**UNIVERSITY OF SÃO PAULO
SÃO CARLOS SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DEPARTMENT**

Álvaro Roberto Gardenghi

**Transient modeling of vapor compression
refrigeration systems for domestic applications**

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Álvaro Roberto Gardenghi

Transient modeling of vapor compression refrigeration systems for domestic applications

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**UNIVERSIDADE DE SÃO PAULO
ESCOLA DE ENGENHARIA DE SÃO CARLOS
DEPARTAMENTO DE ENGENHARIA MECÂNICA**

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aplicações domésticas**

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Abstract

The increasing on energetic efficiency of household vapor compression refrigeration systems brings about a substantial impact in the energy consumption: about 17% of the overall electricity consumption worldwide is attributed to the refrigeration sector (including air-conditioning), being 45% the residential demand. A case study showing it is the Brazilian panorama, where such systems are responsible for approximately 27% of the residential electric consumption, representing about 8% of the whole country's demand. This issue is intensified due the low thermodynamic efficiency presented by these products. Therefore, industry and research institutes are dedicating increasingly efforts and time to develop and apply solutions to promote advances on systems' operation. In this work, two mathematical models are presented: one based on a thermal analysis with the application of the first law of thermodynamics and other including the evaluation of the refrigerant mass distribution in the system. It is also developed an experimental procedure to calculate the thermal conductance and capacity of each component of a domestic refrigerator (compressor, condenser, capillary tube, evaporator, cabinet), which are necessary input data for the models. Experimental data describing the transient behavior of the refrigeration system are also obtained to validate the mathematical models. Two types of cabinets were studied: one with two compartments, operating with R134a and associated to constant speed and variable speed compressors; and a horizontal freezer, with one compartment and operating with R290. The simulation results follow the same experimental trends and are very satisfactory when compared to the transient and mean time experimental results. Two variable speed control strategies were evaluated, with gains up to 31% in consumption reduction by using them. An entropy generation analysis was performed for each system component and the overall system. Parametric analysis were conducted to identify the influence of ambient temperature, refrigerant charge and goods inside the compartments on the refrigeration system performance. The presented models are very appropriate for the transient simulation of vapor compression refrigeration systems for domestic applications.

Keywords: Refrigeration systems. Refrigerator modeling. Simulation. Experimental procedure.

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Resumo

O aumento da eficiência energética de sistemas de refrigeração por compressão de vapor domésticos causa um impacto substancial no consumo de energia: cerca de 17% de todo consumo elétrico mundial é atribuído à área da refrigeração (incluindo ar-condicionado), sendo 45% a demanda residencial. Um estudo de caso é o panorama brasileiro, onde estes sistemas são responsáveis por aproximadamente 27% do consumo elétrico residencial, representando aproximadamente 8% de toda demanda do país. Este problema é intensificado pela baixa eficiência termodinâmica apresentada por estes produtos. Deste modo, várias indústrias e instituições de pesquisa concentram esforços e tempo para desenvolver e aplicar soluções que promovam avanços na eficiência dos sistemas. Neste trabalho, dois modelos matemáticos são apresentados: um baseado numa análise energética com a aplicação da primeira lei da termodinâmica e outro incluindo a avaliação da distribuição da massa de refrigerante no sistema. Também foi desenvolvido um procedimento experimental para calcular a condutância e a capacidade térmica de cada componente de um refrigerador doméstico (compressor, condensador, tubo capilar, evaporador e gabinete), em que são necessários dados de entrada nos modelos. Os resultados experimentais, descrevendo o comportamento transiente do sistema de refrigeração, também são usados para validar os modelos matemáticos. Dois tipos de gabinetes foram estudados: um de dois compartimentos, operando com R134a e associado a compressores de velocidade constante e outro variável; e um *freezer* horizontal, com um compartimento e operando com R290. Os resultados de simulação seguem a mesma tendência que os experimentais e são bastante satisfatórios quando comparados ao comportamento transiente e médias no tempo dos experimentos. Duas estratégias de controle de velocidade variável foram avaliadas, com ganhos de até 31% em redução de consumo. Uma análise da geração de entropia foi realizada para cada componente do sistema e para o todo. Análises paramétricas foram feitas para identificar a influência da temperatura ambiente, da carga de refrigerante e de mercadorias no interior dos compartimentos na performance do sistema. Os modelos apresentados são bastante apropriados para a simulação transiente de sistemas de refrigeração por compressão de vapor para aplicações domésticas.

Palavras-chaves: Sistemas de refrigeração. Modelagem de refrigeradores. Simulação. Metodologia experimental.

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List of symbols

Símbolos Arábicos

A	Area, m^2
C	Thermal capacity, J/K
c_p	specific heat, J/kgK
COP	Coefficient of performance
D	Diameter, m
$f_{\Delta P}$	Pressure loss on adiabatic portion of capillary tube
G	Mass flux, kg/m^2s
h	Enthalpy, J/kg
L	Length, m
M	Mass, kg
$\dot{m}$	mass flow rate, kg/s
n_p	Polytropic exponent
N_{rpm}	Compressor rotation, rpm
P	Pressão, Pa
Q	Heat transferred, J
$\dot{Q}$	Heat transfer rate, W
s	Entropy, J/kgK
$\dot{S}$	Entropy generation, J/Ks
T	Temperature, $^{\circ}C$
t	Time, s
U	Global heat transfer coefficient, internal energy, W/m^2K , J
u	specific internal energy J/kg

UA	Thermal conductance, W/K
V	Volume, m^3
v	specific volume, m^3/kg
V_s	Compressor volumetric displacement, m^3
W	Work, J
$\dot{W}$	Power, W
x	Quality
Z	Compressibility factor
z	Distance, m

Símbolos gregos

α	void fraction
ΔT_{sc}	Degree of sub-cooling, $^{\circ}C$
ΔT_{sh}	Degree of superheating, $^{\circ}C$
ΔT	Temperature difference, $^{\circ}C$
η	Efficiency
ρ	Density, kg/m^3
σ	Solubility
τ	Time constant, s

Sub-índices

1	Compressor inlet/suction line outlet
2	Compressor outlet/condenser inlet
3	Condenser outlet/capillary tube inlet
3i	End of capillary tube adiabatic portion
4	Capillary tube outlet/evaporator inlet
5	Evaporator outlet/suction line inlet

<i>amb</i>	Ambient
<i>avg</i>	Average
<i>c</i>	Related to condenser
<i>cap</i>	Related to capillary tube
<i>Carnot</i>	Related to Carnot cycle
<i>com</i>	Related to compressor
<i>cond</i>	Condensation
<i>conv</i>	Convection
<i>crit</i>	Critical
<i>d</i>	Discharge
<i>dis</i>	Dissolved
<i>e</i>	Related to evaporator
<i>eff</i>	Evaporator on fresh-food compartment
<i>efr</i>	Evaporator on freezer
<i>evap</i>	Evaporation
<i>ff</i>	Related to fresh-food compartment
<i>fr</i>	Related to freezer
<i>g</i>	Global/goods
<i>gen</i>	Generation
<i>H</i>	Hot source
<i>i</i>	Internal
<i>ihx</i>	Related to the internal heat exchanger
<i>in</i>	Inlet
<i>l</i>	Liquid
<i>L</i>	Cold sink
<i>n</i>	New

<i>o</i>	Oil
<i>off</i>	Off
<i>on</i>	On
<i>out</i>	Outlet
<i>real</i>	Related to the real system
<i>s</i>	Isentropic process/suction
<i>sat</i>	Saturation
<i>sc</i>	Sub-cooling
<i>sh</i>	Superheating
<i>tot</i>	Total
<i>v</i>	Volumetric/vapor
<i>w</i>	Wall between freezer and fresh-food compartment
<i>wc</i>	Condenser surface
<i>we</i>	Evaporator surface
<i>weff</i>	Surface of the evaporator on fresh-food compartment
<i>wefr</i>	Surface of the evaporator on freezer

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1 Introduction

Domestic vapor compression refrigeration systems are widely used, ensuring mainly food conservation and thermal comfort. It is estimated that there are approximately 1.5 billion units of domestic refrigerators in operation worldwide. The refrigeration sector is responsible for about 17% of the overall world's electricity consumption, being 45% of it attributed to the residential demand (IIR, 2015). In Brazil, these systems are responsible for approximately 27% of the residential electric consumption (PROCEL & Eletrobrás, 2007), corresponding about 8% of the total country's demand (Empresa de Pesquisa Energética, 2017). These facts emphasize the importance of improvements on such systems. The majority of these systems are equipped with compressor (controlled by a mechanical or electronic thermostat), condenser, capillary tube and a heat exchanger with the suction line (internal heat exchanger), evaporator and cabinet. A traditional control logic on household refrigerators is the on/off, where two limit temperatures are preset: when the temperature inside the refrigerated compartment reaches the inferior limit, the system is turned off and, on the other hand, reaching the superior one, the system is turned on. The optimization of such systems can be obtained through improvements on insulation, compressor efficiencies and heat exchangers heat transfer coefficients, applying variable speed compressors associated to control logics and fitting the correct refrigerant charge. These improvements have reduced the energy consumption significantly in recent years (Binneberg *et al.*, 2002). However, in the global operation, these improvements on each component separately cannot ensure the expected effects (Negrão & Hermes, 2011), due the complex interactions between them. To evaluate the refrigerator's behavior with modifications and even project parameters, several experimental tests are carried out, frequently based on trial and error, becoming this process expensive and time demanding. This way, a well validated mathematical model of household refrigerators is very important, allowing engineers to obtain reliable analysis faster and giving support to the experiments.

In the present work, two lumped, discrete, semi-empirical and transient mathematical models based on the work presented by Jakobsen (1995) are developed. The first one, called thermal model, considers mainly the energetic aspects on the system, with the application of the First Law of Thermodynamics, and can provide important parameters such as COP, consumption, internal air of compartments and components surfaces temperatures, compressor power, refrigeration capacity, duration of on and off periods, pressures on condenser and evaporator when the system is on and an entropy generation analysis. The second model, called capacitive, provides the mentioned results for the thermal model as well, besides the results for the pressures on condenser and evaporator on the

whole operation, sub-cooling and superheating, mass of refrigerant on the heat exchangers and mass flow rate on compressor and on capillary tube. The thermal model is processed faster, as it works with higher time steps, however, it considers only the mass flow rate on compressor, and the sub-cooling and superheating are inputs (acquired experimentally). The capacitive model takes into account the refrigerant mass variation inside the system components, so, it can describe better the transient behavior when the system starts and turns off. This models needs more inputs concerning the geometric parameters for the calculations.

In order to obtain the required input data of the above models, an experimental procedure is described for getting parameters such as thermal conductance (UA) and capacity (C) for each system component. Also it is developed an experimental procedure to acquire experimental data describing the transient behavior of the refrigeration system. These data are used for comparing the numerical results and in some way for model validation. Several simulations of pull-down and cycling tests were performed and the results were compared to the experimental ones. Besides, an entropy generation analysis was evaluated for each system component and for the whole system. Parametric analysis were conducted to identify the influence of ambient temperature, refrigerant charge and goods inside the compartments on the refrigeration system performance.

The components of the refrigerators tested were:

- i) two compartments, hermetic reciprocating compressor (with 163g of oil POE10) and a wire-and-tube condenser, both cooled by natural convection; a capillary tube and a heat exchanger with the suction line (internal heat exchanger); roll-bond evaporators (“box” type on freezer and “plate” on fresh-food compartment). The system is filled with 105g of R134a as refrigerant and controlled by an on/off mechanical thermostat. Two compressor were tested: a constant speed and a variable speed. Also, on the test with the first compressor, the defrost electric resistor was deactivated, and on the second, activated. A scheme of a two compartment refrigerator is shown in figure 1;
- ii) one compartment, hermetic reciprocating compressor with ventilation, skin condenser, capillary tube without internal heat exchanger, skin evaporator, 103g of R290 (propane) and an electric resistor for defrosting purposes when the system turns off;

There is also an analysis of different compressor control strategies and the possible gains that can be reached with its application. Using the variable speed compressor associated with variable speed strategies, gains up to 31% in consumption reduction, in comparison to the standard on/off constant speed compressor, were verified.

Figure 1: Scheme of a two compartment refrigerator.

Fonte: adapted from Klein, F. H. (1998).

1.1 Objectives

The main objective of this work is the development of transient numerical modeling procedures to simulate vapor compression refrigeration systems, focusing on domestic applications, such as refrigerators and freezers with one or two compartments.

In order to attain the main objective of the work, the following specific aims are proposed:

- Implementation of the simulation procedures in an open source programming language. This procedures include two models: the thermal model and the capacitive model;
- Consolidation of an experimental procedure to evaluate the necessary input parameters for the developed models;
- Study of the refrigeration system behavior with several parametric analysis including: influence of the ambient temperature, influence of refrigerant charge and goods inside the cabinets, analysis of entropy generation on system components, identification of the gains in performance by the use of variable speed compressors with desired control strategies.

The mathematical models and the computational programs involve the solution of mass and energy conservation equations as well as rate equations and heat transfer correlations. This way, a reliable and fast-processing computational program was developed,

which is able to provide important results for researchers, engineers and professionals of the refrigeration sector.

2 Literature review

2.1 Refrigeration systems

Refrigeration can be defined as the art or science related to the solids or fluids cooling for temperatures in lower levels than the ones available in a place and in a moment (Gosney, 1982).

According to Wang (2001), the refrigeration phenomena is defined as the process of removal and transference of heat from a thermal source, substance or ambient to be cooled in a specific temperature to another thermal reservoir at greater temperature, causing an energetic demand.

Among the types of refrigeration system, the following can be detached:

- Refrigeration by mechanical vapor compression
- Refrigeration by absorption
- Air cooled systems
- Refrigeration by ejection of vapor
- Thermoelectric refrigeration (Seebeck-Peltier effect)

Where mechanical vapor compression refrigeration systems are the most popular, on domestic, commercial and industrial applications

2.1.1 Thermodynamic concepts and principles related to Refrigeration

It is common knowledge that the heat transfer occurs naturally and spontaneously from the region with higher temperature to the lower one. But, on refrigeration, the interest is to remove heat from the colder thermal reservoir and transfer it to the warmer one, operating on the inverse direction of the temperatures gradient.

According to Clausius, the thermal energy cannot be transferred from a low temperature reservoir to another one in high temperature without an external energy supply to the process. Therefore, for each type of refrigeration system there is an energy cost and, different devices are responsible to insert this energy on the process. On the mechanical vapor compression refrigeration systems, this device is the compressor. In figure 2 is shown a scheme of the refrigeration process.

Figure 2: Diagram of the thermodynamic principles of a refrigeration system.

Source: adapted from Klein & Nellis (2012).

To evaluate the refrigeration system performance, it is used the parameter coefficient of performance (COP), defined as the ratio between the cooling effect (heat removed from the cold reservoir) and the net work (both in J) inserted on the system to its operation (Stoecker & Jones, 1985).

$$COP = \frac{\text{cooling effect}}{\text{net work}} = \frac{Q_L}{W} \quad (2.1)$$

With an energy balance on a control volume around the system, it is obtained:

$$Q_H = Q_L + W \quad (2.2)$$

So, rearranging the Eq. 2.1:

$$COP = \frac{Q_L}{Q_H - Q_L} \quad (2.3)$$

The ideal refrigeration system works with an working fluid following the Carnot cycle, this way, the COP is the maximum that can be reached, as the Carnot cycle is composed by processes with no irreversibilities. These processes, on figure 3, are:

- 1-2: Heat absorption at constant temperature (isothermal process)
- 2-3: Isentropic compression
- 3-4: Heat rejection at constant temperature (isothermal process)

- 4-1: Isentropic expansion

Figure 3: T-s diagram for the Carnot refrigeration cycle.

Source: adapted from Klein & Nellis (2012).

From the diagram:

$$Q_L = T_L(s_2 - s_1) \quad (2.4)$$

$$Q_H = T_H(s_3 - s_4) \quad (2.5)$$

Therefore,

$$COP_{Carnot} = \frac{T_L(s_2 - s_1)}{T_H(s_3 - s_4) - T_L(s_2 - s_1)} \quad (2.6)$$

As the processes 2-3 and 4-1 are isentropic:

$$COP_{Carnot} = \frac{T_L}{T_H - T_L} \quad (2.7)$$

Note that the refrigeration system has its efficiency reduced when the difference between the reservoir temperatures is increased.

Now, the efficiency of a real refrigeration system can be defined, as the maximum value for it is known. Observe Eq. 2.8.

$$\eta_{real} = \frac{COP_{real}}{COP_{Carnot}} \quad (2.8)$$

Other important parameters to evaluate the performance of a system are the cooling capacity and the power demand on compressor. Observe the P-h diagram of the cycle, on figure 4.

Figure 4: P-h diagram for the Carnot refrigeration cycle.

Source: created by the author.

The cooling capacity, in Watts, is defined as the product between the mass flow rate of the fluid and its enthalpy variation on the process of heat removal (1-2). The power demand on compressor, also in Watts, is the product between the fluid mass flow rate and its enthalpy variation on the compression process (2-3). Observe the Eqs. 2.9 and 2.10, respectively.

$$\dot{Q}_L = \dot{m}(h_2 - h_1) \quad (2.9)$$

$$\dot{W} = \dot{m}(h_3 - h_2) \quad (2.10)$$

This way, the *COP* can be also calculated according to Eq. 2.11.

$$COP = \frac{\dot{Q}_L}{\dot{W}} = \frac{h_2 - h_1}{h_3 - h_2} \quad (2.11)$$

2.1.2 Mechanical vapor compression refrigeration system

A mechanical vapor compression refrigeration system is normally composed by four main components: compressor, expansion device, condenser and evaporator (heat exchangers). So, the heat is removed from the refrigerated ambient, under lower temperature, and rejected to the external ambient, at high temperature, through the occurrence of thermodynamic processes with a working fluid, called refrigerant. Observe figure 5, showing the components layout.

Figure 5: Components layout of a vapor compression refrigeration system.

Source: adapted from Klein & Nellis (2012).

The component responsible for provide mass flow rate, adding energy to the system is the compressor. The compressor can be classified in positive displacement and dynamic, where, the most popular ones, among these groups, are: reciprocating (composed by a piston moving alternatively inside a cylinder, with the suction and discharge valves disposed to allow the compression), screw, scroll (based on rotating elements which reduce the gas volume) and centrifugal (based on a rotating element operating under centrifugal forces). The compressors can also be hermetic (where all of the components are inside a closed housing, and the vapor of refrigerant, which enters at low temperatures pass through the electric motor, to its cooling; the most common applications are domestic/commercial refrigerators and freezers and residential air conditioning systems), semi-hermetic (present removable head, being possible the execution of maintenance on valves and pistons) or opened (its crankshaft pass throughout the housing to couple with an external electric motor). The most popular compressors on domestic refrigeration are the hermetic reciprocating ones (Stoecker & Jones, 1985).

The component, in association with the compressor, that promote a pressure difference on the system is the expansion device. It can be a capillary tube or a valve (existing several types, where the most common are the thermostatic and the electronic). Capillary tubes are tubes with 1 to 6 *m* length and inner diameter of 0.5 to 2 *mm* and are the most popular expansion device on small refrigeration systems (domestic, light commercial and residential air conditioners). The thermostatic valves are mass flow rate regulators controlled by the degree of superheating of the gas on evaporator outlet (Stoecker & Jones, 1985) and, on the other hand, the electronic valves are controlled electronically.

