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Research Publications

DYNAMIC MODEL OF A HOUSEHOLD REFRIGERATOR BASED ON A QUASI-STEADY APPROACH

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Abstract

The present work presents a model for the dynamic simulation of a household refrigerator. This model is based on a quasi-steady state approach and it is divided into two sub-models: cabinets and refrigeration loop. The cabinets and the refrigerator loop are modeled following transient and steady-state approaches respectively. This model applies a lumped-capacitance model for each cabinet, which describes the air evolution inside refrigerator. The refrigerator loop is simulated by the simulation tool IMST-ART. This software models the steady-state performance of the whole system. The electronics have been also implemented in the model as temperature control device. Empirical parameters needed by the cabinet models were experimentally obtained such as the thermal capacitances and conductances. The paper presents a novel methodology to model the whole system by coupling both sub-models trough a performance map of the unit. The unit map is previously generated by IMST-ART and the cabinet models only interpolate from this map the required performance parameters for each time step, allowing a fast, accurate and robust dynamic calculation. Cabinets model requires some empirical parameters to perform calculations over the time. These parameters are read from a unit map generated by IMST-ART which contains performance results of the refrigerator loop.

1 Introduction

The purpose of a refrigerator is to preserve the quality of perishable food products. Usually, household refrigerators have two main compartments to preserve food in two temperature levels: fresh food compartment and freezer. Freezer allows storing food products for large periods. The system to cool down or just to keep the low temperatures of the air inside the compartments is based on vapor compression systems. The electric energy consumption of these systems represents the highest electric consumption for a standard house in Spain [1], 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 are going to be always 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. Furthermore, a simulation tool allows exploring easily the potential of new ideas, designs, components…

From a modeling point of view, a refrigerator consists of two main elements: cabinets and the vapor compression system. The phenomena which takes place in refrigeration system is quite complex: two-phase flow in heat exchangers, choked flow in the capillary tube, refrigerant flowing through the compressor… 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.

Performance simulations should be able to evaluate the impact of the different integrating components on the dynamical and static performance results and parameters. Dynamical performance parameters and results are those which have an impact on the global energy consumption and are consequence of a cycling system operation: off time, run time ratio, cycling energy losses, and global energy consumption. Static performance parameters and results are those which correspond to the steady-state evaluation of the refrigeration system given indoor and outdoor conditions, namely: COP, power input, refrigerant charge, cooling capacity, evaporator superheat, condenser subcooling… Parameters like superheat or condenser are very important in the vapor compression system design or selection of compressor condenser, capillary tube and other elements. Refrigerant charge is a design parameter which always tends to be optimized.

In order to select a suitable simulation tool for designing purposes it is necessary to understand the main assumptions applied in the model. First approaches classification is: numerical analysis via commercial CFD packages, and semi-empirical models. CFD models have a large computational cost and they are not a good option for simulating the whole refrigeration system since the computational cost would be unfeasible. However they can be useful if either air flow distribution or air temperature distribution is studied instead of global performance parameters [2], [3].

Semi-empirical models are the best option for global performance simulations. For a semi-empirical model, can be also applied different approaches attending to the time dimension: transient, steady-state, and quasi-steady state. An extensive review of the state of art regarding transient modeling of refrigerators was carried out by Hermes and Melo [4]. 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 [4] 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. [5]. 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. [6] developed a model for a refrigerator, following a steady-state approach, which consisted of same components as [4], 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. Borges et al. [7] 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. [6] for the vapor compression system, while an algebraic transient model was devised for the refrigerated compartments, based on the work of Hermes and Melo [4]. 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. Just one dynamic result that cannot be assessed with such an approach is the cycling energy loss.

In present work, authors present a model for a household two-compartment refrigerator frost-free following a quasi-steady approach for modeling a dynamic operation. The model consists of two coupled sub-models: cabinet model and refrigeration loop model. Cabinet model uses an algebraic set of equations similar to those used by Borges et al. [7], however for the refrigeration loop model a novel approach is introduced 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 compartments air temperature and air flow rate. This map is generated with a high detailed tool for steady state simulation of vapor compression systems: IMST-ART [8], [9]. In this way the map contains accurate data and by interpolation is obtained quite fast the corresponding study-state.

