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Pros & Cons of Using Hot-Wall Condensers in Household Refrigerators

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Pros & Cons of Using Hot-wall Condensers in Household Refrigerators

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ABSTRACT

This work focuses on the pros and cons of using hot-wall condensers in household refrigerators, based on both numerical and experimental approaches. To this end, seven refrigerators were manufactured with distinct designs of hot-wall condensers. The design parameters were the following: (i) adhesive tape (aluminum or polyethylene), (ii) tube outer diameter (4 or 4.76 m), (iii) total length (10 or 11.5 m) and (iv) lay-out. An in-house mathematical model for hot-wall condensers was added to an in-house system simulation tool to predict the refrigerators performance. Experiments were conducted in a climate-controlled test chamber. It was found that the model predictions are close to the energy consumption measurements with deviations of the order of ±10%. It has also been found that the heat load is increased by 7.7% when a hot-wall condenser is added to the system. An extensive sensitivity analysis was also carried out, showing that the hot-wall condenser and thus the refrigerator performance is very much affected by the outer sheet thermal conductivity and thickness, but mainly by the tape thermal conductivity. The contact area between tape and outer sheet also plays a significant role in the heat transfer, meaning that a cheaper polymeric tape might be used if enough contact area is provided. Additionally, it has been found that there is a tube pitch which minimizes the energy consumption in despite of the condenser geometry.

KEYWORDS: hot-wall condenser, household refrigerator, heat exchanger, energy consumption

1. INTRODUCTION

Household refrigerators are one of the most important appliances in a residence. In Brazil, refrigerators are responsible for almost 23% of the energy consumption in the domestic sector (EPE, 2013). This means that an improvement on refrigeration systems can significantly reduce the energy consumption of the country. Therefore, industries are constantly being pushed to invest in research and to improve their products.

A refrigeration system is composed by four main components: compressor, condenser, evaporator and expansion device. The condenser is the component responsible for the heat rejection to the external environment. There are several types of condenser geometries applied to domestic refrigerators, depending on economic, social and thermodynamic factors. Currently, hot-wall condensers (also known as skin condensers) are being widely used around the world, especially in the Asian market, mainly due to aesthetic and cost reasons. In this type of heat exchanger, the condenser tubes are attached to the inner surface of the refrigerator outer steel sheet by an adhesive tape (see Figure 1), so that the external walls act as fins and enhance the heat transfer to the ambient. One of the main disadvantages of hot-wall condensers is that the increased wall temperature can lead to a higher thermal load over the refrigerated compartments. Also, the manufacturing process is a key variable on the heat exchanger performance since the tubes must be well attached to the refrigerator wall in order to avoid any increase in the thermal resistance.

A few reports were found on this topic in the open literature. Rebora and Tagliafico (1998) carried out a numerical finite element analysis on the simultaneous use of skin condensers and skin evaporators in chest freezers and proposed some design recommendations. However, the only experimental validation concerned the temperature profile inside the insulation foam. Also, it was found that the evaporator behavior is independent of that of the condenser. Bansal and Chin (2002) developed and validated a simulation model for hot-wall condensers used in domestic refrigerators. The authors disregarded the effect of the aluminum adhesive tape and considered that all the heat was rejected through the contact between the tube and the outer sheet, which was modeled as a fin. While recognizing that part of the heat is released to the refrigerated compartments, the mathematical model neglected any heat infiltration. The model predictions were compared to a set of in-house experimental data and deviations within a ±10% error band were observed. Gupta and Gopal (2008) carried out some modifications in the model proposed by Bansal and Chin (2002), including the effect of the heat transfer through the aluminum tape, which was treated as a fin. The model was validated against Bansal and Chin (2002) experimental data and a better agreement was observed, with deviations within ±2% error band. Colombo *et al*. (2016) developed a mathematical model that took into account both the heat transfer to the ambient and to the refrigerated compartments. The model predictions for the condenser heat transfer rate were compared to a set of in-house experimental data with deviations within a ±2% error band. The authors pointed out that, for a specific refrigerator tested on specific test conditions, on average 68% of the condenser heat released rate is transferred to the ambient while the remaining 32% is transferred to the refrigerated compartments.