Among the heat exchangers, the one responsible for the heat rejection to the external ambient is the condenser, normally wire-and-tube type in domestic refrigeration, associated with natural convection on lower heat exchange situations and finned tube for higher heat transfer cases, commonly applied on condensing units under forced convection with the compressor. The heat exchanger responsible for the heat removal from the refrigerated ambient is the evaporator, which, in domestic appliances, normally is roll-bond type with natural convection or finned tube type with forced convection, in frost-free refrigerators. Another configuration for both heat exchangers is the skin one, consisting on a coil passing through the internal structure of the refrigerator walls, where the condenser is on the external portion and, the evaporator, on the internal one.

The cooling effect in a vapor compression refrigeration system is based on the principle of evaporate the refrigerant at low pressure and, so, low temperature, where it should be lower than the one on the refrigerated compartment, therefore, by the fact that the fluid is two-phase, the heat transfer is maximized, due the high internal heat transfer coefficient. The fluid on evaporator outlet is vapor, so, to reestablish the cycle, it should return to liquid and reenter evaporator. The heat rejection to this aim occurs at a superior temperature, in comparison to the external ambient, therefore, the condensation is carried out at high temperature and pressure.

The pressure elevation from the evaporation to the condensation ones is through the compressor. Analogously, on condenser outlet, the liquid is at high pressure and, to reduce this parameter to the one necessary on the evaporation, there is the action of the expansion device.

The standard refrigeration cycle, on figure 6, is composed by the following processes:

- 1-2: Isentropic compression (compressor)
- 2-3: Heat rejection at constant pressure (condenser)
- 3-4: Isenthalpic expansion (expansion device)
- 4-1: Heat absorption at constant pressure (evaporator)

A complicating factor on such systems is the possibility of incomplete evaporation until the fluid reaches the compressor, this way, the compressor will receive liquid, and, it is possible that this liquid enters the compression cylinder, which is extremely dangerous for this component. Then, in some cases, a heat exchanger is assembled between the capillary tube and the suction line tube, which connects the evaporator outlet to the compressor inlet (internal heat exchanger). The hot fluid, coming from the condenser, exchange heat with the fluid on the suction line, warming it. Note that on this configuration there is

Figure 6: P-h diagram of the standard refrigeration cycle.

Source: created by the author.

a reduction of the enthalpy of the refrigerant entering the evaporator (observe figure 7). Depending on the project, it is possible to increase the cooling capacity of the system with this additional heat exchanger, since the higher temperature on compressor inlet reduces the mass flow rate, but there is the increasing of the enthalpy variation on evaporator. Normally, this gain occurs in lower applications, as domestic refrigeration.

Figure 7: P-h diagram of the refrigeration system with the internal heat exchanger and the configuration of components.

Source: adapted from Jakobsen, A. (1995).

There are several construction types for the heat exchanger between the capillary tube and the suction line, commonly considered counterflow for modeling. They are: capillary tube passing around the suction line, as on figure 7; capillary tube welded on the suction line and capillary tube passing by the interior of the suction line tube concentrically, observe figure 8.

Figure 8: Internal heat exchanger configurations.

Source: adapted from Mendonça K. C. (1996).

Another characteristic presented on figure 7 are the degree of sub-cooling and superheating. The first can be identified from the saturated liquid curve to the state 3 and the second has two parts: the useful superheating, that occurs on evaporator, from the saturation vapor curve to the state 5 and the non-useful, from state 5 to state 1.

The cycle of a real refrigeration system has different thermodynamic processes and characteristics, caused by losses due to irreversibilities present in several portions of the system, as heat transfer through a finite temperature difference; friction between mechanical elements and flow. Besides there is the pressure loss along the tubes, capillary tube and local, such as on compressor valves. There is also the pre-heating of refrigerant on compressor shell, decreasing the mass flow rate due to the increasing on specific volume; reflux in valves and leakages on circuit. See figure 9 showing a diagram more similar to the real.

Figure 9: P-h diagram of a real refrigeration system.

Source: adapted from Rangel, S. (2007).

According to Stoecker & Jones (1985), the components of a vapor compression refrigeration system present a great interdependence and complex interactions, this way, the performance of each one influence, in a difficult-to-predict way, the global performance. On figure 10, it is presented two diagrams showing how the components respond to some important parameters. The refrigerant charge and the ambient temperature influence the condensing and evaporating pressures and, these ones have a direct relation to the compressor operation and the capillary tube restriction. A system like this can be described by the condensing and evaporating pressures and the degrees of sub-cooling and superheating.

Figure 10: Interdependence between components and some parameters influence.

(a) Source: adapted from Mendonça, K. C. (1996).

(b) Source: adapted from Jakobsen, A. (1995).

2.2 Experimental procedure

Normally, on the modeling and simulation of refrigeration systems, some parameters related to the heat transfer must be measured. It can be accomplished with cabinet

characterization experiments.

Melo *et al.* (2000) compared two methods to determine the thermal conductance of the cabinet compartment: using the reverse heat loss rate measurement (NTB00119, 1992) and using heat flux sensors. The first consists on, with the refrigerator (one compartment) off in a climate chamber, put electric resistors inside the cabinet to dissipate power and warm its internal air. With the measurements of ambient and internal air temperatures, in steady-state, the thermal conductance can be calculated, according to Eq. 2.12. Observe figure 11.

$$UA = \frac{\dot{Q}}{\Delta T} \quad (2.12)$$

Where $\dot{Q}$ is the heat transfer rate through the walls, in W , which is equal to the power dissipated by the resistors (in steady-state) and, ΔT is the difference between the internal cabinet air temperature and the ambient one.

Figure 11: Representation of the the cabinet characterization test through the reverse heat transfer method.

Source: Melo *et al.* (2000).

The second method consists on distribute heat flux sensors along the cabinet walls. The authors used 31 sensors: 6 located on the gasket and 25 on the walls. With the sensors, the heat transfer rate ($\dot{Q}$) is acquired and the calculations are analogous. Both methods returned close results. With the heat flux sensors, the authors identified the regions where more heat is transferred, decreasing the efficiency of the refrigerator. These regions are the door and the side walls of the cabinet.

Several works evaluating the heat transfer on the cabinet were carried out. Gonçalves *et al.* (2000) measured the thermal resistances, and then, the conductances (which are the inverse of the resistances), through the reverse heat transfer method for a frost-free refrigerator with two compartments. The authors analysed the energy balance on each compartment to this aim. Thiessen *et al.* (2016) studied the performance of a similar

refrigerator comparing the standard configuration to the one with the presence of vacuum panels around the walls. The consumption tests showed that with extra insulation, gains up to 11% can be reached. They also evaluated the efficiency loss of the vacuum panels, due the air infiltration.

De Rossi *et al.* (2011) evaluated the influence of the refrigerant charge, on steady-state, of a vertical freezer with a cabinet, capillary tube, reciprocating compressor and wire-and-tube condenser with natural convection, wire-and-tube evaporator and R600a as working fluid. Experiments were carried out with the refrigerant charge in a range of 55 to 190g. The authors verified that with 190g, there was liquid return to compressor. They evaluated parameters such as the T-s diagram; cabinet internal air temperature; condensing and evaporating pressures; suction line temperature; sub-cooling and superheating; mass flow rate; compressor electric and heat power; refrigeration capacity and COP.

Yusof *et al.* (2018) tested a split-unit type air conditioner, with R22 focusing on the performance of the condenser under refrigerant charge variations and Datta *et al.* (2014) evaluated the performance of an automotive air conditioning system operating with R134a, with variations on refrigerant charge, compressor speed and air flow through evaporator. The authors investigated the influence of these inputs on the temperatures and pressures on compressor suction and discharge, cooling capacity, compressor work and COP. The results presented on De Rossi *et al.* (2011), Datta *et al.* (2014) and Yusof *et al.* (2018) show similar responses when the refrigerant charge is manipulated, according to the system size.

2.3 Refrigeration systems modeling

On the numerical modeling implementations, in general, three conflicting characteristics are evaluated and one of them is more valued according to the application: stability, rapidness and accuracy (Ding, 2007). The models for vapor compression refrigeration systems simulation are usually classified on groups such as: steady-state or transient; continuous or discrete; lumped or distributed parameters; theoretical, empirical or balanced (semi-empirical). Steady-state are the models which describe the system since the moment where the parameters are established, all energy fluxes and mass flow rates are constant. Transient models can represent the system dynamics since the condition of non-stable values to the steady-state. Continuous models are normally solved analytically and the mathematical formulation has no discontinuities, on the other hand, discrete models are normally treated on time domain and solved iteratively. Some simulation characteristics, such as the on/off operation of the refrigerator, can introduce discontinuities to the simulation. Lumped models treat the parameters through average values that represent

the phenomena in coherent ways. Pressures on heat exchangers and surface temperatures are commonly approached on this way in such models. Modeling with distributed parameters describes the components dividing them into several volumes, calculating the rates and applying the conservation equations on each one, therefore, with this model, the characteristics can be studied in function of the position on the element. Theoretical models, also called “white-box”, are the ones composed purely by physical formulations, with the application of the phenomena governing equations for mass, energy and momentum, becoming this approach more general and robust. Empirical (or “black-box”) models are more direct in terms of processing due the use of experimental values and correlations acquired experimentally to simulate the phenomena, but only works on the tested conditions. Finally, the balanced (or “gray-box”) models describes the system through a physical formulation with some parameters acquired experimentally, usually heat transfer coefficients, thermal conductances and capacities. As its name says, this type of modeling is more equilibrated, gathering part of the positive aspects of the other ones (Rasmussen, 2012).

Jakobsen (1995) developed two transient, discrete, lumped and semi-empirical models for transient simulation of a household refrigerator: the first, called as DynTherm model, consists on energy analysis, with the application of the First Law of Thermodynamics in control volumes on system components and another, called DynFlo model, including some aspects of the fluid dynamics inside the control volumes. The refrigerator on this work has one compartment (fresh-food) with 325 *L*, hermetic reciprocating compressor and wire-and-tube condenser with natural convection, capillary tube with internal heat exchanger, roll-bond “plate” evaporator and 80*g* of R134a as refrigerant, operating under on/off logic. The models follow a solution procedure composed by “blocks”, each one brings the modeling of a component, being the blocks co-dependent as well as the components on the refrigeration system functioning. The heat exchangers are modeled with a lumped approach, where at their sub-regions (superheated, two-phase and sub-cooled) the heat transfer and transport properties are considered constant and uniform for each time step. The equations are solved iteratively in an equation system, on each time step. Besides, the author also studied ways to interfere on the operation to obtain energetic gains. For example, the “Micloss” control, where a solenoid valve blocks the flow from condenser to evaporator when the system turns off; control of the duration of periods on and off and a steady-state analysis of the compressor rotation. Also discussed about the influence of the capillary tube length, refrigerant choice and use of fan.

Klein (1998) proposed a steady-state and discrete modeling of a domestic refrigerator (composed by one compartment, wire-and-tube condenser, roll-bond evaporator, capillary tube with internal heat exchanger and refrigerant R134a). Two types of a hermetic reciprocating compressor models were tested: an empirical one and other considering polytropic compression (semi-empirical), based on a considerable amount of experiments,

including conventional (standardized) and special ones, measuring values such as temperature and pressure inside the suction and discharge chambers. These tests were carried out with different combinations of condensing and evaporating temperatures, where parameters as electric power consumption, mass flow rate, housing temperature, suction temperature and pressure loss in suction and discharge were evaluated and related. Klein concluded that the semi-empirical model presented the best results.

On the modeling of condenser, it was divided into four regions: discharge line, superheated, two-phase and sub-cooled and the pressure loss along the tube was considered. Another characteristic of this modeling is a global approach for the capillary tube and the internal heat exchanger simultaneously, using the numerical modeling proposed by Dirik *et al.* (1994). The evaporator was treated as an isothermal and vertical plate, with the surface temperature equal to the evaporating one. Finally, on the cabinet model, the following heat transfer ways were considered: conduction through the gasket, one-dimensional conduction through the walls and the radiation on the external side of walls.

The external heat transfer on compressor was modeled as natural convection, comparing correlations for cylinder and sphere, where the second fitted better. For the condenser, the author concluded that the external heat transfer coefficient is dominant and compared the calculations using the correlations of Cyphers *et al.* (1958), Papanek *et al.* (1958) and Tanda (1997), obtaining better results with the first one. For the capillary tube and internal heat exchanger, the software CAPHEAT (Mezavilla, 1995) was used and, finally, on evaporator, the external heat transfer coefficient was obtained through the Churchill & Chu (1975) correlation, and the radiation was modeled through the classical theory, using view factors.

Also a mass inventory was proposed, considering the volumes of components and the fluid properties, analysing the optimal refrigerant charge. For the two-phase regions the following void fraction models were tested: homogeneous, Rigot (1973), Zivi (1964), Smith (1969), Premoli *et al.* (1971), Lockhart-Martinelli (1949), Tandon *et al.* (1985) and Hughmark (1962), where the last one performed better.

The model developed by Hermes (2000), considered separated sub-models for each component, providing an individual or a global analysis of the system. Besides, it is a model for steady-state or transient simulations. Several experiments were used to adjust this predominantly theoretical model. The compressor model also describes phenomena occurring inside it, including heat transferred between internal mechanical components and the refrigerant inside the housing, refrigerant conditions inside the suction and discharge chambers and temperature of internal mechanical components. The wire-and-tube condenser was considered a straight tube with uniformly distributed fins. The mass flow rate and heat transfer were modeled as one-dimensional and the pressure loss effect was

taken into account. A distributed model was elaborated for the roll-bond evaporator, allowing the calculation of the temperature distribution and the superheating of refrigerant at the outlet. A similar analysis was adopted for the internal heat exchanger. The cabinet model provides parameters as internal air, external and internal surfaces temperatures and the heat transferred by radiation.

Porkhial *et al.* (2005) proposed a distributed parameters transient model for a household refrigerator wire-and-tube condenser that describes the pressure, mass flow rate, heat transfer rate, temperatures of the wall and the refrigerant, quality and the mass inventory, in function of time and position. They neglected effects of heat conduction on axial direction of tube, the two-phase flow was considered one-dimensional, vapor and liquid are incompressible in thermal equilibrium and the pressure drop along tube is null.

Hermes (2006), in his thesis, proposed a model for a two compartment vapor compression refrigeration system. This is a "frost-free" refrigerator, equipped with a finned tube evaporator with forced ventilation and the temperature levels in fresh-food compartment and in freezer are determined by the acting of a thermostatic damper. There is a wire-and-tube condenser and a hermetic reciprocating compressor. With some experimental data, the author analysed topics such as start up transient behavior, based on the monitoring of temperatures and pressures since the start to the steady-state (Hermes, 2006) and periodic transient operation, evaluating the energy consumption, the transient operation and the controller limits. The author also worked on a model for the internal heat exchanger, using the effectiveness method (Kays & London, 1984) and solving analytically the enthalpy profile on the non-adiabatic region of the capillary tube. The heat exchangers simulation was carried out with a distributed model, considering the states that can occur isolated or simultaneously: superheated, two-phase and sub-cooled. Two approaches for the compressor modeling were tested: an analytical and a differential. Finally the author also investigated the dynamics of the air and the heat transfer inside the cabinet compartments and the influence of some constructive and geometric parameters.

On the work of Rangel (2007) several analysis of the system behavior were carried out based on the first model of Jakobsen (1995). He studied the heat transfer rate, thermodynamic losses, surface temperatures and electric power consumption in several parametric analysis: changing the thermal load, the overall heat transfer coefficient of condenser and the displacement of compressor. The analysis showed the potential of the model, with coherent results according to the new operation conditions.

Hermes & Melo (2008) developed a semi-empirical model to simulate the start and the cyclical transient of a frost-free refrigerator with two compartments, reciprocating compressor, wire-and-tube condenser with natural convection, capillary tube and internal heat exchanger, finned tube evaporator with forced convection and refrigerant R134a. The heat exchanger approach was with distributed parameters, based on the work of

Rossi and Braun (1999), that presents an explicit formulation on time domain providing the pressure values. The fluid flow was treated as one-dimensional, in a straight and horizontal tube with constant transversal section. Diffusive effects and pressure loss were neglected. The capillary tube was modeled as one-dimensional, viscous, compressible, homogenous and quasi-steady flow, with diffusive effects, heat conduction on the walls and metastability neglected. The compressor model allows the calculation of the mass flow rate on suction, the amount of refrigerant dissolving/separating from oil, the heat transfer through the housing and the electric power consumption. On the cabinet, the heat transfer was evaluated in 4 portions: conduction through the walls, heat transfer on the gasket, internal energy generation and air infiltration.

The work of Gonçalves *et al.* (2009) presents a steady-state modeling of a frost-free refrigerator with two compartments, hermetic reciprocating compressor, wire-and-tube condenser, finned tube evaporator with forced ventilation and refrigerant R134a. To adjust the model, several tests were carried out varying parameters as ambient temperature, compressor rotation, refrigerant charge, the restriction of an auxiliary expansion device, fan speed and thermal load. The heat exchangers were described by regions (superheated, two-phase and sub-cooled) and the heat transfer was calculated based on the effectiveness-NTU method. For the capillary tube and internal heat exchanger, the modeling was based on π -type correlations for sub-cooled and two-phase regions, according to (ASHRAE, 2002), to determine the mass flow through the tube and the effectiveness of the internal heat exchanger and, then, the heat transfer rate. Hermes *et al.* (2009) proposed a model based on this one, adding the calculation of a parameter to estimate the operation time of the system in one cycle of the on/off logic, using the thermal load and the refrigeration capacity, this way, the consumption of the system on these operational conditions can be determined on a steady-state analysis.

On the steady-state model presented by Holanda & Duarte (2009), the compression power is calculated through the values of condensing, evaporating and housing temperatures, where the mass flow rate is determined by the definition of volumetric efficiency, according to Gosney (1982), compressor rotation and displacement and the heat transfer through the housing is obtained with correlations for natural convection and radiation, considering the compressor spherical. The housing temperature is determined with a linear regression in function of the ambient temperature, using the experimental data obtained in pull-down tests, according to NTB00048 (1992) standards. The wire-and-tube condenser was divided into four regions: discharge line, superheated, two-phase, and sub-cooled. The first was assumed to be a straight, vertical and isothermal tube with negligible longitudinal conduction. For single-phase regions, the internal heat transfer coefficient was calculated through the Gnielinski (1976) correlation and for two-phase regions, through Shao & Granyrd (1995) one. The external coefficient was obtained with Tagliafico & Tanda (1997) correlations. The capillary tube mass flow rate is calculated using experi-

mental data and with a dimensional analysis according to Melo *et al.* (1999), while the temperature at the internal heat exchanger outlet is determined using the effectiveness method and experimental data evaluation, according to Stoecker (1989). The roll-bond evaporator (similar to a box without two opposite faces) was treated as four plates, two vertical and two horizontal, where the surface temperature was considered constant, with convection on internal and external surfaces and radiation on the external ones. Finally, the cabinet had its thermal conductance determined through the reverse heat transfer method. The calculations of COP , Second Law efficiency and exergy destruction were also carried out.

Pereira (2009) developed a model classified as "quasi-steady", where the refrigeration system is treated in steady-state and the refrigerated compartment in a transient approach, being possible to estimate the thermal load, the thermal inertia of compartments and the mass flow rate of air inside them. It is a frost-free refrigerator with two compartments, reciprocating compressor, wire-and-tube condenser and finned tube evaporator (both with forced ventilation), capillary tube with internal heat exchanger and R245fa refrigerant.