2 System Description

The system analyzed is a typical frost-free household refrigerator with two compartments designed for European market. The main purpose of this system is to keep the air inside the cabinets with a temperature according to the set point imposed by the control device (electronics for the studied system). For the present work the doors are going to be assumed always as closed and cabinets without food. When electronics detects that air temperature is too warm to satisfy the set point, the compressor starts working. Once the compressor is running, the air stream is made to flow over the evaporator where the air is cooled down, dehumidified and then supplied to the compartments. Compressor is working until indoor temperature is low enough to assure set point, and then electronics switches off the compressor. This cycle keeps on repeating over the entire regime of operation.

The whole system can be studied as two coupled sub-systems: cabinets and refrigerant loop. A more detailed description of the components of the system is described below.

2.1 Cabinets

Cabinets can be defined as compartments of the refrigerator and freezer designed to store food items at low temperatures. They are thermally insulated and can be considered as separate units but in contact through one wall. The refrigerator compartments can be defined as fresh-food cabinet with 223 l of volume in the top, and the freezer compartment with 66 l in the bottom. The fresh-food cabinet is maintained at an average temperature around 4 ºC, while the freezer cabinet has to keep an average temperature of -18 ºC in the food.

The air temperatures in the fresh-food and in the freezer cabinets are controlled by electronics which ensure that each cabinet is maintained at a certain temperature called set point temperature. To achieve this purpose, electronics switches the compressor on or off and controls when the damper is open or closed. In the analyzed system, the control hysteresis of air temperature is defined as $T_{set\ point} \pm \Delta T$ K. This means that the compressor starts-up when the temperature of the freezer cabinet is a ΔT K higher than the set point temperature and it is turned off when this temperature is ΔT K lower than the set point temperature.

In the freezer cabinet there is a centrifugal fan which blows a constant air flow rate into the freezer and fresh-food cabinets. Fan is working provided that compressor is working, and it is always blowing approximately the same air flow rate. This fan is designed to ensure that the air reaches every space in the refrigerator. As a consequence, the air temperature stratification is reduced.

The fresh-food cabinet temperature is controlled by a damper that allows or not to the air flow passing from evaporator to the fresh-food cabinet. The damper remains closed unless the fresh-food cabinet needs to be cooled down. In that case, the damper opens and a certain fixed ratio of the total air flow rate is supplied to the fresh-food cabinet while the rest of air flows into the freezer. In the present analysis the air ratio that is blown into the fresh-food cabinet represents the 30% of the total air flow rate.

2.2 Refrigeration Loop

Refrigeration loop is the system which is employed to cool down the cabinets by using a vapor compression system. Fig. 1 shows a schematic of the refrigeration loop that consists of the following components: a single-speed 9.8 cm^3 reciprocating-hermetic compressor; a finned-tube no-frost evaporator that is fully made of aluminum; a natural draft wire-and-tube condenser; and a concentric capillary tube-suction line heat exchanger, illustrated in Fig. 2. The application of this capillary tube increases the cooling capacity and the outlet temperature of the suction line. As a consequence, sweating in suction line and slugging of the compressor are reduced. The system is charged with 46 g of R600a as refrigerant.

3 Sub-Models description

The model proposed in this paper is based on a quasi-steady state approach and it is divided into two sub-models: cabinets and refrigeration loop.

3.1 Cabinets Model

The cabinets 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 cabinets model is replaced as a "black box", which will be analyzed in next sub-section.

In the present model, the refrigerator is divided into two separated cabinets: fresh-food (ff) and freezer cabinet (fz). Fig. 3 shows the control volumes considered in this model, which are rounded. The air inside cabinets is getting warmer due to heat transfer phenomena from surroundings. First, heat load from outside gets into the refrigerator the refrigerator walls via free convection and radiation. Notice that radiation for these applications is not negligible due to the very low value of heat transfer coefficients. Secondly, heat is transferred across the cabinet walls by heat conduction. It is also considered the heat transfer via conduction between cabinets through the wall (*m*) that separates both compartments due to the temperature difference between them.