It can be seen that in all the previous works only one hot-wall condenser geometry was tested to validate the mathematical models. Also, the models were only used to predict the condenser heat transfer rate and not the system energy consumption. In this context, the aim of the present study was to extend the analysis of hot-wall condensers applied to household refrigeration systems. To this end, seven refrigerator samples of the same model were manufactured with distinct condenser designs. Furthermore, an in-house mathematical model for hot-wall condensers (Colombo *et al.*, 2016) was coupled to an in-house system simulation tool (Hermes *et al.*, 2009) to predict the samples energy consumption. Finally, a parametric analysis was carried out in order to evaluate the pros and cons of using hot-wall condensers in household refrigerators.

2. EXPERIMENTAL WORK

Seven bottom-mount refrigerators of the same model were manufactured with distinct designs of hot-wall condensers. The refrigeration system is equipped with a single speed reciprocating compressor charged with isobutane. The evaporator is a finned-tube type and the capillary tube is placed in contact with the suction line in a counter-flow arrangement. Several condensers geometries were manufactured varying the following parameters: (i) adhesive tape material (aluminum or polyethylene), (ii) tube outer diameter (4 or 4.76 mm), (iii) condenser total length (10 or 11.5 m) and (iv) lay-out. The geometric parameters of each condenser are summarized in Table 1.

A sketch of the condensers circuitry is presented in Figure 2. They are all identical in the top wall and have 5 passes of tubes symmetrically distributed in the side walls. The maximum and minimum height of each condenser (H_{max} and H_{min}, respectively) vary to match the desired condenser positioning and its total length. The refrigerated compartments are divided by the freezer height, H_{fz}, which is equal to 0.76 m. Products 5 to 7 have H_{min} higher than H_{fz}, which means that the condenser is positioned at the fresh-food region only. This is an attempt to reduce the heat infiltration rate since the temperature gradient is lower between the condenser tubes and the fresh-food compartment. On the other hand, H_{min} is lower than H_{fz} in products 1 to 4, so that the condenser is distributed in both compartments. The refrigerator general dimensions and the components thermal properties are listed in Table 2.

Table 1: Geometric parameters of the condensers

Type	Adhesive tape	Tube diameter mm	l_c m	H_{max} m	H_{min} m
1	Al	4.00	10.0	1.13	0.53
2	Pe	4.76	10.0	1.13	0.53
3	Pe	4.00	11.5	1.24	0.42
4	Al	4.76	11.5	1.24	0.42
5	Pe	4.00	10.0	1.58	0.98
6	Al	4.00	11.5	1.66	0.85
7	Pe	4.76	11.5	1.66	0.85

Table 2: Refrigerators general dimensions

Outer dimensions	Height	1.8 m
	Depth	0.8 m
	Width	0.7 m
Outer sheet properties	Material	Steel
	Thickness	0.5 mm
	Conductivity	50 W/(m·K)
Thermal insulation properties	Material	Polyurethane
	FF thickness	55 mm
	FZ thickness	73 mm
	Conductivity	0.0214 W/(m·K)

