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EFFECT OF SOME DESIGN PARAMETERS AND REFRIGERANT FLUID ON THE PERFORMANCE OF A LIGHT COMMERCIAL FREEZER

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ABSTRACT

The present work presents a set of numerical studies, for a light commercial freezer, about the influence on the performance of some design parameters. In particular, the influence of the combination of charge-capillary tube diameter has been analyzed. The results do not only analyze parameters related with the energy consumption but with other operation parameters. The results are shown in two dimensional maps for each parameter. These maps have the enormous advantage to provide detailed and important information regarding the sensitivity of the system to variations of the refrigerant charge and capillary tube diameter allowing, therefore, to determine their absolute optimum values. Moreover, the current trend to replace the synthetic refrigerants with natural fluids, has suggested the realization of simulations aiming at determining the performance of the system with different refrigerant fluids. It has been compared the performance of the application running with R600a, R600, R290 and R134a. Results show that for this application the propane has the best performance in terms of COP with a reduced charge.

1. INTRODUCTION

Today, the design of the refrigeration vapor compression systems and their performance optimization means solving problems related to some different topics. The first one is about the energy consumption reduction. Indeed, the current European legislation is pushing to manufactures to reduce the energy consumption of their products due to the adoption of the energy labels such as: A+ + +, A+ + and A+.

The second important aspect concerns the replacement of synthetic refrigerant fluids considered to be causing of ozone depletion and global warming. Actually the research is directed towards the use of natural fluids such as CO_2 and hydrocarbons (HC). In particular, the hydrocarbons have excellent thermodynamics properties, but have the disadvantage of being flammable. For this reason, the amount of refrigerant charged in the system doesn't have to cause danger of fire or explosion in case of accidental loss or accidents.

During the tests, the necessity to change many system configurations, try different refrigerant fluids and change a lot of operating conditions led the researchers to find new methods of work aimed to reduction of computational time and costs typical of the experimental approaches. Recently, some authors, such as Hermes et al. (2008), Gonçalves et al. (2009) and Borges et al. (2011), suggested the use of analytical methods based on experimental data. Today, the semi-empirical approach is widely used and allows obtaining good results in relatively short times for this kind of systems.

Recently Björk and Palm (2006) and Boeng and Melo (2012) analyzed experimentally for household refrigerator the influence on the performance of the refrigerant charge and the capillary tube. Björk and Palm (2006) measured directly the energy consumption of the tested system, whereas Boeng and Melo (2012) calculated the energy consumption from the power consumption measured in steady-state tests. Both works varied the capillary tube by using a needle valve. This device affects the expansion device capacity (EDC) but was not directly related with a length or diameter variation. Acting on the position of needle valve, Boeing and Melo (2012) proposed a method to individuate the optimal couple of capillary tube length and refrigerant charge. Actually, not choosing this optimal couple, the thermodynamic cycle realized by the system varies significantly. In particular the evaporator temperature, the superheat and sub cooling assumes so different values. Consequently also the performances of the system undergo a sensible decrease causing until to 30% power consumptions increase.

The present paper presents a set of numerical studies similar to Boeng and Melo (2012) but the studies were carried out directly as function of the capillary tube diameter. The results have been obtained by using the model (IMST-ART, 2010) presented and validated by Pisano et al, (2012).
In addition to the previous studies, the performance of the same unit was evaluated changing the refrigerant fluid. The comparison has been realized for R600a, R600, R290 and R134a. The unit was optimized previously for each refrigerant in order to get a fair comparison.

2. EXPERIMENTAL SET-UP

2.1 Experimental test bench description

The tests were conducted in a light commercial freezer based on a vapor compression cooling system. The experimental system set-up is accurately depicted in Figure 1 and Figure 2.

It consists of the following components: a single-speed 14.32 cm3 reciprocating-hermetic compressor; the evaporator is a fin-and-tube heat exchangers, while the condenser is a tubeless heat exchanger model STVF 124 furnished by LU-VE. The capillary tube is brazed with the suction line forming a lateral counter-flow heat exchanger. The application of this capillary tube reduces sweating in suction line and slugging of the compressor in addition to an improvement on efficiency. The refrigerant employed is propane (R290). The specific data of components was described by Pisano et al. (2012) in detail.

The air temperature in the cabinet is controlled by the electronics which switches the compressor on or off to keep the temperature equal to the setting point with a certain hysteresis. An axial fan supplies a constant cold air flow rate into the freezer.

