INSTITUTO DE INGENIERÍA ENERGÉTICA
UNIVERSIDAD POLITÉCNICA DE VALENCIA

CPI, Edificio 8E, Cubo F, 5ª planta, Camino de Vera s/n, 46022 Valencia, Spain Tel. (34) 963877270, Fax (34) 963877272, Email: energeti@upvnet.upv.es

INSTITUTO DE INGENIERÍA ENERGÉTICA
(Institute for Energy Engineering)

Research Publications

NUMERICAL ASSESMENT OF INFLUENCE OF REFRIGERATOR CHARGE IN A LIGTH COMMERCIAL FREEZER

A. PISANO[(a)], S. MARTÍNEZ-BALLESTER[(b,*)], José-M. CORBERÁN[(b)], A. W. MAURO[(a)]

(a) DETEC, University of Naples "Federico II", p.leTecchio 80,
Napoli, 80125, Italy
(b) Instituto de Ingeniería Energética,
Universidad politécnica de Valencia, Camino de Vera, s/n,
Valencia, 46022, España
(*) Tel: +34 963 879 121, e-mail:sanmarba@iie.upv.es

ABSTRACT

This work analyzes in a light comercial freezer the influence on the performance of varying the operating conditions, refrigerant charge and the system configuration. This analysis was conducted employing a simulation software (IMST-ART), based on a semi-empirical approach that assumes steady-state conditions, for the simulation of refrigerant vapor compression systems. The simulated vapor compression system uses propane as working fluid. The system consists of a single speed hermetic compressor, finned tube heat exchangers and a non-adiabatic capillary tube. The paper briefly describes the model used for the simulations. In particular, there is a description of the sub-models of the compressor, heat exchangers and the expansion device. The validation of the model was performed with experimental data obtained by testing the freezer.

The purpose of the simulations is to identify the amount of refrigerant which minimizes energy consumption under varying operating conditions. It was also assessed the benefits of a system with variable charge against a fixed charge one. Results show that a system with a fixed charge, which uses the optimum charge for an specific ambient condition, can get a similar perfomance to a variable charge system when ambient conditions change.

1. INTRODUCTION

In recent years, the growing interest on the issues concerning to the environment conservation and energy resources is pushing a lot of researchers to identify technology solutions focused on reducing energy consumption and pollutant emissions.

In this context, the refrigeration industry is also committed to find innovative and challenging solutions for the refrigeration field. Cecchinato and Corradi (2011) pointed out that the refrigerators and freezers were affecting negatively world energy consumption. They also described accurately the most recent developments in the refrigeration field until that moment. Recently, Melo and Da Silva (2010) presented a paper about the current technological challenges in the design of household refrigerators, giving a global overview about topics such as latest technologies, development on compressors and heat exchangers, research about components matching and thermal energy losses.

Traditionally, the design of high efficiency refrigeration systems is mainly based on the careful selection of the best components on the market and the evaluation the most efficient control and management of the facility. To this end, experimental campaigns are always necessary to obtain knowledge and data for determine the performance of the unit for different operating conditions. This is a very expensive and time-consuming task due to the high cost of laboratory.

The use of simulation tools is increasing due to its costs reduction by reducing considerably the number of experimental tests. The two main modeling approaches are numerical analysis via CFD package and semi-empirical models. The use of commercial CFD packages can provide a very detailed description of the refrigerator operation, but they have the disadvantage of requiring large computational cost for calculation. Semi-empirical models are simple but can provide a good ratio between accuracy and computational cost for simulation of vapor compression systems, thus they are strongly recommended for such systems

Regarding the time dimension, the semi-empirical models can be classified in three different approaches: transient (Melo et al., 2008, Tagliafico et al., 2012), steady-state (Gonçalves et al., 2009, Hermes et al., 2009) and quasi-steady (Borges et al., 2011, Martínez-Ballester et al., 2012a). Transient approach is the most accurate since it reproduces the phenomena really as it is, but large calculation time is required compromising also the robustness of the model due to the complex phenomena modeled. Steady-state approach (Gonçalves et al., 2009, Hermes et al., 2009) simplifies the problem to analyze the refrigerant loop working in steady conditions. This simplification reduces greatly the computational cost besides increase the robustness of the model. Boeng and Melo (2012) estimated, with predictions of the refrigerant loop performance, the energy consumption of a household refrigerator assuming steady conditions.