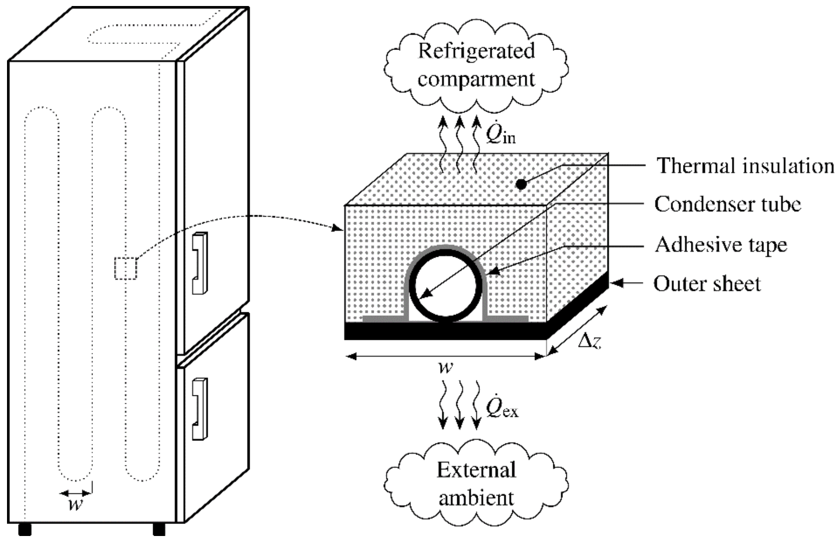

Figure 1: Condenser discretization

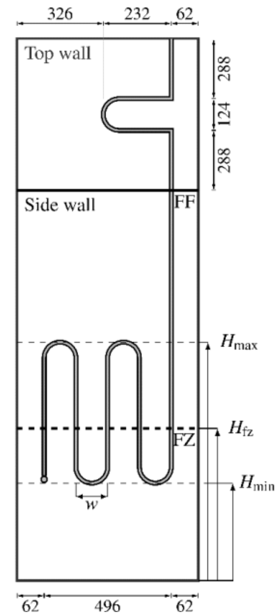

Figure 2: Sketch of the condensers circuitry

The experimental work consisted of manufacturing, instrumenting and testing the refrigerators. All samples were submitted to cyclic energy consumption tests. Due to inherent difficulties in carrying out time-consuming standardized ISO 62552 (2015) energy consumption tests (e.g. tylose packages, +24h operation), a simplified methodology was adopted. The refrigerators were placed inside a climatic controlled chamber kept at 32°C and turned on in a thermostat-guided cyclic operation. The system overall energy consumption was calculated as the time-integrated system power during 5h of cyclic operation, always considering complete on-off cycles. The refrigerated compartment temperatures were given by the arithmetic average of the readings of 3 thermocouples placed at the geometric center of each compartment shelf. Each refrigerator was submitted to two energy consumption tests, one keeping the temperature of the compartments above and another below reference values, which are -18°C for the freezer compartment and 5°C for the fresh-food compartment.

3. MATHEMATICAL MODELLING

The numerical approach consisted in the implementation of the in-house steady state simulation tool introduced by Hermes *et al.* (2009), in which the refrigeration system is divided into the following sub-models: compressor, evaporator, condenser and capillary tube-suction line heat exchanger. A few modifications needed to be made in the condenser model and also on the thermal load calculation. The condenser was simulated by another in-house mathematical model for hot-wall condensers (Colombo *et al.*, 2016), which is capable of predicting the condenser heat transfer rate by taking into account the rates exchanged with the external ambient and the refrigerated compartments. The modelling approach consists in the condenser discretization into small elemental units of length Δz. As shown in Figure 1, each element consists of a portion of the condenser tube, the outer sheet, the adhesive tape and the thermal insulation.

The heat transfer rate on the elemental unit is calculated based on the temperature profile in the adhesive tape and the outer sheet, which were both treated as independent one-dimensional fins (see Figure 3). Both the outer sheet and adhesive tape fins were divided into sections as they exchange heat with different mediums along their extension (Figure 4). The first section of the adhesive tape is the region glued to the condenser tube. Due to the inherent variability of the manufacturing process, it was assumed that the tape is attached to half of the tube perimeter and follows a 90º angle with the outer sheet. Thus, section II is the region in contact with the air cavity and section III represents the region attached to the outer sheet. Section I of the outer sheet represents the contact area with the condenser tube. It was assumed a very small contact of 1 mm. Section II is the region in contact with the air cavity.

Section III represents the region in contact with the aluminum tape and section IV the region in contact with the thermal insulation.

