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MODELING OF A VARIABLE SPEED COMPRESSOR APPLIED TO A DOMESTIC REFRIGERATOR

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Abstract. *Increasing the energetic efficiency of domestic refrigerators will bring direct impact in Brazilian energetic matrix, since this kind of application is responsible for approximately 8% of the electric consumption in Brazil. This issue is intensified due the low thermodynamic efficiency of such products. Improvements can be obtained by advances on compressor efficiency, insulation and heat exchangers heat transfer coefficient or changing common capacity control strategy from compressor on/off cycling to a variable capacity control strategy, which can be obtained with variable speed compressors. However, the improvements mentioned on each component, cannot guarantee this improvement on entire system because of the complexity of their interactions. Usually, to evaluate and analyze the performance of the system, several tests occur at standard points, which are expensive and slow. Therefore, it is very important to have a well validated model of household refrigerators to represent the system and allow engineers to perform reliable analysis faster. In this work, it is shown an experimental method to obtain the thermal conductance and thermal capacity of the components of a domestic refrigerator (compressor, condenser, evaporator, cabinet). A mathematical model of the refrigerator was developed and validated with experimental data. With this model, it was carried out studies to compare the performance of a standard on/off versus a variable speed compressor. In the ideal strategy, it was obtained gains of up to 34% in consumption, using the variable speed compressor.*

Keywords: *domestic refrigerator, energetic efficiency, variable speed compressor, refrigerator modeling*

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). In your great majority, these products are equipped with a thermostat and an on/off compressor, that remains functioning at a constant speed until the refrigerated volume temperature reaches the minimum target temperature defined by the thermostat, when it is shuttled-off. Improvements in insulation, compressor efficiency and optimization of refrigerant charge have reduced energy consumption significantly in recent years (Binneberg, Kraus and Quack, 2002). However, according to Negrão and Hermes (2011), increase the efficiency of each component separately does not guarantee itself an improvement in the performance of the overall system due a poor component matching.

Usually, this matching among the components is obtained by several tests of the refrigerator at standard points, which are very costly and time consuming (Negrão and Hermes, 2011). This way, a well validated mathematical model of domestic refrigerators is very important to give agility to the process, reduce costs and provide reliable results to engineers. In this work, an experimental methodology to find the thermal conductance and thermal capacity of components and a mathematical model, based on the work developed by Jakobsen (1995), is shown. The domestic refrigerator simulated has two compartments: refrigerator with 207 L and freezer with 53 L capacity, a roll-bond "plate" on refrigerator and a roll-bond "box" evaporator on freezer, a wire-and-tube condenser and an alternative compressor, with R134a. The mathematical model was validated with experimental data, and then, used to carried out studies comparing the performance of a standard on/off versus a variable speed compressor, which proved to be more efficient.

2. METHODOLOGY

To provide inputs to the mathematical model, two sets of tests were performed to obtain the thermal conductance and the thermal capacity for each component of the system. These tests are: conductance tests for cabinet called "cabinet characterization"; and a "modified pull-down test".

2.1 Cabinet characterization

In this test, the refrigerator is kept turned off and electric resistors are put inside the compartments, as shown in Fig. 1, warming them. The average temperature of both compartments should be above the ambient temperature approximately 25 °C, as recommended by the NTB00119 (1992). It is also defined that the total power provided to the resistors should not be greater than 50 W, to not damage the refrigerator. The ambient temperature selected was 25 °C, so the temperature of each compartment would not be so high. These temperatures were monitored with thermocouples installed, as shown in Fig. 1, with 3 thermocouples distributed in the freezer (top, middle and bottom positions) and 5 thermocouples on the refrigerator, distributed from the top to the bottom. To the electric resistors were fixed aluminium plates, to facilitate the natural convection and obtain a more uniform temperature distribution.

The compartments are closed and the electric resistors turned on, rising the temperature of the compartmets until a steady state condition is established. At this moment, the heat transfer from the electric resistors to the internal air of the compartments is equal to electric power on them. Also, there is heat flow through the cabinet walls between each compartment. So, measuring the electric power and temperatures, we find the heat transferred through the walls. Therefore, with the Eq. (1), the thermal conductances (UA, in W/°C) can be calculated.

$$\dot{Q} = UA \cdot \Delta T \tag{1}$$

Figure 1. Instrumentation of the refrigerator for the "cabinet characterization test".