Figure 1. Picture of the cabinet in the climati chamber

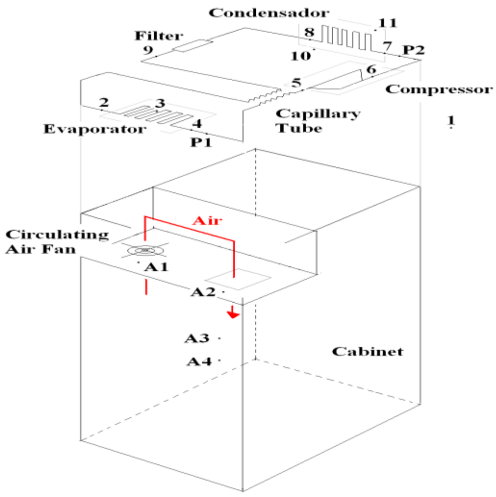

Figure 2. System layout: position of thermoresistance, piezoresistive transducers and NTC sensors

As depicted in figure 2, pressure and temperature measurements along the circuit are performed by means of piezoresistive transducers (P1 and P2) and of thermoresistances PT100 model (1-10), respectively. Ambient temperature and relative air humidity are monitored by means of a NTC sensor (A1-A4) and a hygrometer. The power is measured by means of a wattmeter installed in the cycle. The experimental apparatus is equipped with a system for data acquisition and processing.

In order to obtain useful data to validate correctly the mathematical model used in the simulations studies, experiments were carried out with modifications in some components definition and working conditions. Table 1shows the definition of those parameters modified with regard to the baseline system (Pisano et al.,

2012) for each test besides measurements of the most relevant parameters. The total capillary tube length remained constant for all the tests. Cabinet was always empty and doors closed over all the tests.

Table 1. Experimental parameters

Experiment	Charge [g]	Diabatic length [cm]	T_{amb} [°C]	W [W]	T_{ev} [°C]	T_{co} [°C]	$T_{cabinet}$ [°C]
Test 1	95	120	30	340,4	-33,1	38,4	-25,7
Test 2	95	105	32	341,0	-32,4	39,3	-22,1
Test 3	95	80	32	352,7	-31,8	39,7	-23,4
Test 4	95	45	32	351,9	-31,1	40,0	-24,2
Test 5	85	120	32	343,0	-31,6	40,1	-21,0

The model used in the simulation studies assumes steady-state regime, so that these conditions have to be reached in the test in order to use the measurements to validate the model. To get steady conditions, the system starts-up with ambient air inside cabinets and works with the electronics was switched off and closed doors. In this way, the system cools down the air until heat gain equalizes the cooling capacity. Therefore, the compressor is running continuously and steady state hypothesis is verified.

2.2 Data reduction

The previous experimental tests have permitted to evaluate the temperature and pressure of some points of the refrigerant loop and the real power consumption of the system. Processing these data, inserting them in the mass conservation equation and in the energy balances, it was possible to determine all operative parameters of refrigeration system including cooling capacity and COP. All these parameters, as later shown, have been used to the model validation.

The mass flow rate was estimated by using manufacturer data of the compressor. This data allows obtaining the volumetric efficiency of compressor as a function of the pressure ratio. Therefore, the mass flowrate is estimated as follows,

$$\dot{V} = V \cdot n \cdot \eta_v \tag{1}$$

$$\dot{m} = \dot{V} \cdot \rho_5 \tag{2}$$

Where the volumetric flow rate ($\dot{V}$) is equal to the product of compressor displacement (V), number of revolution per minute (n) and compressor volumetric efficiency (η_v). The mass flow rate is calculated considering the compressor suction density (ρ_5).

The capacity is defined using the energy balance on the evaporator as follows,

$$\dot{Q} = \dot{m} \cdot \left(h_4 - h_2\right) \tag{3}$$

However, for a capillary tube heat exchanger with the suction line, the enthalpy at evaporator inlet cannot be calculated with temperature and pressure, neither assuming same enthalpy as condenser outlet, so next equation is used instead of (3)

$$\dot{Q} = \dot{m} \cdot \left(h_5 - h_4\right) \tag{4}$$

Each enthalpy value is estimated using REFPROP (NIST, 2012)] as function of experimental temperature and pressure values. The COP is obtained with the following equation:

$$COP = \frac{\dot{Q}}{\dot{W}} \tag{5}$$

3. MODEL DESCRIPTION

The simulation study has been performed by means of IMST-ART software (Corberán at al., 2002, IMST, 2010). A short description of the main characteristics of the model in what matters the present paper is given in this section. For a full description of its characteristics and capabilities the reader is referred to www.imst-art.com.