An interesting compromise between accuracy and computational cost could be a quasi steady-state approach (Borges et al., 2011; Martínez-Ballester et al., 2012a), i.e. when the air temperature changes, the refrigerant loop immediately reach steady conditions. Martinez-Ballester et al. (2012b) concluded that these kinds of approaches cannot predict correctly the air temperature evolution over the time for an On-Off system. They pointed out as main reason the impact of the transient phenomena, which turns out to be very important in this kind of systems. However, Martinez-Ballester et al. (2012b) showed that a quasi-steady approach for the refrigerant loop predicts correctly its performance if the experimental air temperature at evaporator inlet is used instead of the calculated one with the model. Results of Martinez-Ballester et al. (2012b) showed that the variation of refrigerant loop performance, once quasi-steady conditions are reached, is negligible. Hermes et al. (2012) proposed to estimate consumption of household refrigerators based on the steady state performance obtained for standard rating conditions. This methodology provides very good results in systems that transient phenomena have negligible effect.

Authors, such as Mauro-De Rossi (2009), Bjorn and Palm (2012), Boeng and Melo (2012) analyzed experimentally the performance of household refrigerators and freezers for different working conditions and by modifying different components of the refrigerant loop.

In this work, the authors present a numerical analysis of a commercial freezer by using a steady-state model, the commercial software IMST-ART. The work presents studies on the trend of important refrigerant performance parameters when the charge amount is modified for different environmental temperature.

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.

Figure 1. Picture of the cabinet in the climatic chamber

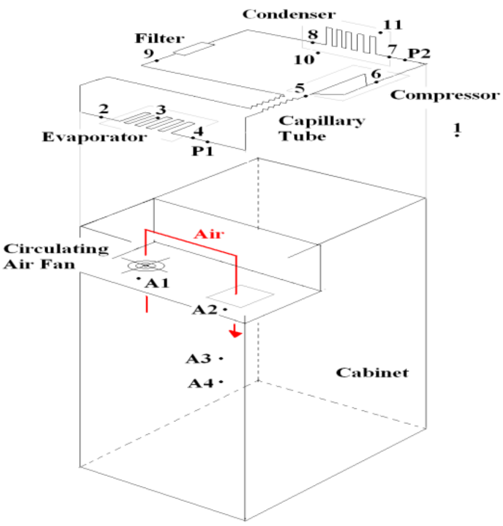

Figure 2. System layout: position of thermoresistance, piezoresistive transducers and NTC sensors

Table 1. Definition for the components and operating conditions of baseline system

Heat Exchangers	**Evaporator**	**Condenser**
Type	Fins&Tube	Tubeless
Model	-	STVF 124
Company	SEST	LUVE
Number of circuits	1	1
Number of rows	6	1
Number of tubes for rows	8	5
Inside tube diameter (mm)	7.5	-
Outside tube Diameter (mm)	9	-
Longitudinal Tube spacing (mm)	13.5	-
Transverse tube spacing (mm)	16	-
Fin spacing (mm)	4.2	-
Fin thickness (mm)	0.13	-
Finned length (mm)	185	-
Heat exchange surface (m^2)	2.102	1.25
Internal volume (dm^3)	0.52	0.3
Tube material	Copper	Copper
Fin material	Aluminum	Aluminum
Fan nominal flow rate (m^3/h)	200	650
Fan power (W)	80	45
Compressor		
Type	Reciprocating	
Model	NPY14LAa	
Company	ACC	
Displacement (cm^3)	14.32	
Nominal speed (rpm)	2900	
Expansion device		
Type	Regenerative capillary	
Diameter (mm)	0.76	
Total length (mm)	147	
Regenerative length (mm)	45-105-120	
Nominal conditions		
Ambient temperature (°C)	25	
Cabinet temperature (°C)	-22	
Ambient relative humidity (%)	50	
Cabinet relative humidity (%)	90	

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 is described in Table 1.
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.