In this case, with a two-compartment refrigerator, after measure the temperatures, some procedures must be done to find the thermal conductance. Performing energy balances in each compartment, we obtain:

$$\dot{W}_{rr} = UA_{rr}(T_{rr} - T_{amb}) - UA_{w}(T_{rf} - T_{rr}) \tag{2}$$

where $\dot{W}_{rr}$ is the electric power provided to electric resistors inside the refrigerator compartment; UA_{rr} is the thermal conductance between the refrigerator walls and external ambient; UA_{w} is the thermal conductance of the wall between refrigerator and freezer compartments; T_{rr} is the temperature of the air inside the refrigerator and T_{rf} is the temperature of the air inside the freezer. Analogously, for the freezer:

$$\dot{W}_{rf} = UA_{rf}(T_{rf} - T_{amb}) + UA_{w}(T_{rf} - T_{rr}) \tag{3}$$

where UA_{rf} is the thermal conductance between the freezer walls and external ambient. So, summing the equations (2) and (3), we obtain:

$$\dot{W}_{rr} + \dot{W}_{rf} = UA_{rr}(T_{rr} - T_{amb}) + UA_{rf}(T_{rf} - T_{amb}) \tag{4}$$

Thus, $\dot{W}_{rr}$, $\dot{W}_{rf}$, T_{rr}, T_{rf} and T_{amb} are known. Therefore, with two tests, changing the electric power in both compartment, an equation system is obtained and the conductance values are found. Another method to do this is getting many experiments, changing the electric power on compartments and, to find the conductance values, apply the ordinary least squares method with the Eq. (4).

2.2 Modified pull-down test

In this test, the refrigerator is left to stay in thermal equilibrium with the ambient first. Then, with the thermostat by-passed, the compressor is turned on and remains working without cycling, pulling-down the temperatures inside the refrigerator until they reach a steady state condition, which takes around 8 hours. Initially, the experiment was conducted with a conventional on/off compressor running at constant rotation speed of 3600 rpm and with the ambient temperature at 32 °C. Posteriorly, this test was repeated to evaluate the system with a variable speed compressor, running at: 1600, 2000, 3000, 3600, 4500 rpm. The data measured on both cases were: time; voltage; current; electric power consumption; frequency; suction and discharge pressures; and several temperatures (compressor suction and discharge; condenser inlet, middle and outlet; compressor housing (top and bottom); top, middle and bottom of freezer; 5 thermocouples on refrigerator distributed from the top to the bottom; refrigerator roll-bond “plate” inlet, middle and outlet; ambient temperature; heat exchanger outlet on suction line side; heat exchanger inlet on capillary tube side; capillary tube inlet; and discharge line.). Figure 2 shows the instrumentation setup used in this experiment.

Figure 2. Instrumentation of the refrigerator for the “modified pull-down test”.

At the steady state condition, with the pressure and temperature measurements, it is possible to calculate the refrigerant properties at each point of the refrigeration cycle. Knowing also the mass flow rate given by the compressor, an energy balance can be conducted in each component of the refrigeration system, where the heat transfer between the refrigerant and the component itself (internal heat transfer) is equal to the one between the component and the external ambient air (external heat transfer). So, using the Eq. (1), the conductance can be calculated.

To provide sufficient inputs to the model, the thermal capacities (MC, in J/°C) were also necessary. These values can be calculated if the conductance values and the time constants (τ, in s) for each component are known, as follows from Eq. (5) to Eq. (7):

$$\tau = R_t \cdot MC \tag{5}$$

$$R_t = 1/UA \tag{6}$$

$$MC = \tau \cdot UA \tag{7}$$

Time constants were calculated with the tests results, considering the time when 63.2% of the system dynamic is completed. It was also considered that five times the time constant is approximately the time for the system reach steady state. So, the time and temperatures measured in modified pull-down test were used to calculate these parameters.

2.3 Modeling

The refrigeration cycle can be represented in a pressure-enthalpy diagram, as shown in Fig. 3, where point 1 corresponds to the compressor inlet and suction line outlet; point 2 to the compressor outlet and condenser inlet; point 3 to the condenser outlet and capillary tube inlet; point 3o to the initial part of the capillary tube that is connected to the suction line to exchange heat; point 4 to the capillary tube outlet and evaporator inlet; and point 5 to the evaporator outlet and initial part of the suction line that is connected to the capillary tube to exchange heat.

