

MAPPING THE ENERGY CONSUMPTION OF HOUSEHOLD REFRIGERATORS BY VARYING THE REFRIGERANT CHARGE AND THE EXPANSION RESTRICTION

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Abstract. *In this work the thermodynamic behavior of a household refrigerator was experimentally studied by simultaneously varying the refrigerant charge and the expansion restriction. A special charging device was designed and constructed for this purpose comprised of a cylinder, a load cell and two solenoid valves. In addition, the original capillary tube was replaced with a larger-diameter capillary tube and installed in series with a metering valve. The expansion restriction was varied by adjusting the capillary tube-metering valve pair to settings higher and lower than that of the original system. A total of 95 measurements were recorded with different combinations of refrigerant charge and expansion restriction. The energy consumption was measured according to a steady-state methodology which substantially reduced the testing time. A minimum energy consumption region comprised of several combinations of refrigerant charge and expansion restriction was clearly identified. It was also observed that an improper combination of capillary tube and expansion restriction may increase the energy consumption by up to 30%.*

Keywords: household refrigerator, expansion device, capillary tube, refrigerant charge, energy consumption

1. INTRODUCTION

In Brazil, household refrigeration is responsible for 30% of the average household energy consumption and this accounts for approximately 6% of the national electricity consumption. This relatively high fraction is due to the large number of units in operation – more than one billion – and also to the inherent low thermodynamic efficiencies of such products.

It is well known that household refrigerators have highest efficiency when operating with certain combinations of expansion restriction and refrigeration charge (Gonçalves and Melo, 2004, Vjacheslav et al., 2001). Such parameters are normally determined at a late stage in the development process after all other system components have been designed or selected. The common procedure is to use a trial-and-error approach, with the energy consumption measured according to the ISO 15502 (2007) standard. However, such a procedure is time consuming – an energy consumption test may last more than 48 hours – and does not always produce the results required.

Dmitriyev and Pisarenko (1982) suggested a method to calculate the optimum refrigerant charge for household refrigerators, based only on the evaporator and condenser internal volumes. The authors concluded that regardless of the ambient temperature, there is a particular refrigerant charge that maximizes the system coefficient of performance (COP).

Jakobsen (1995) simulated the effect of varying the expansion restriction and the refrigerant charge on the COP of a particular household refrigerator. It was found that the COP was more sensitive to a deficit of charge and to an excess of expansion restriction than the opposite scenario.

Vjacheslav et al. (2001) presented a model to estimate the optimum refrigerant charge for refrigeration systems. The authors modeled each component separately, exploring the effect of refrigerant charge on each of them and consequently on the system performance. Similarly to Dmitriyev and Pisarenko (1982), they found that the system performance is strongly dependent on the refrigerant charge.

Björk and Palm (2006) conducted a series of experiments with a household refrigerator working under cyclic conditions in an attempt to identify the ideal combination of expansion restriction and refrigerant charge. Based on a database of 600 data points they concluded that the energy consumption had a minimum for certain combinations of expansion restriction and refrigerant charge and that this minimum was flat and wide.

This work follows very closely the research of Björk and Palm (2006), but with the following differences: i) top-mount refrigerator with a no-frost evaporator and without a low side accumulator; ii) energy consumption measured under steady-state conditions; and iii) expansion restrictions much larger than the original value.

2. EXPERIMENTAL APPARATUS

The experimental apparatus is basically comprised of a charging device connected to the suction and discharge lines, through which refrigerant charge is added to or removed from the system in a controllable way. In addition, the original internal heat exchanger, also known as the capillary tube-suction line heat exchanger, was modified by replacing the original capillary tube with another of the same length but with a larger internal diameter. A metering valve was also installed in series with the capillary tube in order to vary the expansion restriction. This arrangement, which provides mass flow rates higher and lower than that provided by the original system, is shown in Figure 1.