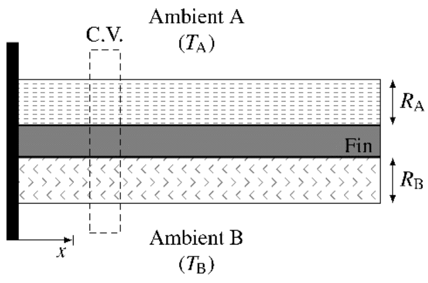

Figure 3: Independent one-dimensional fin

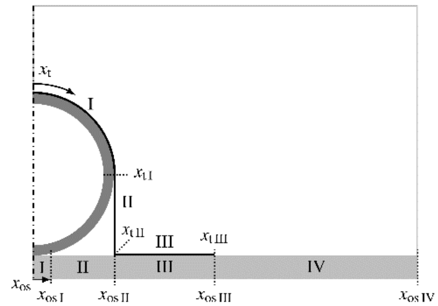

Figure 4: Independent one-dimensional fin

The heat transfer rate on each section of the fins is calculated through energy balances and by determining the respective temperature profile. Then, the heat released to the external ambient, Q_{ex}, is given by summing the heat transfer rates on the external surface of the outer sheet (bottom parts of sections I, II, III and IV). On the other hand, heat released to the refrigerated compartments, Q_{in}, is given by summing the upper parts of the adhesive tape of sections I, II and III and the upper part of the outer sheet section IV.

Calculating the thermal load over the refrigerated compartments is mandatory since it directly affects the compressor run-time ratio. As already mentioned, the thermal load model presented by Hermes *et al.* (2009) needed to be modified due to the presence of the hot-wall condensers. In order to account for the thermal load from the environment, the modelling strategy consisted in pondering the original global thermal conductance of each compartment (UA_{ff} and UA_{fz}) in the area, discounting the area of the outer sheet that contains condenser tubes. Therefore, new conductances, $UA_{ff,nc}$ and $UA_{fz,nc}$, were found for the refrigerator walls without condenser tubes. Thus, the total thermal load was given by summing the portions from the environment, the condenser, Q_{in}, and also the evaporator fan, Q_{fan}:

$$Q_t = UA_{ff,nc}\left(T_{ex} - T_{ff}\right) + UA_{fz,nc}\left(T_{ex} - T_{fz}\right) + Q_{in} + Q_{fan} \tag{1}$$

where the parameters $UA_{ff,nc}$ and $UA_{fz,nc}$ are calculated by:

$$UA_{ff,nc} = UA_{ff} - k_{pu} l_{c,ff} w \left(t_{pu,ff}\right)^{-1} \tag{2}$$
$$UA_{fz,nc} = UA_{fz} - k_{pu} l_{c,fz} w \left(t_{pu,fz}\right)^{-1} \tag{3}$$

being $l_{c,ff}$ and $l_{c,fz}$ respectively the length of condenser tubes in the fresh-food and freezer compartments, w the elemental unit width in the condenser discretization, k_{pu} the polyurethane thermal conductivity and $t_{pu,ff}$ and $t_{pu,fz}$ respectively the thermal insulation thickness in the fresh-food and freezer compartments.

Assuming that both thermal load (Q_t) and cooling capacity (Q_e) are constant along a periodic on-off cycle, the compressor run-time ratio can be calculated as follows:

$$RTR \equiv t_{on}(t_{on}+t_{off})^{-1} \approx Q_t(Q_e)^{-1} \tag{4}$$

Finally, the energy consumption (EC), in kWh/month, can be calculated through the following equation:

$$EC \approx 0.72(W_k + W_{fan}) \tag{5}$$

where W_k is the compressor power and the constant 0.72 is a conversion factor from W to kWh/month.

4. RESULTS

The test results and the input data for the model are shown in Table 3. The model validation is shown in the next section, as well as a parametric analysis exploring the effect of the condenser design parameters on the refrigerator energy consumption.

Table 3: Experimental results

Test #	Sample	T_{ex} [°C]	T_{ff} [°C]	T_{fz} [°C]	*RTR* [-]	*EC* [kWh/month]
1	1	31.3	1.8	-19.8	65.7	55.00
2	1	31.4	6.1	-16.3	53.4	46.38
3	2	30.9	2.3	-19.1	51.5	48.84
4	2	31.2	5.6	-15.4	44.1	44.16
5	3	32.7	1.5	-18.6	55.1	51.05
6	3	32.9	5.0	-14.8	43.3	43.19
7	4	32.2	2.6	-18.4	51.7	49.89
8	4	32.4	6.1	-14.4	41.7	43.51
9	5	31.8	1.9	-19.5	58.7	52.89
10	5	31.9	5.5	-16.7	51.4	47.93
11	6	33.5	1.6	-19.3	57.0	53.05
12	6	32.9	5.9	-15.4	44.7	45.56
13	7	32.6	1.3	-19.0	60.1	58.95
14	7	32.8	4.3	-15.1	48.1	51.96

4.1 Model validation

Figures 5 and 6 compare the experimental results to the model predictions. A reasonable agreement was observed and the model was capable of predicting the refrigerator energy consumption and runtime ratio within a ±10% error band for all tests. In general, the model proved to be robust and was capable of capturing the experimental trends related to variations on the condensers design, such as the influence of the type of adhesive tape and the tubes positioning. The effect of the operational parameters, such as the compartments internal temperatures, could also be captured by the mathematical model.