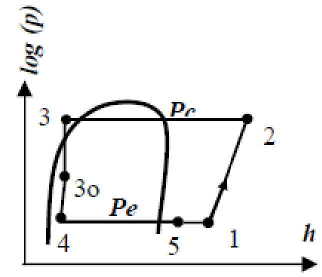

Figure 3. Pressure-enthalpy (*p*-*h*) diagram that represents an ideal refrigeration cycle.

To develop the mathematical model of the problem, some simplifying hypothesis were adopted:

- Hypothesis for physical and 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 neglegible field forces.

- Hypothesis for the components modeling. We disregarded: 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.

The model consists, basically, of an energy balance in an 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. Following are presented these expressions. 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.

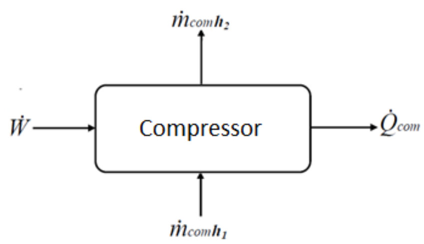

Figure 4: Control volume and energy balance on the compressor.

Figure 4 shows the energy balance on the compressor. Initially, the volumetric and isentropic efficiencies (η_v and η_s, respectively) must be calculated from compressor curves, which are polynomial expressions that provide the cooling capacity, power consumption, mass flow rate and current, as function of the condensing and evaporating temperatures (and rotation speed also, for variable speed compreesors). Thus, an expression for the efficiencies that depends of the condensing and evaporating pressures can be found. The compressor mass flow is given by:

$$\dot{m}_{com} = \eta_v \rho_1 V_s \frac{N_{rpm}}{60} \tag{8}$$

where 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 was modeled by two different strategies: one considering a polytropic compression process and other that approximates this temperature to the compressor's housing surface. It was verified that the second method presented better results.

The heat transferred through the compressor housing $\dot{Q}_{com}$ and the power consumption $\dot{W}_{com}$ are given by:

$$\dot{Q}_{com} = UA_{com}(T_{com} - T_{amb}) \tag{9}$$

$$\dot{W}_{com} = \dot{m}_{com} \frac{(h_{2s} - h_1)}{\eta_s} \tag{10}$$

where UA_{com} is the compressor thermal conductance; T_{com} and T_{amb} are the compressor housing and ambient temperatures, respectively; h_1 is the enthalpy at compressor inlet and h_{2s} is the enthalpy at compressor outlet if the compression process was isentropic. Finally, the transient behavior of the compressor housing temperature can be obtained applying a transient energy balance to the compressor, considering the assumptions already detached:

$$MC_{com} \frac{dT_{com}}{dt} = \dot{W} - \dot{Q}_{com} - \dot{m}_{com}(h_2 - h_1) \tag{11}$$

where MC_{com} is thermal capacity of the compressor.

The same way as for the compressor, the condenser can be modeled, as shown in Fig 5.

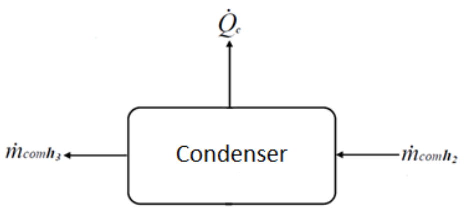

Figure 5. Control volume and energy balance on the condenser.

The pressure on condenser is established as the saturation pressure at the condensing temperature T_{cond}. The degree of sub-cooling ΔT_{sc} was obtained from "modified pull-down test" results, considering the difference between condensing temperature and the condenser outlet T_3. Therefore, the heat transfer through the condenser wall $\dot{Q}_c$ can be calculated from:

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

where UA_c is the condenser thermal conductance and T_{wc} is the condenser wall temperature, which is considered to be equal to the condensing temperature. Applying the transient energy balance to the condenser, we have:

$$MC_c \frac{dT_{wc}}{dt} = \dot{m}_{com}(h_2 - h_3) - \dot{Q}_c \tag{13}$$

where MC_c is thermal capacity of the condenser.

In the capillary tube, it is considered an isenthalpic expansion until the flow reach the part of the capillary tube heat exchanger connected to the suction line, which control volume is presented in Fig. 6.