The volumetric and isentropic efficiencies and the thermal conductance of compressor were determined through the regression of experimental data. The effectiveness method was used to model the heat exchangers, which were divided in regions. The hypothesis adopted were: steady-state, straight and horizontal tubes with constant transversal section, axial heat conduction, kinetic and potential energy variations, viscous dissipation and radiation heat transfer neglected. For the internal heat exchanger, the considerations were: straight and horizontal tubes with constant transversal section; one-dimensional, viscous and compressible flow; fully developed flow, steady-state, null heat diffusion effects, pressure loss on inlet and on outlet and metastability neglected; two-phase homogeneous flow. The degree of superheating on evaporator outlet was imposed through experimental data, in order to represent the effects of refrigerant charge. Finally, on the cabinet modeling, one-dimensional heat conduction through the walls was taken into account, and five thermal resistances were considered: convection on the external air side, conduction on the steel of the wall, conduction on the PU insulation, conduction on the internal plastic wall and convection on the internal air side. The internal energy dissipation from the fan motor and the defrost resistor are present in the model. The author also evaluated the heat transfer through the gasket, with a simplified model, air infiltration, the mass flow rate of insufflated air and the thermal inertia of internal accessories such as drawers and shelves.

Negrão and Hermes (2011) proposed a model focused in optimize the operation of the domestic refrigeration system to reduce the energetic consumption and costs related to refrigerators. They tested a vertical freezer with one compartment, hermetic reciprocating

cating compressor, wire-and-tube condenser, capillary tube with internal heat exchanger, C type roll-bond evaporator on top and a plate in the middle and refrigerant R134a. The internal heat exchanger was considered counterflow type and modeled with the effectiveness method. The condenser and the evaporator were described through the model by regions and the pressure losses, neglected. The thermal conductances were acquired experimentally, including the reverse heat transfer method. The authors verified errors inside the range of $\pm 10\%$ for refrigeration capacity, electric consumption, and the ratio between the time on and the time off in a cycle. The modeling consider steady-state, but, the transient effect (refrigerator turning on and turning off) was inserted through the time ratio determined with a global energy balance during an on-off cycle. The evaluation of consumption and costs reductions were based on the optimization of the heat transfer area of heat exchangers and the cabinet insulation thickness.

The dynamic performance, of a household refrigerator was studied by Martínez-Ballester *et al.* (2012a), with a quasi-steady approach modeling. It was used a transient and lumped model developed by Martínez-Ballester *et al.* (2012b) with a black-box approach on compressor which uses a performance map containing the main data for the compression unit, depending on the compartments air conditions. The model is able to simulate a two-compartment frost-free refrigerator and the energy losses related to the closing of the damper and the compressor start-up were monitored.

Borges (2013) developed a semi-empirical model for a frost-free refrigerator considering the effects of door openings. The refrigerator tested has two compartments, hermetic reciprocating compressor, wire-and-tube condenser with natural convection, capillary tube with internal heat exchanger, finned tube evaporator with forced convection and 100g of R134a. The refrigeration system was treated in steady-state, but the evaporator, which had a transient approach as well the compartments, being possible to study the frost formation on evaporator and the humidity on the refrigerated ambient. The pressure loss in tubes was neglected, except in the capillary tube. On the heat exchangers, the thermal inertia of the structure material and the thermal resistance for both structure material and refrigerant side convection were also neglected. The flow, mass and heat transfer were considered one-dimensional. The compressor model is the one presented by Hermes *et al.* (2009). The condenser was divide into regions according to the fluid state. The capillary tube model was based on the work of Hermes *et al.* (2010), where the flow is considered one-dimensional at axial direction, two-phase, homogeneous, with no metastability and straight and horizontal tube with constant transversal section. For the internal heat exchanger the effectiveness method for counterflow heat exchangers was applied. In order to take into account the evaporator superheating variation with the door openings, the method presented by Wedekind (1968) was used, where the position of the boundary between single-phase and two-phase regions is determined. To model the heat and mass transfer, the pressure loss on the air side and the formation, growth and densification of

the frost layer, the works of Hermes & Melo (2009), about frost formation in plates, and Knabben (2010), adapting this feature for finned tube heat exchangers, were used.

Guzella (2013) developed a steady-state model using the software EES® and a transient model using the GT-SUITE®. The refrigerator modeled was the same presented by Klein (1998). The heat exchangers description was a great focus on this work: on the steady-state modeling, the condenser was divided in sub-regions (superheated, two-phase and sub-cooled), being a simplification of what was adopted on the transient evaluation, where the condenser was divided in several control volumes with a length Δz . The evaporator modeling in analogous, with two sub-regions in steady-state analysis (two-phase and superheated). The capillary tube was divided in an adiabatic and a non-adiabatic portion. Posteriorly a modified capillary tube model was proposed by Guzella *et al.* (2016). The flow modeling was based on the works of Yilmaz & Unal (1996) and Zhang & Ding (2001) and the heat transfer was determined through the effectiveness-NTU method. For the compression, a polytropic process approach was applied and the refrigerant amount dissolved on oil was taken into account.

Qureshi *et al.* (2013) presented an experimental investigation to identify the effects of a dedicated sub-cooler system operating with R12 on a R22 vapor compression refrigeration system. The experimental facility was a 1.5 *ton* air-conditioning system with a finned serpentine as condenser, expansion valve, evaporator and compressor. The sub-cooler system is also a vapor compression refrigeration system, where the evaporator is the sub-cooler which receives the outlet liquid of the condenser on the main system. The refrigeration capacity on evaporator was increased by 0.5 *kW* with a sub-cooling about 5 to 8°C on R22 and the second law efficiency raises up to 21%, but, this increasing is inversely proportional to the ambient temperature variation.

Dilay *et al.* (2014) developed a semi-empirical method to model and simulate engineering energy systems, called volume element method (VEM), combining the laws of conservation with some empirical and theoretical correlations. The components of the system are discretized in space, producing a system of algebraic and ordinary differential equations, in time (Dilay *et al.*, 2014). Then, Nunes *et al.* (2015) proposed a transient and semi-empirical mathematical model to simulate and optimize (seeking the maximum steady-state *COP* and second law efficiency) the dynamic response of a vapor compression refrigeration system. The authors developed normalized charts through convenient dimensionless groups that can be applied to similar devices. The experimental setup was an industrial chiller with a thermal load of 300W. The simulations were carried out comparing the results with R12, R134a and R1234yf. They concluded that the application of R1234yf in systems originally designed for R12 returns close results to the one with R12 or R134a. The optimal results, in terms of minimum pull-down time were obtained with an evaporator corresponding to 55% of the total system heat exchanger area and, in

terms of COP, 40%.

Zsembinszki *et al.* (2017) presented a steady-state semi-empirical model. They tested a walk-in freezer with one compartment, a condensing unit (reciprocating compressor and finned tube condenser with forced convection), an electronic expansion valve, a finned tube evaporator with forced convection and refrigerant R404A. The heat exchangers were divided into regions according to the fluid phase and average parameters were evaluated for each region. The inputs are external and internal air temperatures and the degree of superheating on compressor inlet. For the heat exchangers, the pressure loss was neglected, uniform external heat transfer properties, constant specific heat on sub-cooled and superheated regions, airflow uniformly distribute on the external surface and, all the heat transfer from the fluid is exchanged with the external air.

Yang & Ordonez (2018) proposed a methodology to simulate, using the volume element method, proposed on Dilay *et al.* (2014) and experimentally validated on Nunes *et al.* (2015), and optimize the performance of vapor compression refrigeration systems, seeking the best combination of evaporator and condenser sizes. The higher *COP* and second law efficiencies and lower pull-down times were reached with smaller evaporator areas, where there was a little increasing on the compressor power and exergy destruction. The model evaluate the system performance and the dynamics of the vapor compression refrigeration system and the refrigerated compartment on both transient and steady-state conditions and can provide conclusions of what configuration is better for each performance parameter. On the same reasoning, Yang *et al.* (2017) proposed a mathematical model and a procedure to optimize the internal heat transfer regions of the heat exchangers and the pressure ratio on a vapor compression refrigeration system and, through the calculations, the coefficient of performance (*COP*), the refrigeration rate and the second law efficiency can be optimized. The heat exchangers were modeled with the effectiveness-NTU method, considering the regions according to the fluid phase (sub-cooled, two-phase and superheated) and the methodology focus on the improvement of the pressure ratio and the heat exchanger areas. The working fluids tested were R1234yf, R22 and R410A. The authors reported the significance of this optimization with a variation of approximately 400% on the second law efficiency between the initial tested conditions and the optimized parameters.

2.4 Considerations about the system components

Hermes *et al.* (2010a,b) presented models for the mass flow rate through capillary tube and for the heat transfer on the internal heat exchanger. The considerations on the first are: straight and horizontal tube with constant transversal section; pressure loss at inlet and outlet neglected; homogeneous two-phase flow and no metastability. Besides,

the flow is considered isenthalpic and the pressure loss due the flow acceleration was considered null, compared to the one caused by friction. They used experimental data for R600a and R134a and adjusted a model to predict and relate the mass flow rate to the length and the diameter of the tube. The model presented 91% of the points in the range of $\pm 10\%$ error, compared to the experiments.

For the second model, they evaluated experimental data for diabatic flow with R134a and R600a. The effects of heat diffusion, conduction through the walls and pressure loss in suction line were neglected. Besides, the suction line was considered perfectly insulated from the external ambient and that there is only superheated vapor inside it. The heat transfer rate is calculated using the effectiveness method. To correct the mass flow rate on the diabatic section, the authors used the Buckingham π theorem to adjust a factor called diabatic multiplier.

Ablanque *et al.* (2015) proposed an iterative, discrete and one-dimensional model for non-adiabatic capillary tubes. This way, several control volumes were defined, with length Δz and nodes at the inlet and outlet sections, so, the equations of mass, momentum and energy conservation are applied, in steady-state, for each volume. This model is considered quasi-homogeneous, since the calculations are based on average properties at the volumes interfaces, i.e., at the nodes, but the gas and liquid velocities are considered different. The authors applied several empirical correlations to determine parameters such as void fraction, heat transfer coefficient, and pressure loss by friction. The metastability effects were neglected. Some highlights of the work are related to the numerical solution, where there is more computational cost, but also more stability on convergence, besides, the methodology adopted avoid the presence of control volumes with very different sizes.

Guzella *et al.* (2016) developed a steady-state model for the capillary tube and the internal heat exchanger. They considered one-dimensional flow and heat transfer; compressible flow; conduction through the walls, metastability, viscous dissipation and pressure loss on tube inlet and outlet and in suction line were neglected. The hydrodynamic model, for the mass flow rate and capillary tube length calculation was based on the hypothesis of adiabatic tube and the thermal model, on the effectiveness method. Also the Fauske criterion for choked flow verification was applied.

On the work of Suguimoto (2011), it was proposed a model for roll-bond (plate) evaporators and the interactions between this component and refrigerated ambient. The heat transfer mechanisms considered were natural convection and radiation. To the refrigerant flow, the finite volume method was applied, dividing the evaporator in regions according to the fluid phase (two-phase and superheated). On the external side of the plate, the author also considered two regions: the tubes and the space between tubes, which is analogous to a single fin, promoting heat conduction through the surface. On the cabinet model, external air convection, conduction through the walls and convection

and radiation on internal surfaces and evaporator were considered. The hypothesis were: evaporator as a single vertical plate with uniform thickness; two-dimensional heat conduction; convection on the two faces of evaporator; one-dimensional and fully developed flow; homogeneous flow in two-phase region and null pressure loss in the curves of the tube. The cabinet model admits that the walls are plates with one-dimensional heat conduction; constant ambient temperature; radiation between the external surfaces and the neighborhood is similar to the one between a cavity and much smaller objects inside; the internal and external environments do not take part on the radiation; only the properties of air vary with the temperature and the internal surfaces are considered opaque, gray and with null transmissivity.

2.5 Second Law analysis

Yataganbaba *et al.* (2015) analysed the exergy on systems designed for R134a operating with R1234yf and R1234ze, focusing on the evaluation of the condenser and evaporator temperatures influence. The cycle studied counts with two evaporators on different temperatures, one compressor, individual expansion valves for each evaporator and one condenser. The analysis was carried out in steady-state, the pressure variation on the heat exchangers was disregarded, and the flow was considered adiabatic on the other system components. The authors concluded that the R1234yf can replace R134a since its performance is only a little worst and, for the R1234ze some slight modifications on the systems are convenient to the correct operation; the majority of exergy destruction occurs on compressor.

Concerning to the second law analysis, Ma *et al.* (2017) studied the entropy generation on the several processes that occur in the system, in order to obtain an analytical expression to predict the coefficient of performance (COP), in function of the evaporating and condensing temperatures, compressor isentropic efficiency, evaporator and condenser outlet temperatures and the fluid properties on these points and on saturation. The evaluated processes in terms of entropy generation were: heat transfer on the two-phase and superheated regions of evaporator; compression; heat transfer between the compressor discharge state to the state if compression was isentropic, between this “isentropic” state and the saturation, on the two-phase, on the sub-cooled regions; on the throttling process. The expression obtained by the authors was compared to the other ones proposed by Alefeld (1987), Shelton & Grossmann (1985) and the definition of COP and proved to be a good way to predict the COP in steady-state, through simulations using several working fluids, with a maximum relative error of $\pm 5\%$.

3 Methodology

3.1 Experimental work

In this work are provided experimental and numerical simulation results for two refrigerators, one working with the R134a and other with R290. The first refrigerator, operating with R134a, has two compartments (207 *L* on fresh-food compartment and 53 *L* on freezer). The refrigeration system is composed by a hermetic reciprocating compressor (with 163g of POE10 oil) and a wire-and-tube condenser, both cooled by natural convection; a capillary tube and a heat exchanger with the suction line (internal heat exchanger); and roll-bond evaporators (“box” type on freezer and “plate” on fresh-food compartment). The system is filled with 105g of R134a as refrigerant. The control is attributed to an on/off mechanical thermostat. Two compressors were tested: a constant speed and a variable speed. Also, on the test with the first compressor, the defrost electric resistor was deactivated, and on the second, activated. In Table 1 are presented the dimensions of the components ¹.

Table 1: Dimensions of the R134a system components.

Condenser volume (10^{-5} m^3)	7.91
Condenser external area (m^2)	0.404
Condenser internal area (m^2)	0.095
Capillary tube inner diameter (10^{-4} m)	6.5
Evaporator volume (10^{-5} m^3)	12.9
Evaporator external area - fresh-food compartment (m^2)	0.17
Evaporator external area - freezer (m^2)	0.52
Evaporator internal area (m^2)	0.07
Compressor displacement (10^{-6} m^3)	5.54
Compressor shell volume (10^{-3} m^3)	1.34
Compressor discharge line volume (10^{-5} m^3)	3.77

For the R134a refrigerator with the variable speed compressor, the compressor displacement is 5.15 (10^{-6} m^3).

The second refrigerator, operating with R290, is a horizontal freezer, with a single compartment. The refrigeration system is composed by a variable speed reciprocating compressor with ventilation, skin condenser, capillary tube without internal heat exchanger, skin evaporator, 103g of propane and an electric resistor for defrosting purposes when the system turns off. The dimensions of the components are presented in table 2.

¹ Capillary tube diameter estimated according to the common value for these systems on the market.

Table 2: Dimensions of the R290 system components.

Condenser volume (10^{-4} m^3)	3.63
Condenser external area (m^2)	4.13
Condenser internal area (m^2)	0.43
Capillary tube inner diameter (10^{-4} m)	8.8
Evaporator volume (10^{-4} m^3)	12.17
Evaporator external area (m^2)	3.46
Evaporator internal area (m^2)	0.76
Compressor displacement (16^{-6} m^3)	12.47
Compressor shell volume (10^{-3} m^3)	1.34
Compressor discharge line volume (10^{-5} m^3)	3.77

The experiments were developed in a climate chamber at Tecumseh do Brasil LTDA. The scheme of the experimental facility with an indication of the experimental measurement locations is shown in figure 12, for the R134a refrigerator. Also the states evaluated of the refrigeration cycle are presented: 1 corresponds to the compressor inlet and suction line outlet; 2, to the compressor outlet and condenser inlet; 3, to the condenser outlet and capillary tube inlet; 3i, to the initial part of the capillary tube that is connected to the suction line to exchange heat; 4, to the capillary tube outlet and evaporator inlet; and 5, to the evaporator outlet and initial part of the suction line that is connected to the capillary tube to exchange heat.

The refrigerator was instrumented with T-type thermocouples fixed with aluminum tape on components external surfaces: compressor inlet and outlet sections; compressor housing (top and bottom); condenser inlet, middle and outlet sections; fresh-food roll-bond evaporator inlet, middle and outlet sections; internal heat exchanger outlet on suction line side; internal heat exchanger inlet on capillary tube side. Besides, there were 5 thermocouples distributed inside the fresh-food compartment and 3 on freezer. Pressure measurements were carried out by absolute pressure transducers at the suction and discharge sections of compressor. The compressor power consumption was acquired through an electric power transducer. For the cabinet characterization test, 3 electric resistors were positioned inside the compartments (1 on freezer and 2 on fresh-food) and the power on them was measured in the same way of the compressor power consumption.

The scheme of the experimental facility for the R290 refrigerator is shown in figure 13. Now 1 is the compressor inlet and evaporator outlet, 2, the compressor outlet and condenser inlet, 3, the condenser outlet and capillary tube inlet and 4, the capillary tube outlet and evaporator inlet. Again T-type thermocouples were used to measure the temperatures on compressor housing (top and bottom); condenser inlet, middle and outlet; evaporator middle and outlet and some measurements of the temperature on the internal wall of the cabinet. To calculate an average temperature of the internal air, 7 thermocouples were distributed inside the compartment. Finally, the pressures were determined

through the measurement of absolute pressure transducers at suction and discharge of compressor. The refrigerator power consumption was acquired with electric power transducers and, as this refrigerator has only one compartment, the cabinet characterization test was not necessary (more discussions on section 3.1.1).

Figure 12: Experimental facility and measured experimental data and points in the R134a refrigeration system.

Source: created by the author.

Figure 13: Experimental facility and measured experimental data and points in the R290 refrigeration system.

Source: created by the author.

The experimental data are used for two purposes. The first aim consists in the experimental characterization of the refrigeration system operation. These results include the transient characterization of components surface temperatures variation, as well as, the computation of experimental system power consumption, refrigeration capacity and coefficient of performance, COP , among other parameters (observe section 3.1.1).

The second aim is to provide the necessary experimental data for computing the input variables needed by the developed simulation models. The models' input parameters computed through experimental tests are: the thermal conductance, UA (W/K), and heat capacity, C (J/K), of the system components measured by a thermal cabinet characterization and pull-down operation tests; the fraction of pressure loss occurring on the adiabatic portion of capillary tube, $f_{\Delta P}$, considering that this component has an adiabatic (isenthalpic expansion) and a non-adiabatic (internal heat exchanger) parts; and the degrees of sub-cooling and superheating, ΔT_{sc} and ΔT_{sh} ($^{\circ}C$).

3.1.1 Pull-down tests for thermal conductance and capacity determination

In these tests, the refrigerator is put into thermal equilibrium with the external ambient at first. Then, with the thermostat by-passed, the compressor is turned on and remains working without cycling, pulling-down the temperatures inside the refrigerator until the steady-state condition is attained. This operation takes about 8 hours. The experiments were conducted with the conventional constant speed compressor running at constant rotation speed of 3600 *rpm* and with the external ambient temperatures of 25, 32 and 43 $^{\circ}C$ and with the variable speed compressor at ambient temperature of 32 $^{\circ}C$ and 1600, 2000, 3000, 3600 and 4500 *rpm* for the R134a system. For the R290 system the conditions were rotation of 4500 *rpm* and 25 $^{\circ}C$ ambient. Figure 14 shows the locations where the instrumentation was placed to measure the desired thermal conductance on the first device. These locations are schematically shown in figure 12.

