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ABSTRACT

The paper presents a quasi-steady approach for modeling a household refrigerator. The model applies a novel methodology to perform the dynamic simulation. It uses a map of unit which contains high accuracy performance data of the refrigeration loop. This map is used by the model of the freezer and fresh-food air cabinets to determine the transient evolution of the air inside. This methodology allows obtaining high accuracy results, with high robustness and low computational cost. The validation of the model with experimental data is shown, by comparison of cabinet temperature and global power input. In order to understand some transient phenomena of the actual system operation, the model has been used to perform comparison studies between the operation of a real system and the equivalent quasi-steady system. Efficiency of both systems has been compared in order to detect energy losses sources. The energy losses analyzed are those related to controlling actions such as: compressor start-up and closing of the damper that supplies air flow to fresh-food cabinet. A discussion about their impact on performance and phenomena involved is presented.

1. INTRODUCTION

The electric energy consumption of these systems represents the highest electric consumption for a standard house in Spain (IDAE, 2011), which turns to be up to 18% of the global electric energy consumption, even though its power is very small compared with rest of appliances. The design of a high efficiency refrigerator will imply the suitable selection of the components of the refrigeration system and control system definition. To this end, knowledge of the phenomena taking place is necessary but experimental energy assessments will always be required. These tests are expensive and takes much time, hence reliable simulation models can provide substantial cost and time savings during the design and optimization process of heat exchangers.

From a modeling point of view, a refrigerator consists of two main elements: cabinets and the vapor compression system. Attending to the phenomena that takes place inside each elements modeling of the cabinets and refrigeration system is quite different, so that it requires different numerical approaches. In order to select a suitable simulation tool for designing purposes it is necessary to understand the main assumptions applied in the model. Semi-empirical models are the best option for global performance simulations. For a semi-empirical model, can be applied different approaches attending to the time dimension: transient, steady-state, and quasi-steady. An extensive review of the state of art regarding transient modeling of refrigerators was carried out by Hermes and Melo (2008).

A transient approach for the whole system is the best option with regard to the accuracy since it models what actually happens. Hermes and Melo (2008) developed and validated an accurate semi-empirical model by applying a transient approach to a two-compartment refrigerator. A transient approach is the one that has the ability to assess the energy losses due to the cycling working mode, which are hard to assess even experimentally. Its worst feature is the high required computational cost, which is not reliable for practical designing or optimization purposes, as explained Hermes et al. (2009).

On the other hand a steady-state approach is the fastest one in terms of simulation time, but its accuracy is also much lower. This approach assumes the air inside compartments to be in steady-state conditions of temperature, therefore for dynamic results, e.g. run time ratio, only approximations can be obtained. For the vapor compression system, this approach assumes steady-state running mode, which is actually rather correct. In fact, it can predict accurately static parameters and results. Gonçalves et al. (2009) developed a model for a refrigerator, following a steady-state approach, which consisted of same components as Hermes and Melo (2008), but now the modeling approach of each of these sub-models was much simpler: all the sub-models were formulated according to lumped models, excepting the heat exchangers which were discredited in the number of refrigerant states (three for condenser and two for the evaporator).

Quasi-steady approach consists on modeling a transient process as a serial of steady states. It is a very useful assumption since it applies just the required approach to each system, saving a lot of computational cost compared with a transient approach but retaining most of its accuracy. A transient approach is necessary to describe the air temperature evolution in the cabinets, cycling operation of compressor and influence of controlling system. However, the vapor compression system does not need a transient approach since the time it needs to get steady-state conditions is much lower than cabinets system. Borges et al. (2011) argued same ideas and developed a quasi-steady approach for a dynamic simulation of a refrigerator by using same lumped sub-models as used by Gonçalves et al. (2009) for the vapor compression system, while an algebraic transient model was devised for the refrigerated compartments, based on the work of Hermes and Melo (2008).

In present work, authors use a model developed by Martínez-Ballester et al. (2012) for a household two-compartment refrigerator frost-free following a quasi-steady approach for modeling a dynamic operation. It uses a novel approach for the refrigeration loop model which consists in using a performance map of the vapor compression unit. This map contains the main performance data of the unit which depends on cabinets air conditions. This map is generated with a high detailed tool for steady state simulation of vapor compression systems: IMST-ART (2010). In this way the map contains accurate data and by interpolation is obtained quite fast the corresponding steady-state performance for each operating condition.