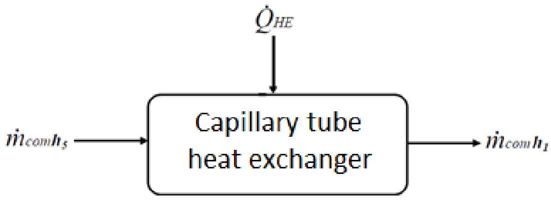

Figure 6. Control volume and energy balance on the capillary tube heat exchanger.

The analysis starts calculating the inlet pressure P_{3o} and temperature T_{3o} of refrigerant on the heat exchanger in the capillary tube inlet as:

$$P_{3o} = P_{cond} - f_{he} \cdot (P_{cond} - P_{evap}) \tag{14}$$

$$T_{3o} = f(P_{3o}, h_3) \tag{15}$$

where f_{he} is the fraction of pressure loss that occurs in the adiabatic part of capillary tube. Verifying the tests, it was observed that the suction temperature T_1 is approximately equal to T_{3o}, which was adopted in the present model. Finally, the heat exchange in the capillary tube is given by:

$$\dot{Q}_{he} = \dot{m}_{com}(h_1 - h_5) \tag{16}$$

where h_1 and h_5 are the enthalpy at compressor inlet and evaporator outlet, respectively.

The evaporator parts of the refrigerator and of the freezer were considered as a unique evaporator. Thus, the cooling capacity ($\dot{Q}_e$) is the sum of the heat transfer in both parts ($\dot{Q}_e = \dot{Q}_{er} + \dot{Q}_{ef}$) and there with an equivalent thermal capacity (MC_{eq}). The control volume over the evaporator is show in Fig. 7.

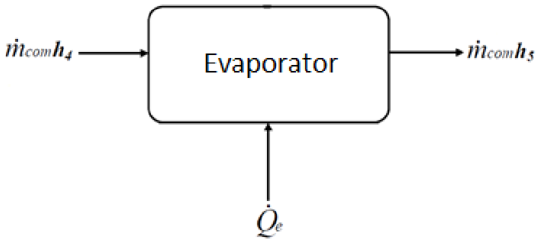

Figure 7. Control volume and energy balance on the evaporator.

The pressure on evaporator is the saturation pressure at the evaporating temperature. The temperature at the evaporator outlet is given by summing the evaporator temperature and the degree of superheat ΔT_{sh}.

Temperatures of the air inside the refrigerator and freezer are T_{rr} and T_{rf}, respectively, and the heat transferred between the internal ambient air of each compartment and the evaporator are given by:

$$\dot{Q}_{er} = UA_{er}(T_{rr} - T_{we}) \tag{17}$$

$$\dot{Q}_{ef} = UA_{ec}(T_{rf} - T_{we}) \tag{18}$$

where T_{we} is the temperature in the evaporator surface, which was considered to be equal to the evaporating temperature. The transient energy balance applied to the evaporator is given by:

$$MC_{eq}\frac{dT_{we}}{dt}=\dot{Q}_e-\dot{m}_{com}(h_5-h_4) \tag{19}$$

where MC_{eq} is the equivalent thermal capacity of the evaporator.

When the system turns off, the temperatures of the evaporator surfaces located in the refrigerator and the freezer become very different, because of the temperature difference between the air inside each of these compartments. Thus, the surface temperatures are treated separately at these moments:

$$MC_{er}\frac{dT_{we,r}}{dt}=\dot{Q}_{er} \tag{20}$$

$$MC_{ef}\frac{dT_{we,f}}{dt}=\dot{Q}_{ef} \tag{21}$$

where $T_{we,r}$ is the surface temperature of the roll-bond plate and $T_{we,f}$ is the surface temperature of the roll-bond box.

Finally, are modeled the refrigerator and freezer compartments. The heat transfers of these compartments are shown in Fig. 8, where $\dot{Q}_{rr}$ is the heat transfer between refrigerator wall and the external ambient; $\dot{Q}_{rf}$ is the heat transfer between freezer wall and the external ambient and $\dot{Q}_w$ is the heat transfer through the wall between the two compartments.