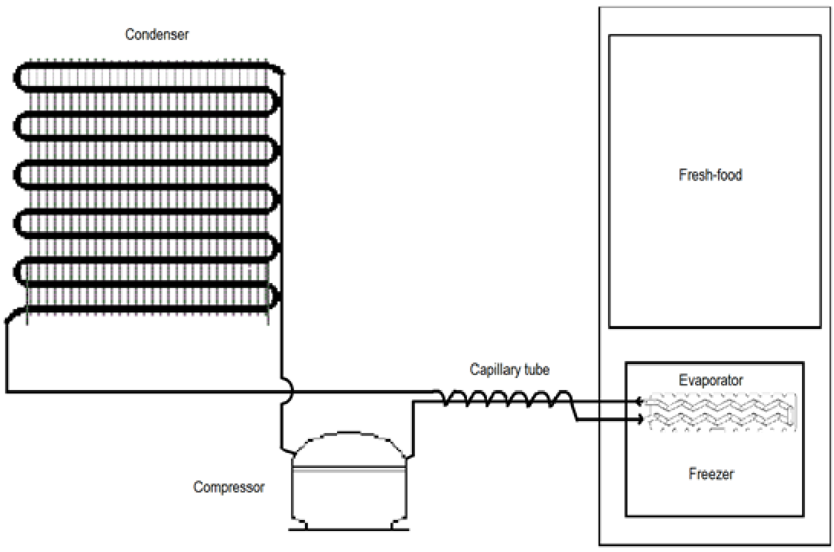

Figure 1. Schematic of a household refrigerator

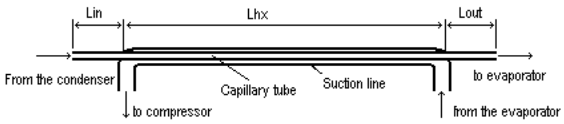

Figure 2. Schematic of a concentric capillary tube-suction line heat exchanger

Once the compressor is running, the warm air streams from these cabinets are made to flow towards the evaporator where the air is cooled down and dehumidified. Subsequently, the fan blows the cold air which is delivered to the freezer cabinet and to the fresh-food cabinet, if the damper is open, according to an air flow ratio defined in section 2. Finally, the system has a centrifugal fan that heats up the air flow that the fan blows when it is working. Hence, it can be considered as other heat source.

The energy conservation equation for an air volume can be expressed as follows,

$$\sum \dot{Q}_i - \sum \dot{W}_i = \frac{dE}{dt} \tag{1}$$

Applying the equation above to each cabinet, the differential Eq. (2) and Eq. (3) are obtained,

$$UA_{ff}\left(T_{room} - T(t)_{ff}\right) - \rho(t)_{ff}\dot{V}(t)_{ff}\left(i(t)_{ff} - i(t)_{evap,out}\right) + UA_m\left(T(t)_{fz} - T(t)_{ff}\right) + \dot{W}(t)_{fan\to ff} = \frac{dT(t)_{ff}}{dt}C_{ff} \tag{2}$$

$$UA_{fz}\left(T_{room} - T(t)_{fz}\right) - \rho(t)_{fz}\dot{V}(t)_{fz}\left(i(t)_{fz} - i(t)_{evap,out}\right) + UA_m\left(T(t)_{ff} - T(t)_{fz}\right) + \dot{W}(t)_{fan\to fz} = \frac{dT(t)_{fz}}{dt}C_{fz} \tag{3}$$

where Eq. (2) is referred to the fresh-food cabinet and the Eq. (3) to the freezer cabinet. The first term of both equations corresponds to the heat transfer through cabinet walls by convection and radiation, the second one is related to the cooling capacity of the refrigeration system and the third one represents the heat transmission through the wall that separates the cabinets. Notice that the values of fan power W_{fan} and the cooling capacity Q_{cool} are zero when the compressor is not working. The air flow rates are calculated by using the Eq. (4) and Eq. (5), where R_v is the air flow ratio that is supplied to the fresh-food cabinet when the damper is open.

$$\dot{V}_{fz} = \dot{V}_{fan}(1 - R_v) \tag{4}$$

$$\dot{V}_{ff} = \dot{V}_{fan}R_v \tag{5}$$

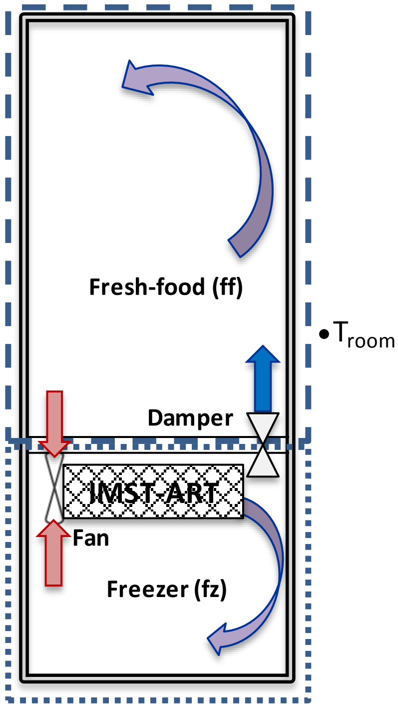

Figure 3. Schematic of the cabinets and elements considered in the cabinets model