An experimental validation of the model was worked out. In this validation studies were detected interesting transient phenomena that could be interpreted as energy losses (Coulter and Bullard, 1997; Krause and Bullard, 1996; Rubas and Bullard, 1995; Jansen et al., 1992). These phenomena happened when any control action of electronics took place. In order to understand reasons of these phenomena, a comparison between the real performance of a system and the equivalent quasi-steady performance of the same system is analyzed.

2. MODEL DESCRIPTION

The model proposed in this paper in is based on a quasi-steady approach and is divided into two coupled sub-models: cabinets and refrigeration loop. The paper only presents a brief description of the model, for a full description of the sub-models, reader is referred to Martínez-Ballester et al. (2012).

2.1 System description

The system analyzed is a typical frost-free household refrigerator with two compartments designed for the European market. The whole system can be studied as two sub-systems: cabinets and refrigeration loop.

2.1.1 Cabinets

Cabinets can be defined as compartments of the refrigerator and freezer designed to store food items at two temperature levels, 4ºC and -18ºC respectively. The refrigerator or fresh-food compartment is in the top whereas the freezer in the bottom. They are thermally insulated and can be considered as separate units but in contact through one wall. Doors are going to be assumed always as closed.

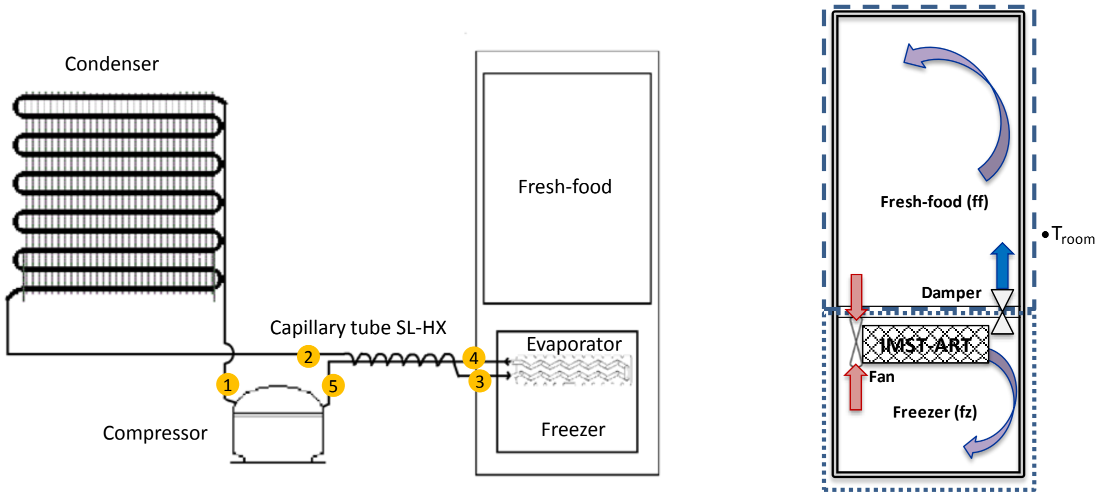

Figure 1: **(a)** Schematic of the household refrigerator studied. **(b)** Schematic of the cabinets and elements considered in the cabinets model

2.1.2 Refrigeration Loop

Refrigeration loop is the system which is employed to cool down the cabinets by using a vapor compression system. Figure 1(a) shows a schematic of the refrigeration loop that consists of: a single-speed reciprocating-hermetic compressor; a finned-tube no-frost evaporator; a natural draft wire-and-tube condenser; and a concentric capillary tube-suction line heat exchanger. The system uses R600a as working fluid.

2.1.3 Electronics: Temperature and air flow control

The air temperatures in the fresh-food and in the freezer cabinets are controlled by electronics. The electronics controls damper, fan and compressor. The compressor operation (on-off) only depends on the freezer temperature. In the freezer cabinet there is a centrifugal fan which supplies an air flow to the freezer and fresh-food cabinets through a distribution system. The fan is always working provided that compressor is running. Fan has also a single speed motor.

The fresh-food cabinet temperature is controlled by a damper that allows or not flowing the air to the fresh-food cabinet. When damper is open, 30% of the total air flow comes into the cabinet. This value was experimentally estimated. The damper remains closed unless the fresh-food cabinet needs to be cooled down regardless of the compressor operation. The damper position affects slightly to the hydraulic performance of the fan, so the air flow rate blown by the fan varies slightly when damper is closed.