The calculation methodology of thermal conductance consists in the application of an energy balance in each refrigerator component according to the next steps. First, at the steady-state operational condition, with the pressure and temperature measurements, the refrigerant properties at each point of the refrigeration cycle shown in figures 12 and 13 are calculated (neglecting pressure losses in the condenser and evaporator). Second, an energy balance is conducted on each component of the refrigeration system, where the heat transfer between the refrigerant and the component itself (internal heat transfer), defined as the product between the mass flow rate and the enthalpy variation, is equal to the one between the component and the external ambient air (external heat transfer). The mass flow rate is computed using the compressor efficiencies definitions (obtained through the data provided by the manufacturer, see forthcoming section 3.1.3). At this point, it is possible to determine cooling capacity, heat rejection on condenser and compressor and,

Figure 14: Instrumentation of the refrigerator for the pull-down test: thermocouples on (a) freezer, (b) fresh-food compartment and evaporator, (c) condenser; (d) thermocouples and pressure transducers on compressor and suction and discharge line; (e) thermocouples on suction line and capillary tube.

Source: created by the author.

then, with the measured electric power, the COP and consumption. So, using the Eq. 3.1, the conductance (UA) can be calculated.

$$UA = \frac{\dot{Q}}{\Delta T} \quad (3.1)$$

The thermal capacities (C) are also necessary to provide sufficient input data to the mathematical models. The thermal capacities are calculated from the conductance values and the time constants (τ , in s) for each component. The time constants are defined as a fifth of the time to reach steady-state and can be determined through the graphs obtained on pull-down tests that show the temperature distribution until the system stabilization, i.e., the attainment of stationary working operation. The time constant of a thermal system is the product between the thermal resistance and capacity and the thermal resistance is the inverse of the thermal conductance so, C is calculated as shown on Eq. 3.2.

$$C = UA\tau \quad (3.2)$$

The numerical values of these two parameters are provided in section 3.1.4.

For a refrigerator with a single compartment, this test is sufficient to determine all the UA and C , including the cabinet ones. On steady-state, the heat transferred through

the evaporator is the same that the one through the cabinet walls. So, with the temperature measurements (internal air and ambient), these parameters can also be determined. It is the case of the refrigerator with R290. For a two-compartment refrigerator, the test on the next section is necessary.

3.1.2 Thermal conductance determination of cabinet compartments

To characterize the cabinet compartments, it was carried out a reverse heat loss rate measurement test, according to NTB00119 Standards (1992). Melo *et al.* (2000) also used this experimental method to determine the thermal conductance of a cabinet compartment. In this test, the refrigerator, in a climate chamber, is kept turned off and electric resistors are conveniently put inside the compartments, warming them as shown in Figure 15. The difference between the average temperature of compartments should be about 25°C higher than the ambient one and the maximum power dissipated on the resistors, 50W . The electric resistors were fixed to aluminium plates, to facilitate the natural convection and obtain a more uniform temperature distribution.

Figure 15: Instrumentation of the refrigerator for the cabinet characterization test.

Source: created by the author.

The compartments are closed and the electric resistors turned on, rising the temperature of the compartments until a steady-state condition is established. At this moment, the electric power on the resistors is equal to the heat transfer from them to the internal air of the compartments, which is equivalent to the heat transfer through the walls to external air. Another heat flux is the one occurring from one compartment to another, through the wall between them. So, measuring the electric power and the temperatures, energy balances can be established according to Eq. 3.3, for the fresh-food compartment and 3.4, for the freezer.

$$\dot{W}_{ff} = UA_{ff}(T_{ff} - T_{amb}) - UA_w(T_{fr} - T_{ff}) \quad (3.3)$$

$$\dot{W}_{fr} = UA_{fr}(T_{fr} - T_{amb}) + UA_w(T_{fr} - T_{ff}) \quad (3.4)$$

In Eqs. 3.3 and 3.4 $\dot{W}_{ff}$ and $\dot{W}_{fr}$ (W) are the power dissipated inside the fresh-food compartment and the freezer, UA_{ff} and UA_{fr} , the thermal conductance of these compartments, UA_w , the conductance of the wall between them and T_{ff} , T_{fr} , T_{amb} ($^{\circ}C$) are the fresh-food compartment, freezer and ambient temperatures, respectively. So, summing the equations above the Eq. 3.5 is obtained.

$$\dot{W}_{ff} + \dot{W}_{fr} = UA_{ff}(T_{ff} - T_{amb}) + UA_{fr}(T_{fr} - T_{amb}) \quad (3.5)$$

Where the unknown variables are UA_{ff} and UA_{fr} . Therefore, with at least two tests, varying the power dissipated on each compartment, an equation system is obtained and these variables are determined. Five tests were performed and, combining the equations in pairs, ten values were calculated for each variable. Finally, taking the averages, the process was concluded. Now, using Eq. 3.3 or 3.4, UA_w is determined.

3.1.3 Compressor efficiencies measurements

The compressor efficiencies (volumetric and “global”) were obtained through experimental data from calorimeter tests provided by the manufacturer. These data are about mass flow rate, electric power consumption and refrigeration capacity, in steady-state. The volumetric η_v and global η_g efficiencies can be calculated as follows:

$$\eta_v = \frac{60\dot{m}_{com}}{\rho_1 V_s N_{rpm}} \quad (3.6)$$

$$\eta_g = \frac{\dot{m}_{com}(h_{2s} - h_1)}{\dot{W}} \quad (3.7)$$

In Eqs. 3.6 and 3.7, $\dot{m}_{com}$ (kg/s) is the compressor mass flow rate, ρ_1 (kg/m^3), the refrigerant density at the compressor inlet, V_s (m^3), the compressor displacement, N_{rpm} , the compressor rotation in rpm , h_{2s} (J/kg), the enthalpy on discharge if the compression process was isentropic, h_1 , the enthalpy on compressor inlet and $\dot{W}$ (W), the electric power consumption.

Therefore, with the data for mass flow rate and electric power consumption, the efficiencies can be determined and polynomials as a function of the ratio between condensing and evaporating pressures (in Pa) for each efficiency are fitted. The Eqs. (3.8) and (3.9) present the results for the constant speed compressor.

$$\eta_v = -0.0013 \left(\frac{P_{cond}}{P_{evap}} \right)^2 + 0.0118 \left(\frac{P_{cond}}{P_{evap}} \right) + 0.6798 \quad (3.8)$$

$$\eta_g = -0.0007 \left(\frac{P_{cond}}{P_{evap}} \right)^2 + 0.0109 \left(\frac{P_{cond}}{P_{evap}} \right) + 0.58 \quad (3.9)$$

For the variable speed compressor:

$$\eta_v = -0.0204 \left(\frac{P_{cond}}{P_{evap}} \right) + 0.8824 \quad (3.10)$$

$$\eta_g = -0.0004 \left(\frac{P_{cond}}{P_{evap}} \right)^2 + 0.0021 \left(\frac{P_{cond}}{P_{evap}} \right) + 0.633 \quad (3.11)$$

Another approach was adopted for the compressor operating with R290, where a set of calculated efficiencies were included on the Python program and interpolation features from this language were in charge to determine the correct value on the operational conditions.

3.1.4 Experimental uncertainties

The uncertainty of measured temperatures, pressures and electric power are obtained directly from the instruments, and are shown at Table 3. The mass flow rate uncertainty is obtained from the calorimeter test data of compressor provided by the manufacturer, associated to an average error of 7%.

Table 3: Uncertainty of instruments.

Instrument	Uncertainty
T-type thermocouples	0.34°C
Suction line pressure transducer	0.2% full scale (150 <i>psia</i>)
Discharge line pressure transducer	0.2% full scale (500 <i>psia</i>)
Electric power transducer	1.5% measurement

The uncertainties for the other parameters (thermodynamic and thermal properties and component parameters) are calculated according to the method presented by Taylor and Kuyatt (1994). The details of uncertainty calculation for each variable are described in Appendix A. Tables 4 to 6 present the numerical values of all parameters and their uncertainties, obtained with the steady-state pull-down tests of the R134 refrigerator with the constant speed compressor in three different ambient temperatures.

The Table 7 presents the values and uncertainties for thermal conductances and capacities obtained through the cabinet characterization tests with the reverse heat loss measurement method.

Table 4: Temperatures, pressures, enthalpies, electric power and mass flow rate acquired on the pull-down test (steady-state) for the thermodynamic states of the refrigerator cycle (refrigerator with R134a and constant speed compressor).

Ambient	25°C	32°C	43°C
T_1 (°C)	34.02 ± 0.34	41.51 ± 0.34	51.38 ± 0.34
T_2 (°C)	74.29 ± 0.34	87.88 ± 0.34	106.32 ± 0.34
T_3 (°C)	35.31 ± 0.34	44.34 ± 0.34	56.71 ± 0.34
T_{3i} (°C)	31.6 ± 0.34	39.87 ± 0.34	52.07 ± 0.34
T_4 (°C)	-39.08 ± 0.34	-33.57 ± 0.34	-29.65 ± 0.34
T_5 (°C)	-36.83 ± 0.34	-32.72 ± 0.34	-26.33 ± 0.34
P_{evap} (kPa)	53.75 ± 2.07	64.29 ± 2.07	85.77 ± 2.07
P_{cond} (kPa)	918.38 ± 6.89	1166 ± 6.89	1582 ± 6.89
P_{3i} (kPa)	806.65 ± 7.69	1014 ± 9.12	1389 ± 11.56
h_1 (kJ/kg)	433.05 ± 0.29	439.36 ± 0.29	447.73 ± 0.3
h_2 (kJ/kg)	457.66 ± 0.36	468.41 ± 0.37	483.17 ± 0.38
h_3 (kJ/kg)	249.46 ± 0.49	262.93 ± 0.51	282.17 ± 0.54
h_{3i} (kJ/kg)	249.46 ± 0.49	262.93 ± 0.51	282.17 ± 0.54
h_4 (kJ/kg)	192.68 ± 0.63	202.55 ± 0.64	217.57 ± 0.68
h_5 (kJ/kg)	376.28 ± 0.27	378.98 ± 0.27	383.14 ± 0.27
$\dot{W}$ (W)	72.57 ± 1.09	78.33 ± 1.18	88.99 ± 1.34
$\dot{m}_{com}$ (g/s)	0.45 ± 0.03	0.48 ± 0.03	0.54 ± 0.04

Table 5: Surface and air temperatures on components on the steady-state pull-down test (refrigerator with R134a and constant speed compressor).

Ambient	25°C	32°C	43°C
T_{com} (±0.34°C)	72.26	84.48	99.48
T_{wc} (±0.34°C)	40.15	50.53	65.15
T_{weff} (±0.34°C)	-37.16	-32.97	-26.49
T_{ff} (±0.34°C)	-6.54	-0.62	7.805
T_{fr} (±0.34°C)	-34.13	-30.18	-24.45
T_{amb} (±0.34°C)	24.40	32.30	43.70

Table 6: Thermal conductance and capacity and pressure loss factor calculated values (refrigerator with R134a and constant speed compressor).

Ambient	25°C	32°C	43°C
UA_{com} (W/K)	1.29 ± 0.03	1.23 ± 0.03	1.25 ± 0.04
UA_c (W/K)	5.91 ± 0.45	5.45 ± 0.41	5.08 ± 0.34
UA_{eff} (W/K)	1.15 ± 0.04	1.16 ± 0.04	1.20 ± 0.04
UA_{efr} (W/K)	3.84 ± 0.46	3.76 ± 0.42	4.25 ± 0.48
C_{com} (kJ/K)	6.64 ± 0.16	6.36 ± 0.16	6.46 ± 0.19
C_c (kJ/K)	21.88 ± 1.67	20.18 ± 1.51	18.78 ± 1.38
C_{eff} (kJ/K)	0.97 ± 0.03	0.98 ± 0.03	1.01 ± 0.03
C_{efr} (kJ/K)	9.22 ± 1.09	9.03 ± 1.01	10.20 ± 1.16
C_{eq} (kJ/K)	8.76 ± 0.94	8.59 ± 0.87	9.63 ± 1.01
$f_{\Delta P}$	0.13 ± 0.01	0.14 ± 0.01	0.13 ± 0.01

Table 7: Refrigerator compartments thermal conductance and capacity values and their uncertainties obtained by the cabinet characterization test. (refrigerator with R134a).

UA_{ff} (W/K)	UA_{fr} (W/K)	UA_w (W/K)	C_{ff} (kJ/K)	C_{fr} (kJ/K)
1.14 ± 0.02	0.32 ± 0.02	0.07 ± 0.004	6.854 ± 0.131	1.363 ± 0.086

Tables 8 to 10 present these results for the R134a refrigerator with the variable speed compressor in five different rotations and $T_{amb} = 32^\circ C$.

Table 8: Temperatures, pressures, enthalpies, electric power and mass flow rate acquired on the pull-down test (steady-state) for the thermodynamic states of the refrigerator cycle (refrigerator with R134a and variable speed compressor).

Rotation	1600 rpm	2000 rpm	3000 rpm	3600 rpm	4500 rpm
T_1 ($^\circ C$)	37.82 ± 0.34	37.03 ± 0.34	38.09 ± 0.34	37.46 ± 0.34	37.83 ± 0.34
T_2 ($^\circ C$)	63.83 ± 0.34	64.70 ± 0.34	74.21 ± 0.34	77.75 ± 0.34	81.08 ± 0.34
T_3 ($^\circ C$)	39.95 ± 0.34	40.57 ± 0.34	41.76 ± 0.34	42.26 ± 0.34	42.52 ± 0.34
T_{3i} ($^\circ C$)	36.26 ± 0.34	36.61 ± 0.34	37.87 ± 0.34	38.03 ± 0.34	38.17 ± 0.34
T_4 ($^\circ C$)	-25.86 ± 0.34	-28.26 ± 0.34	-31.86 ± 0.34	-33.93 ± 0.34	-34.69 ± 0.34
T_5 ($^\circ C$)	-16.26 ± 0.34	-22.31 ± 0.34	-31.03 ± 0.34	-33.21 ± 0.34	-33.96 ± 0.34
P_{evap} (kPa)	102.32 ± 2.07	91.60 ± 2.07	77.19 ± 2.07	69.74 ± 2.07	67.19 ± 2.07
P_{cond} (kPa)	1030 ± 6.89	1049 ± 6.89	1086 ± 6.89	1102 ± 6.89	1112 ± 6.89
P_{3i} (kPa)	918.91 ± 8.48	927.73 ± 8.54	960.35 ± 8.76	964.46 ± 8.79	968.14 ± 8.82
h_1 (kJ/kg)	435.54 ± 0.29	435.03 ± 0.29	436.19 ± 0.29	435.76 ± 0.29	436.12 ± 0.29
h_2 (kJ/kg)	445.04 ± 0.37	445.63 ± 0.37	455.08 ± 0.37	458.58 ± 0.37	461.96 ± 0.37
h_3 (kJ/kg)	256.33 ± 0.50	257.25 ± 0.50	259.03 ± 0.51	259.79 ± 0.51	260.18 ± 0.51
h_{3i} (kJ/kg)	256.33 ± 0.50	257.25 ± 0.50	259.03 ± 0.51	259.79 ± 0.51	260.18 ± 0.51
h_4 (kJ/kg)	211.34 ± 0.64	208.31 ± 0.64	202.64 ± 0.64	202.43 ± 0.64	201.98 ± 0.64
h_5 (kJ/kg)	390.56 ± 0.28	386.09 ± 0.27	379.79 ± 0.27	378.39 ± 0.27	377.92 ± 0.27
$\dot{W}$ (W)	39.12 ± 0.59	45.26 ± 0.68	61.07 ± 0.92	66.06 ± 0.99	72.89 ± 1.09
$\dot{m}_{com}$ (g/s)	0.39 ± 0.03	0.42 ± 0.03	0.50 ± 0.04	0.50 ± 0.04	0.50 ± 0.04

Table 9: Surface and air temperatures on components on the steady-state pull-down test (refrigerator with R134a and variable speed compressor).

Rotation	1600 rpm	2000 rpm	3000 rpm	3600 rpm	4500 rpm
T_{com} ($\pm 0.34^\circ C$)	60.92	61.84	68.29	69.67	73.30
T_{wc} ($\pm 0.34^\circ C$)	41.25	42.04	44.54	45.55	46.29
T_{weff} ($\pm 0.34^\circ C$)	-20.08	-24.98	-31.14	-33.26	-33.97
T_{ff} ($\pm 0.34^\circ C$)	6.60	2.90	-1.64	-3.23	-3.74
T_{fr} ($\pm 0.34^\circ C$)	-23.86	-26.02	-29.28	-31.08	-31.74
T_{amb} ($\pm 0.34^\circ C$)	32.70	32.29	32.66	32.16	32.07

Table 10: Thermal conductance and capacity and pressure loss factor calculated values (refrigerator with R134a and variable speed compressor).

Rotation	1600 rpm	2000 rpm	3000 rpm	3600 rpm	4500 rpm
UA_{com} (W/K)	1.26 ± 0.03	1.38 ± 0.03	1.45 ± 0.04	1.46 ± 0.04	1.46 ± 0.04
UA_c (W/K)	8.51 ± 0.76	8.15 ± 0.69	8.34 ± 0.67	7.41 ± 0.58	7.08 ± 0.55
UA_{eff} (W/K)	1.12 ± 0.04	1.20 ± 0.04	1.33 ± 0.04	1.35 ± 0.04	1.35 ± 0.04
UA_{efr} (W/K)	9.19 ± 2.31	8.47 ± 1.92	7.79 ± 1.55	7.20 ± 1.31	7.05 ± 1.25
C_{com} (kJ/K)	7.54 ± 0.19	8.28 ± 0.21	8.68 ± 0.23	8.75 ± 0.24	8.73 ± 0.23
C_c (kJ/K)	45.94 ± 4.11	44.02 ± 3.76	45.04 ± 3.63	39.98 ± 3.14	38.23 ± 2.97
C_{eff} (kJ/K)	1.54 ± 0.06	1.66 ± 0.06	1.83 ± 0.06	1.86 ± 0.06	1.87 ± 0.06
C_{efr} (kJ/K)	14.21 ± 3.43	18.64 ± 4.22	17.13 ± 3.40	15.83 ± 2.88	15.49 ± 2.75
C_{eq} (kJ/K)	20.21 ± 5.08	13.19 ± 2.85	12.23 ± 2.30	11.36 ± 1.95	11.14 ± 1.86
$f_{\Delta P}$	0.12 ± 0.01	0.13 ± 0.01	0.13 ± 0.01	0.13 ± 0.01	0.14 ± 0.01

Finally, for the R290 refrigerator the uncertainties, at $N_{rpm} = 4500$ rpm, are:

Table 11: Temperatures, pressures, enthalpies, electric power and mass flow rate acquired on the pull-down test (steady-state) for the thermodynamic states of the refrigerator cycle (refrigerator with R290).

Ambient	25°C
T_1 (°C)	-24.58 ± 0.34
T_2 (°C)	81.69 ± 0.34
T_3 (°C)	36.27 ± 0.34
T_4 (°C)	-46.43 ± 0.34
P_{evap} (kPa)	83.43 ± 2.07
P_{cond} (kPa)	1342.53 ± 6.89
h_1 (kJ/kg)	552.57 ± 0.51
h_2 (kJ/kg)	706.11 ± 0.74
h_3 (kJ/kg)	297.02 ± 0.096
h_4 (kJ/kg)	297.02 ± 0.96
$\dot{W}$ (W)	250.99 ± 3.76
$\dot{m}_{com}$ (g/s)	0.80 ± 0.06

Table 12: Surface and air temperatures on components on the steady-state pull-down test (refrigerator with R290).