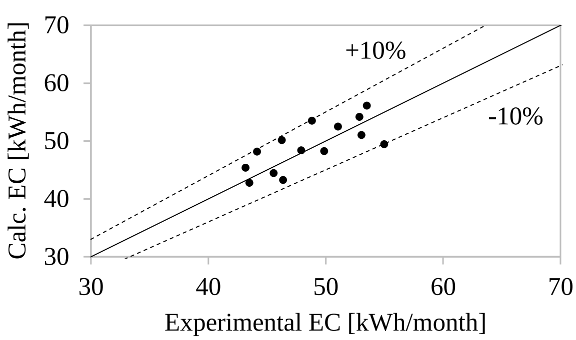

Figure 5: Validation of the energy consumption

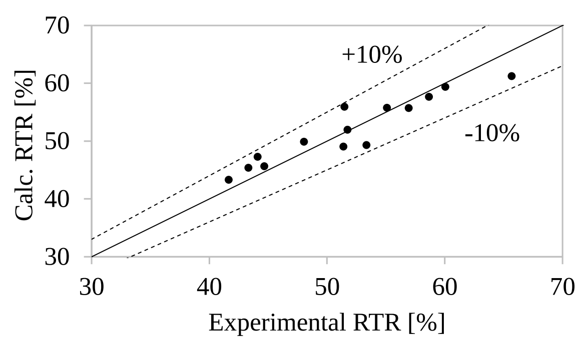

Figure 6: Validation of the run-time ratio

4.2 Mathematical analysis

As previously mentioned, the model is capable of calculating the thermal load over the refrigerator considering the contribution of the hot-wall condenser. A comparison was made between the thermal load calculated by the model for the refrigerator with hot-wall condensers and a hypothetic refrigerator, without heated walls. In this hypothetical case, the thermal load would be only due to the temperature difference between the external environment and the refrigerated compartments. This is the common case for the majority of refrigerators mounted with other condenser types, such as wire-on-tube or microchannel. The results are shown in Table 4. It can be seen that the presence of the hot-wall condenser increased, on average, 7.7% the thermal load over the refrigerator. This was expected since the heated walls increase the potential for heat infiltration. It must be pointed out here that usually the choice for skin condensers is made based on aesthetics and cost reasons, so, in order to overcome the increased thermal load, the thermal insulation should be improved or the condensing temperature reduced.

Table 4: Thermal load comparison

Test #	$\dot{Q}_{t,HW}$ [W]	$\dot{Q}_{t,hyp.}$ [W]	Increase [%]
1	77.7	73.0	6.4
2	70.7	65.7	7.7
3	76.4	71.2	7.3
4	71.0	65.5	8.5
5	80.3	75.3	6.7
6	72.6	67.2	8.1
7	78.1	72.8	7.2
8	72.1	66.5	8.4
9	79.0	73.8	7.1
10	73.3	67.7	8.2
11	81.3	76.1	6.8
12	74.0	68.3	8.3
13	81.8	76.0	7.6
14	76.1	69.9	8.9

Next, a sensitivity analysis was carried out in order to evaluate the impact of some of the condenser design parameters on the refrigerator energy consumption. To this end, the geometry of sample 1 was selected, and the following operating parameters were considered: ambient temperature of 32°C, fresh-food temperature of 5°C, freezer temperature of -18°C and superheating degree of 2°C at the evaporator outlet.