2.2 Numerical Scheme

The cabinet sub-model describes the transient temperature evolution of the refrigerator compartments over time adopting a lumped-capacitance model for each control volume. The vapor compression system in the cabinet model is replaced with a "black box", which will be analyzed in next sub-section.

The refrigerator is divided into two separated cabinets: fresh-food (*ff*) and freezer cabinet (*fz*). Figure 1(b) shows the control volumes considered in this model. The energy conservation equation is applied to each air volume. The system of linear differential equations is discretized following a finite-difference based method resulting in an explicit equation for each air cabinet temperature. For full description of equations reader is referred to Martínez-Ballester et al. (2012).

In order to represent the refrigeration loop, the simulation software IMST-ART (2010) for vapor compression systems has been used. This model assumes steady performance. For further information about this model reader is referred to Corberán et al. (2002) and Martínez-Ballester et al. (2012).

For dynamic simulation of the whole refrigerator, both sub-models are coupled resulting in a quasi-steady approach, by assuming steady conditions for the refrigeration loop sub-model at each time step. First step of the global dynamic model is the unit map generation that is worked out by simulating a parametric study in IMST-ART. The parameters for this parametric study have to be the variables that could vary the working operation regime of the refrigeration system such as air temperature and humidity at evaporator inlet, and even air flow rate. In the current paper only parameter is the air temperature at evaporator inlet which is calculated according to Equation (1).

$$T_{in} = R_v\, T_{ff} + (1 - R_v)\, T_{fz} \tag{1}$$

In Equation (1) R_v corresponds to the air flow ratio that is supplied to the fresh-food cabinet. IMST-ART simulates and exports all the desired performance results as function of the parameters analyzed. Once the unit map is available, no more simulations are needed for the refrigeration loop since they content all the required data. The unit maps represent simple and accurate data about refrigeration loop.

The cabinet model needs from refrigeration loop sub-model just the air temperature at evaporator outlet. Other performance parameters will be used for energetic analysis. The cabinet model searches in the unit map file the operating point corresponding to the inlet temperature, calculated with Equation (1), in such a time. Then, the outlet conditions and performance parameters (COP, compression power) are obtained by linear interpolation.

This novel methodology does not requires that refrigeration loop model runs each time since the unit map contains all the possible working points along the simulation time saving a huge amount of simulation time. Notice that refrigeration loop model is the model that more computational cost requires since it is the most complicated one. Most of models available in literature include the refrigeration loop model calculations inside the global dynamic model, regardless the used approach or assumptions used. From the authors' point of view, the use of a unit map containing the refrigeration loop performance for the whole range of operating conditions has many advantages with regard to: accuracy, robustness and computational cost.

3. MODEL VALIDATION

3.1 Experimental Set-Up

The system tested corresponds to a system present in the European market like the one previously described. The system was measured in a climatic chamber where temperature was kept constant to 25 °C. Electronics was operating to satisfy the set point of 4 °C and -18 °C in fresh food cabinet and freezer respectively. The cabinets were empty and the system was running sufficient time to assume that air inside cabinets was quite dry and no frost appeared in evaporator along the test. The test started (t=0) 48 hours after first compressor start-up and the model started simulating from this point.

In the air loop all the temperature values were measured by using T-types thermocouples, installed with copper cylinders in the terminal. Air temperature at following locations was measured: evaporator inlet and outlet; three thermocouples distributed along the fresh-food cabinet height; one thermocouple in each drawer of the freezer; ducts of returning air from fresh-food cabinet. Regarding refrigeration loop, T-type thermocouples were used to measure the temperature in the numbered points of Figure 1(a). All the temperatures measurements have an uncertainty of ±0.5 K. For refrigeration loop, piezometric transducers were used to measure pressure at discharge and suction of the compressor with an uncertainty of ±1.4 kPa and ±0.524 kPa respectively. Power input of the whole system was measured along each test with an uncertainty of ±0.5%. It was also monitored the controlling signals of fan and damper.

The temperatures of fresh-food (T_{ff}) and freezer (T_{fz}) cabinets were evaluated as the arithmetic mean value of all the corresponding air temperature measurements in the cabinet. The evaporator air inlet temperature ($T_{a,in}$) is calculated according to Equation 1 assuming for R_v a value of 0% (closed) or 30% (open).