Calculation of the air thermodynamic properties is performed with REFPROP subroutines from NIST [10] as function of its temperature and pressure, which is always assumed to be atmospheric. Finally, the values of thermal capacitances C_{ff} and C_{fz}, and thermal conductances UA_{ff}, UA_{fz}, UA_m are determined by experimental tests which are explained in the section 4.

Eq. (2) and (3) define a system of linear differential equations and a finite-difference based method has been used in order to discretize these equations. The numerical integration of Eq. (2) and Eq. (3) resulted in the following explicit equations,

$$T_{ff}^{i} = \frac{C_{air,ff}T_{out,evap}^{i-1} + UA_{ff}T_{room} + UA_{m}T_{fz}^{i-1}}{C_{air,ff} + UA_{ff} + UA_{m}} - \left(\frac{C_{air,ff}T_{out,evap}^{i-1} + UA_{ff}T_{room} + UA_{m}T_{fz}^{i-1} + R_{v}\dot{W}_{fan}}{C_{air,ff} + UA_{ff} + UA_{m}} - T_{ff}^{i-1}\right) e^{\left(\frac{C_{air,ff} + UA_{ff} + UA_{m}}{C_{ff}}\Delta t\right)} \tag{6}$$

$$T_{fzz}^{i} = \frac{C_{air,fz}T_{out,evap}^{i-1} + UA_{fz}T_{room} + UA_{m}T_{ff}^{i-1} + \dot{W}_{fan}}{C_{air,fz} + UA_{fz} + UA_{m}} - \left(\frac{C_{air,fz}T_{out,evap}^{i-1} + UA_{fz}T_{room} + UA_{m}T_{ff}^{i-1} + (1-R_{v})\dot{W}_{fan}}{C_{air,fz} + UA_{fz} + UA_{m}} - T_{fz}^{i-1}\right) e^{\left(\frac{C_{air,fz} + UA_{fz} + UA_{m}}{C_{fz}}\Delta t\right)} \tag{7}$$

where the coefficients $C_{air,ff}$ and $C_{air,fz}$ can be calculated as follows,

$$C_{air,ff} = \rho_{ff}^{i-1}\dot{V}_{ff}^{i-1}Cp^{i-1} \tag{8}$$

$$C_{air,fz} = \rho_{fz}^{i-1}\dot{V}_{fz}^{i-1}Cp^{i-1} \tag{9}$$

Eq. (6) and (7) provide the value of the temperature in an instant indicated by the superscript *i*. This state is calculated always as function of the temperature at the end of the previous time step, indicated by the superscript *i-1*, which is assumed to be constant over the time interval Δt.

When the compressor is switched off the cooling capacity Q_{cool} is zero therefore the $T_{out,evap}$ will be the same as the air temperature at evaporator inlet. Regardless the compressor operation, the fan can be running or not depending on the refrigerator temperature. When the fan is not running the values of V_{ff}, V_{fz} and W_{fan} are zero.

3.2 Refrigerant Loop model

The simulation study has been performed by means of IMST-ART software ([8], [9]). 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 heat pump. The global set of equations forms a complex system of non-linear equations AEs and DAEs, which is solved globally by a Newton-like solver. The global system of equations is solved using a standard solver based on the MINPACK subroutine HYBRD1, which uses a modification of M.J.D. Powell's hybrid algorithm.

Calculation of the refrigerant thermodynamic and transport properties is performed by REFPROP subroutines from NIST [10] for each refrigerant. The corresponding properties are then conveniently stored in a refrigerant data library. The required properties during the simulations are estimated by interpolation from the corresponding data file. Additionally, built-in tables allow the calculation of the properties of any usual secondary fluid, i.e. dry air, humid air, water and common brines.

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. Alternatively, IMST-ART allows definition of compressor by using the ARI performance polynomials provided by the compressor manufacturer, or even introduce directly catalogue performance data.

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 as a function of the pressure at the compressor suction and of the oil temperature. For this purpose, the compressor model also includes an empirical correlation of the oil temperature as a function of the condensing temperature and the compressor discharge temperature, Navarro et al. [11].The solubility curves of refrigerant in oil have been extracted from ASHRAE Handbook [12].