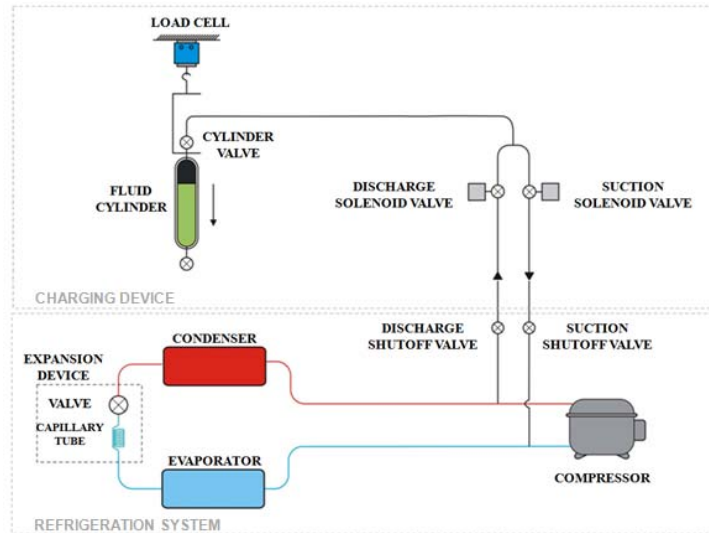


Figure 1. Schematic of the experimental apparatus

The experiments were carried out with a top-mount refrigerator, originally comprised of a reciprocating compressor, a natural draft wire and tube condenser, a tube-fin forced-draft no-frost evaporator, a concentric internal heat exchanger (2.05 m length) and a capillary tube (3.32 m length, 0.70 mm I.D.).

The charging device consisted essentially of a cylinder suspended in a load cell from which refrigerant enters and exits under the control of two solenoid valves connected to the discharge and suction lines. The refrigerant pressure inside the cylinder, which corresponds to the saturation pressure at ambient temperature, always lies between the suction and discharge pressures. Thus, refrigerant exits to feed the system through the opening of the solenoid valve connected to the suction line and returns from the system through the opening of the solenoid valve connected to the discharge line.

The expansion restriction was varied by a metering valve installed in series with a capillary tube. To this end, the original capillary tube was replaced with a 0.83 mm I.D. capillary tube. This allowed higher or lower restrictions with respect to that imposed by the original capillary tube. The opening of the metering valve was controlled by a step motor in steps of 1.8 °.

The air temperatures in the refrigerated compartments and the room air temperature were measured by T-type thermocouples embedded in standard-sized copper cylinders, with a maximum uncertainty of $\pm 0.1^\circ\text{C}$. The refrigerant temperatures along with the refrigeration loop were also measured by T-type thermocouples, attached to the pipe wall with plastic straps and dielectric adhesive tape. Thermal paste was used to decrease the contact resistance between the thermocouples and the pipe wall. The suction and discharge pressures were measured by absolute pressure transducers, with a maximum uncertainty of ± 0.04 bar. The refrigerant mass flow rate was measured by a Coriolis-type mass flow meter installed in the discharge line, with a maximum uncertainty of ± 0.06 kg/h. The power of the compressor, evaporator fan and the electric heaters was measured by a power transducer, with a maximum uncertainty of ± 1.5 W. The refrigerant mass contained in the charge cylinder was measured by a 10 N maximum capacity load cell, with a maximum uncertainty of ± 0.3 g. The transducer readings were acquired by a computerized data acquisition system. A computer program to manage both the charging device and the metering valve was developed using the LabVIEW ® platform.

All experiments were carried out in a climate-controlled chamber, constructed in accordance with the recommendations of the ISO8561 (1995) standard. Such a chamber is capable of maintaining the air temperature at any value ranging from 18 ° C to 50 ° C, with a maximum uncertainty of ± 0.2 ° C. The relative humidity is also maintained

at any value within the range of 30% to 95%, with a maximum uncertainty of $\pm 1\%$. The air velocity at any point inside the chamber does not exceed 0.25 m/s, as required by the standard.

3. EXPERIMENTAL PROCEDURE

The experimental procedure consisted of measuring the energy consumption for different pairs of expansion restriction and refrigerant charge. The experiments started with the metering valve fully open and with a refrigerant charge large enough to reach the reference temperatures of 5°C and -18°C in the fresh food and freezer compartments, respectively. Next the metering valve was gradually closed until the reference temperatures be could no longer attained. The valve was then reset to the fully open position, a small amount of refrigerant – around 3g – was added to the system and the test was restarted. The experiments ended when the refrigerant charge reached a limit, above which liquid droplets appear in the evaporator outlet and are ingested into the compressor cylinder. The valve position increments were not constant, since the refrigerant mass flow rate is almost unaffected by the valve position at the beginning of the closing cycle and strongly affected at the end.

3.1. Measurement of the energy consumption

The cooling capacity of a household refrigerator is always higher than that needed to match the system thermal load, mainly to decrease the pull-down time – the length of time needed to reach the reference temperatures after the compressor start-up. The system then normally works under on-off cyclic conditions after the initial transient period due to the excess of cooling capacity provided by the compressor.