Ambient	25°C
T_{com} ($\pm 0.34^\circ\text{C}$)	60.31
T_{wc} ($\pm 0.34^\circ\text{C}$)	37.70
T_{we} ($\pm 0.34^\circ\text{C}$)	-37.59
T_{cab} ($\pm 0.34^\circ\text{C}$)	-24.58
T_{amb} ($\pm 0.34^\circ\text{C}$)	25.60

Table 13: Thermal conductance and capacity and pressure loss factor calculated values (refrigerator with R290).

Ambient	25°C
$UA_{com} (W/K)$	3.74 ± 0.28
$UA_c (W/K)$	26.85 ± 2.17
$UA_e (W/K)$	15.62 ± 1.24
$UA_r (W/K)$	4.05 ± 0.29
$C_{com} (kJ/K)$	4.48 ± 0.33
$C_c (kJ/K)$	48.32 ± 3.91
$C_e (kJ/K)$	18.74 ± 1.49
$C_{cab} (kJ/K)$	23.33 ± 1.66

Note how all the thermal conductances are greater on this refrigerator. The explanations are: on compressor, the fact that the fan is imposing forced convection raises the global heat transfer coefficient (U); on condenser, evaporator and cabinet, the greater area increases the conductance.

3.1.5 Experimental results

In this section transient experimental data are provided to evaluate the measured experimental results. The data can be used also by other researchers for mathematical model comparison and development. In section 4 these results will be compared to the results obtained from the simulation models. The results for the R134a refrigerator are the ones using the constant speed compressor. The data for the R290 system are also presented.

The components of a refrigeration system have very dynamic interactions between them. During the transient operation, the increasing on compressor mass flow rate promotes the increasing on condenser pressure and a decreasing on evaporator one. On the other hand, with this greater difference between the high and low pressures (and temperature) regions, the increasing of the mass flow on compressor becomes smaller and the capillary mass flow rate (which depends on the pressure difference) rises, so the pressure on condenser is reduced, as well the pressure on evaporator is increased. These processes lead the system to a stabilization, in terms of mass flow rate.

In the period where the compressor mass flow rate is higher, the condenser is being filled and the evaporator emptied, this way, both sub-cooling and super-heating degrees are increased. As the capillary tube mass flow rate increases, the opposite occurs, letting condenser with less and evaporator with more refrigerant content, so, both, sub-cooling and super-heating degrees decrease.

In figure 16 are presented the transient temperature distributions in the high temperature regions of the system. These are the compressor housing and condenser wall temperatures, for both refrigerators. The compressor housing temperatures are shown

for the up and bottom compressor surfaces. The temperature differences between these regions are explained by the more intense circulation of cold refrigerant on the bottom region from the suction line, which also contributes to cool the electric motor. On the other hand, the discharge line and the compression cylinder are located more close to the top region, causing higher temperatures in the upper housing surface. The temperature difference increases when the system is on, i.e., when there is mass flow rate through the compressor, and can be noted on both systems tested.

The condenser can be divided into three regions regarding the internal fluid phase: a superheated region with a superheated gas from compressor; a two-phase region where the condensation occurs; and a sub-cooled region, which may exists or not. In figure 16 are presented the condenser wall temperatures at condenser inlet (superheated region), middle region (two-phase), and outlet region (slightly sub-cooled). After minute 10, on figure 16a, and on the beginning, on figure 16b, when the systems start, the compressor mass flow rate is by far greater than the capillary tube's one, which causes a quickly elevation of the pressure and temperature. The inlet section is much warmer than the other two because of the hot superheated gas received from compressor discharge. The temperature on the middle region is close to the condensing one, due the high two-phase heat transfer coefficient and the temperature of the outlet region is almost the same, because of the small sub-cooling that exists and the relative high liquid heat transfer coefficient value. When the system turns off (about minute 36 – figure 16a – and minute 70 – figure 16b), the mass flow rate through compressor ceases and the pressure has a strong decrease with the mass flowing through the capillary tube to evaporator until the “pressure equalization” between the condenser and evaporator.

This abrupt pressure diminution leads to a flash effect in the condenser, decreasing almost instantly the whole condenser temperature, i.e., the measured temperatures in the three condenser regions. The flash effect is related to the cooling effect by the abrupt reduction of the condenser saturation temperature determined by the pressure reduction. This thermodynamic expansion process, plus the fact that during some time the mass flow rate is significant, leads to a quick cooling of the wall on the three measured points and in the whole condenser. After this period, there is only gas inside condenser and the external heat transfer coefficient becomes dominant, maintaining the condenser surface temperature close to the ambient one. It should be noted that the condenser wall temperature used for comparison with the simulated one is the average value of the above commented three temperatures.

In Figure 17 are presented the experimental measures for temporal temperature distributions in the low temperature regions of the system. These are the evaporator wall, evaporation, and the air inside compartments temperatures, for both refrigerators. The evaporator commonly has two parts: the two-phase region where the convective evap-

Figure 16: Compressor housing and condenser wall transient temperature measured distributions.

(a) R134a

(b) R290

Source: created by the author.

oration process occurs (present on inlet region) and a superheated single-phase region (present on outlet region), that may exists or not.

For the R134a system (figure 17a), the degree of superheating presented is low, which means that the two-phase region is by far greater than the superheated one and that the temperatures of the inlet and outlet regions are close. When the system starts, the temperature in the inlet region is the saturation one which decreases rapidly following

Figure 17: Evaporator wall, evaporation, fresh-food compartment and freezer internal air transient temperature measured distributions.

(a) R134a

(b) R290

Source: created by the author.

the pressure reduction, and the surface temperature become close to this one due the high two-phase heat transfer coefficient. The outlet region temperature is close, due the small superheating. When the system turns off, the evaporator on fresh-food compartment gets considerably warmer than the refrigerant fluid (which is in a two-phase state) temperature because of the reduction of the heat transfer coefficient and due to the great temperature difference between its surface and the fresh-food compartment air. The surface tempera-

ture of the evaporator in the freezer is very close to the refrigerant fluid one, both when the system is on and off: when it is on, the high two-phase heat transfer coefficient maintains these close temperature values and, when the system is off, the temperatures are still close due the low temperature difference between the air inside the freezer, and the refrigerant temperature.

For the R290 system (figure 17b), the degree of superheating is greater, this way, there is a considerable superheated region and the outlet temperature has a great increasing related to the evaporating one. Again, when the system starts, the temperature decreases quickly and, due the great superheating, the temperature on the middle of evaporator is considerably higher than the evaporating one. As the system turns off and the mass flow ceases, the temperature on evaporator surface raises and become close to the cabinet internal air one.

Concerning to the temperatures on the fresh-food compartment, figure 17a, the values were acquired for five positions from the top to the bottom. This last measurement occurred inside the vegetable drawer, which explains the higher temperature. The other ones follow the expected behavior, decreasing from the top to inferior positions. The temperature of the air inside the freezer, on three positions from the top to the bottom, decreases on inferior positions. The same temperature dynamics is presented on the R290 refrigerator (figure 17b).

Figure 18: Refrigerator electric power consumption measurement.

Source: created by the author.

Figure 18 shows the results for the instantaneous compressor electric power consumption. It is evident the peak when compressor starts, period where an auxiliary coil on electric motor is activated to increase the torque. After this period, the auxiliary coil is deactivated and the electric power consumption decreases until the system stops. On the system with R290, Fig. 18b, the presence of defrost electric resistance can be noted, as the power when the system turns off do not becomes null.

Figure 19: Pressures measurement (on condenser and on evaporator).

Source: created by the author.

Figure 19 presents the behavior of the pressures on condenser and evaporator. When the system starts, the pressure difference increases rapidly, due the compressor mass flow rate, which is higher than that from the capillary tube on this period, filling up the condenser. When the system is deactivated, the condenser pressure has a strong decrease due the mass flow through the capillary tube reducing the mass on it and increasing the mass on evaporator, with two-phase fluid. When this mass flow rate ceases, the pressures on condenser and evaporator become very close, with a little difference maintained by the capillary tube hydraulic resistance.

3.2 Mathematical models

In this section are presented the developed mathematical models, which are based on the work of Jakobsen (1995). The main purpose of the models is their use in the numerical simulation of the transient behavior of the refrigeration systems. Both models are composed by “sub-models” for each system component (compressor, condenser, capillary tube and internal heat exchanger, evaporator and cabinet compartments), where the input variables for one of them are the outputs provided by the previous one, following the refrigerant flowing direction. Figures 20 and 21 show the control volumes that are considered in the simulation of both refrigeration systems. The details of each mathematical model are presented in sections 3.2.1 and 3.2.2, respectively. The two models are applied for simulating each refrigeration system.

The first model denominated as the “thermal” model consists on the application of the first law of thermodynamics and heat transfer rate equations to each control volume representing each component of the refrigeration system. A second law analysis is performed when the steady-state is numerically established, calculating the entropy gen-

Figure 20: Control volumes and the interaction between the components of the R134a refrigeration system.

Source: created by the author.

Figure 21: Control volumes and the interaction between the components of the R290 refrigeration system.

Source: created by the author.

eration on each component.

The second model is denominated as a “capacitive” model because this model considers aspects and effects of the refrigerant mass distribution and flow dynamic in the system. In this model are used the same tools of the thermal model, including additional expressions to compute the working fluid mass distribution. This model capability allows the transient characterization of the refrigerant mass distribution in the refrigeration system.

Results such as refrigeration capacity, electric power consumption, condenser heat rejection, COP, monthly energy consumption, component surfaces and compartments internal air temperatures can be obtained with both models. However, only the capacitive model is able to reproduce the pressures behavior when the system turns off, pressure peaks when the system starts, dynamic behavior of sub-cooling and superheating degrees, influence of geometric characteristics of capillary tube and of refrigerant charge.

The following simplifying hypothesis were adopted in the development of the mathematical models:

- Hypothesis for physical and mathematical description: Control volumes around the system components have only one inlet and one outlet; kinetic and potential energy variations inside and at the open boundaries are neglected; Thermodynamic and transport properties are uniform in each control volume; and the force fields are neglected.
- Hypothesis for the components modeling: Delays on transport, pressure losses and refrigerant accumulation on connector tubes; pressure losses on condenser and evaporator; spatial variation of temperature at condenser, evaporator and compressor surfaces and inside the compartments of cabinet are all neglected.

The simulations do not consider door openings, following the conditions of the experimental tests. The air infiltration is not taken into account. Temperatures are expressed in $^{\circ}\text{C}$; pressure in Pa ; density in kg/m^3 ; volume in m^3 ; mass flow in kg/s ; enthalpies in J/kg ; entropies in J/kgK ; and heat transfer rates and power consumption in W .

3.2.1 Thermal model

Compressor model: The control volume used to simulate the compressor is shown in Figures 20 and 21. In the compressor sub-model, the inputs are the inlet temperature, T_1 , the inlet pressure (considered equal to the evaporating one, $P_1 = P_{\text{evap}}$), the inlet enthalpy, h_1 , and outlet pressure (equals to the condensing one, $P_2 = P_{\text{cond}}$). Other input parameters are the external ambient temperature, T_{amb} , the compressor thermal

conductance and capacity, UA_{com} and C_{com} , the volume displacement, V_s , the compressor rotational speed, N_{rpm} , and the polytropic exponent, n_p . The output results obtained from the compressor are the mass flow rate, $\dot{m}_{com}$, the electric power consumption, $\dot{W}_{com}$, the heat transferred through the housing, $\dot{Q}_{com}$, the compressor housing surface temperature, T_{com} , the outlet temperature, T_2 , and enthalpy, h_2 .

As presented on section 3.3, on the first time step, the state 1, the condensing (T_{cond}) and evaporating (T_{evap}) temperatures (and consequently, the pressures) and the initial components surface and compartments internal air temperatures are guessed, so it is possible to start the calculations. The mass flow rate through compressor is obtained with Eq. 3.12.

$$\dot{m}_{com} = \eta_v \rho_1 V_s \frac{N_{rpm}}{60} \quad (3.12)$$

Where ρ_1 is the density on inlet.

Now, it is possible to determine the compressor electric power, Eq. 3.13, after the calculation of the outlet enthalpy if the compression process was isentropic, h_{2s} .

$$\dot{W}_{com} = \dot{m}_{com} \frac{(h_{2s} - h_1)}{\eta_g} \quad (3.13)$$

The heat transferred through the compressor housing is obtained based on the experimental thermal conductance.

$$\dot{Q}_{com} = UA_{com}(T_{com} - T_{amb}) \quad (3.14)$$

The new compressor housing temperature is calculated by the solution of the energy balance differential equation for the compressor, represented by Eq. 3.15:

$$C_{com} \frac{dT_{com}}{dt} = \dot{W}_{com} - \dot{Q}_{com} - \dot{m}_{com}(h_2 - h_1) \quad (3.15)$$

The enthalpy at the compressor outlet, h_2 , is calculated as a function of discharge pressure and the discharge temperature, T_2 . Two strategies were evaluated for calculating this temperature: first the consideration of a polytropic compression considering a preheat of refrigerant on the housing before enter the compression chamber, according to Jakobsen (1995), Eq. 3.16; and second the consideration of this temperature approximately equal to the housing one, T_{com} , Eq. 3.17, as presented by Negrão & Hermes (2011).

$$T_2 = \left(\frac{T_1 + T_{com}}{2} \right) \left(\frac{P_{cond}}{P_{evap}} \right)^{\frac{n_p - 1}{n_p}} \quad (3.16)$$

$$T_2 \approx T_{com} \quad (3.17)$$

For the R134a system, the Eq. 3.17 was adopted because of the compressor with natural convection cooling and, for the R290 system, the Eq. 3.16 was applied, with a polytropic exponent of 1.08. Other non-cited thermodynamic variables are calculated as a function of the respective thermodynamic state known variables.

Condenser model: The control volume for modeling this component is shown in figures 20 and 21. The inputs in this sub-model are: the compressor outlet temperature and enthalpy, T_2 and h_2 , the compressor mass flow rate, $\dot{m}_{com}$, the condenser thermal conductance and capacity, UA_c and C_c and the degree of sub-cooling, ΔT_{sc} , obtained experimentally. The outputs are: the condenser heat transfer rate, $\dot{Q}_c$, the condenser surface temperature, T_{wc} , and the condenser outlet temperature and enthalpy, T_3 and h_3 . This last property is easily computed as a function of the condensing pressure, $P_{cond} = P_{sat}(T_{cond})$ and the condenser outlet temperature, $T_3 = T_{cond} - \Delta T_{sc}$.

The heat transfer through condenser wall is calculated analogously to the compressor one.

$$\dot{Q}_c = UA_c(T_{wc} - T_{amb}) \quad (3.18)$$

The calculation of the new condenser surface temperature, T_{wc} , involve the solution of the energy conservation equation in the control volume on figure 20 and 21:

$$C_c \frac{dT_{wc}}{dt} = \dot{m}_{com}(h_2 - h_3) - \dot{Q}_c \quad (3.19)$$

Finally, the new condensing temperature is determined evaluating the condenser internal heat convection.

$$T_{cond,n} = T_{wc} + \frac{\dot{m}_{com}(h_2 - h_3)}{h_{conv,c}A_{ic}} \quad (3.20)$$

In Eq. 3.20 $h_{conv,c}$ means the average internal heat transfer coefficient on condenser which is the area averaged value of heat transfer coefficient in each region, calculated by Shah (2016) correlation for the two-phase flow, and by Gnielinski (1976) correlation for the superheated and subcooled regions, using the pull-down experimental tests. The stationary average heat transfer coefficient value, of the whole condenser, is $1216 \text{ W/m}^2\text{K}$. In Eq. 3.20 A_{ic} is the internal surface area of the condenser.

Capillary tube model (R134a system – figure 20): The capillary tube is divided into two portions (an adiabatic and a non-adiabatic, with the internal heat exchanger) and is considered thermally insulated from outside. In the sub-model of capillary

tube/internal heat exchanger the inputs are the condensing and evaporating pressures, P_{cond} and P_{evap} , the enthalpy on condenser outlet, h_3 , the enthalpy on evaporator outlet, h_5 , the mass flow rate provided by compressor, $\dot{m}_{com}$, the thermal conductance of the internal heat exchanger UA_{ihx} and the parameter $f_{\Delta P}$. The outputs are the temperature on compressor inlet, T_1 and the enthalpy on evaporator inlet, h_4 .

The state 3i is located at the refrigerant inlet section of the internal heat exchanger on capillary tube side (see figure 12). The temperature of this state needs to be found to compute the inlet thermodynamic state of the compressor, state 1. The pressure in this point is calculated through Eq. 3.21.

$$P_{3i} = P_{cond} - f_{\Delta P}(P_{cond} - P_{evap}) \quad (3.21)$$

The initial portion of the capillary tube is considered adiabatic and the enthalpy in the heat exchanger inlet is equals to h_3 . Then, it is carried out a verification of the fluid condition in the heat exchanger inlet to identify if it is sub-cooled liquid or two-phase fluid. For this purpose a saturation curve for R134a on liquid side was fitted. This way, it is possible to obtain the saturation temperature ($T_{sat,liq}$), in K, as a function of the enthalpy by Eq. 3.22. This expression is valid for pressures between 50 and 2500 kPa :

$$T_{sat,liq}(h) = -7 \cdot 10^{-10}h^2 + 0.001h + 102.91 \quad (3.22)$$

Next, the saturation temperature of state 3i can be found as a function of pressure as $T_{3i,sat} = T_{sat}(P_{3i})$. If $T_{3i,sat}$ is greater than $T_{sat,liq}$, the temperature of state 3i is $T_{3i} = T(P_{3i}, h_{3i})$, else, $T_{3i} = T_{3i,sat}$.

The compressor inlet temperature can be obtained by a differential energy balance in the heat exchanger, Eq. 3.23, and some algebraic steps, according to Jakobsen (1995):

$$\dot{m}_{com}c_{pv}dT_v = UA'_{ihx}(T_{cap} - T_v)dz \quad (3.23)$$

Where, c_{pv} is the specific heat of vapor on the differential element (suction line side), T_v , the vapor temperature, UA'_{ihx} (W/mK), the thermal conductance of the internal heat exchanger per length unit, T_{cap} , the fluid temperature in capillary tube side and dz , the length of the differential element, being the direction of the fluid on suction line, i.e., from the evaporator outlet to the compressor inlet, the positive one.

Assuming that the internal heat exchanger is a counterflow type and the temperature of fluid in capillary tube varies linearly from T_{3i} to T_5 (evaporator outlet temperature):

$$T_{cap} = T_5 + \frac{(T_{3i} - T_5)}{L_{ihx}}z \quad (3.24)$$

Where L_{ihx} is the length of the internal heat exchanger.