In IMST-ART the expansion device can be modeled as a thermostatic expansion valve or as a capillary tube, either adiabatic or non-adiabatic. In case of a household is widely used as expansion device a capillary tube which forms a counter-flow heat exchanger with the suction line. The capillary tube to suction line heat exchanger (CT/SL HX) modeled in IMST-ART corresponds to a concentric type. The model for the CT/SL HX, first uses an empirical correlation specific for this kind of capillary tubes (ASHRAE Handbook [12]) 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.

The model of heat exchangers is the most important sub-model of the whole system in order to get accurate results, and it is also the most complicated. 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 [13].

Regarding refrigerant side, in the case of the evaporator or condenser a 2-phase flow with phase change occurs. A steady 2-phase flow is considered to occur along the refrigerant path. The separated two-phase flow model is assumed. At a fluid cell, the number of equations to be considered is three: mass, energy and momentum conservation. Correlations from literature are employed for the evaporation and condensation heat transfer and friction coefficients. Both condenser and evaporator in a frost-free refrigerator are air-to-refrigerant heat exchangers. However condenser is a free-convection heat exchanger while evaporator is forced-convection. Nevertheless, refrigerant side for both heat exchangers are modeled identically and only air-side is differently modeled.

- Evaporator air side: In a frost-free refrigerator, the evaporator is a finned-tube heat exchanger. In an evaporator either dehumidification or frost can be present and these effects are modeled in IMST-ART, but without taking into account either ice layer or ice growth. The governing equations are those stated for the mass, energy and momentum conservation. The approach followed to treat the dehumidification process is the one proposed by Threlkeld [14].

Due to costs reduction, usually evaporators for household refrigerators are fully made of aluminum and use collarless fins. Collarless fins introduce a contact resistance between fin base and tube wall. To this end contact resistance is taken into the model by using a thermal contact conductance. The value for the thermal contact conductance is constant and equal to 9.5 kW m^{-2} K^{-1}, which is extracted from ElSherbini et al.'s experimental investigation [15].

- Condenser air side: The type of condenser modeled in the proposed model is a natural draft tube-and-wire heat exchanger, which is widely used for household refrigerators. Basically, this heat exchanger is like a finned-tube heat exchanger but with free-convection in the air side instead of forced-convection as occurs in a finned-tube heat exchanger. For the air-side uniform room temperature is assumed and no air flow is modeled. For the heat transfer coefficient between air and tube wall, Melo and Hermes' correlation [16] was employed, which includes fin efficiency and radiation effects.

4 Experimental characterization

Experimental steady-state and transient tests were carried out in order to obtain the empirical parameters of the model: C_{ff}, C_{fz}, UA_{ff}, UA_{fz}, UA_{m}. Steady tests were performed to obtain the overall thermal conductances whereas the thermal capacitances were determined by transient tests.

The objective of steady tests is to get a steady-state in both the freezer and the fresh-food cabinet. For these conditions, the overall thermal conductances are the only unknown parameters since the thermal capacitances are not present in a steady-state problem.

In order to carry out these tests two electrical heaters were placed inside the refrigerator compartments, therefore a heat flux was established from cabinets to surroundings. Thermal load could be varied by an electrical regulator to get different and independent temperatures values in each cabinet. These tests were carried out by using a different heat load while the air temperature was measured for each cabinet. Finally the UA_i values were obtained by fitting Eq. (2) and Eq. (3) to the experimental measurements. Table 1 shows the overall thermal conductances obtained.

In order to get empirical values of C_{ff} and C_{fz}, transient tests were carried out with the refrigerator running in standard conditions. However, only data corresponding to the period of time when compressor was switched off and the damper was closed were used to obtain the thermal capacitances. In these conditions both cabinets were heating up, and the refrigeration loop could be ignored. During these tests, temperature of each cabinet was measured and the thermal capacitances were obtained by fitting temperatures measured along the time to the Eq. (6) and Eq. (7), where the values of UA_i obtained above were used. The values obtained are shown in Table 1.

	Steady-state tests	Transient tests
UA_{ff} (W/K)	1.08 →	1.08
UA_{fz} (W/K)	0.46 →	0.46
UA_{m} (W/K)	0.037 →	0.037
C_{ff}(J/K)	--------------	18970
C_{fz} (J/K)	--------------	38744

Table 1. Overall thermal conductances and thermal capacitances experimentally obtained.