The global model of the whole system is divided in sub-models: compressor, heat exchangers, expansion device, accessories, and piping. Each sub-model involves a series of non-linear equations and in the case of the heat exchangers, a system of ODEs, which is discretized with a finite volume technique. Then, the sub-models are coupled to form a global model of the system. The global set of equations forms a complex system of non-linear equations, which is solved globally by a Newton-like solver.

3.1 Compressor

IMST-ART models the compressor performance as function of the volumetric efficiency, the compressor efficiency and the fraction of power input which is lost to the environment from the outer shell of the compressor. The volumetric and compressor efficiencies can be obtained from catalogue data as a function of the pressure ratio.

Of special importance for a simulation and optimization tool of refrigerators is the estimation of the amount of refrigerant dissolved in the lubricant oil. The software includes built-in correlations with the refrigerant into oil solubility for some typical combination of refrigerant and oil, which allow the estimation of the amount of refrigerant dissolved in the oil. The solubility curves of refrigerant in oil have been extracted from ASHRAE Handbook (2006) and Henderson's work (1994).

3.2 Expansion device

In case of household refrigerators or commercial freezers, expansion device widely used is a capillary tube that forms a counter-flow heat exchanger with the suction line. The model for the capillary tube to suction line heat exchanger (CT/SL HX), first uses an empirical correlation specific for this kind of capillary tubes (ASHRAE Handbook, 2006) to obtain the refrigerant mass flow rate along the CT/SL HX. Once the mass flow rate is known the heat transfer problem is solved by using a moving boundary model. Each phase-region is calculated using a –NTU based lumped approach, where average heat transfer coefficients are be evaluated for each region.

3.3 Heat exchangers

The model of heat exchangers is the most important sub-model of the whole system in order to get accurate results, and it is the most complicated one. Heat exchangers are modeled applying a segment-by-segment approach. The numerical method employed for the heat exchangers solution is called SEWTLE (Semi Explicit method for Wall Temperature Linked Equations). Basically, this method is based on an iterative solution procedure, which consists in an iterative series of explicit calculation steps. For further description, see Corbèran's work (2001).

Regarding refrigerant side, in the case of the evaporator or condenser a 2-phase flow with phase change occurs. The separated two-phase flow model is assumed. Correlations from literature are employed for the evaporation and condensation heat transfer and friction coefficients. In the system analyzed, the evaporator is a finned tube heat exchanger while the condenser is a tubeless heat exchanger. The condenser has been modeled as a finned tube heat exchanger with identical circuitry and fin geometry but using a diameter hydraulically equivalent to the actual duct.

Regarding the air side, the evaporator either dehumidification or frost can be present. For these studies, either ice layer or ice growth was not taken into account. The approach followed to treat the dehumidification process is the one proposed by Threlkeld (1970).

In both heat exchangers model the governing equations are those stated for the mass, energy and momentum conservation.

4. MODEL VALIDATION

First, all the data required by the model was defined in IMST-ART for the baseline system (Pisano et al., 2012) from catalogues data or drawings: compressor curves, fan curves, geometric data of capillary tube,

heat exchangers and pipes, After definition of components introducing the operating conditions is necessary: refrigerant charge and air temperature and humidity at the inlet of both evaporator and condenser.
The air temperature at condenser inlet is assumed to be the same as the surrounding air. The inlet temperature and humidity at evaporator could be considered as the temperature of air inside cabinet which is constant for the experimental tests analyzed. For each experimental test (Pisano et al., 2012), the corresponding modified parameters were updated in the model. Figure 3(a), (b), (c) and (d) show the results of the model after a global adjustment work. Figure 3(a) and 3(b) show how both experimental cooling capacity and power consumption are predicted IMST-ART within a ±10% error band.

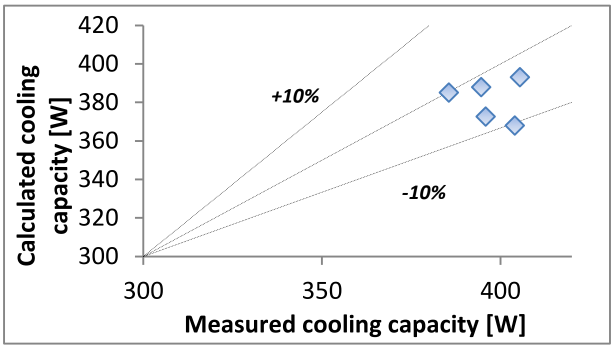

Figure 3a.