The energy consumption of any household refrigerator sold in the Brazilian market must be measured according to the recommendations of the ISO 15502 (2007) standard. According to this standard two tests are performed, one above and another below the reference temperatures, each of them lasting at least 24 hours after steady-state periodic conditions are reached. In order to avoid such a time consuming procedure an alternative steady-state methodology (Hermes et al., 2012) was adopted in this work. In this procedure the excess of cooling capacity was balanced by PID-driven electric heaters installed in the refrigerated compartments, while the thermostat was turned off and the damper was maintained at the maximum position.

According to this procedure, the cabinet thermal load, \dot{Q}_t , should firstly be calculated using the fresh food and freezer temperatures, T_{ff} , T_{fz} , the room temperature, T_a , and the overall thermal conductances of each compartment, UA_{ff} , UA_{fz} . The thermal conductances are obtained beforehand from the so-called reverse heat leakage tests (Sin and Ha, 2011). Under steady-state conditions the product thermal load can then be expressed simply as:

$$\dot{Q}_t = UA_{ff}(T_a - T_{ff}) + UA_{fz}(T_a - T_{fz}) + \dot{W}_{fan} \quad (1)$$

where \dot{W}_{fan} is the evaporator fan power.

The cooling capacity, \dot{Q}_e , is then obtained by taking into account the thermal load and also the electrical power released by the heaters, \dot{W}_{ff} , \dot{W}_{fz} :

$$\dot{Q}_e = \dot{Q}_t + \dot{W}_{ff} + \dot{W}_{fz} \quad (2)$$

In addition, when the refrigerator is running according to an on-off cycling pattern, all energy transferred to the refrigerated compartments during a cycle must be removed by the cooling system through the compressor action. The following energy balance over a whole cycle then applies (Hermes et al. 2009):

$$\bar{Q}_e \cdot t_{on} = \bar{Q}_t \cdot t_{on} + (\bar{Q}_t - \bar{W}_{fan}) \cdot t_{off} \quad (3)$$

where t_{on} and t_{off} are the times of compressor on and off, respectively.

Rearranging the previous equation for the compressor duty-cycle, τ , yields:

$$\tau \equiv \frac{t_{on}}{t_{on} + t_{off}} \cong \frac{\bar{Q}_t - \bar{W}_{fan}}{\bar{Q}_e - \bar{W}_{fan}} \quad (4)$$

The energy consumption (EC), in turn, can be obtained by multiplying the compressor duty-cycle by the system average power consumption ($\bar{W}_{sist} = \bar{W}_{comp} + \bar{W}_{fan}$), as follows:

$$EC = 0.72 \tau \bar{W}_{sist} \quad (5)$$

where the coefficient of 0.72 stands for the conversion factor from [W] to [kWh/month].

Finally, the coefficient of performance can be calculated as:

$$COP = \frac{\dot{Q}_e}{\dot{W}_{sist}} = \frac{\bar{Q}_t + \bar{W}_{fan}(\tau - 1)}{EC} \quad (6)$$

4. RESULTS

The experiments were carried out according to the previously described methodology. In total, 95 experimental data points were gathered. The minimum and maximum refrigerant charges were 36.7 g and 64.7 g, respectively. The metering valve opening was varied between the fully open position (9.75 turns) and the opening corresponding to 0.425 turns.

4.1. Effect of the expansion restriction

The expansion restriction was varied through the opening of the metering valve. The valve opening was related to the remaining number of turns required for total closure. The fully-open position corresponds to the 9.75 opening, while the fully-closed position corresponds to the zero opening. As the refrigerant mass flow rate does not vary linearly with the valve opening, the number of turns was correlated with a position scale, varying between 0 and 11. The opening-position relationship, obtained applying variations in the mass flow rate of around 0.04 kg/h, is shown in Table 1.

Table 1. Metering valve opening vs. position

Opening [turns]	9.75	1.750	1.000	0.750	0.650	0.600	0.550	0.525	0.500	0.475	0.450	0.425
Position [-]	11	10	9	8	7	6	5	4	3	2	1	0

Figure 2a shows the compression work and the refrigerant mass flow rate as a function of valve position when the refrigerator was charged with 46.9 g of HC-600a. It is shown that the compression work increases while the refrigerant mass flow rate decreases with increasing expansion restriction. This is so because with greater restriction the refrigerant tends to build-up in the condenser, which starves the evaporator. The evaporation pressure then decreases, the evaporator superheating increases, the specific volume at the inlet of the compressor increases and consequently the mass flow rate decreases. Moreover, the system pressure ratio increases, thereby increasing the compression work and decreasing the volumetric efficiency and mass flow rate.