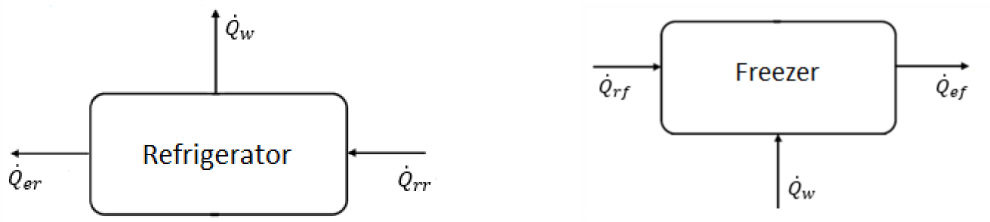

Figure 8. Control volume and energy balance on refrigerator and freezer.

$$\dot{Q}_{rr}=UA_{rr}(T_{amb}-T_{rr}) \tag{22}$$

$$\dot{Q}_{rf}=UA_{rc}(T_{amb}-T_{rf}) \tag{23}$$

$$\dot{Q}_{w}=UA_{w}(T_{rr}-T_{rf}) \tag{24}$$

To conclude, the temperatures inside these compartments are calculated performing an energy balance applied to both compartments:

$$MC_{rr}\frac{dT_{rr}}{dt}=\dot{Q}_{rr}-\dot{Q}_{er}-\dot{Q}_w \tag{25}$$

$$MC_{rf}\frac{dT_{rf}}{dt}=\dot{Q}_{rf}-\dot{Q}_{ef}+\dot{Q}_w \tag{26}$$

All the equations of the mathematical model were implemented in a numerical code developed using Python. Thermodynamic properties were calculated using CoolProp library. To run the program, initially, an initial guess was given for all temperatures and the constants. The ordinary differential equations were solved using 4th order Runge-

Kutta method with a timestep of 15 s, and the set of algebraic equations obtained for each time step was calculated using a Newton-Rapson method. Both methods are available into Python´s libraries.

It was simulated the refrigeration system in three ways, to compare a standard on/off versus a variable speed compressor: First, with the on/off compressor; then, secondly, with a variable speed compressor (VSC) that provides the same refrigeration capacity of the on/off one (at ASHRAE conditions) operating at on/off logic and, finally, with the variable speed compressor operating under a classic proportional control logic, schematized in Fig. 9.

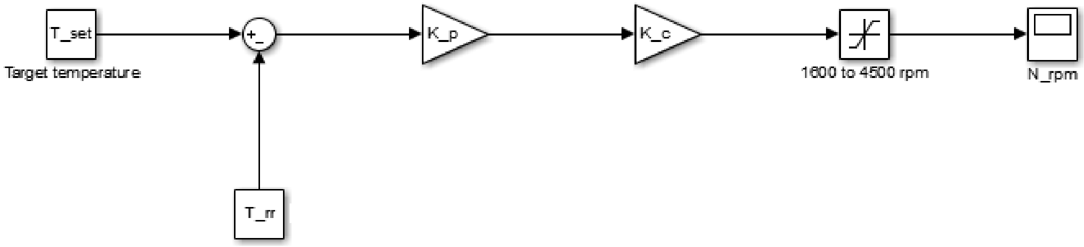

Figure 9. Diagram that shows how the proportional control works.

The product between K_p and K_c was adopted as 1500 in the simulations, this way, when the temperature inside the refrigerator reaches 3°C greater than the set temperature, the system will be saturated, which means the rotation will be the superior limit of 4500 rpm. It was stablished 1600 and 4500 rpm as limits running speeds. The temperature set was the inferior limit of the thermostat, when it turns off the system.

3. RESULTS

Table 1 contains the thermal conductance and thermal capacity values, obtained from the characterization tests with the on/off compressor:

Table 1. Thermal conductance and thermal capacity values calculated from the tests.

Component	Thermal conductance (W/ºC)	Thermal capacity (J/ºC)
Compressor	1.23	6356.64
Condenser	5.45	20173.72
Evaporator (refrigerator)	1.16	975.7
Evaporator (freezer)	3.76	9026.66
Evaporator (equivalent)	--	8593.81
Refrigerator	1.14	6853.2
Freezer	0.32	1363.57
Cabinet wall	0.065	--

It was also found $\Delta T_{sc} =$ 0.87 °C and $\Delta T_{sh} =$ 2.85 °C from experiments. Same data were obtained as function of the rotation speed, using the tests with the variable speed compressor, and they are presented in Tab. 2

Table 2. Expressions for the component properties in function of rotation.