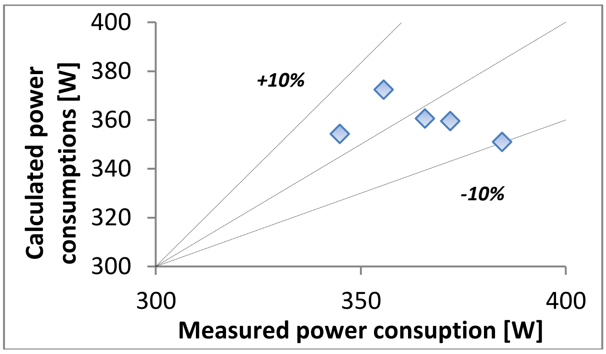

Figure 3b.

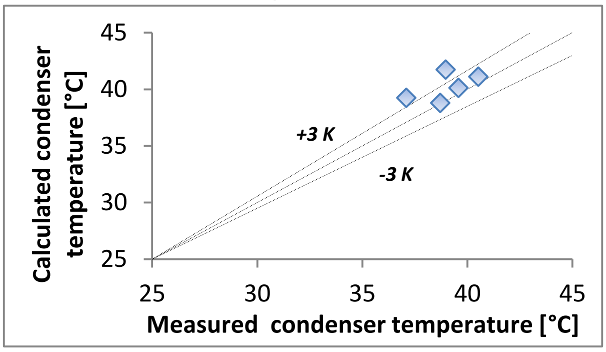

Figure 3c.

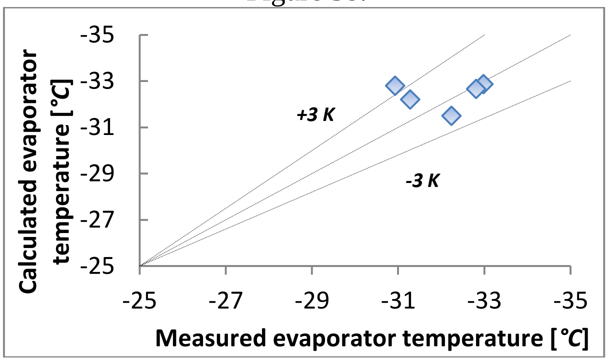

Figure 3d.

Figure 3(c) and 3(d) show that the experimental evaporation and condensation temperature are predicted with an error band of ±3 K. The error in condenser is larger than for evaporator, it could be due to the hydraulic analogy applied in the tubeless heat exchanger.

5. SIMULATION RESULTS

The system analyzed, in normal operation, would work cycling because of the electronics action. Thus, the cabinet temperature varies over the time within the hysteresis values.

The studies will be done assuming steady regime so that just one temperature will be used. The air temperature chosen corresponds to the setting point. Martinez-Ballester et al. (2012) showed that once quasi-steady state is reached, the performance variation due to this temperature change is negligible. In fact, this idea can be observed in Figure 4 that shows the baseline system running in real conditions.
The freezer operation studied in the present work corresponds to any “on period” of a serial of “on-off periods”. This assumption is quite realistic for systems with forced convection heat exchangers, where transient phenomena are minimized. This fact can be also observed from Figure 4. In this regime, the parameter referred to as duty-cycle ratio assumes a lot of importance. It is defined as:

$$\tau = \frac{\Delta t_{on}}{\Delta t_{on} + \Delta t_{off}} \cong \frac{\dot{Q}_{load}}{\dot{Q}_{evap.}} \tag{6}$$

The energy consumptions are calculated as Melo (2012) suggests. The equation is given as following:

$$CE = \tau \cdot \dot{W} \tag{7}$$

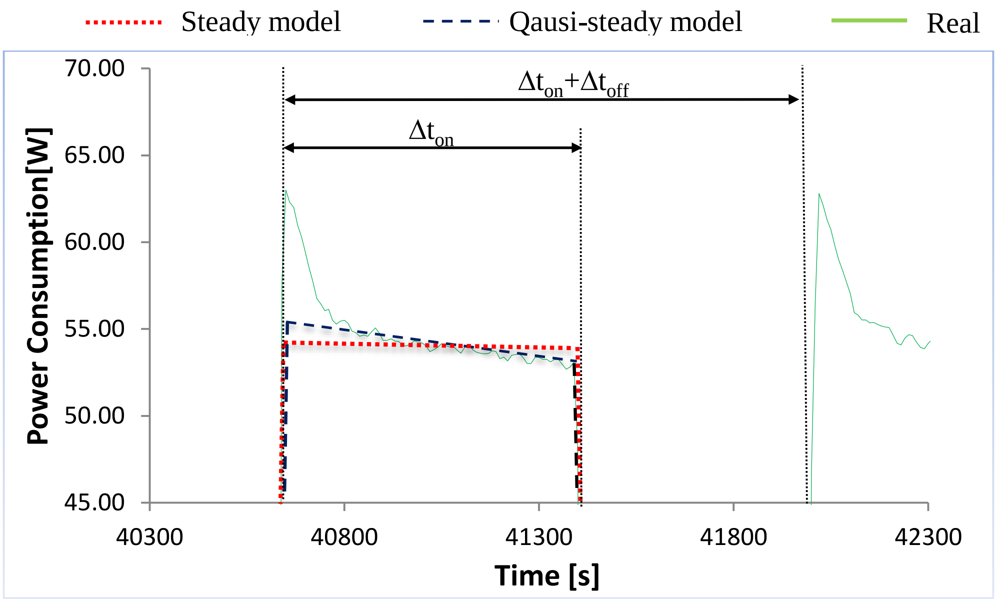

Figure 4. Compressor power consumptions

Thus, in order to calculate the freezer energy consumption, the evaluation of the heat gain into the cabinet is required. It can be evaluated as follows,

$$\dot{Q}_{load} = UA\,(T_{amb} - T_{cabinet}) \tag{8}$$

The UA value was experimentally determined which corresponds to a value of 4,5 W/K.

5.1 Two-dimensional influence of refrigerant charge- capillary tube diameter on the performance

This section analyzes the effect on performance of two important design parameters of a commercial freezer: the refrigerant charge and the capillary tube diameter. Usually, the dimensions of the system fix the total capillary tube length and then the diameter is chosen for optimizing the performance in terms of COP. Anyhow, what really is looked for in the selection of a capillary tube is an expansion device capacity (EDC), which can be obtained with either capillary diameter or length. Once the system is fully defined the last step is choosing the optimum refrigerant charge (in terms of COP). There will be always an optimum charge given the design of the capillary tube.

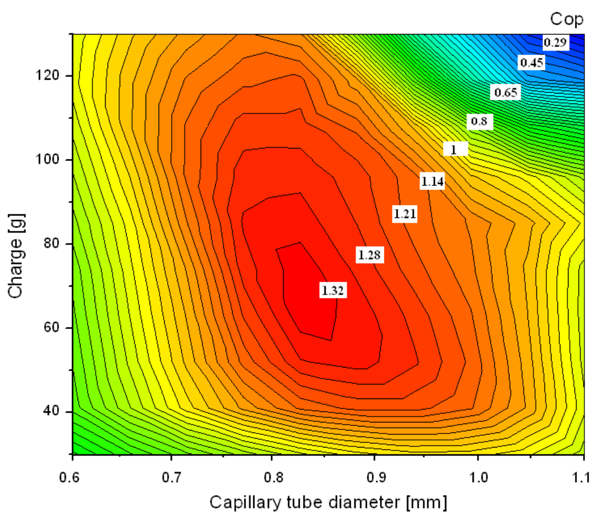

Figure 5a.

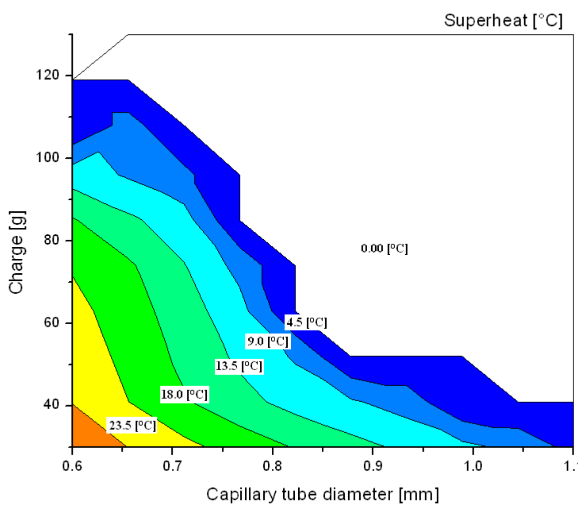

Figure 5b.