The compression power, \dot{W}_k , results from the compression work multiplied by the mass flow rate. Due to the contrasting response of these two parameters to the restriction variation, the compression power passes through a maximum as illustrated in Figure 2b. The cooling capacity, \dot{Q}_e , and the coefficient of performance (COP) are also shown in Figure 2b as a function of the expansion restriction.

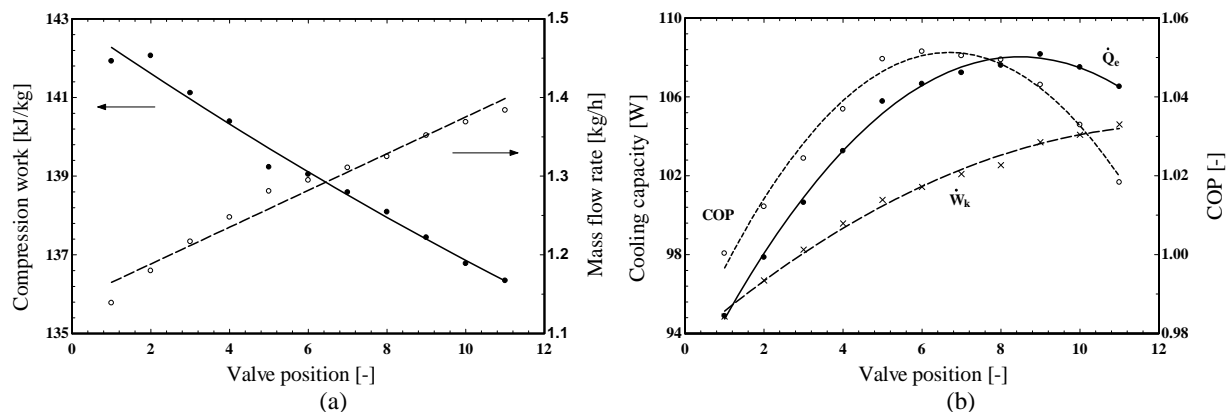


Figure 2. (a) Compression work and mass flow rate vs. valve position and (b) cooling capacity, compression power and COP vs. valve position.

4.2. Effect of the refrigerant charge

The refrigerant charge was varied using a specially designed and constructed charging device as previously mentioned. Figure 3a shows the mass flow rate as a function of refrigerant charge and valve position (VP). As more refrigerant is added to the system, more refrigerant is accumulated in the heat exchangers, thereby increasing the condensing and evaporating pressures and decreasing the evaporator superheating. The combined effect of evaporator superheating and evaporating pressure decreases the specific volume at the compressor inlet, thereby increasing the refrigerant mass flow rate. It should be noted that this behavior is more pronounced at lower restrictions when the evaporator overflows and liquid refrigerant is carried to the suction line, reducing its temperature and the specific volume at the inlet of the compressor, as shown in Figure 3b.

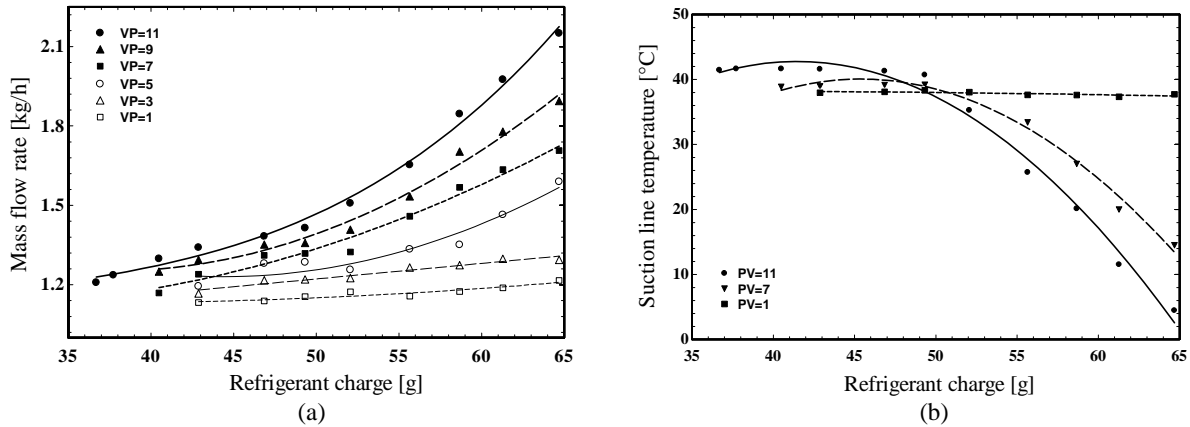


Figure 3. (a) Mass flow rate and (b) suction line temperature vs. refrigerant charge.