3.2 Validation

Figure 2 shows the comparison between modeled and experimental cabinets temperature evolution along the time, where the compressor and damper signals are also plotted. While compressor is off, the predicted temperature is very good, which means that the thermal inertias (C_i) and thermal conductance (UA_i), which were experimentally determined by Martínez-Ballester et al. (2102), are correctly adjusted. Tough the hysteresis of control system looks to be between -21°C and-23°C, actually is between -21.5°C and -22.5°C, since it is when compressor starts-up and stops. When compressor starts-up the model prediction is interrupted (region 1) because of transient effects that has a large impact on the temperature evolution. Phenomena that cause the transient response will be discussed in next section. Since the model assumes a quasi-steady operating regime, the modeled temperature would start to drop as soon as the compressor and fan are switched on. Predicted results would cause a significant delay in the temperature evolution that makes the model to fail when compressor starts up. That is the reason why model was not used for region 1 of figure 2. When transient effects disappear (between regions 1 and 2), prediction of model is good.

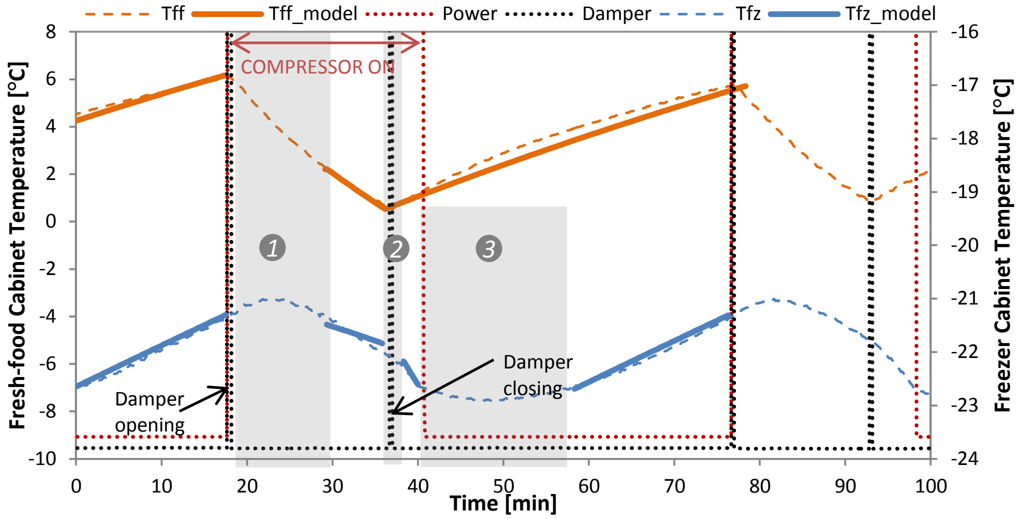

Figure 2: Comparison of modeled and experimental values for the temperature in freezer and fresh-food cabinets.

Due to same reasons, when the temperature is actually dropping, the model curve for freezer temperature is again interrupted when the flap closes (region 2). This damper closing induces a discontinuity in the air temperature at evaporator inlet. It changes from being a mixture of air from freezer and fresh-food cabinet to be an air flow that comes only from the freezer. The result is again a transient response of the refrigeration loop, which a prior is less drastic. This interruption in predicted results is shorter than before, and its prediction is good when model is again running (between regions 2 and 3). The air of fresh-food cabinet is now isolated form evaporator and its prediction is quite good until next damper opening.

When compressor stops, damper is also closed but freezer temperature is still decreasing due to thermal inertia of evaporator. This effect is also neglected by the model which cannot predict correctly the freezer temperature in region 3 due to these effects not taken into account. However, prediction of fresh-food cabinet is well predicted in region 3, since damper is closed and there is not connection with the evaporator.

Another important source of error in the predicted results is due to the slow dynamic of cabinets. A small error in definition of hysteresis temperatures definition would introduce an important delay in the results with regard to the actual values.

Figure 3 shows the comparison between experimental and predicted power input for the same time window as shown in Figure 2. Since the illustrated power input is directly related to previous showed temperatures, the predicted power only appears where the temperature has been simulated; out of transient conditions. While damper is open, the model matches very well with experimental values. After damper closing the difference is larger and looks like actual power input is still decreasing due to some transient phenomena, the steady value is not reached yet.

An interesting fact, which will be analyzed below, is the increase in power input with regard to a quasi-steady performance when there is any control action, either compressor start-up or damper closing. The power increase when the compressor starts up was analyzed by Coulter and Bullard (1997), who identified them as On-Cycle losses. Second increase could be understood in same way since phenomena are the same.