But there will be only one absolute maximum if we analyze together the capillary tube and the charge. This two-dimensional design will provide a much better performance because we can explore many other operating points, and maybe different from the optimum value if we take into account other operating factors.
This section studies both parameters in the system performance but not only attending to efficiency parameters but also to important operating variables. Compressor suction temperature is not a relevant parameter from an efficiency point of view but is a very important to factor for preventing quality issues like sweating when its temperature is below dew point.
Figure 5(a), (b), (c), (d), (e) show the results of this analysis, where100 operating points have been evaluated. Figure 5(a) shows the maximum value of COP within a small area of the map characterized by the following ranges of values: 0.81-0.86 mm and 58-81 g. The map for the energy consumption is shown in Figure (6). It is interesting to observe that the most dominant variable is the capillary tube diameter, expecting for extreme charges.

Charge only has influence in the region close to optimum and rather small, it only has large influence for very low charges. Another interesting fact is that the gradient in the region close to the optimum value is small, i.e. is quite flat. This fact means that there are many combinations of charge and diameter for obtaining a similar COP and close to the optimum one.

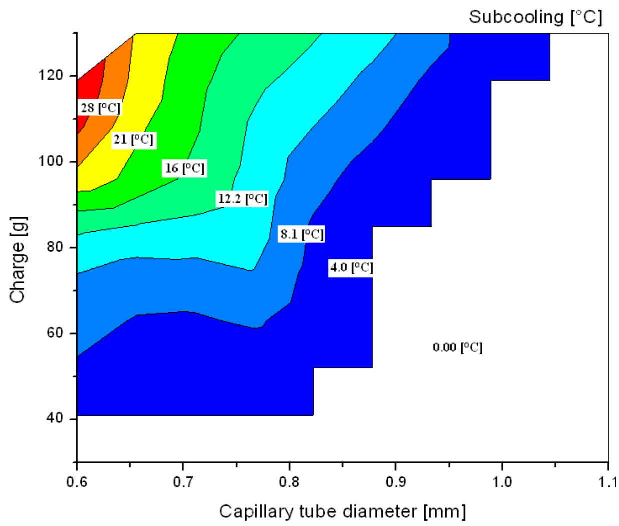

Figure 5c.

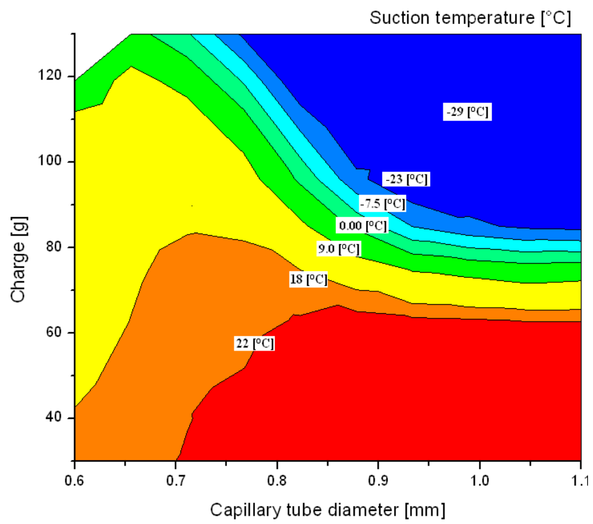

Figure 5d.

These combinations correspond to large charge with small diameter or large diameter with low charge. These results agree quite well with the results already obtained experimentally by Boeng and Melo (2012). However there is an important difference in the methodology applied them. Boeng and Melo (2012) modified the EDC by using a needle valve, so there is no direct relationship between the results and the capillary tube diameter or length.
Figure 5(b) shows the evaporator superheat. If Figure 5(a) and (b) are analyzed together an interesting relationship can be established between the evaporator superheat and the combination of diameter-charge that optimizes the COP. The combination that optimizes the COP corresponds to the same one that produces saturated vapor at the evaporator outlet. This relationship is quite common in this kind of systems. Fixed a capillary tube diameter and increasing the refrigerant charge, the evaporator superheat decreases and the total sub-cooling (Figure 5(c)) increases.
Figures 5(c), (d) and (e) aid the selection of the capillary tube and charge because critical operating points can be easily avoided by observation of the figures. Figure 5(c) has to be consulted in order to select a combination that ensures a sub-cooling greater than 1 K for prevention of slugging in the capillary tube inlet.