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 2 shows the definition of those parameters modified with regard to the baseline system (Table 1) for each test besides measurements of the most relevant parameters. The total capillary tube length remained constant for all the tests. The cabinet was always empty and doors closed over all the tests.

Table 2. Experimental parameters

Experiment	Charge [g]	Diabatic length [cm]	T_{amb} [°C]	W [W]	T_{evap} [°C]	T_{cond} [°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 (figure 2) as follows,

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

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),

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

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 were 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 ε–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 Corberán'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). The fan curve for each heat exchanger was used in the model.

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

4. MODEL VALIDATION

First of all, the data required by the model was defined in IMST-ART for the baseline system (Table 1) 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 know the refrigerant charge, the air temperature and the 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 (Table 2), the corresponding modified parameters were updated in the model.

In order to validate the predictions of the model, it was chosen as the most suitable comparison parameters: the cooling capacity, power consumption, condensation and evaporation temperature. The first two parameters describe the overall behavior of refrigerator system, while the others show the goodness of the heat exchangers sub-models.

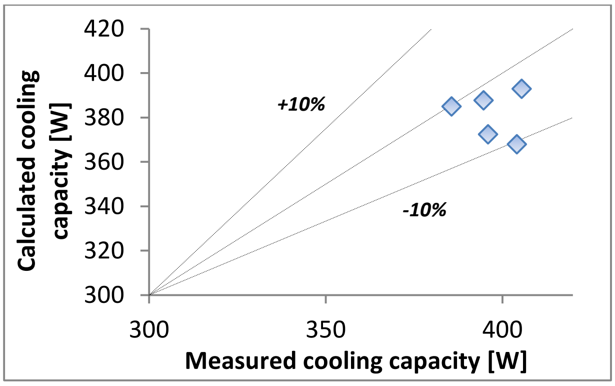

Figure 3a.

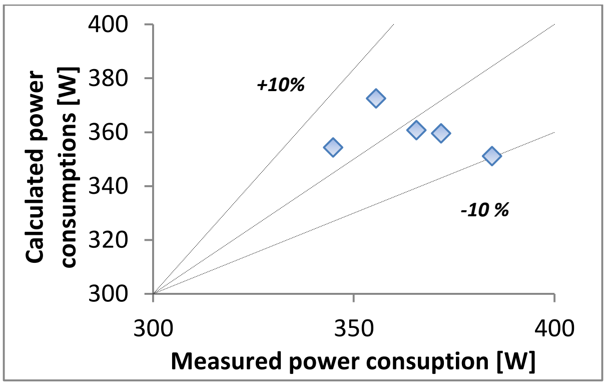

Figure 3b.

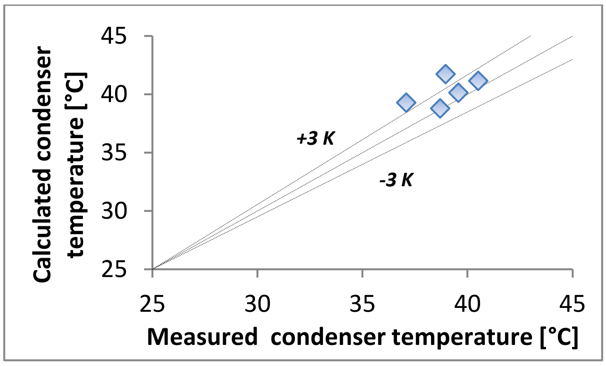

Figure 3c.