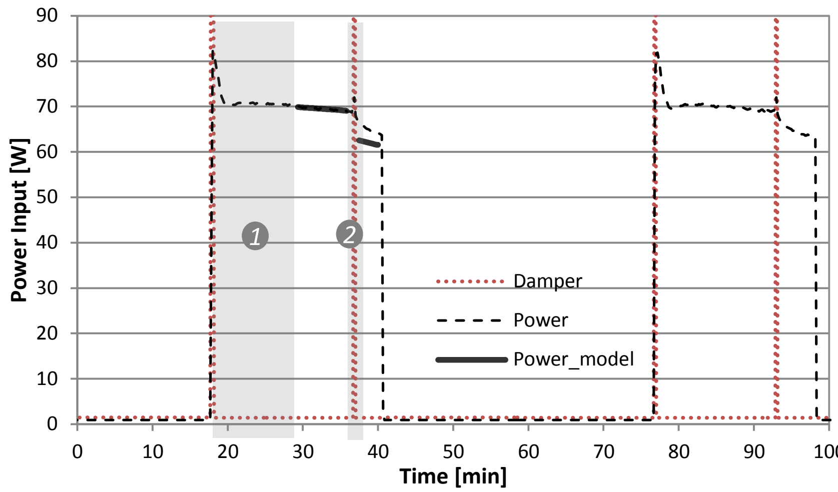

Figure 3: Comparison of modeled and experimental values for the power input.

4. IMPACT OF REGULATION ON PERFORMANCE

Regarding strategies for energy consumption reduction, a useful analysis is comparison of the actual performance against a quasi-steady model. This analysis can provide interesting regions with potential to reduce the energy consumption. To this end, the quasi-steady approach, previously described is a powerful tool. Main disadvantage of the model (prediction along the time) is not here a problem since the model now uses the experimental value of the air temperature at evaporator inlet. Thus, only data from the unit map is used.

For energy consumption analysis of this kind of systems, the best parameter is the COP. Notice that no cooling capacity is required but a thermal load. For this kind of systems an energy consumption reduction is directly related with an efficiency increase. This section compares the real COP (COP_{exp}) and the COP (COP_{st}) in steady-state conditions by using the simulation tool described in the previous sections. For steady-state conditions both values should be the same, whereas for transient periods the difference could be considered as an energy loss or not depending on the phenomenon involved. The COP_{exp} is defined as follows,

$$\mathrm{COP}_{\mathbf{exp}} = \dot{Q}_{air} \big/ \dot{W} \qquad (7)$$

COP_{exp} depends on the capacity and the total power input. For capacity evaluation, the air-side heat transfer has been used because is the useful energy that is released from the air of cabinets. It is only equal to the cooling capacity in the refrigerant-side for steady-state conditions. The cooling capacity has been calculated by using the experimentally estimated value of the air flow rate and the air temperature at both inlet and outlet evaporator.

Figure 4 shows the results, where two regions appear: in the first one, which takes 18 min, the compressor starts-up and the damper is open; whereas in second one that takes 4 min the damper closes at the beginning and compressor is still running. When compressor starts-up a significant difference between COP_{exp} and COP_{st} is observable. If this difference is understood as efficiency losses, these energy losses were estimated to be 18% of the steady-state total energy consumption. This value agrees quite well with those reported by Coulter and Bullard (1997), who estimated that their real refrigerator operation was between 5% and 25% less efficient than corresponding quasi-steady machine. Reasons for these differences are the following:

- The thermal inertia of several devices such as compressor and heat exchangers. In the compressor start-up, the evaporator wall absorbs energy from the refrigerant capacity so that only the difference is released from the air. This idea is depicted in Figure 4, where experimental COP from refrigerant side ($COP_{exp,ref}$) is greater than COP_{exp} (notice that power input is same for both COP definitions). In the figure is not plotted any value of $COP_{exp,ref}$ close to the start-up because estimation of refrigerant mass flow-rate (based on compressor's performance tables) during the star-up is not reliable.

- The redistribution of the refrigerant that migrated during the off-cycle. To reestablish the pressure the difference between evaporator and condenser requires a pumping energy that is consumed each time the compressor switches on.
- The process of redistribution of refrigerant charge in the whole system together with the temperature evolution of heat exchangers, imposed by thermal inertia, can lead to the system to work in condition less efficient than corresponding steady operation.