Replacing T_{cap} from Eq. 3.24 on 3.23:

$$dT_v = \frac{UA'_{ihx}}{\dot{m}_{com}c_{pv}} \left(\frac{T_{3i} - T_5}{L_{ihx}} z + T_5 - T_v \right) dz \quad (3.25)$$

Considering $a = \frac{UA'_{ihx}}{\dot{m}_{com}c_{pv}}$; $b = \frac{T_{3i} - T_5}{L_{ihx}}$ e $c = T_5$:

$$dT_v = a(bz + c - T_v) \quad (3.26)$$

With a substitution of variables: $n = b - a(bz + c - T_v)$, is obtained:

$$\frac{dn}{dz} = -an \quad (3.27)$$

Therefore, integrating the Eq. 3.27:

$$n = C_0 \exp(-az) \quad (3.28)$$

So, with the new value of n on Eq. 3.28:

$$b - a(bz + c - T_v) = C_0 \exp(-az) \quad (3.29)$$

Adopting that for $z = 0$, $T_v = T_5$, as a boundary condition it is possible to find $C_0 = b$. This way:

$$b - a(bz + c - T_v) = b \exp(-az) \quad (3.30)$$

$$\Rightarrow T_v = \frac{b}{a} (\exp(-az) - 1) + az + b \quad (3.31)$$

Considering a second boundary condition of $z = L_{ihx} \Rightarrow T_v = T_1$:

$$T_1 = T_{3i} - \frac{(T_{3i} - T_5)}{aL_{ihx}} [1 - \exp(-aL_{ihx})] \quad (3.32)$$

Which corresponds to:

$$T_1 = T_{3i} - \frac{(T_{3i} - T_5)}{\frac{UA_{ihx}}{\dot{m}_{com}c_{pv}}} \left[1 - \exp \left(-\frac{UA_{ihx}}{\dot{m}_{com}c_{pv}} \right) \right] \quad (3.33)$$

In the case of presence of liquid in the suction line (two-phase), an experimental expression was acquired for the heat transferred on the internal heat exchanger, $\dot{Q}_{ihx}$, in W .

Finally, with an energy balance on the internal heat exchanger it is possible to determine the enthalpy on evaporator inlet, h_4 .

$$h_4 = h_{3i} + h_5 - h_1 \quad (3.34)$$

The guessed evaporating temperature leads to the evaporator outlet temperature, T_5 , and, so, the enthalpy on its outlet, h_5 is defined.

Capillary tube model (R290 system – figure 21): This refrigerator does not have the internal heat exchanger, so, the capillary tube is assumed to be adiabatic in totality, causing an isenthalpic expansion. The only input is the enthalpy on condenser outlet and the output, the enthalpy on evaporator inlet. Therefore, $h_4 = h_3$.

Evaporator model (R134a system – figure 20): The evaporator sub-model is similar to the condenser one, however, it is divided in two parts: one on fresh-food compartment and other on freezer. Its inputs are mass flow rate, $\dot{m}_{com}$, temperatures of the air on fresh-food compartment and freezer, T_{ff} and T_{fr} , enthalpy on state 4, h_4 , thermal conductances (UA_{eff} , for the one on fresh-food compartment, UA_{efr} , on freezer), thermal capacities (C_{eff} , for the one on fresh-food compartment, C_{efr} , for the one on freezer and, C_{eq} , for an equivalent value considering both parts of evaporator) and the degree of superheating, ΔT_{sh} . The outputs are: the evaporator heat transfer rate, $\dot{Q}_e$, the evaporator surface temperature, T_{we} , and the evaporator outlet temperature and enthalpy, T_5 and h_5 . This last property is analogous to the condenser case, where it is function of the evaporator pressure, $P_{evap} = P_{sat}(T_{evap})$ and the evaporator outlet temperature, $T_5 = T_{evap} + \Delta T_{sh}$.

The heat transferred on each part of the evaporator is:

$$\dot{Q}_{eff} = UA_{eff}(T_{ff} - T_{weff}) \quad (3.35)$$

$$\dot{Q}_{efr} = UA_{efr}(T_{fr} - T_{wefr}) \quad (3.36)$$

Where $\dot{Q}_{eff}$ and $\dot{Q}_{efr}$ are the heat transfer rates on the evaporator on fresh-food compartment and on freezer, respectively, T_{weff} , the surface temperature of the evaporator on fresh-food compartment and T_{wefr} , the surface temperature of the evaporator on freezer. The total heat transfer rate is:

$$\dot{Q}_e = \dot{Q}_{eff} + \dot{Q}_{efr} \quad (3.37)$$

When the system is on, the surface temperature on each evaporator are very close, because of the low internal thermal resistance compared to the external one. Therefore, the evaporator is considered as a unit. So, to calculate the total evaporator surface temperature, T_{we} , the Eq. 3.38 is used.

$$C_{eq} \frac{dT_{we}}{dt} = \dot{Q}_e - \dot{m}_{com}(h_5 - h_4) \quad (3.38)$$

However, when the system turns off, with the end of the mass flow rate, the surface temperatures of each part of evaporator become considerably different, due the temperature difference on fresh-food compartment and freezer, which are the neighborhood of them. This way, the energy balance on each portion of evaporator are as presented on Eq. 3.39 and 3.40.

$$C_{eff} \frac{dT_{weff}}{dt} = \dot{Q}_{eff} \quad (3.39)$$

$$C_{efr} \frac{dT_{wefr}}{dt} = \dot{Q}_{efr} \quad (3.40)$$

On the simulation results all the temperatures plotted are the average values.

The new evaporating temperature is:

$$T_{evap,n} = T_{we} - \frac{\dot{m}_{com}(h_5 - h_4)}{h_{conv,e} A_{ie}} \quad (3.41)$$

Where $h_{conv,e}$ is the internal area averaged heat transfer coefficient on evaporator, calculated through the correlation of Liu & Winterton (1991) on the two-phase region and Gnielinski (1976) correlation for the superheated region. The average heat transfer coefficient on steady-state was $500 \text{ W/m}^2\text{K}$. A_{ie} is the internal surface area of evaporator (sum of the one on fresh food compartment, A_{ieff} , and on freezer, A_{iefr}).

Evaporator model (R290 system – figure 21): In this case, the refrigerator has only one evaporator, so, the modeling is analogous to the R134a case, but Eqs. 3.39 and 3.40. Also, despite the fact that there is no internal heat exchanger, the state 1 is equals to the state 5.

Cabinet model (R134a system – figure 20): The cabinet was also divided in two parts, the compartments. Inputs for this sub-model are the heat removed by the evaporator on each one, $\dot{Q}_{eff}$ and $\dot{Q}_{efr}$, and the thermal conductances and capacities

(UA_{ff} and C_{ff} for the fresh-food compartment, UA_{fr} and C_{fr} for the freezer and UA_w for the wall between the compartments).

The heat transfer rates from the ambient to the fresh-food compartment, $\dot{Q}_{ff}$, to the freezer, $\dot{Q}_{fr}$, and between compartments, $\dot{Q}_w$, are given by Eq. 3.42, 3.43 and 3.44.

$$\dot{Q}_{ff} = UA_{ff}(T_{amb} - T_{ff}) \quad (3.42)$$

$$\dot{Q}_{fr} = UA_{fr}(T_{amb} - T_{fr}) \quad (3.43)$$

$$\dot{Q}_w = UA_w(T_{ff} - T_{fr}) \quad (3.44)$$

Finally, with the energy balances on compartments, their internal temperature can be calculated:

$$C_{ff} \frac{dT_{ff}}{dt} = \dot{Q}_{ff} - \dot{Q}_{eff} - \dot{Q}_w \quad (3.45)$$

$$C_{fr} \frac{dT_{fr}}{dt} = \dot{Q}_{fr} - \dot{Q}_{efr} + \dot{Q}_w \quad (3.46)$$

Cabinet model (R290 system – Figure 21): In the refrigerator with R290, the cabinet modeling is exactly the same, but, with only one compartment, this way, there is no $\dot{Q}_w$.

In order to identify the effects of an extra thermal load inside the cabinet compartments, the presence of goods was also taken into account. It was considered the presence of goods with the same properties of meat in packages of 1 *kg*. Each package has a thermal conductance (UA_g) of 0.416 *W/K* and a thermal capacity (C_g) of 3400.0 *J/K*. This way, the heat transferred from the food to the air inside the cabinet (Q_{ffg} , on fresh-food compartment and Q_{frg} , on freezer) and the temperature of it (T_{ffg} and T_{frg} , respectively) can be determined with the following equations:

$$\dot{Q}_{ffg} = UA_g(T_{ffg} - T_{ff}) \quad (3.47)$$

$$\dot{Q}_{frg} = UA_g(T_{frg} - T_{fr}) \quad (3.48)$$

$$C_g \frac{dT_{ffg}}{dt} = -\dot{Q}_{ffg} \quad (3.49)$$

$$C_g \frac{dT_{frg}}{dt} = -\dot{Q}_{frg} \quad (3.50)$$

3.2.2 Capacitive model

In the capacitive model the sub-models for the condenser, capillary tube and evaporator change, in order to take into account effects of the fluid distribution in each component. In this model the transient version of the mass conservation equation is applied. The previous energy conservation equations applied in each of these three components also change to incorporate the effect of the consideration of fluid distribution. Besides the condenser and evaporator pressures are modeled.

Condenser model (R134a system – Figure 20): The new input is the mass flow through capillary tube, $\dot{m}_{cap}$, and the new outputs are: the refrigerant mass inside condenser, M_c , the quality on outlet (if two-phase), x_3 , the sub-cooling degree, ΔT_{sc} , and, the pressure on condenser, P_{cond} , and the condensing temperature, T_{cond} . The refrigerant mass in the condenser, M_c , can be determined using the continuity equation:

$$\frac{dM_c}{dt} = \dot{m}_{com} - \dot{m}_{cap} \quad (3.51)$$

The mass in the condenser is found by the integration of the equation above, knowing first the thermodynamic state at the condenser outlet. This state strongly depends on the condenser working regime, i.e., a condenser full of superheated vapor, a condenser with superheated vapor and a two-phase fluid, and a condenser with these three regions, superheated vapor, two-phase fluid and single-phase subcooled liquid.

Two parameters can be defined analysing the condenser isolated, with different amount of fluid, starting from zero. With very few refrigerant mass, the condenser will be completely filled with superheated vapor, therefore, on its outlet (state 3), there is only vapor. Increasing refrigerant mass until a value denoted by M_{vc} , the outlet will be saturated vapor, with quality $x_3 = 1$. Increasing again the refrigerant mass, the pressure will raise and a two-phase region will appear, this way, $0 < x_3 < 1$. When the condenser outlet become saturated liquid ($x_3 = 0$), the refrigerant mass will be denoted by M_{vlc} . These masses are calculated as follows:

$$M_{vc} = \rho_{vc} V_c \quad (3.52)$$

$$M_{vlc} = V_c [\alpha_c \rho_{vc,sat} + (1 - \alpha_c) \rho_{lc,sat}] \quad (3.53)$$

Where ρ_{vc} is the average density at the moment where the quality on condenser outlet is 1, V_c , the internal volume of condenser, $\rho_{vc,sat}$, the density of saturated vapor at

condensing temperature, $\rho_{lc,sat}$, the density of saturated liquid at same condition and α_c , the average void fraction when the appearance of sub-cooled liquid is imminent.

Still increasing the refrigerant mass inside condenser above M_{vlc} , the value denoted as M_{lc} , is reached and the condenser will be full of liquid, being its inlet saturated liquid:

$$M_{lc} = \rho_{lc} V_c \quad (3.54)$$

where ρ_{lc} the liquid density.

Now, considering that the quality in the condenser outlet varies linearly with the mass of refrigerant, the situation presented on figure 22 is obtained:

Figure 22: Quality on the condenser outlet in function of the refrigerant mass in it.

Source: created by the author.

By similarity of triangles, between the gray and the biggest one:

$$x_3 = \frac{M_{vlc} - M_c}{M_{vlc} - M_{vc}} \quad (3.55)$$

Thus, if the condenser outlet state is two-phase, this temperature will be $T_3 = T_{cond}$ and, if, according to Eq. 3.55, the quality is “negative”, it corresponds to outlet sub-cooled state. In this case, there is a portion of condenser filled with sub-cooled fluid. The external area of this portion, A_{sc} , can be calculated, considering that it varies linearly with the refrigerant mass inside condenser. Observe figure 23.

Again, by similarity of triangles between the gray and the biggest one:

$$A_{sc} = A_c \left(\frac{M_c - M_{vlc}}{M_{lc} - M_{vlc}} \right) \quad (3.56)$$

In the case of sub-cooled outlet, this temperature can be determined through an energy balance on a differential element in this region:

$$\dot{m}_{cap} c_{pl} dT = -U_{sc} (T - T_{amb}) dA_{sc} \quad (3.57)$$

Figure 23: Sub-cooled area on condenser, in function of the refrigerant mass in it.

Source: created by the author.

$$\Rightarrow \frac{dT}{(T - T_{amb})} = -\frac{U_{sc}dA_{sc}}{\dot{m}_{cap}c_{pl}} \quad (3.58)$$

Where c_{pl} is the specific heat of the liquid on the differential volume, T , its temperature and U_{sc} , the global heat transfer coefficient on the element.

Adopting $\theta = (T - T_{amb})$:

$$\frac{d\theta}{\theta} = -\frac{U_{sc}dA_{sc}}{\dot{m}_{cap}c_{pl}} \quad (3.59)$$

Integrating the Eq. 3.59 from $\theta_{cond} = (T_{cond} - T_{amb})$ to $\theta_3 = (T_3 - T_{amb})$, to take into account all the portion with sub-cooled liquid, it is obtained:

$$\theta_3 = \theta_{cond} \exp\left(-\frac{U_{sc}A_{sc}}{\dot{m}_{cap}c_{pl}}\right) \quad (3.60)$$

$$\Rightarrow T_3 = T_{amb} + (T_{cond} - T_{amb}) \exp\left(-\frac{UA_{sc}}{\dot{m}_{cap}c_{pl}}\right) \quad (3.61)$$

In Eq. 3.61, UA_{sc} is the thermal conductance on the sub-cooled region, determined as:

$$UA_{sc} = \left(\frac{UA_c}{A_c}\right) A_{sc} \quad (3.62)$$

In Eq. 3.62 A_c , is the external surface area of condenser. Applying an energy balance on condenser, the variation of the internal energy (U_c) with time is calculated by:

$$\frac{dU_c}{dt} = \dot{m}_{com}h_2 - \dot{m}_{cap}h_3 - h_{conv,c}A_{ic}(T_{cond} - T_{wc}) \quad (3.63)$$

For a two-phase fluid, the following expressions for total internal energy (U) and quality (x) are valid:

$$U = M[xu_{v,sat} + (1 - x)u_{l,sat}] \quad (3.64)$$

$$x = \frac{v - v_{l,sat}}{v_{v,sat} - v_{l,sat}} \quad (3.65)$$

$$\Rightarrow (1 - x) = \frac{v_{v,sat} - v}{v_{v,sat} - v_{l,sat}} \quad (3.66)$$

Where u is the specific internal energy, v , the specific volume and M the total mass.

Rearranging Eq. 3.64:

$$U = M \left(\frac{v_{v,sat}u_{l,sat} - v_{l,sat}u_{v,sat}}{v_{v,sat} - v_{l,sat}} + v \frac{u_{v,sat} - u_{l,sat}}{v_{v,sat} - v_{l,sat}} \right) \quad (3.67)$$

Following Jakobsen (1995), with $f_1 = \frac{u_{v,sat} - u_{l,sat}}{v_{v,sat} - v_{l,sat}}$ and $f_2 = \frac{v_{v,sat}u_{l,sat} - v_{l,sat}u_{v,sat}}{v_{v,sat} - v_{l,sat}}$:

$$U = Mvf_1 + Mf_2 = Vf_1 + Mf_2 \quad (3.68)$$

As $u_{v,sat}$, $u_{l,sat}$, $v_{v,sat}$ e $v_{l,sat}$ are properties on saturation conditions, they only depends on the pressure, P . So, applying the derivative in time to Eq. 3.68:

$$\frac{dU}{dt} = V \frac{df_1}{dt} + f_2 \frac{dM}{dt} + M \frac{df_2}{dt} \quad (3.69)$$

$$\Rightarrow \frac{dU}{dt} = V \frac{df_1}{dP} \frac{dP}{dt} + f_2 \frac{dM}{dt} + M \frac{df_2}{dP} \frac{dP}{dt} \quad (3.70)$$

Therefore:

$$\frac{dP_{cond}}{dt} = \frac{\frac{dU_c}{dt} - f_2 \frac{dM_c}{dt}}{M_c \frac{df_2}{dP_{cond}} + V_c \frac{df_1}{dP_{cond}}} \quad (3.71)$$

Where $\frac{dU_c}{dt}$ and $\frac{dM_c}{dt}$ are already known. The terms $\frac{df_1}{dP}$ and $\frac{df_2}{dP}$ can be calculated through polynomial fit, as Jakobsen (1995) did for R134a or explicit, with low time steps on simulations.

When the system is turned off, after a period of time, the condenser becomes filled with only superheated vapor. In this case, an approximation with the law of ideal gases is carried out, using the compressibility factor, Z , to do the correction, by the fact that the gas is not ideal.

$$Pv = ZRT \Rightarrow Z = \frac{Pv}{RT} \quad (3.72)$$

As $U = Mc_vT$:

$$\frac{dU}{dt} = c_v \frac{d(MT)}{dt} \quad (3.73)$$

Using the law of ideal gases:

$$MT = \frac{PV}{ZR} \Rightarrow \frac{d(MT)}{dt} = \frac{V}{ZR} \frac{dP}{dt} \quad (3.74)$$

Therefore,

$$\frac{dP}{dt} = \frac{ZR \frac{dU}{dt}}{c_v V} \Rightarrow \frac{dP_{cond}}{dt} = \frac{ZR \frac{dU_e}{dt}}{c_v V_c} \quad (3.75)$$

Condenser model (R290 system – Figure 21): The same approach is used. The calculations of f_1 and f_2 are explicit.

Evaporator model (R134a system – Figure 20): The new input is the mass flow through capillary tube, $\dot{m}_{cap}$, and the new outputs are: the refrigerant mass inside evaporator, M_e , the quality at outlet (if two-phase), x_5 , the superheating degree, ΔT_{sh} , and, the evaporator pressure, P_{evap} , and evaporating temperature, T_{evap} .

Analogously to the condenser, for the evaporator, the continuity equation is applied to determine the mass of refrigerant.

$$\frac{dM_e}{dt} = \dot{m}_{cap} - \dot{m}_{com} \quad (3.76)$$

Again, with an isolated analysis, a great amount of fluid inside the evaporator promote filling up of it with sub-cooled liquid. Decreasing this mass, until a value denoted by M_{le} , the outlet refrigerant state becomes a saturated liquid, i.e., $x_5 = 0$. Maintaining the mass diminution, the presence of two-phase occurs and, with an amount denoted by M_{vle} , the outlet is saturated vapor, i.e., $x_5 = 1$. Observe Eqs. 3.77 and 3.78.

$$M_{le} = \rho_{le} V_e \quad (3.77)$$

$$M_{vle} = V_e [\alpha_e \rho_{ve,sat} + (1 - \alpha_e) \rho_{le,sat}] \quad (3.78)$$

Where V_e is the total volume of evaporator (sum of the one on fresh-food compartment and freezer), and α_e , the average void fraction when the appearance of superheated

vapor is imminent. With a low mass of refrigerant, the evaporator will be completely filled with vapor and, with the mass M_{ve} (Eq. 3.79), the inlet is saturated.

$$M_{ve} = \rho_{ve} V_e \quad (3.79)$$

Again, analogously to the condenser case, assuming linear variation of the quality on evaporator outlet, x_5 , with the mass inside it:

$$x_5 = \frac{M_{le} - M_e}{M_{le} - M_{vle}} \quad (3.80)$$

If $0 \leq x_5 \leq 1$, the outlet temperature of evaporator will be $T_5 = T_{evap}$. On the other hand, if the quality is “greater than one”, according to Eq. 3.80, there is superheated vapor on the outlet, and the evaporator area with superheated fluid, A_{sh} , assuming linear variation with the mass, is:

$$A_{sh} = A_e \left(\frac{M_{vle} - M_e}{M_{vle} - M_{ve}} \right) \quad (3.81)$$

Where A_e is the total external area of evaporator (sum of the one on fresh-food compartment and freezer).