Figure 4a shows the cooling capacity, compression power and coefficient of performance as a function of refrigerant charge at the maximum valve opening position (VP = 11). It can be seen that the minimum refrigerant charge at which the system is still able to reach the reference temperatures is 36.7 g. At this charge the cooling capacity is quite small since a considerable fraction of the evaporator is occupied by vapor refrigerant. As more refrigerant is added to the system the evaporator is more properly filled with refrigerant and consequently the cooling capacity increases. However, after a certain limit, the amount of refrigerant in the condenser also increases, thereby increasing the system pressure ratio and the compression power and decreasing the cooling capacity. In short, at a fixed restriction, as more refrigerant is gradually added to the system the coefficient of performance passes through a maximum, as shown in Figure 4a.

4.3. Combined effect of the refrigerant charge and expansion restriction

The cooling capacity is driven by the air temperature at the inlet of the evaporator and by the mean evaporator temperature. The latter varies with the evaporation temperature and superheating, while the former is held constant. Thus, it can be concluded that the lower the evaporation temperature and superheating the lower the evaporator mean temperature and the higher the cooling capacity. Figure 4b shows the mean evaporator temperature, T_{coil} , and the evaporation temperature, T_e , as a function of refrigerant charge. It can be observed that as more refrigerant is added to the system, liquid builds-up in the evaporator, reducing the superheating and increasing the cooling capacity up to a maximum point. From this point on the evaporator remains fully activated, but with a higher mean temperature, which reduces the cooling capacity.

Figure 5a shows the cooling capacity against the refrigerant charge and valve restriction. It is worth noting that at the lowest restriction (VP = 11) the maximum cooling capacity is reached with a charge of approximately 48 g. It can also be noted that the maximum point is shifted to the right, i.e., towards higher refrigerant charges, as the expansion restriction increases.

Figure 5b shows the valve position at the maximum cooling capacity against the refrigerant charge. It can be observed that the optimum restriction decreases continuously with increasing refrigerant charge. When the charge increases the refrigerant tends to accumulate in the evaporator, increasing the cooling capacity until the superheating reaches a minimum. Thereafter, an increase in the refrigerant charge increases the evaporation pressure and decreases the cooling capacity. Thus, when more refrigerant is added to the system the expansion restriction must be increased in order to reduce the evaporation pressure and keep the cooling capacity at its maximum value.

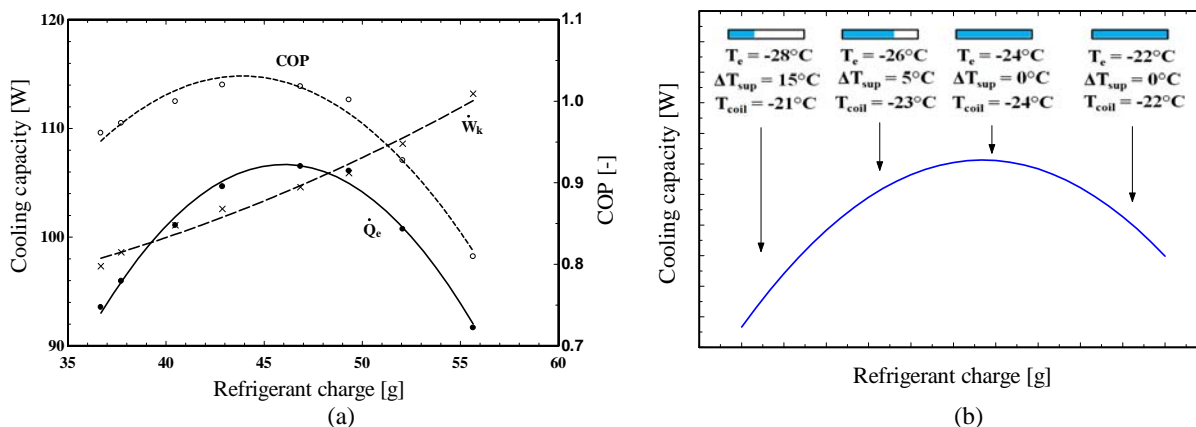


Figure 4. (a) Cooling capacity, compression power and COP vs. refrigerant charge and (b) mean evaporator temperature and cooling capacity vs. refrigerant charge