The refrigerator outer sheet is responsible for diffusing the heat released by the condenser tubes, mainly to the external environment. Therefore, it's expected that its design parameters are important to the system performance. Figure 7 shows the influence of the outer sheet thermal conductivity in the refrigerator energy consumption. It can be verified that its thermal conductivity is crucial. The original steel outer sheet proved to be suitable for the application (k = 50 W/m·K). Materials with a better thermal conductivity could improve the system performance (200 W/m·K reduces the energy consumption in 2.5%), but would be much more expensive. On the other hand, polymeric materials shall not be used in the outer sheet of hot-wall condensers, because their lower thermal conductivity strongly reduce the condenser heat transfer rate, which leads to an increase in the condensing pressure and consequently in system energy consumption.

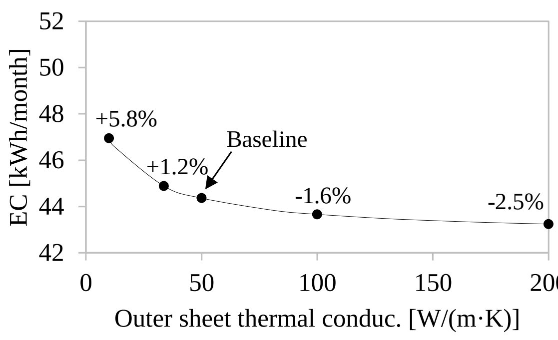

Figure 7: Energy consumption as a function of the outer sheet thermal conductivity

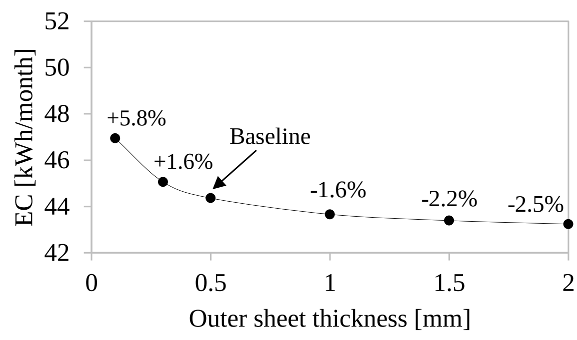

Figure 8: Energy consumption as a function of the outer sheet thickness

In line with the previous analysis, Figure 8 shows the influence of the outer sheet thickness. As the outer sheet behaves as a fin, an increase in its thickness leads to an increase in the fin cross-sectional area, which enhances the heat diffusion process. The original thickness of the outer sheet is 0.5 mm, which is also suitable for this kind of application. Thicker plates of 2 mm, for instance, could reduce up to 2.5% the energy consumption, but would be much more expensive and would also complicate the manufacturing process. Thinner plates could be cheaper, however, they can strongly increase the energy consumption and may also hinder the refrigerator physical structure.

Bansal and Chin (2002) stated that almost 70% of the heat dissipated by the outer sheet is related to radiation. Therefore, the outer sheet emissivity is very important to the system performance. As can be seen in Figure 9, the energy consumption can increase by up to 13.2% if the emissivity drops from 0.95 to 0.2. However, this should not be a big concern since the materials commonly used in the outer sheet have an emissivity higher than 0.8.

The adhesive tape also plays an important role on the heat transfer process. Figure 10 shows the system energy consumption as a function of the adhesive tape thermal conductivity, considering three different contact areas (1 mm, 2 mm and 3 mm). It can be noted that for a small contact area between the condenser tube and the outer sheet (1 mm), the use of a polyethylene (k = 0.5 W/m·K) instead of an aluminum tape (k = 170 W/m·K) would increase the energy consumption by 8%. On the other hand, for a larger contact area (3 mm), the increase in energy consumption is only of 2%. The heat released from the refrigerant must be dissipated through the adhesive tape and/or the contact area between tube and outer sheet. Therefore, there is a tradeoff between the adhesive tape thermal conductivity and the contact area. It is possible then to manufacture a variety of condenser designs with the same performance by combining these parameters. As the condenser tube is commonly circular (i.e very small contact area), aluminum tapes are generally used to provide a better performance. However, the aluminum tape could be replaced by a cheaper polymeric tape as long as an alternative to increase the contact area is used, such as a D-profile tube. Furthermore, the tubes must be well attached to the outer sheet in order to avoid an increase in the contact thermal resistance.