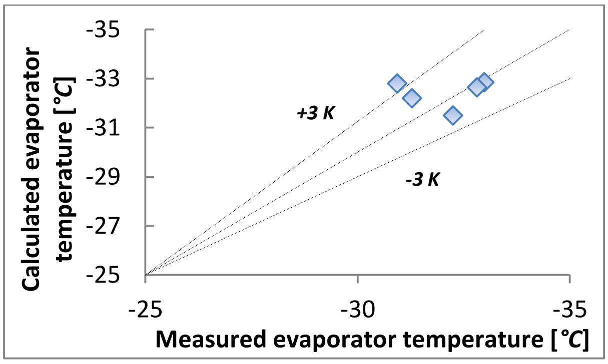

Figure 3d.

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 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 simulations will focus in evaluating the effect of the refrigerant charge in this kind of systems and the efficiency penalization of not using the optimal value for different conditions. This study was conducted for the baseline system (Table 1).
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 band. 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. (2012b) 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 is 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:

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

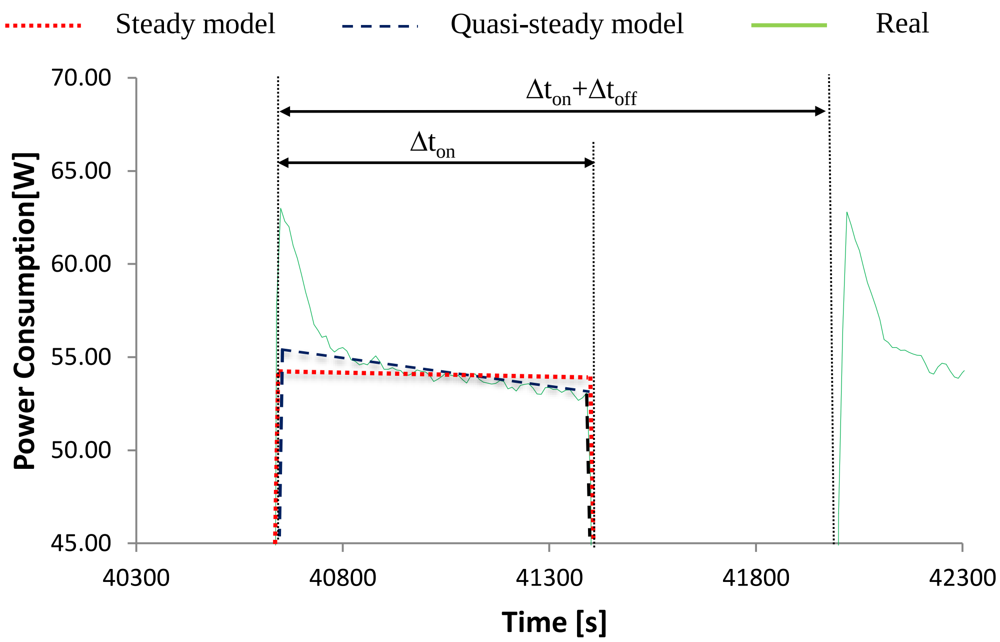

Figure 4. Compressor power consumptions

Indeed, as suggest Boeng and Melo (2012), knowing the duty-cycle ratio, the energy consumption can be calculated as follow:

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

5.1 Analysis of the effect of refrigerant charge in performance

An interesting design parameter of these kinds of systems is the refrigerant charge. This section studies its effect 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 factor for preventing quality issues like sweating when its temperature is below dew point.

Figure 5(a), (b), (c) and (d) show the results of this analysis, where the effect of varying the refrigerant charge in the baseline system (Table 1) has been assessed for different ambient temperature. Each plot also shows the point corresponding to the optimal charge that maximizes the COP.

Figure 5(a) shows that the impact of refrigerant charge on the COP is up to 10%. A system undercharged looks to be less affected by the charge. It can be also observed that the larger the ambient temperature, the lower the optimal refrigerant charge. The variation in the optimal charge within the range of the ambient temperature studied is about 40 grams.