In the same manner, Figure 5(d) informs about the suction temperature at the compressor inlet, essential parameter for prevention of suction line sweating. In the area of map close to the optimum COP, the suction compressor temperature is in a range of values equal to 18-22 ºC, therefore, greater than the dew point temperatures calculated in the nominal conditions. Figure 5(e) plots the duty-cycle ratio, whose definition is established in Eq. (6). If we analyze this parameter with a steady approach, its variation will be related only to the cooling capacity variation because the heat gain remains constant.

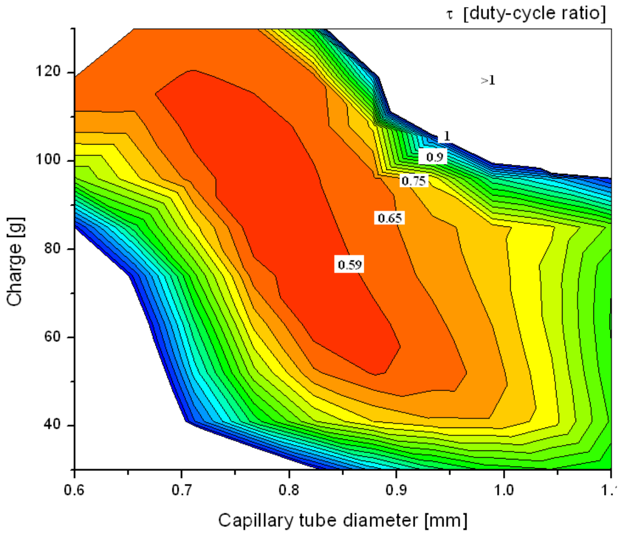

Figure 5e.

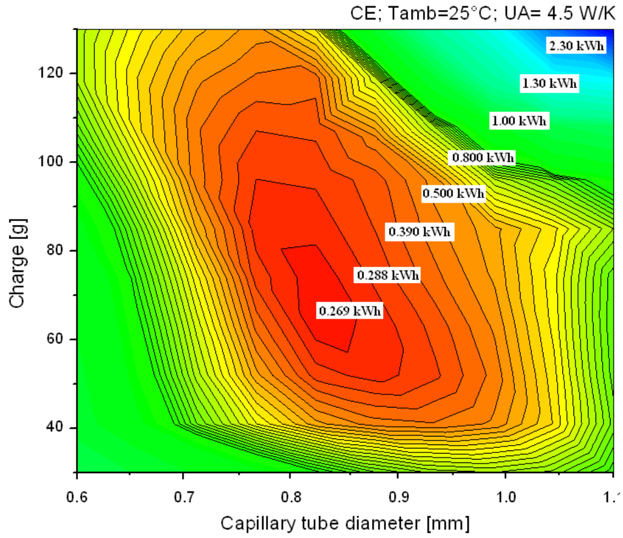

Figure 6.

However this parameter provides useful information: the closer the duty-cycle ratio is to one, the more continuous the operation will be. Values larger than one makes the system not able to cool down the cabinet to the required setting point. Even though this parameter is evaluated applying a steady approach, important information about transient phenomena can be analyzed. In an on-off system as analyzed in the present work, a large value of the duty-cycling ratio means a reduced frequency of start-up and shut-down. Each start-up produces energy loses due to transient phenomena (Coulter and Bullard, 2012, Martinez-Ballester et al., 2012), which means a penalization of the total energy consumption of the system.
The main conclusion of these figures is that choosing a pair diameter-charge attending only to the COP map could derive in important operation problems.

5.2Refrigerant comparison
The experimental apparatus uses propane (R290) as working fluid. The performance of a refrigerator is strongly influenced by chemical and physical properties of the fluid. Changing the refrigerant, the same system may achieve higher COP, providing the same cooling capacity but with a lower power consumption. Thus, the performance of the system has been compared with four different refrigerants: R600a, R600 and R290 as natural; and R134a as synthetic.

Obviously, the comparison is going to be performed for the same operating conditions, which are those corresponding to the baseline conditions. But, a fair comparison does not correspond to compare same system with same components by just replacing the refrigerant. A components optimization for each refrigerant has to be carried out, in order to compare each fluid with the best components definition, specific to such a refrigerant.