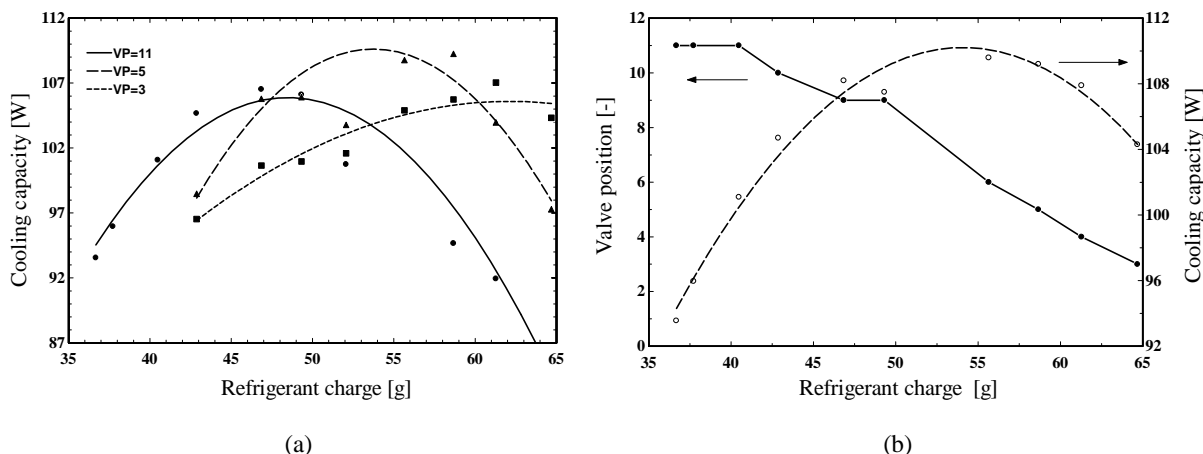


Figure 5. (a) Cooling capacity vs. refrigerant charge and (b) valve position at maximum cooling capacity vs. refrigerant charge.

Figure 6a shows the coefficient of performance for several refrigerant charges and expansion restrictions. As also shown in Figure 4a, the coefficient of performance passes through a maximum as more refrigerant is added to the system. As the expansion restriction increases, the maximum COP point is shifted to the right, i.e., towards higher refrigerant charges.

Figure 6b illustrates the valve position at the maximum COP against the refrigerant charge. It can be seen that the expansion restriction must be increased in order to keep the COP at its maximum value when more refrigerant is added to the system. This behavior is similar to that shown in Figure 5b. It can also be seen that the COP again passes through a maximum. This occurs because a higher restriction decreases the mass flow rate, which has two contrasting effects on the COP: (i) it decreases the compression ratio; and (ii) it decreases the cooling capacity. Until the maximum point, the effect of lowering the compression ratio prevails over the effect of the reducing the cooling capacity, and after this point the opposite behavior is observed.

Figure 7a shows the valve position at the maximum COP and maximum cooling capacity against the refrigerant charge. It is worth noting that the charge-restriction pairs which provide the maximum capacity and the maximum COP do not match. For a given refrigerant charge, the restriction that maximizes the COP is always less than that which maximizes the cooling capacity. Likewise, for the same restriction the charge that maximizes the COP is always less than that which maximizes the cooling capacity. This is because the refrigerant charge affects both the cooling capacity and the compression power, parameters which have contrasting effects on the coefficient of performance.

Figure 7b shows an energy consumption contour map plotted using the valve position and the refrigerant charge as the graph coordinates. This map is composed of contour lines – lines which delimit regions of equal energy consumption – obtained by the Multi-quadric Radial Basis Function Interpolation Method (Wright, 2003).