$UA_{com} = 6(10^{-5})N_{rpm} + 1.2208$	$UA_{ec} = -0.0007N_{rpm} + 10.091$
$MC_{com} = 0.3645N_{rpm} + 7324.7$	$MC_{eq} = -1.0497N_{rpm} + 15513$
$UA_{c} = -0.0005N_{rpm} + 9.2872$	$\Delta T_{sc} = 0.00013N_{rpm} + 0.4118$
$MC_{c} = -2.5541N_{rpm} + 50151$	$\Delta T_{sh} = 32.847e^{-5(10^{-4})N_{rpm}}$
$UA_{er} = 8(10^{-5})N_{rpm} + 1.0315$	--

To verify and validate the model, a cycling test (on/off compressor) without load, with ambient temperature of 32 °C and duration of 8 h was made. Figure 10 shows the comparison between the experiment (dashed line) and the simulation (solid line). It was obtained a good description of the system parameters. Even with initial values settled out of the stabilization range of the cycles (showed by the data collected from the experiment), the simulation converged to the experimental range with good agreement.

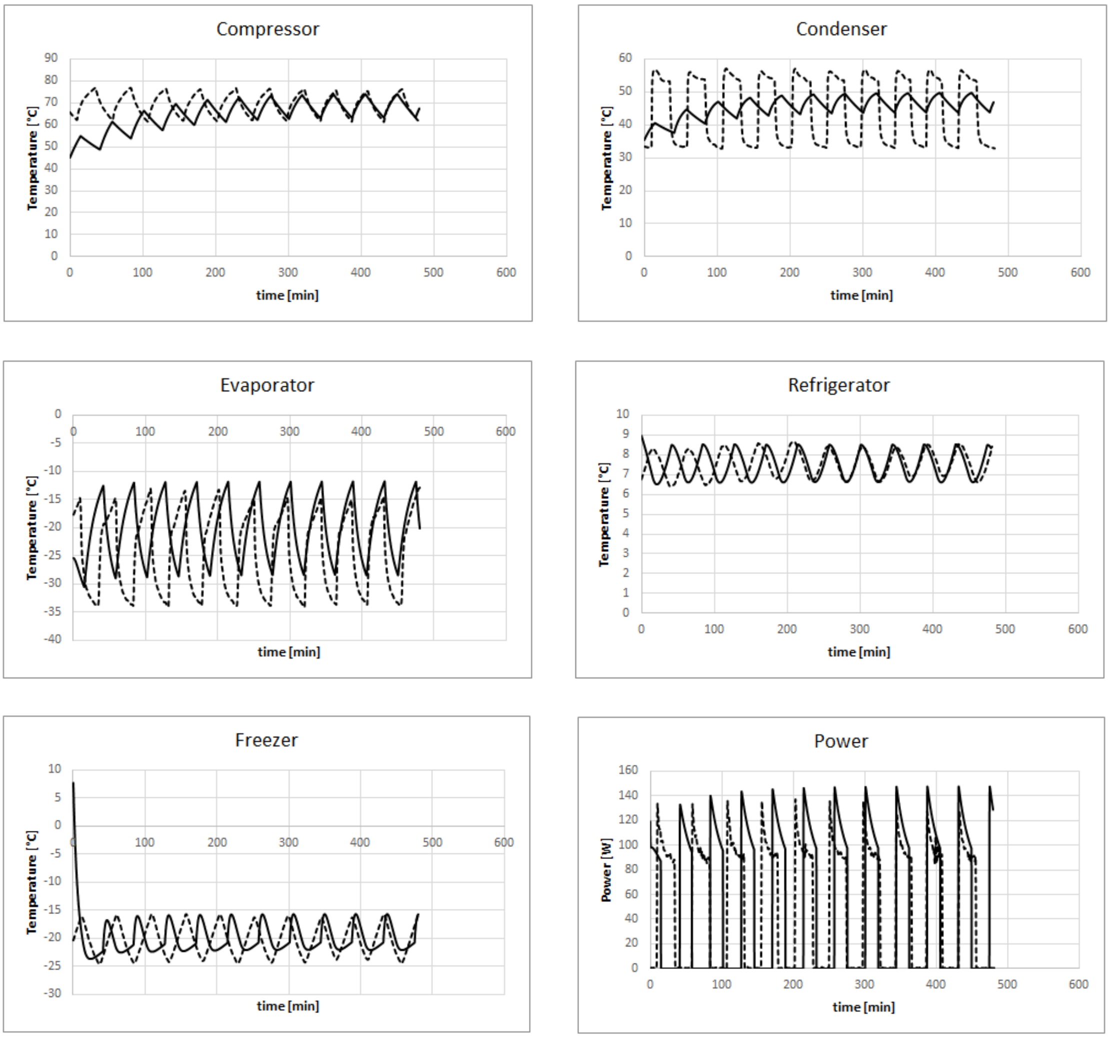

Figure 10. Behavior of the components temperature and power.

A comparison between different compressor running strategies were performed at two ambient temperatures and three conditions of the thermostat: (min.: 3.0 to 4.5 °C, med.: 5.0 to 6.5 °C and max.: 7.0 to 8.5 °C), as shown in Tab. 3. The results showed that the variable speed compressor operating under on/off logic at the same rotation of the standard on/off compressor already shows better efficiency and less consumption. This can be explained mainly due differences on the electric motor efficiencies, which was higher for the brushless motor of the variable speed compressor than for the induction motor of the on/off compressor. However, when we apply the proportional control logic to vary the rotation speed (and the refrigeration capacity), we can observe an improvement on efficiency, reaching values up to 34%, as consequence of the consumption decrease. Nevertheless, it is important detach that the control adjustments to attend different requirements on a practical case may decrease the obtained gains.

4. CONCLUSION