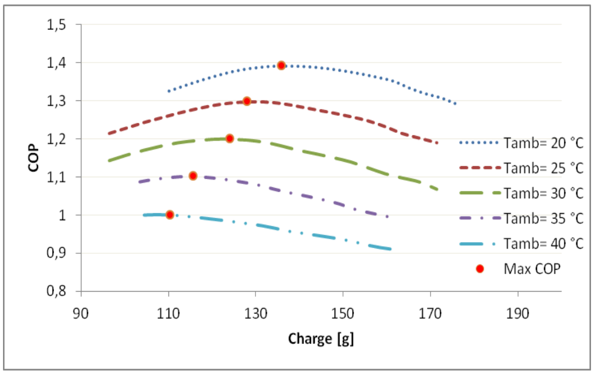

Figure 5a.

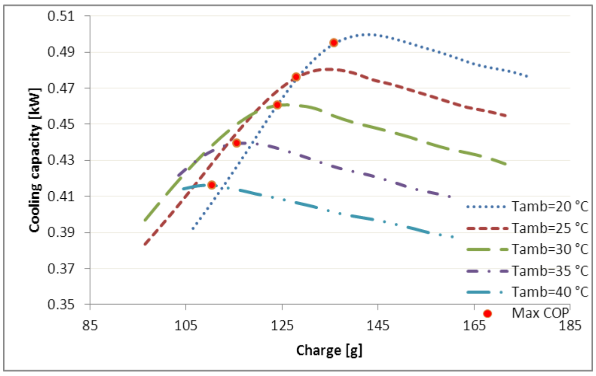

Figure 5b.

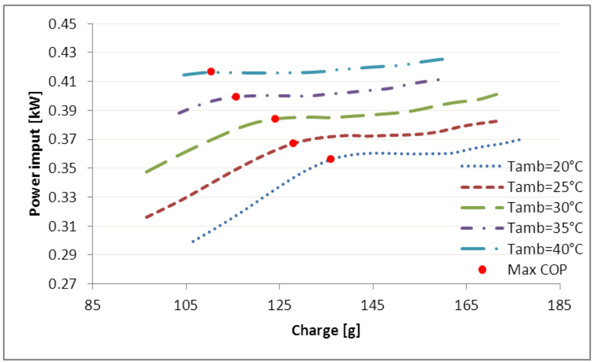

Figure 5c.

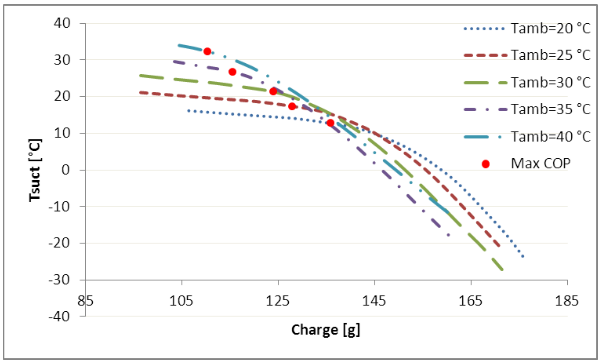

Figure 5d.

The trend of the cooling capacity shown in Figure 5(b) is rather the same regardless the ambient temperature. What is interesting to note is the drastic reduction in the capacity for an undercharged system. A priori, the only requirement that there is for the cooling capacity is that has to be larger than the heat gain in the cabinet, which can be evaluated knowing the external temperature and the parameter UA [W/K]. A variation in the cooling capacity will affect directly to the duty-cycle ratio (Equation 5). This factor has important influence in the energy consumption of this kind of system, but unfortunately it cannot be evaluated with a steady state model since it affects directly to the transient phenomena. However, in global terms, the shorter the duty-cycle ratio the larger the energy losses are due to transient phenomena. The power input, depicted in Figure 5(c), increases as the refrigerant charge is increased.

Figure 5(d) shows that the compressor suction temperature decreases strongly when the refrigerant charge exceeds a certain value, which depends on the ambient temperature. It is interesting to notice that the optimal charge is in the safety region, on the left of the strong decrease of the suction temperature.