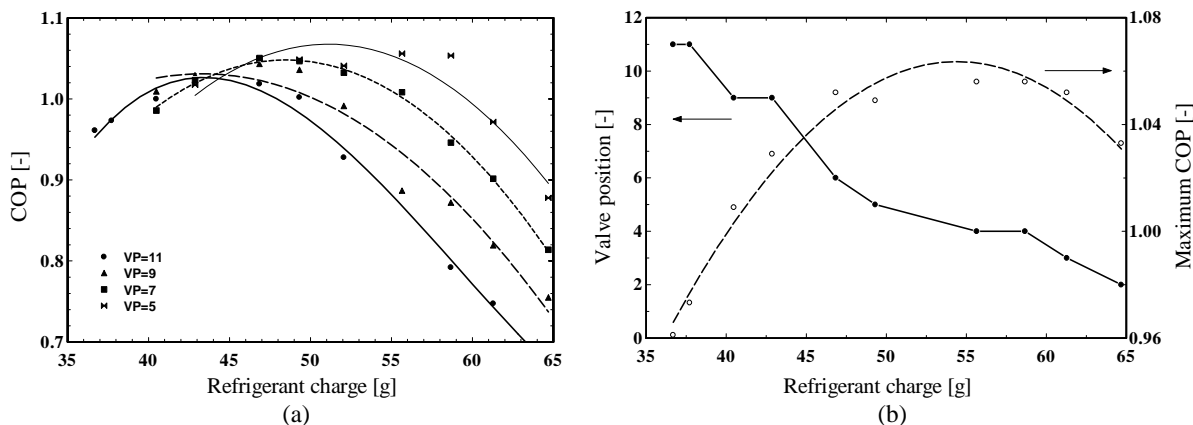


Figure 6. (a) COP vs. refrigerant charge and (b) valve position at maximum COP vs. refrigerant charge.

Each contour line, represented by a color scale ranging from blue to red, corresponds to a range of energy consumption values. The blue regions, which are predominant in the central part of the figure, correspond to regions of low energy consumption. The successive contour lines correspond to regions of higher energy consumption which increase with the distance from the central region.

It is worth noting that the minimum energy consumption region (50-52 kWh / month) is quite wide. It can also be noted that at the lowest restrictions, between positions 4 to 11, the system accepts changes of up to 6 g in the refrigerant charge without any appreciable effect on energy consumption. It can also be seen that at the highest restrictions, between positions 0 and 4, there is a nearly linear relationship between the optimum restriction and the refrigerant charge. Such behavior is quite similar to that observed by Björk and Palm (2006), despite the dissimilarities of the refrigerator models.

Below position 4, any additional restriction causes an accumulation of refrigerant in the condenser and a lack of refrigerant in the evaporator, thereby increasing the superheating and decreasing the evaporation pressure and the system performance. This effect is compensated for by adding more refrigerant to the system. From another perspective, an increased refrigerant charge results in evaporator overflow and decreases the suction line temperature, with a negative effect on the system performance. To compensate for this effect the restriction of the expansion device is increased.

At the lowest restrictions the system is less sensitive to the refrigerant charge. Björk and Palm (2006) explained this behavior focusing on the low-side accumulator that would prevent both superheating and the suction line being too cold, both having negative effects on the system performance. However, it should be emphasized that the refrigerator used in this study does not have a low-side accumulator, and so another explanation is needed in this case.

The distinct behavior above and below position 4 can be explained by the refrigerant state at the entrance of the capillary tube. At higher restrictions there is a small fraction of liquid at the entrance of the capillary, and therefore this component effectively “governs” the refrigerant mass flow rate and the system performance. At lower restrictions, the liquid seal at the entrance of the capillary tube disappears and the mass flow rate and system performance are now “governed” by the refrigerant state at the inlet of the capillary tube, which oscillates intermittently, thus ensuring a self-regulating characteristic of the system (Boeng, 2012).