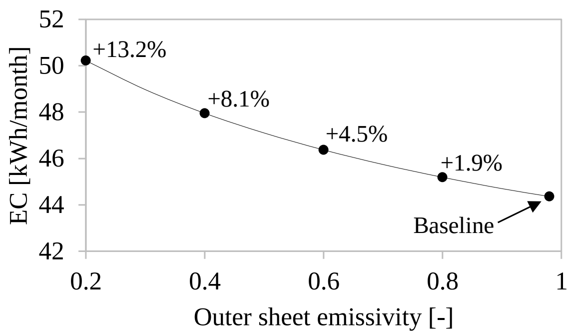

Figure 9: Energy consumption as a function of the outer sheet emissivity

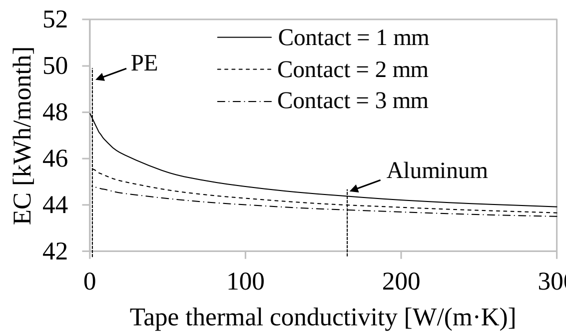

Figure 10: Energy consumption as a function of the tape thermal conductivity

Regarding the outer sheet, it is important to use the largest possible area to attach the condenser tubes, as long as the surfaces are metallic (higher thermal conductivity). However, it is important to optimize the thickness of the thermal insulation of the refrigerated compartments in order to avoid an increase in the thermal load. It must be considered that the thermal gradient is higher in the freezer compartment, and therefore the thermal insulation must be properly compensated. Keeping this in mind, another analysis was carried out to evaluate this effect. Two different condenser designs were simulated, case A and B, as shown in Figures 11 and 12. Both have similar design characteristics (same tube type, adhesive tape and tube length), but in configuration A the condenser was distributed in both fresh-food and freezer compartments, while in configuration B the condenser was positioned only in the fresh-food (lower thermal gradient). Then, the thickness of the thermal insulation of the freezer compartment in configuration A was varied, always keeping the thickness of the fresh-food walls constant and equal to 54 mm. As can be seen in Figure 13, the lower the thickness of the freezer insulation, the higher the refrigerator energy consumption due to the increased thermal load. It can also be noted that there is a point in which configurations A and B present nearly the same energy consumption. This point is close to 73 mm, which is the original thickness of the freezer insulation. Therefore, for this specific case, the walls insulations were already reasonably balanced and the condenser tubes could be positioned anywhere in the refrigerator outer sheet without penalizing the system overall energy consumption. However, if the insulation was thinner than this, the positioning of the condenser tubes in the region of the freezer compartment would decrease the system performance due to the increased thermal load.