5.2 Performance evaluation and comparison of a system with charge control

This section analyzes the benefits of a system that could work always with the optimum refrigerant charge when the ambient conditions change (system A). This is just a concept of system to illustrate the maximum benefits of using the optimum refrigerant charge. Such a system would need to save or input charge into the system, when ambient temperature changes, with e.g. a liquid receiver. To perform the study, the system A is going to be compared with a system with a fixed charge (system B). The system B will be charged with the optimum charge corresponding to the baseline conditions (Table 1). Both system A and B use the same components as described in Table 1.

Figure 6(a) and 6(b) show the results, where the corresponding optimal charge for each ambient temperature can be observed in the right axis. The relationship between the optimum refrigerant charge and the ambient temperature is rather linear. Figure 6(a) shows that the improvement when ambient temperature is equal to the baseline temperature is zero, because both systems have the same charge. However, for other ambient temperatures, the improvement is very small. What really means this fact is that using a system with the optimum charge for an intermediate temperature is enough to work effectively over the ambient temperature variation. If it would not be like this if used an optimum charge for an extreme temperature or a non-optimum charge for any temperature; which would have penalization up to 10%, as explained above.

The analysis for the cooling capacity shown in Figure 6(b) is slightly different. Now, the difference in performance is larger for extreme temperatures. This difference in cooling capacity would report a difference in the duty-cycle ratio, which affects the energy consumption, due to transient phenomena, as exposed in previous section.

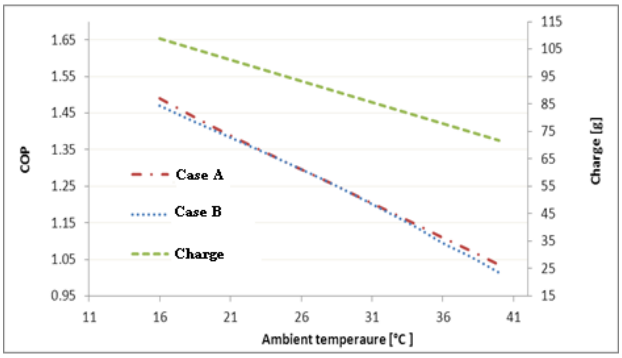

Figure 6a.

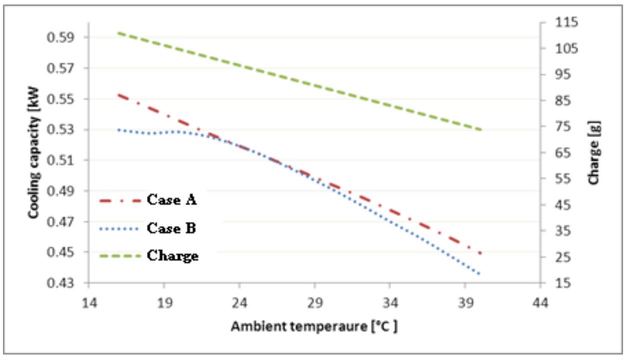

Figure 6b.

CONCLUSIONS

The paper proposes a steady model for simulation of a light commercial freezer. An experimental validation of the model has been carried out showing prediction errors within ±10% for capacity and power consumption.
The model has been used to study the influence of the refrigerant charge for this kind of systems resulting in the following conclusion:

- A different optimum charge, which maximizes the COP, exists for each ambient temperature. The penalisation in the COP, when the optimum charge is not used, resulted to be up to 10%.
- In the analyzed system, using the optimum charge does not affect negatively to other operating parameters, not only related to energy consumption.
- Using a system with a variable charge control is not clearly justified if compared with a fixed charge system that uses the optimum charge for a specific ambient condition

REFERENCES

1. ASHRAE, 2006. ASHRAE Handbook: Refrigeration (SI)
2. Boeng J., Melo C., 2012, Uma metodologia para a seleção do par tubo capilar -carga refrigerante que maximiza o desempenho de refrigeradores domésticos -Parte I: Mapeamento do consumo de energia, CYTEF-2012, VI Congreso Ibérico y IV Congreso Iberoamericano de Ciencias y Técnicas del Frío.
3. Borges B. N., Hermes C. J. L., Gonçalves J. M., Melo C., 2011, Transient simulation of household refrigerators: A semi-empirical quasi-steady approach, Applied Energy 88, pp 748-754.