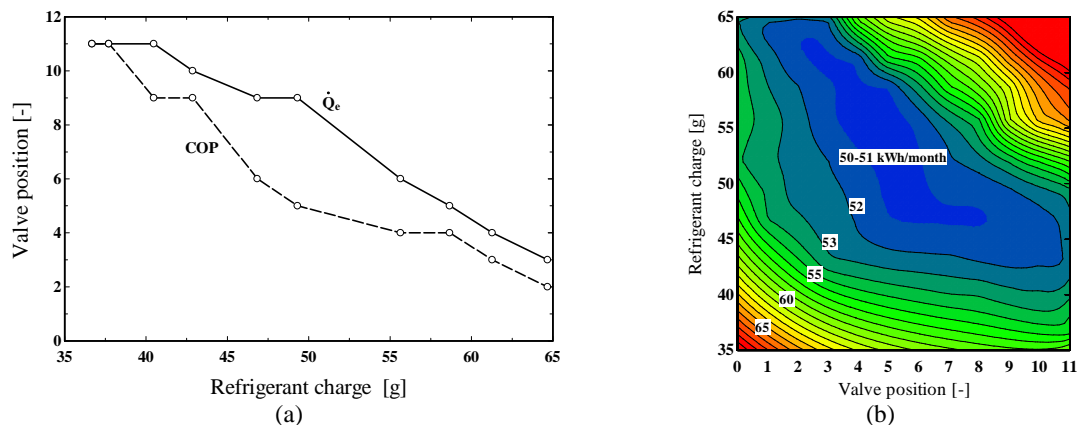


Figure 7 (a) Valve position at maximum COP and maximum cooling capacity vs. refrigerant charge and (b) energy consumption vs. refrigerant charge and valve position

5. CONCLUDING REMARKS

This research provided insights related to the household refrigeration sector. Some of them are listed below:

- The steady-state energy consumption method was found to be practical, reliable and much faster than that recommended in the ISO 15502 (2007) standard;
- A minimum energy consumption region, comprised of several combinations of refrigerant charge and expansion restriction, was found. It was also observed that an inappropriate combination of expansion restriction and refrigerant charge may increase the energy consumption by up to 30%;
- A deficit of charge or an excess of restriction increases the evaporator superheating. On the other hand, an excess of charge or a small restriction leads to evaporator overflow and decreases the suction line temperature. In both cases the cooling capacity and the energy consumption are penalized;
- The product energy consumption was completely mapped since the replacement of the capillary tube resulted in mass flow rates much higher than those supplied by the original system. This was a limitation of the work of Björk and Palm (2006), who were unable to substantially decrease the expansion restriction;
- Despite the dissimilarities between the refrigerator used in this study and that used by Björk and Palm (2006), there is clearly a similarity in the energy consumption maps, which suggests a common behavior for most household refrigerators;
- A methodology to find the optimum operation point in terms of refrigerant charge and expansion restriction, based on a small amount of experimental data, was also developed, but it is out of the scope of this paper.

6. REFERENCES

- Boeng, J., 2012, "A methodology to select the optimum pair capillary tube-refrigerant charge for household refrigerators", M.Sc. Dissertation, Federal University of Santa Catarina.
- Björk, E., Palm, B., 2006, "Performance of a domestic refrigerator under influence of varied expansion device capacity, refrigerant charge and ambient temperature". *International Journal of Refrigeration*. Vol. 29, No. 5, pp. 789-798
- Dmitriyev, V.I., Pisarenko, V.E., 1984, "Determination of optimum refrigerant charge for domestic refrigerator units". *International Journal of Refrigeration*, Vol. 7, No. 3, pp. 178-180.
- Gonçalves, J.M., Melo, C., 2004, "Experimental and numerical steady-state analysis of a top-mount refrigerator". 10th International Refrigeration and Air Conditioning Conference at Purdue, West Lafayette, USA.
- Hermes, C.J.L., Melo, C., Knabben, F. T., 2012, "Alternative test method to assess the energy consumption of frost-free household refrigerating appliances". 14th International Refrigeration and Air Conditioning Conference at Purdue, West Lafayette, USA.
- Hermes C.J.L., Melo C., Knabben F.T., Gonçalves J.M., 2009, "Prediction of the energy consumption of household refrigerators and freezers via steady-state simulation". *Applied Energy*, Vol. 86, pp. 1311-1319.
- ISO/FDIS 15502, 2007, "Household refrigerating appliances – Characteristics and test methods". International Organization for Standardization, Geneva. Switzerland.
- Jakobsen, A., 1995, "Energy optimization of refrigeration systems. The domestic refrigerator – a case study", Ph.D. Thesis, The Technical University of Denmark.
- Sim J.S., Ha J.S., 2011, "Experimental study of heat transfer characteristics for a refrigerator by using reverse heat loss method". *International Communications in Heat and Mass Transfer*, Vol. 38, pp. 572-576.
- Vjacheslav, N., Rozhentsev, A., Wang, C., 2001, "Rationally based model for evaluating the optimal refrigerant mass charge in refrigerating machines". *Energy Conversion and Management*, Vol. 42, No. 18, pp. 2083-2095
- Wright, G.B., 2003, "Radial basis function interpolation: numerical and analytical developments. Ph.D. Thesis, University of Colorado, USA.

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