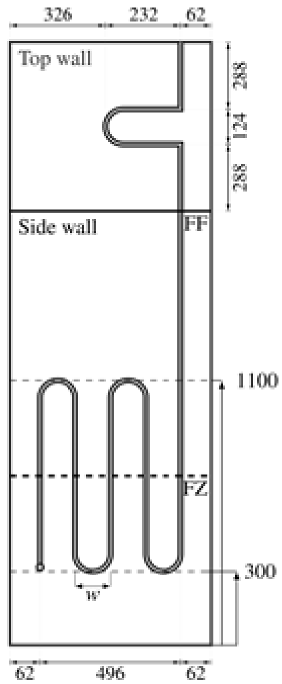

Figure 11: Configuration A

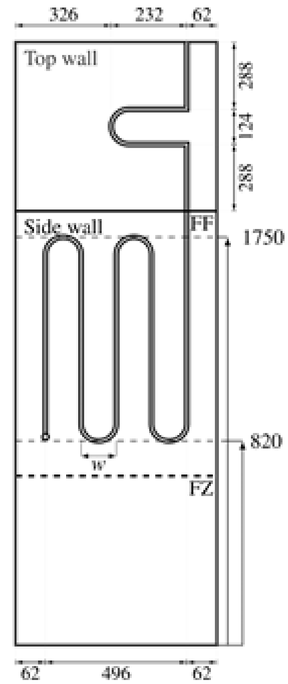

Figure 12: Configuration B

An additional analysis was also put forward with configuration A (see Figure 11) in order to check the effect of the tube pitch on the system performance. The simulation results are shown in Figure 14. It can be seen that there is a tube pitch that minimizes the energy consumption. For this specific geometry, the minimum energy consumption is achieved when the number of tube passes is close to 5, which is the original value. Initially, the increase in the number of tubes causes an increase in the average temperature of the outer sheet, resulting in a higher heat transfer rate to the external environment and a better performance of the system. However, as the number of tube passes further increases, the tubes become very close to each other, which gradually reduces the fin width, responsible for the heat diffusion. Consequently, the condensing temperature starts to increase leading to a higher energy consumption.

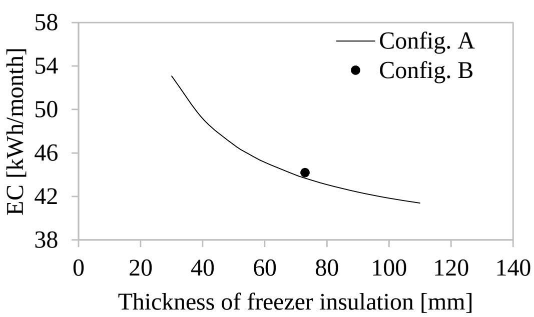

Figure 13: Energy consumption as a function of the thickness of the freezer insulation

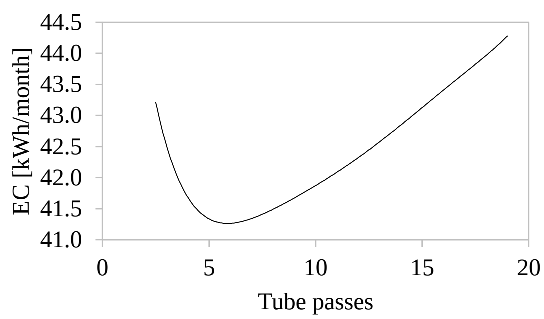

Figure 14: Energy consumption as a function of the number of tube passes

5. CONCLUSIONS

This work addressed an investigation on the pros and cons of the use of hot-wall condensers in household refrigerators, based on both numerical and experimental approaches. Seven different configurations of hot-wall condensers were manufactured in a specific household refrigerator model. Cyclic energy consumption tests were carried out and the results were compared to the mathematical model predictions. A good agreement was observed, with deviations within a ±10% error band. It was noted that the outer sheet thermal conductivity, thickness and emissivity are very important to the refrigerator performance, and that polymeric materials shall not be used. It was verified that the adhesive tape plays a very important role on the condenser performance, and that a trade-off must be considered when selecting the tape. Either the tape must have a good thermal conductivity, such as aluminum tapes, or the contact area between the condenser tube and the outer sheet must be increased in order to select a cheaper adhesive tape with a lower thermal conductivity, such as polyethylene. It was noticed that the largest area of the outer sheet shall be used to place the condenser tubes, however the thickness of the thermal insulation of both refrigerated compartments must be properly balanced in order to avoid an increase in the thermal load. It also could be seen that for a specific condenser geometry, there is a tube pitch that minimizes the system energy consumption. Finally, it must be pointed out that these analyses are always product-dependent. Thus, any specific new refrigerator mounted with a hot-wall condenser must be simulated, and the mathematical models proposed herein become a valuable tool for this task. Despite the relevance of the proposed recommendations, the manufacturing feasibility and costs must be carefully analyzed before any implementation.

NOMENCLATURE

Δz	elemental unit length	(m)
H	height	(m)
k	thermal conductivity	(W/(m·K))
l	length	(m)
Q	heat transfer rate	(W)
RTR	compressor run-time ratio	(-)
t	thickness, time	(m, s)
T	temperature	(°C)
UA	global thermal conductance	(W/K)
w	elemental unit width	(m)
W	power	(W)

Subscripts

c	condenser
e	evaporator
ex	external
fan	evaporator fan
ff	fresh-food compartment
fz	freezer compartment
HW	hot-wall
hyp	hypothetical
in	internal
k	compressor
max	maximum
min	minimum
nc	with no condenser tubes
os	outer sheet
PU	polyurethane
r	refrigerant
t	tape, thermal load

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