4. 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.
5. Cecchinato L., Corradi M., 2011, Transcritical carbon dioxide small commercial cooling applications analysis, International Journal of Refrigeration, Volume 34, Issue 1, January 2011, Pages 50-62.
6. Corberán J.M., Fernández de Córdoba P., Gonzálvez J., Alias F., 2001, Semiexplicit method for wall temperature linked equations (SEWTLE): a general finite-volume technique for the calculation of complex heat exchangers. Numerical Heat Transfer Part B-Fundamentals 40 (1), 37-59
7. Corberàn J.M., Gonzàlves J., Montes P., Blasco R., 2002, 'ART' a Compiuter Code Assist The disign of Refrigeration and A/C Equipment, Internation Refrigeration and Air Conditioning Conference at Purdue, IN, USA.
8. de Rossi F., Mauro A.W., Musto M., Vanoli G.P., 2009, Experimental investigation of operating characateristics and energy consuptions for a vertical freezer varying the refrigerant charge, IIR 1st Workshop on refrigerant Charge Reduction, Cemagref Antony, France.
9. Gonçalves J. M., Melo C., Hermes C.J.L., 2009, A semi-empirical model for steady-steate simulation of household refrigerators, Applied Thermal Engineering 29, pp 1622-1630.
10. Henderson David R., 1994, Solubility, viscosity and density of refrigerant/lubrificant mixtures, The Air-Conditioning and Refrigeration Technology Institute, p.144 Stockbridge.
11. Hermes C.J.L., Melo C, Knabben F.T., 2012, Alternative Energy Test Method for Frost-Free Refrigerators and Freezers, International Refrigeration and Air Conditioning Conference at Purdue.
12. Hermes C. J. L., Melo C., 2008, A first-principles simulation model for the start-up and cycling transient of household refrigerator, Internation journal of refrigeration, 31, pp 1341-1357.
13. 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, Volume 86, Issues 7–8, pp 1311-1319.
14. IMST-ART, 2010, Simulation tool to assist the selection, design and optimization of refrigerator equipment and components, http://www.imst-art.com, Universitat Politècnica de Valéncia, Instituto de Ingenieria Energética, Spain.
15. Martínez-Ballester S., Lèon-Moya B., Herráiz J.N., Gonzalvèz-Maciá J., 2012a, Dynamc modeof a household refrigerator based on a quasi-steady approach, CYTEF-2012. VI Congreso Ibérico y VI Congreso Iberoamericano de Ciencias y Técnicas del Frío, Spain.
16. Martínez-Ballester S., León-Moya B., Vesson M., Gonzálvez-Maciá J., Corberán J.M., 2012b, Dynamic Performance Simulation of a Household Refrigerator with a Quasi-Steady Approach, International Refrigeration and Air Conditioning Conference, Purdue (France).
17. Melo C., da Silva L.W., 2010, A Perspective on energy saving in household refrigerators, Sustainable Refrigeration and Heat Pump technology Conference, Stockolm, Sweden.
18. Navarro E., Corberán J.M., Martínez-Galván I.O., Gonzálvez J., 2011. Oil sump temperature in hermetic compressors for heat pump applications, International journal of Refrigeration, In Press.
19. NIST. Reference Fluid Thermodynamic and Transport Properties Database (REFPROP): Version 8.0
20. Tagliafico L.A., Scarpa F., Tagliafico G., 2012, A compact dynamic model for household vapor compression refrigerated systems, Applied Thermal Engineering, Volume 35, pp 1-8.
21. Threlkeld J.L., , 1970, Thermal Environmental Engineering. Prentice Hall, Englewood Cliffs, NJ.