The outlet temperature in this case is calculated through the expression below, obtained with an energy balance on a differential element on evaporator outlet, analogous to the condenser.

$$T_5 = T_{ff} + (T_{ff} - T_{evap}) \exp \left(-\frac{U A_{sh}}{\dot{m}_{com} c_{pv}} \right) \quad (3.82)$$

Where c_{pv} is the specific heat of the vapor, and the thermal conductance on the area with superheated fluid, $U A_{sh}$:

$$U A_{sh} = \left(\frac{U A_{eff}}{A_{eff}} \right) A_{sh} \quad (3.83)$$

In Eq. 3.83 A_{eff} is the external area of the evaporator on fresh-food compartment, which is the region where the superheating occurs (the final portion of evaporator). Applying an energy balance on evaporator, the variation of the internal energy (U_e) with time is calculated:

$$\frac{dU_e}{dt} = \dot{m}_{cap} h_4 - \dot{m}_{com} h_5 + [h_{conv,e} A_{ieff} (T_{weff} - T_{evap}) + h_{conv,e} A_{iefr} (T_{wefr} - T_{evap})] \quad (3.84)$$

And the pressure is calculated through Eq. 3.85.

$$\frac{dP_{evap}}{dt} = \frac{\frac{dU_e}{dt} - f_2 \frac{dM_e}{dt}}{M_e \frac{df_2}{dP_{evap}} + V_e \frac{df_1}{dP_{evap}}} \quad (3.85)$$

Evaporator model (R290 system – Figure 21): Again, the same approach is used, but focusing on only one evaporator and compartment. The calculations of f_1 and f_2 are explicit.

Capillary tube model (R134a system – Figure 20): The capillary tube mass flow rate, can be modeled following the Eq. 3.86, according to Christensen & Knudsen (1993). The new inputs are the condensing and evaporating pressures, P_{cond} and P_{evap} , the specific volume in the capillary tube inlet, v_3 , and the sub-cooling degree, ΔT_{sc} . The new output is the mass flow rate, m_{cap} .

$$\dot{m}_{cap} = a \sqrt{\frac{(P_{cond} - P_{cap,out})}{v_3}} + b \Delta T_{sc} + c \quad (3.86)$$

The coefficients a , b and c are determined through experimental results. In the pull-down tests the mass flow rate on capillary tube was evaluated through the compressor mass flow rate in steady-state, where both values are the same. Through the polynomial provided by the manufacturer described in the experimental section, the compressor mass flow rate was acquired, so, the coefficients on Eq. 3.86 were fitted as: $a = 0.004384$, $b = 0.300938$, and $c = 0$.

At the capillary tube outlet it is possible to occur a critical flow due the high acceleration of fluid in the device. So, the Fauske's criterion (Fauske, 1962) was implemented to determine the critical pressure.

$$P_{crit} = G \sqrt{v_f P_f \phi} \quad (3.87)$$

Where,

$$\phi = 1.63(10^5) P_f^{-0.72} \quad (3.88)$$

The variables v_f , P_f are the specific volume and pressure, respectively, on the flash point, and G is the mass flux. Thus, the effective pressure at the tube outlet is:

$$P_{cap,out} = \max(P_{crit}, P_{evap}) \quad (3.89)$$

Capillary tube model (R290 system – Figure 21): The same approach was used. In this refrigerator the coefficients are $a = 0.0050416$, $b = 0.3460787$, and $c = 0$.

Compressor model (R134a system – Figure 20): To the compressor model, new calculations were added, to evaluate the mass of refrigerant in the shell, in the

discharge line and dissolved on oil. The inputs are the evaporator pressure, P_{evap} , the properties at states 1 and 2, and the compressor housing temperature, T_{com} . The output is the amount of refrigerant inside compressor, M_{com} .

The mass on compressor housing (M_s) is calculated through the average density (ρ_s), considering an average temperature between the inlet (state 1) and the housing temperatures, T_{1av} . The calculation on the discharge line (M_d) is analogous, based on the state 2.

$$M_s = \rho_s V_s \quad (3.90)$$

$$M_d = \rho_d V_d \quad (3.91)$$

The mass of refrigerant dissolved on oil (M_{dis}) is based on the work of Hermes (2006). The expression for this calculation, derived from the definition of solubility is:

$$M_{dis} = \frac{\sigma M_o}{1 - \sigma} \quad (3.92)$$

Where M_o is the mass of oil and σ , the solubility of refrigerant (R134a) on the oil (poliolester – POE 10). The data provided by the oil manufacturer was fit in a polynomial, presented on Hermes (2006), Eq. 3.93.

$$\sigma = a_0 P_r^{a_1} \exp[(c_0 + c_1 P_r) + (d_0 + d_1 P_r) \ln(1 + b_0 T_r^{b_1})] \quad (3.93)$$

Where P_r and T_r are the reduced pressure and temperature on refrigerant, based on P_{evap} and T_{1av} , respectively. The coefficients on Eq. 3.93 follows on table 14.

Table 14: Coefficients for Eq. 3.93.

a_0	a_1	b_0	b_1	c_0	c_1	d_0	d_1
0.0599	0.8940	-0.9721	-0.0872	-0.3002	-20.535	-0.6667	-5.6439

So, the total mass of refrigerant in the compressor is:

$$M_{com} = M_s + M_d + M_{dis} \quad (3.94)$$

3.2.3 Entropy generation

Using the second law of thermodynamics, Eq. 3.95, on the control volumes established, the entropy generation can be calculated. These calculations were conducted in steady-state with thermal and capacitive models and also in the transient working

regime with the capacitive model, because only this model provides information about mass content on components.

$$\dot{S}_{gen} = \sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} - \sum \left(\frac{\dot{Q}}{T} \right) + \frac{dS}{dt} \quad (3.95)$$

3.2.4 Control strategies

It is necessary to find the range of compressor rotations (on the refrigerator equipped with the variable speed compressor) that provide the best results in terms of electric consumption and, at the same time, returns an adequate performance according to the standards.

The compressor rotation has as limits 1600 and 4500 *rpm*, in order to ensure its correct operation, so, through the experimental data and simulations, it was verified that, in this refrigerator, with the thermostat maintaining an on/off control with established cycles (here, a cycle is defined as two consecutive periods of operation: one with compressor turned on and other while off), lower rotations provide a correct functioning with less energy consumption, as shown on table 15. Therefore, the evaluated logics were developed to seek lower rotations whenever possible.

Table 15: Simulation results presenting the refrigerator performance.

<i>Rotation (rpm)</i>	<i>COP</i>	<i>Consumption (kWh/month)</i>
1600	1.24	25.16
2000	1.13	27.80
3000	0.94	34.41
3600	0.89	36.33
4500	0.84	38.27

The first strategy studied is the “time-based control”, where the period of time which the compressor is on, t_{on} , in a cycle is evaluated. A target time, t_{target} , and a reference rotation, N_{ref} , are preset. If the compressor reaches this time while on, the controller evaluate the difference between its rotation and the reference one, N_{dif} . If it is greater than another established value, R , which operate as a “tolerance”, the compressor rotation will be increased, else, it is maintained. If the compressor reaches two times the target time, the rotation is increased. This increment is 500 *rpm*. If the compressor turns off before t_{target} , the rotation is decreased on the next cycle, as described by Eq. 3.96. Figure 24 presents a scheme of the logic.

The start up rotation for the next cycle is:

$$N_{rpm} = N_{avg} - 500 \left(1 - \frac{t_{on}}{t_{target}} \right) \quad (3.96)$$

Figure 24: Scheme of the time-based compressor control strategy.

Source: created by the author.

Where N_{avg} is the average rotation on the last cycle, when the compressor was activated. On the simulations, the target time was established as 60 *min*, the reference rotation as N_{avg} , and R as 100 *rpm*.

The second strategy is based on a proportional control and it is necessary the acquisition of the internal air temperature on fresh-food compartment, so, it is indicated to refrigerators with electronic thermostat. Figure 25 presents the diagram of the logic.

The parameters evaluated on this control are a target temperature, T_{set} , a proportionality constant, K_p , and the temperature inside the fresh-food compartment. The controller analyse the difference between T_{rr} and T_{set} , this way an “error” can be obtained. Then, the error is multiplied by K_p , resulting on a new rotation value, which is corrected by the limiter to be in the range of 1600 to 4500 *rpm*. On the simulations K_p was established as 1500 and T_{set} , the inferior temperature of thermostat.

Figure 25: Scheme of the proportional compressor control strategy.

Source: created by the author.

For the R290 system, the target time on the time-based strategy was 150 *min* and

the compressor rotation range, 2500 to 4500 *rpm*.

3.3 Numerical description

A code on Python language was developed to solve the mathematical models. The models equations are solved using an explicit temporal integration method, until the desired preset operation time is reached. Thus, as presented on the last section, the problem consists on solving a set of algebraic and ordinary differential equations (ODE's). Therefore, the 4th order Runge-Kutta method, present on Python's libraries, is applied to solve the ODE's at each time step. Other numerical issue is refrigerant properties calculations, handled with the fluid properties library CoolProp.

Some parameters are provided by the user such as: which compressor is considered, the control strategy to be applied, the presence or not of goods inside cabinet, the ambient temperature, compressor rotation (if on/off or pull-down tests), operation time and the desired time step. At the first step, some parameters are guessed as inputs: state 1, condensing and evaporating temperatures and the temperatures of the refrigeration system components, for the thermal model, in addition to the initial mass of refrigerant on heat exchangers and capillary tube mass flow rate, for the capacitive one.

In the thermal model, the calculations start on compressor sub-model, this way, the mass flow rate, electric power, heat rejection through the housing and the state 2, at the outlet, are determined. Then the calculations sequence are: on condenser sub-model: the heat transfer rate and the state 3 (condenser outlet); on capillary tube sub-model: the state 3i, where the adiabatic portion of the tube is over, the state 4 (evaporator inlet) and the new temperature on state 1; on evaporator sub-model: the cooling capacity and the state 5; and, finally, on the cabinet sub-model: the heat transfer rates through the compartments' walls and on the goods (if considered) are calculated.

Then, the 4th order Runge-Kutta method is applied to determine the temperatures on components surfaces and inside compartments. Now, using these last calculations, the new condensing and evaporating temperatures, and so, pressures, are determined, as well the complete definition of the new state 1. Following, the thermostat evaluate if the compressor should be turned off or maintained on and the controller calculates the new rotation. Finally, if the operation time is lower than the one established by the user, the process restarts, else, the calculations are done.

In the capacitive model, the calculations are the same on compressor, capillary tube and cabinet sub-models. On the condenser and evaporator sub-models, only the condenser heat rejection and the cooling capacity are determined. Then, the ODE's are solved for the temperatures on components surfaces and inside compartments. Now, the new capillary tube mass flow rate, state 3 (condenser outlet) and state 5 (evaporator

outlet) are calculated through Eqs. 3.86, 3.61 and 3.82, respectively. Also, $\frac{dU_c}{dt}$, $\frac{dU_e}{dt}$ and the mass of refrigerant on compressor are determined. Now the ODE's for the mass of refrigerant on condenser and evaporator are solved, being possible to determine the new pressures on condenser and evaporator, using the 4th order Runge-Kutta method. Therefore, the new condensing and evaporating temperatures and the new state 1 can be calculated, so, the thermostat, the controller and the operation time are evaluated deciding if the process restarts or finishes.

An important difference between the models solution structure can be noted: in the thermal model, using the superficial temperatures of condenser and evaporator, the condensing and evaporating temperatures are determined and, finally, the pressures; in the capacitive model, there is the calculation of the quality and the states of fluid on outlet, the mass of refrigerant and the internal energy on condenser and evaporator and mass of refrigerant on compressor. Following, the 4th order Runge-Kutta method is applied again to determine the condensing and evaporating pressures and, finally, the corresponding temperatures. The algorithms are presented in figures 26 and 27, respectively.

Figure 26: Algorithm of the thermal model simulation program.

Source: created by the author.

Figure 27: Algorithm of the capacitive model simulation program.

Source: created by the author.

4 Results

4.1 R134a refrigerator with constant speed compressor

4.1.1 Pull-down simulations (thermal model)

The pull-down tests were performed for ambient temperatures of 25, 32 and 43°C, according to the experiments for a constant compressor rotation of 3600 *rpm*. In figures 28, 29 and 30 are presented the obtained results. The solid symbols refer to the simulation results with the thermal model, while the hollow symbols refer to the experimental data for the R134a refrigerator with constant speed compressor.

It can be noted that the transient and steady-state simulated temperatures behave very similar regarding the experimental data. The highest discrepancies are observed for the condenser wall temperature. It should be noted that the numerical results are compared with the mean values of the experimental results, thus some differences are expected. In the case of condenser, the temperature of the inlet section is much higher than the temperatures of the other two sections (see Figure 16), leading to the observed differences between the simulated and experimental results.

The good agreement between the simulated and experimental temperature values indicates that the UA (crucial factor for the components temperatures in steady-state) and C (which determines how fast is the transient behavior of these temperatures) are coherent.

The temperatures start at the value of ambient temperature, evolving to the steady-state operational regime values, which takes about 8h to be established. The increasing on ambient temperature causes higher temperatures inside the compartments, so, the temperature and pressure on evaporator also raises. On the same way, with higher external temperature on condenser, these parameters also increase on it. The compressor housing temperature follows the ambient one. The increase of ambient temperature also produces an enhancement on the electric power consumed by the compressor and the consequent diminution of the system COP . This is a direct effect of the condenser and compressor temperature augmentations, leading to higher condensation pressure and to the necessity of more work to attain the stationary working regime.

Figure 28: Simulation and experimental results on pull-down test under 25°C : (a) temperatures; (b) compressor electric power.

(a)

(b)

Source: created by the author.

Figure 29: Simulation and experimental results on pull-down test under 32°C : (a) temperatures; (b) compressor electric power.

(a)

(b)

Source: created by the author.

Figure 30: Simulation and experimental results on pull-down test under 43°C : (a) temperatures; (b) compressor electric power.

(a)

(b)

Source: created by the author.

This effect can be seen clearly in figure 31 that presents a comparison between the Pxh diagram for the three ambient temperatures, considering the steady-state simulation results. A consequence of higher ambient temperatures is the increasing on the pressure difference between the condensation and evaporation and consequently the diminution of system COP , as explained.

Figure 31: Refrigeration cycle in steady-state of pull-down test for the ambient temperatures of 25, 32 and 43°C.

Source: created by the author.

The entropy generation results, obtained by the second law analysis, are presented in table 16 for the steady state regime and the three tested ambient temperature values. It is computed for the entire system and compressor, condenser, capillary tube and evaporator, considering $N_{rpm} = 3600 \text{ rpm}$. The entropy generation in the compressor is more intense, due the greater amount of irreversibilities of this component, followed by the capillary tube. As expected the increase in the ambient temperature leads to highest irreversibilities of the whole system and its components.

Table 16: Entropy generation of components in steady-state.

$T_{amb} (^{\circ}C)$	$\dot{S}_{com} (W/K)$	$\dot{S}_c (W/K)$	$\dot{S}_{cap} (W/K)$	$\dot{S}_e (W/K)$	$\dot{S}_{tot} (W/K)$
25	0.0867	0.0095	0.0267	0.0207	0.1436
32	0.1014	0.0113	0.0313	0.0222	0.1663
43	0.1273	0.0155	0.0397	0.0245	0.2071

The capacitive model was also used to determine the transient entropy generation on a pull-down test. The results are displayed in figure 32. At the start, the heat transfer rate on evaporator is very intense, due the great difference of temperature that occur between the air inside compartments (surrounding evaporator) and the evaporator surface.

After this period, the cooling capacity is reduced so, the compressor and the capillary tube gets the expected first and second positions in terms of entropy generation.

Figure 32: Entropy generation of components in a pull-down test.

Source: created by the author.

4.1.2 On/off operation (comparison between thermal and capacitive models)

Both, the simulations and experimental results with the refrigeration system in the on/off working regime were obtained for the following operational conditions: compressor velocity of $N_{rpm} = 3600 \text{ rpm}$, ambient temperature equal to $T_{amb} = 32^\circ\text{C}$, refrigerant charge of 105g and thermostat limits of 7 and 8.5°C on fresh-food compartment. This is the base line condition for the performed on/off experimental tests.

In figures 33, 34 and 35 are presented the high and low component temperature regions and the consumed compressor electric power, respectively, considering both, the thermal and the capacitive models.

Figure 33a and 33b presents the comparison between experimental and simulation results for the high temperature components. The average compressor housing temperatures are in good agreement for both models. Otherwise, the behavior of average condenser wall temperature is not well described with the thermal model. As mentioned above, the fast condenser wall temperature variation depends on the fluid charge variation when system turns off and, to represent it, the temperature of refrigerant must be calculated as a function of fluid pressure. In the thermal model, the pressure is calculated as a function of the condensing temperature. However, because the capacitive model is able to take the dynamics of fluid charge variation in the condenser, this model produces the correct transient behavior of average condenser wall temperature, as can be noted in figure 33b.

Figure 33: Simulation and experimental results for the higher temperatures on on/off operation: (a) thermal model; (b) capacitive model.

(a)

(b)

Source: created by the author.

The results for the regions with lower temperatures are presented on figures 34a and 34b. As the evaporator does not present a fast temperature variation when the system turns on and off, due the constant presence of liquid inside it, the results of both models are very close and present a similar behavior when compared with the experimental results. The same behavior is observed for the freezer, also presenting some differences with the acquired experimental temperature data. The temperature on freezer is maintained

Figure 34: Simulation and experimental results for the lower temperatures on on/off operation: (a) thermal model; (b) capacitive model.

(a)

(b)

Source: created by the author.

about -20°C with the assumed preset of thermostat. The transient behavior of fresh-food compartment temperature is very well simulated by both models.

The compressor electric power consumption is presented in figure 35. The characteristic peak when compressor starts is simulated by both models. Note that this power peak and the periods of time with compressor on and off are better described with the capacitive model, regarding the experimental data.

Figure 35: Simulation and experimental results for the electric power on on/off operation: (a) thermal model; (b) capacitive model.

(a)

(b)

Source: created by the author.

A comparison between the simulation results and the experimental data for some average performance parameters of the refrigeration cycle are shown in Table 17. Both models produce very good results in comparison with the experimental data, being the simulated results obtained with the capacitive model very close to experimental ones.

The thermal model simulates the refrigeration system in few seconds, producing good simulation results, as can be noted from Table 17, for the refrigeration system average

Table 17: Average performance parameters on simulations and experiment.

	COP	Consumption [kWh/month]	$\dot{Q}_{e,avg}$ [W]	$\dot{W}_{avg}$ [W]
Thermal	0.87	36.98	45.09	51.36
Capacitive	0.89	36.18	45.06	50.25
Experimental	0.88 ± 0.04	36.22 ± 0.53	44.31 ± 1.6	50.30 ± 0.74

performance parameters. This is the main advantage of this model, the production of good simulation results very quickly. Using this fact, some interesting parametric analysis was carried out with this model. Table 18 presents an evaluation of the internal and external heat transfer coefficients influence on average performance parameters, using the thermal model simulating the on/off operation with $N_{rpm} = 3600 \text{ rpm}$ and $T_{amb} = 32^\circ\text{C}$.