Table 2. Optimal system configuration for each refrigerant

Refrigerant	[-]	Propane	Isobutane	Butane	R134a
Comp. Displacement.	[cm^3]	14.32	51	71	26
Capillary tube					
Diameter	[mm]	0.83	1	1.16	0.99
Length	[m]	1.47	1.47	1.47	1.47
Diab. Length	[m]	1	1	1	1

We can assume that regardless the refrigerant, the heat exchanger will be chosen as big as possible to fit to the system dimensions. So, these elements are not modified and are the same as shown in Pisano et al. (2012). If we fix as optimization restriction that the cooling capacity has to be the same, i.e. the same duty-cycle ratio, the optimization problem consist of finding the combination of capillary tube diameter and refrigerant charge which maximizes the COP for a system whose compressor displacement has to be chosen to satisfy the previous restriction. This methodology requires somehow analyzing a 2D map for each compressor displacement until the optimum combination diameter-charge gives the same cooling capacity of the baseline system. This tedious task was numerically performed, providing the parameters shown in Table 2.

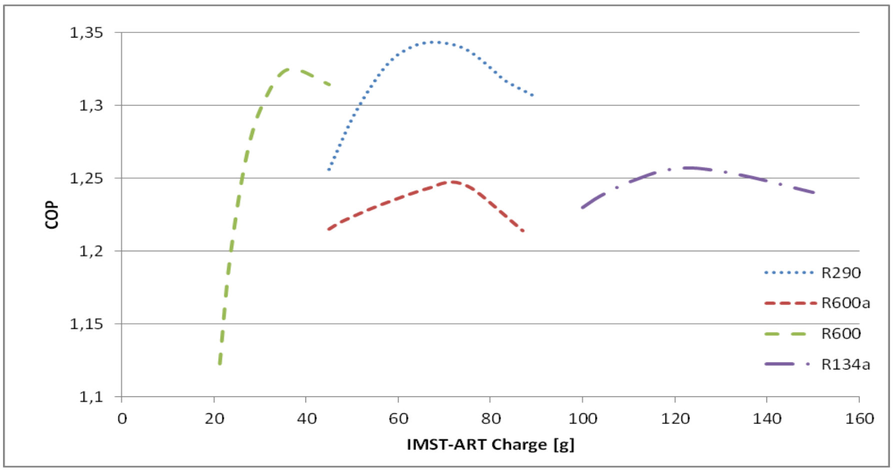

Figure 7.COP as a function of the refrigerant charge

The performance comparison between the different refrigerants using the components previously defined is shown in Figure 7. The figure shows a parametric study of COP as function of the refrigerant charge. It is important notice that the COP values shown in the figure are the maximum values that each refrigerant can get in this system. The propane provides the best absolute COP. Even though the butane reaches good values of COP, the gradient of the curve is especially steep which makes it very sensible to charge variations. Actually, isobutane has the lowest COP, though is almost the same as for R134a. However, R134a has an interesting flat curve.

Another interesting analysis can be performed in terms of how much charge each refrigerant needs to get the maximum COP. This comparison is not only interesting in terms of charge reduction but in terms of energy loses. Martinez-Ballester (2012) scribed that the migration of refrigerant in this kind of systems, when the compressor shuts-down, introduces an additional heat lose. Indeed, when the compressor turns-off, the charge moves from condenser to evaporator carrying thermal energy. This energy is issued in the cabinet of the refrigerator. Future studies could evaluate the strong relationship between the energy release and charge amount. The set of natural refrigerants studied needs much less charge than R134a. Butane is the one who needs the least refrigerant charge.

6. CONCLUSION

The paper presents a set of numerical studies with the aim to analyze the influence of important design parameters in the described system. The parameters analyzed in the paper are the refrigerant charge and the capillary tube diameter. Important conclusions were obtained by observation of the two-dimensional maps, warning that a combination that optimizes the COP can affect negatively other operating parameters such as the suction temperature, sub-cooling and superheat. Main conclusions for this system are the following:

- The parameter combination that optimize the performance in terms of COP corresponds whit the following ranges of values: 0.81-0.86 mm and 58-81 g.
- In the area of map where the COP assumes the highest values, the variability of the refrigerant charge doesn't affect significantly on the performance of refrigerator system. In this area, the system is more sensitive to capillary tube diameter changes.
- In the optimum operating condition, in the evaporator outlet are present saturated vapor condition, while, at the condenser outlet, the optimal combination of charge-capillary tube

ensures a sub-cooling greater than 1 K avoiding the presence of slugging in the capillary tube inlet.

- In the optimum operating condition, the compressor suction temperature is greater than the dew point temperature calculated in the nominal conditions.

Last study evaluated the potential of a set of refrigerants as alternative for using in freezers attending to the energetic performance. The comparison was carried out by using an optimized system for each refrigerant. Propane resulted to be the best option in terms of COP, requiring a low refrigerant charge.

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