In the present work an experimentally validated numerical model to simulate a two compartments refrigerator, with a roll-bond “plate” on refrigerator and a roll-bond “box” evaporator on freezer, was developed. The experimental setup was used to characterize the refrigerator at several compressor running speeds. With the numerical model, it was possible to investigate the refrigerator efficiency using three different compressor running manners and at several environment conditions. It was observed an improvement when using the proportional control logic, obtaining gains up to 34% in consumption, when compared to traditional on/off cycling. Nevertheless, it is important detach that the control adjustments to attend different requirements on a practical case may decrease the obtained gains.

Table 3. Performance comparison between the system with different compressors and logics.

		Ambient: 22 °C			Ambient: 32 °C		
		Compressor on/off	**Var. speed (on/off)**	**Variable speed**	**Compressor on/off**	**Var. speed (on/off)**	**Variable speed**
Thermostat (3 to 4.5 °C)	**COP**	1.19	1.26	1.57	0.87	0.93	1.30
	Consumption (kWh/month)	23.89	22.70	17.50	42.40	39.80	28.02
	% Gain	-	5.0	26.7	-	6.1	33.9
	$\dot{Q}_e$ **(W)**	39.57	39.75	38.18	51.01	51.17	50.75
	No. of cycles	10	9	8	10	9	5
	Time on (min)	14.80	15.81	29.06	28.00	27.97	67.44
	Time off (min)	32.94	32.47	27.96	20.00	20.61	17.25
Thermostat (5 to 6.5 °C)	**COP**	1.39	1.40	1.71	0.89	0.96	1.26
	Consumption (kWh/month)	19.48	19.57	15.24	39.10	36.87	27.37
	% Gain	-	-0.5	21.8	-	5.7	30.0
	$\dot{Q}_e$ **(W)**	37.64	38.09	36.14	48.42	49.24	48.05
	No. of cycles	9	9	8	10	10	7
	Time on (min)	12.89	13.86	24.50	22.18	22.92	49.08
	Time off (min)	37.56	36.88	31.75	22.28	22.58	19.04
Thermostat (7 to 8.5 °C)	**COP**	1.60	1.65	1.84	0.91	0.97	1.26
	Consumption (kWh/month)	16.03	15.63	13.44	36.81	34.38	26.19
	% Gain	-	2.5	16.2	-	6.6	28.9
	$\dot{Q}_e$ **(W)**	35.55	35.90	34.42	46.51	46.50	45.70
	No. of cycles	8	8	8	11	10	8
	Time on (min)	11.50	12.25	20.97	18.40	19.375	38.25
	Time off (min)	43.61	42.75	36.79	25.00	25.03	21.25

5. ACKNOWLEDGEMENTS

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