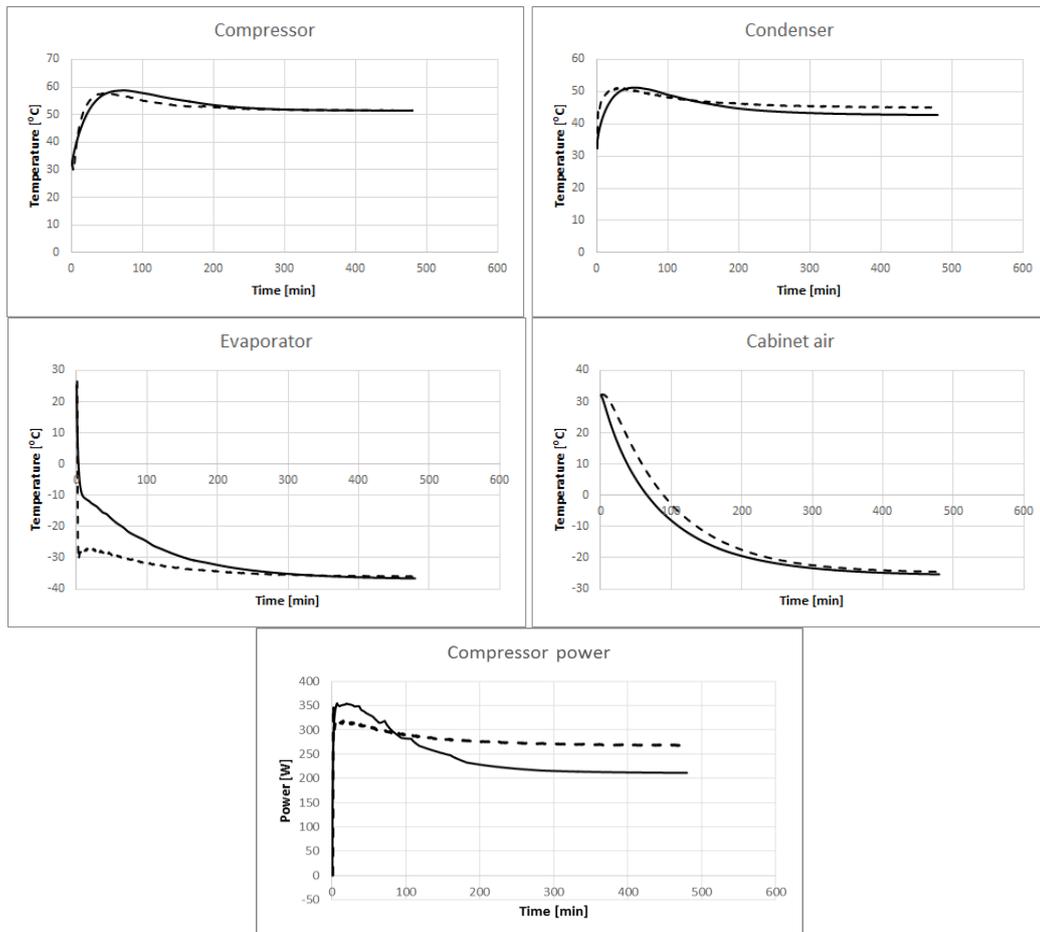

Figure 2. Behavior of the components temperature and power consumption on the pull-down test (experimental vs. simulation).

The Tab. 2 shows the refrigerant charge values computed in some parts of the refrigeration system, obtained by the adopted modeling approach.

Table 2. Refrigerant mass inventory in some parts of the systems calculated by the present model.

Condenser void fraction	0.709
Evaporator void fraction	0.862
Mass of gas on condenser	3.045 g
Two-phase mass on condenser	85.588 g
Mass of liquid on condenser	2.463 g
Mass of gas on evaporator	0.420 g
Two-phase mass on evaporator	49.049 g
Mass in the high pressure compressor region	0.937 g
Mass in the low pressure compressor region	3.674 g
Total mass	145.175 g

The mass of propane measured experimentally on the test was 150 g, so, the model is coherent.

The effect of changing the charge of refrigerant was studied and compared to the base line condition. To do this analysis, the inputs of subcooling (ΔT_{sc}) and superheat (ΔT_{sh}) were modified, as shown on the Tab. 3.

Table 3. Comparison between different operation conditions.

Condition	ΔT_{sc}	ΔT_{sh}	Total mass of refrigerant [g]	COP
Base line	1.0	0.5	145.175	1.117
Higher subcooling	10.0	0.5	158.728	1.193
Higher superheat	1.0	10.0	141.427	1.167
Lower subcooling	0.1	0.5	143.786	1.110
Lower superheat	1.0	0.1	145.343	1.115

These results can show the COP (coefficient of performance) and the tendency of subcooling and superheat when the mass of refrigerant is changed on the system. When this one is increased, the subcooling increases and the superheat decreases, on the other hand, with a reduction in the mass of refrigerant, the subcooling decreases and the superheat increases, which was expected.

Another conclusion is that variations on refrigerant charge cause more effect on degree of superheat in comparison to the effect on degree of subcooling.

4. CONCLUSIONS

In the present work, a numerical simulation model for computing the performance of a mechanical vapor compression domestic refrigeration system and the determination of refrigerant charge is being developed. The system components were characterized through the results of experimental tests and the mathematical model was validated comparing its results to the behavior obtained experimentally. The refrigerant mass inventory already measured and simulated, led to conclude that the model is coherent and has potential to model the behavior of such kind of system.

5. ACKNOWLEDGEMENTS

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