5 Coupling of models: quasi-steady approach

Previous section presented two different sub-models for refrigerated compartments and refrigerant loop respectively. For dynamic simulation of the whole refrigerator, both models are going to be coupled resulting in a quasi-steady approach because the dynamic behavior of the air inside compartments is calculated from results of the vapor compression system which is assumed to be steady for each time step.

Fig. 4 shows the steps and calculation flow for a simulation of the global dynamic model. First step is the unit map generation. Once, all the components of the vapor compression system have been modeled in IMST-ART a parametric study is carried out. The parameters for this parametric study have to be the variables that could vary the working operation regime of the refrigeration system, i.e. air temperature and humidity at evaporator inlet.

In the current paper, relative humidity is assumed always as uniform and corresponding to a quite dry value, 90%. Reasons are that doors are always closed, compartments are free of food and simulations are done in a time long enough to assume almost dry conditions. Thus, parametric studies will depend just on one variable, inlet air temperature at evaporator which is calculated according to Eq. (10).

$$T_{in} = R_v\, T_{ff} + (1 - R_v)\, T_{fz} \tag{10}$$

The parametric study has to be done as function of the air inlet temperature and ranging between the expected extreme values while compressor is running. IMST-ART simulates and exports all the desired performance results as function of the parameters analyzed. For the purpose of the dynamic model, the performance results needed to store are those which are required by the cabinet model in order to allow its calculation. It will be also interesting to generate all data related to power consumption and COP of the vapor compression system along the running time in order to work out future analysis of the performance behavior of the whole system: cycling energy losses, running time ratio, total running time and energy consumption. From now on, no more simulations are needed for the refrigeration loop, since all the data needed is in these maps. An example of a performance map for the unit modeled in the present work is shown in Fig. 5, where COP and air temperature at evaporator outlet are plotted as function of air temperature at evaporator inlet.

Once unit maps are available, cabinet model can be simulated whose calculation process is showed in Fig. 4. Initialization of the model requires data about: cabinet characterization (UA_i, C_i), initial conditions, and electronics parameters. When simulation starts, the modeled electronics determines if compressor is running or not. If compressor is not running the model does not need nothing from the refrigeration loop since it is off and calculation can continue to the following step. In case compressor is working, the evaporator is cooling down and the air outlet conditions depend on the refrigerant loop. Along the time, air crossing the evaporator will only vary its air temperature at inlet, thus this variable can be used as independent variable for the refrigeration loop model instead of the time. This choice allows developing in advance a unit map containing all needed data by the cabinet model to continue with the calculation. In order to perform calculations cabinet model needs from refrigeration loop model just the evaporator outlet temperature, 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 Eq. (10), in such a time. Then, the outlet conditions and performance parameters (COP, compression power) are obtained by linear interpolation.

This novel methodology of the proposed model does not requires that refrigerant 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 refrigerant loop model is the model that more computational cost requires since is the most complicated one.

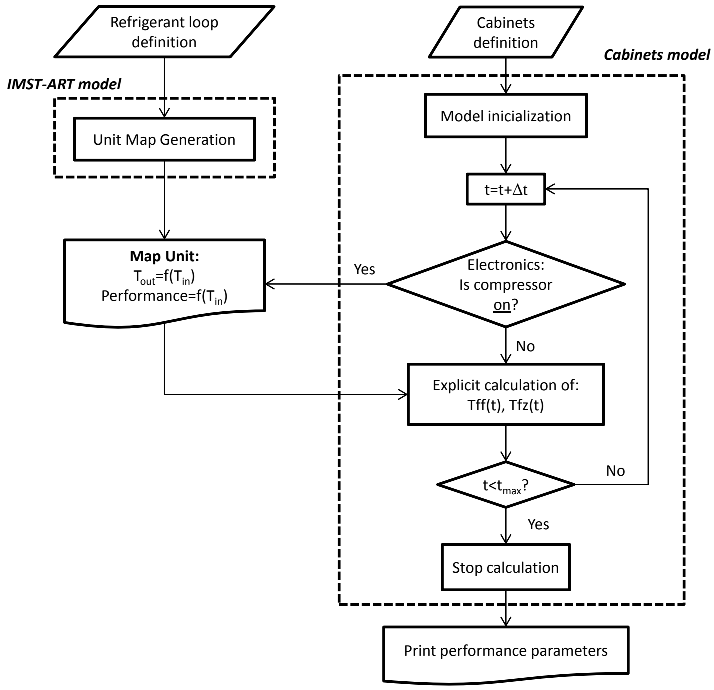

Figure 4. Schematic and calculation flow the global dynamic model.