However, not all the differences in efficiency can be considered as energy losses. Thermal inertia is actually a conservative process though not its impact on the electric consumption through the vapor compression system. We only consider as source of energy losses during the on-cycle those due to the redistribution refrigerant charge. These sources of energy losses have been studied by several authors (Coulter and Bullard, 1997; Krause and Bullard, 1996; Rubas and Bullard, 1995; Jansen et al., 1992).

The quasi-steady evolution shows that COP decreases with the time. This is due to the fact that air is in a closed cabinet so that its temperature decreases with the time.

When the damper closes a discontinuity in air at the evaporator inlet, explained in previous section, occurs. Consequence of this discontinuity, a difference between COP_{exp} and COP_{st} appears again. Reasons could be explained with same phenomena above mentioned.

Furthermore, when the system is working in steady conditions the COP is smaller than the COP when damper is opened. Now, air temperature at evaporator inlet is much lower than before so pressure ratio will rise and COP will decrease. This observation points out the idea that if this regulation system is used, the damper closing should be avoided because clearly introduces energy losses. A solution could be changing the air flow ratio supplied to the fresh-food cabinet with a value that synchronizes the temperature evolution of both freezer and fresh-food cabinets.

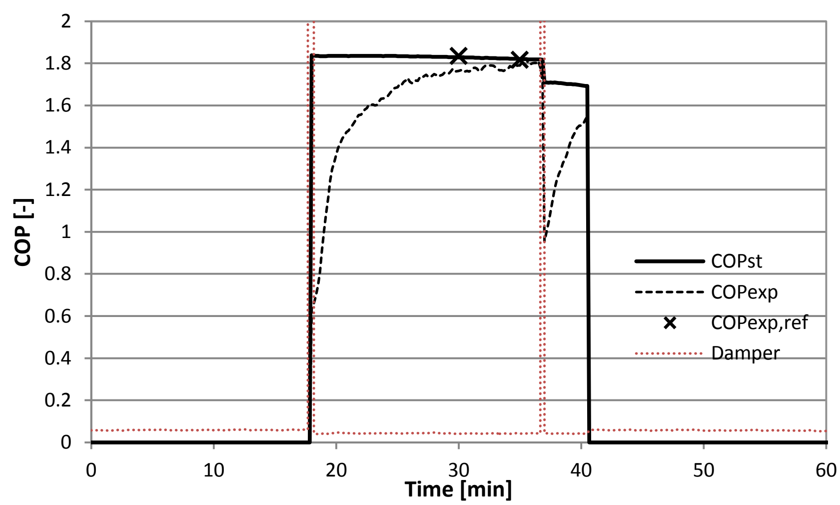

Figure 4: Comparison of steady-state COP_{st} and experimental COP_{exp}.

Finally, other heat losses related to the off-cycle are consequence of the migration of refrigerant when compressor stops. These losses only exist in systems without a solenoid valve that prevents this migration. When refrigerant migrates and compressor is off, refrigerant reaches an equalized pressure and all the refrigerant is condensed at evaporator because is the coldest component. For these conditions the saturation temperature is warmer than the air at freezer cabinet. This effect, what actually produces is an increase in the heat load that the refrigeration loop has to release from air along the whole running period.

5. CONCLUSIONS

The paper employs a novel methodology to simulate the dynamic performance of a household refrigerator frost-free of two compartments. The model applies a quasi-steady approach by coupling two sub-models: cabinets model and

refrigeration loop. For the cabinets model a transient sub-model has been devised whereas for refrigeration loop a commercial simulation tool has been used, IMST-ART. This methodology offers high accuracy, high robustness, and low computational cost.
The model has been validated against experimental data and the regulation has been analyzed in terms of efficiency. Following conclusions can be drawn:

- Impact of transient phenomena when compressor starts-up and damper works is not negligible in the temperature evolution of cabinets. This fact makes that a quasi-steady based approach cannot predict correctly dynamic performance, since large time delays will result in predicted values.
- Nevertheless when quasi-steady conditions are reached, the model predicts results with negligible differences.
- Transient phenomena have been detected as responsible of leading to the system to work less efficiently along almost the whole on-cycle. The maximum value for energy losses was 18%.
- The energy required to redistribute the refrigerant that migrated during the off-cycle is considered as the main energy losses source. Thermal inertia of system had an important impact on the performance.

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