Most of models available in literature include the refrigerant loop model calculations inside the global dynamic model, regardless the approach used to discretize differential equations or assumptions used. From author's point of view, the use of a unit map containing the refrigerant loop performance for the whole range of operating conditions has many advantages with regard to: accuracy, robustness and computational cost.

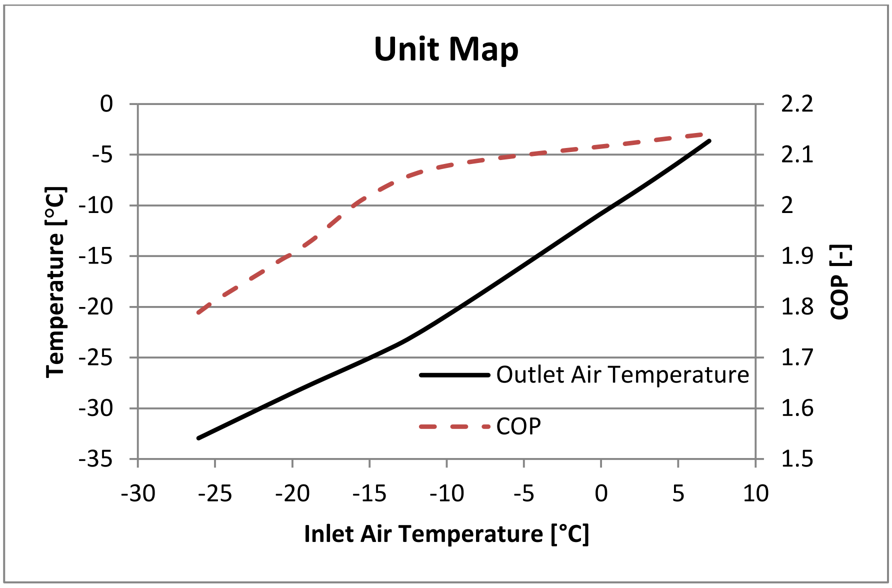

Figure 5. Map of performance parameters for the unit modeled as function of the air temperature at evaporator inlet.

6 Conclusions

The paper has presented a dynamic model for a household refrigerator frost-free of two compartments. This 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.

The model proposed uses a novel methodology to simulate the dynamic performance of the system, which consists of the following steps:

- The refrigerant loop performance is previously characterized by using IMST-ART, which generates a performance map of the unit. The unit map contains the performance parameters as function on the air inlet temperature.
- The cabinets model simulates the air inside compartments as function of electronics. When electronics switches on the compressor, the data required to continue with model's calculations are obtained by linear interpolation from the unit map.
- Performance parameters for each time step are written in a results file.

The global model is a semi-empirical model and the unknown parameters, to obtain experimentally, are: the overall thermal conductances and thermal capacitances. These coefficients were obtained experimentally.

The result is a model with the following advantages against others dynamic models which uses a vapor compression vapor model built-in in the dynamic calculation of air:

- Accuracy: It uses a quite detailed model, IMST-ART, for the most complicated system, i.e. the vapor compression system, which allows obtaining accurate static performance parameters, even parameters not possible to obtain with more simple approaches, e.g. total refrigerant charge.
- Robustness: A complicated model such as the refrigerant loop model may have convergence issues. If the model does not run along the dynamic simulation any convergence issues will be found at this step.
- Computational cost: A lot of simulation time is saved by interpolating the corresponding working conditions from the unit map against a model which have to use it each time step. Furthermore, the simulation time employed would be quite similar to the corresponding time spent by a steady-state approach but with a higher accuracy.

Nomenclature

C	thermal capacitance (J K^{-1})	Greek symbols	
E	internal energy (J)	ρ	density (kg m^{-3})
$\dot{Q}$	heat (W)	Subscript	
R_v	air flow ratio	ff	fresh-food cabinet
T	temperatura, K	fz	freezer cabinet
t	time (s)	in	inlet
UA	thermal conductance, (W K^{-1})	m	mullion wall
$\dot{V}$	volumetric flow rate (m^3 s^{-1})	out	outlet
$\dot{W}$	power (W)		

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