From the results below, it is clear that the external heat transfer is more dominant on the determination of the system's performance. As expected, the increasing of the heat transfer coefficients decreases the heat exchangers' thermal resistances, enhancing its thermal conductance and, so, the *COP*. However, as the thermal model does not consider the refrigerant charge variation inside components, the results below are limited to the model limitations. Nevertheless, the results shown in Table 18 demonstrate the usefulness of the thermal model for the development of a first project of the refrigeration system.

Table 18: Average performance parameters for simulations with the thermal model.

Condition	COP	Consumption [kWh/month]	$\dot{Q}_{e,avg}$ [W]	$\dot{W}_{avg}$ [W]	$\dot{Q}_{c,avg}$ [W]
Increasing 50% on UA_c	0.93	35.08	45.06	48.72	85.55
Increasing 50% on UA_e 's	0.91	34.84	44.10	48.39	80.45
Decreasing 50% on UA_c	0.78	41.43	45.06	57.54	65.01
Decreasing 50% on UA_e 's	0.65	51.70	46.53	71.81	67.77
Increasing 50% on $h_{conv,c}$	0.88	36.70	45.08	50.97	79.72
Increasing 50% on $h_{conv,e}$	0.89	36.37	45.00	50.52	80.76
Decreasing 50% on $h_{conv,c}$	0.86	37.71	45.10	52.38	79.88
Decreasing 50% on $h_{conv,e}$	0.73	44.38	44.99	61.64	78.74

Conversely, while the capacitive model takes much more time to perform the refrigeration system simulations, it produces results related to the refrigerant distribution inside the system components. This information is very valuable for a whole analysis of the system and of the heat exchangers, as well. In order to explore this fact, results provided only by the capacitive model for the on/off operation are presented on figures 36 to 39. Experimental results are also shown for comparison purpose.

Figures 36 and 37 bring the results for the mass flow rate on compressor and capillary tube, and the fluid pressures transient behavior, respectively. It can be noted the

Figure 36: Mass flow rate on compressor and on capillary tube.

Source: created by the author.

Figure 37: Pressures on condenser and on evaporator.

Source: created by the author.

higher mass flow rate on compressor when the system starts, which fills up the condenser and contributes to the pressure increase on it. Then, with the fast augmentation of the system pressure difference, the capillary tube mass flow rate, which depends on the pressure difference, increases, attaining the compressor mass flow rate values after some small period of time. When this occurs, the pressures in condenser and evaporator become more stable, coinciding with the experimental measured data. When the system turns-off, the

Figure 38: Mass of refrigerant on condenser, evaporator and compressor

Source: created by the author.

Figure 39: Sub-cooling and superheating degree.

Source: created by the author.

mass flow through compressor ceases, so the refrigerant migrates to evaporator, which is at lower pressure, through capillary tube till the system “pressure equalization”.

As a consequence of the above phenomena, the mass of refrigerant on the heat exchangers vary according to figure 38. When the mass flow rate on compressor is higher than the capillary tube one, the condenser is filled and, then, when the system turns off, the evaporator is filled through the mass flow rate on capillary tube. Finally, figure

39 shows the behavior of sub-cooling and superheating degrees on the heat exchangers. These results are compared with the experimental ones, showing a reasonable agreement. The experimental data show almost constant values of the sub-cooling and superheating degrees, while the simulation results show an increase of the superheating degree and a decrease of the sub-cooling degree until attaining the experimental values at the end of the turn-on period. But, overall, the differences are very small. From these results it is possible to establish that the condenser has three zones regarding the refrigerant thermodynamic state (superheated, two-phase and sub-cooled) and that the evaporator has two zones (two-phase and superheated).

4.1.3 Refrigerant charge analysis (capacitive model)

In order to explore in more details the capabilities of the capacitive model, in this section it is investigated how the refrigerant charge influences the system performance. Besides, this is an important evaluation of the robustness and accuracy of the capacitive model. This analysis was carried out for a compressor with a fixed velocity equal to $N_{rpm} = 3600 \text{ rpm}$ and $T_{amb} = 32^\circ\text{C}$. The refrigeration system was simulated for eight refrigerant charge amounts, namely: 50, 70, 95, 105, 115, 130, 140 and 160g of refrigerant. It should be commented that the presented experimental results in previous sections were obtained for 105g of refrigerant. The results are shown for the steady-state condition and for time average parameters at the pull-down operation, represented by solid and hollow symbols, respectively.

First it is shown the system COP in figure 40a. Note that, for steady-state, this parameter is higher between 105g and 140g, attaining the maximum value at 140g. For the pull-down operation, this parameter is maximum at about 105g which is the base line configuration, corresponding to experimental charge. This result is coherent, since this amount of refrigerant was chosen by the manufacturer with several pull-down tests to optimize it. The COP results are consistent with literature experimental studies by De Rossi *et al.* (2011) and Yusof *et al.* (2018). In figure 40a, it is also presented a comparison between the simulated COP for the steady state condition, calculated by definition, and two analytical expressions from literature: Jakobsen (1995) and Shelton and Grossmann (1985). The first correlation presented very good results in relation to the simulated ones. The second correlation presented an opposite behavior of the system COP with the refrigerant quantity. Other correlation tested (Ma *et al.*, 2017) does not fit well and is not shown for this reason. This analysis indicates that the theoretical performance relations should be used with care in the analysis of refrigeration systems.

Also the second law efficiency in steady-state is presented on figure 40b. Agreeing with the COP , the best efficiency is obtained using about 140g of refrigerant.

The highest COP values shown for both operational regimes in figure 40a coincide

Figure 40: System COP for steady state and pull-down conditions (a) and 2nd law efficiency for steady-state (b).

(a)

(b)

Source: created by the author.

with the highest cooling capacity values presented in figure 41, where there are also shown the electric power and heat rejection through compressor housing. More refrigerant on the system also causes higher compressor electric power. Combining the cooling capacity and compressor electric power variations are obtained the *COP* curves. As can be noted (see figure 41), the cooling capacity changes much more than the compressor electric power consumption for both operational conditions.

Figure 41: Cooling capacity (a), compressor electric power and heat rejection through the housing (b).

(a)

(b)

Source: created by the author.

The cooling capacity (figure 41a) with a low amount of refrigerant is smaller because, despite the high enthalpy variation, in general, between the inlet and the outlet of evaporator (dashed line, for steady-state as an example), the mass flow rate is insufficient to use this potential. The system mass flow rate is shown in figure 42. As can be noted, for small quantities of refrigerant the system mass flow rate is low. When the system refrigerant mass increases, its mass flow rate also increases, as expected, but the

evaporator enthalpy variation decreases. These facts promote gain on the cooling capacity until a maximum value, where, the decreasing of enthalpy variation overcome the growing system mass flow rate.

Figure 42: System mass flow rate (steady state and pull-down conditions) and fluid density at compressor inlet.

Source: created by the author.

The system mass flow rate has the same behavior of the fluid density at the compressor inlet. High fluid density values at the compressor inlet provide high system mass flow rate, because enter more mass into the compressor, and even if the compressor volumetric efficiency changes, the increase of the inlet fluid density is a preponderant fact.

The fluid density variation at the compressor inlet varies influenced by the superheating degree at suction line. If this parameter decreases, the fluid density at compressor inlet raises, causing the augmentation of the system mass flow rate, as explained. The superheating degrees on evaporator outlet and suction line, as well as, the sub-cooling degree are shown in figure 43 as a function of the system refrigerant charge.

The heat rejection through compressor housing, shown in figure 41, follows the behavior of the combined superheating on evaporator and on the internal heat exchanger (figure 43), so, high values of refrigerant charge cause the reduction on superheating degree, on compressor housing temperature and on the compressor heat rejection.

As the refrigerant amount is increased, the condenser is more filled and its pressure raises, so, the sub-cooling degree is increased. On the other hand at low refrigerant charges, the evaporator is more empty, so there is a great superheated area and then high superheating degree. With more refrigerant, the evaporator is filled and the superheating degree decreases, see figures 43a and 43b.

Figure 43: Sub-cooling degree (a) and superheating degrees on evaporator and suction line(b).

(a)

(b)

Source: created by the author.

In the figure 43b, it is presented also the superheating degree on the suction line. Note that this parameter gets higher values when there is vapor (or more vapor, if two-phase) in the tube. The high values of this parameter with low refrigerant charges are due the internal heat exchanger between the suction line and the capillary tube. Note that, with the highest refrigerant charges, there is two-phase fluid in the suction line. The internal heat exchanger, in these cases, raises the quality.

Figure 44: Condensing and Evaporating pressures.

Source: created by the author.

As expected, parameters such as condensing and evaporating pressures, and the mass on heat exchangers and in the compressor (figures 44 and 45) tends to raise with the increase of the system refrigerant charge. The condensing pressure shows a higher increase because the condenser receives a high pressure fluid from the compressor. As the pumped mass increases with the refrigerant charge, and the density of superheated vapor is low, the pressure also rises to accommodate more fluid in the same volume. The increase of the evaporating pressure is much lower, mainly because the superheated vapor at the evaporator outlet is aspirated by the compressor, creating a low pressure region. The condensing pressure increasing rate causes a rise on the electric power consumption.

As the sub-cooling (if it exists) increases or the quality on the condenser outlet decreases, the presence of liquid on condenser is higher, so, the mass of refrigerant in the condenser increases. The mass on evaporator, as well, is increased as the superheating (if it exists) or the quality on the evaporator outlet decreases, also due to more liquid content. This behavior can be realized on figures 45a for steady-state and pull-down conditions. In figure 45b it is presented the evaluation of the total mass of refrigerant on compressor and the portion dissolved on oil (POE10).

The refrigerant mass distribution in steady-state is shown in figure 46 for the whole refrigeration system. The behavior obtained is coherent with that presented by De Rossi *et al.* (2011), that experimentally studied a similar system. The refrigerant mass difference between the one provided by the sum of masses on compressor, condenser and evaporator and the total mass of the system is located in the other system components such as suction line, discharge line, connector tubes and filters. Excepting for the lower

Figure 45: Refrigerant mass on the heat exchangers (a) and on compressor (b).

(a)

(b)

Source: created by the author.

refrigerant charge of 50g, the evaporator always contains the higher mass of refrigerant. This is a characteristic of domestic vapor compression refrigeration systems.

Figure 47 presents the total entropy generation in steady-state. Note that the minimum entropy generation is at 140g of R134a, which is the refrigerant charge corresponding to the maximum COP value in steady state. In fact the entropy generation behavior should be opposite to the system COP . The same behavior should be obtained for the temperature inside compartments. In steady-state, the temperature established in-

Figure 46: Refrigerant distribution on system components.

Source: created by the author.

Figure 47: Total entropy generation of the refrigeration system in steady state.

Source: created by the author.

side them must be lower when the refrigerator operates with the most efficient refrigerant charges. These simulation results are shown in figure 48.

Figure 48: Temperature values (a) of fresh-food compartment, and (b) of freezer, in steady state.

(a)

(b)

Source: created by the author.

4.1.4 Effect of extra thermal load - goods

In this section the capacitive model is used to study the refrigeration system behavior under the influence of an additional thermal load promoted by the presence of goods inside the compartments. The simulations consider the presence of 6kg of meat in the fresh-food compartment and 3kg in the freezer. The numerical results are obtained

for various refrigerant charges, but it will be shown the system's behavior for 70g and 130g mass of refrigerant.

Figure 49: Temperatures of components surfaces, compartments air and goods for the operation with (a) 70g and (b) 130g of R134a.

Source: created by the author.

In the simulations, the goods, with initial temperature of 30°C , were put into the refrigerator on the minute 200. As soon as the goods are distributed inside the refrigeration compartment, the temperatures inside the compartments increase and starts a longer on period of the system. This behavior can be noted in figure 49. This period is longer in the

case with 70g due the lower cooling capacity.

Figure 50: Mass of refrigerant in the heat exchangers and compressor for the operation with (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

After this on period, when the system turns off, the operating time of each cycle is reduced more and more, as the goods temperature decreases, until their stabilization. At this moment, the temperature of the goods keeps near to the average one inside the compartments. The most elevated temperature of condenser wall with 130g, indicates that the condensing temperature and pressure are higher, as expected when the refrigerant

charge is increased. For this refrigerant charge the goods attain the desired temperature in a less time, mainly by a higher cooling capacity of the system. However, due to this more quickly goods refrigeration process the system cycles more. This can lead to a more energy consumption by the compressor for an specific analyzed time interval, and consequently can result in a smaller *COP*.

Figure 51: Sub-cooling and superheating degrees for the operation with (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

In figures 50a and 50b, the mass of refrigerant in the heat exchangers and in the

compressor are shown as a function of time. As expected, the mass on evaporator is considerably greater for the 130g case. In this case the refrigerant mass in the condenser and compressor oscillate in the same mass interval. When it is used 70g of refrigerant, the mass on evaporator is greater, but in some times the mass in the compressor attains the maximum mass value of the evaporator. The refrigerant mass in the condenser is smaller than in the evaporator, being almost equal when the system shuts down and starts.

The mass behavior shown in figure 50 is coherent to the behavior presented in figures 51a and 51b for the sub-cooling and superheating degrees. As the superheating degree is by far higher for the 70g charge, there is more presence of gas, so, the average density of the fluid on evaporator in this case is lower and, then, its total mass. In the same way, as the sub-cooling is a bit higher for the case with 130g charge, there is more presence of liquid, so, the mass is little greater on condenser in this case.

In figures 52 to 57 are presented average performance parameters related to the operation of the first on/off cycle where the goods are considered inside compartments. The system COP is presented in figure 52. The highest COP value is obtained for the system with 105g of refrigerant. For this case the cooling capacity is the highest, while the electric power is not, as can be seen in figure 53. This provides the best COP evaluated. However this COP value is almost equal to that obtained for 95g of refrigerant. As commented before, the COP for the system with 130g is smaller than that obtained for 70g.

Figure 52: On/off results for COP .

Source: created by the author.

The above commented behavior of the cooling capacity and compressor electric power for these two charges is clearly shown in figure 53. Note that the cooling capacity

of the system increases until $105g$ decreasing after this value, being, however, still higher for $130g$ than for $70g$ of refrigerant.

In figure 53, the same behavior of the previous cases for the steady-state condition can be noted. The cooling capacity grows until some refrigerant charge value, $105g$, and then starts to falls. This is a consequence of the mass flow rate augmentation shown in figure 55 and the evaporator enthalpy difference changing with the refrigerant charge.

Figure 53: On/off cooling capacity (a), compressor electric power and heat rejection through the housing (b).

Source: created by the author.

The compressor electric power increases with the refrigerant charge and the heat rejection reaches a maximum value at 105g and then decreases with the lower degrees of superheating. With more refrigerant and the heat exchangers more filled (figure 56), the sub-cooling degree increases and the superheating degree decreases, as can be noted in figure 54.

Figure 54: On/off sub-cooling degree (a) and superheating degree (b).

Source: created by the author.

As expected, with greater amounts of refrigerant, the mass flow rate (figure 55a) and the condensing and evaporating pressures (figure 55b) increase. Note that the previous explanation for the steady-state regime apply for explaining the condensing and evaporating pressure changes with the refrigerant charge augmentation.

Figure 55: On/off mass flow rate (a) and condensing and evaporating pressures (b).

Source: created by the author.

Figure 56: On/off refrigerant mass: (a) on heat exchangers and (b) on compressor.

(a)

(b)

Source: created by the author.

Finally, in figure 57, the operation time of the first on/off cycle is shown. As well as the cooling capacity increases with the refrigerant charge until some value, 105g, the operation time decreases. This means that the goods are cooled faster. In fact the operation time presents the opposite behavior of the cooling capacity as a function of the refrigerant charge. Remembering again the comparison of results for charges of 70g and 130g, it is noted in this figure that the higher refrigerant charge is characterized by a less operation time, and consequently a faster cooled capacity. However in this case the

compressor electric power is the highest leading to an small *COP* regarding that obtained for the 70g of refrigerant.

Figure 57: Operation time on the first cycle where the goods are put inside compartments.

Source: created by the author.

4.2 R134a refrigerator with variable speed compressor

4.2.1 Pull-down simulations (thermal model)

In this subsection are presented numerical results for pull-down tests performed on the R134a refrigerator with variable speed compressor and the thermal model. The results are compared with experimental data to make a simple validation of it. The tested and simulated conditions were $T_{amb} = 32^{\circ}C$, refrigerant charge of 105g and rotations of 1600, 3600 and 4500 *rpm*. The components temperatures and compressor electric power are shown in figures 58, 59 and 60 for the three analyzed rotations.

The simulation results are coherent with the experimental data. The expected behavior was achieved: with higher a rotation, the temperatures on compressor shell and the electric compressor power were increased. Also, the elevated rotations promote the increasing of the condensing pressure and the decreasing of the evaporating pressure, provoking the augmentation of condenser temperature and the reduction of evaporator temperature. The temperatures on fresh-food compartment and freezer were decreased.

Figure 58: Pull-down under 1600 rpm: (a) temperatures; (b) compressor electric power.

(a)

(b)

Source: created by the author.

Figure 59: Pull-down under 3600 rpm: (a) temperatures; (b) compressor electric power.

(a)

(b)

Source: created by the author.

Figure 60: Pull-down under 4500 rpm: (a) temperatures; (b) compressor electric power.

(a)

(b)

Source: created by the author.

4.2.2 Control strategies comparison

Several simulations of the refrigerator operating with the three control strategies, presented in section 3.2.4, were carried out. Figures 61 to 64 present the results of the system operating with $T_{amb} = 32^{\circ}\text{C}$, and a refrigerant charge of 105g, using the capacitive model.

Figure 61: Simulation of the on/off operation: (a) temperatures; (b) mass flow rate, (c) mass of refrigerant on heat exchangers and compressor; (d) pressures; (e) sub-cooling and superheating degrees.

Source: created by the author.

Figure 62: Simulation of the operation with proportional control strategy: (a) temperatures; (b) mass flow rate, (c) mass of refrigerant on heat exchangers and compressor; (d) pressures; (e) sub-cooling and superheating degrees.

Source: created by the author.

Figure 63: Simulation of the operation with time-based control strategy: (a) temperatures; (b) mass flow rate, (c) mass of refrigerant on heat exchangers and compressor; (d) pressures; (e) sub-cooling and superheating degrees.

Source: created by the author.

Figure 64: Comparison of (a) electric power and (b) compressor rotation with the three strategies.

(a)

(b)

Source: created by the author.

It is observed from the above figures that the temperature variations of the system components are smaller and more gradual when the variable speed control strategies are applied, including the temperatures inside compartments, which is a positive aspect to the food conservation. As a consequence of lower rotations, the mass flow rate decreases with the variable speed control strategies. These operating conditions cause the refrigerant mass diminution and augmentation in the condenser and evaporator, respectively, along the system operation time. This in turn provokes lower pressures on condenser and higher pressures on evaporator, leading also to the reduction of both sub-cooling and superheating degrees. The consequence of this behavior is the increase in system COP .

Some average performance parameters are presented on table 19. It can be noted the great gain on COP and the reduction of electric energy consumption with the application of variable speed control strategies. Also, due the low rotations the instantaneous cooling capacity is decreased, so the time of functioning periods are greater and the off periods are shorter, due the higher surface temperature of evaporator. In this case the time-based control strategy is that with the highest thermal performance. It should be noted that in this case it is simulated the refrigeration system with empty compartments.

Table 19: Average simulated performance parameters with the three control strategies.

Control strategy	COP	$\dot{Q}_{e,avg}$ [W]	$\dot{W}_{avg}$ [W]	Consumption [kWh/month]	t_{on} [min]	t_{off} [min]
On/off	0.85	44.70	52.90	38.09	24.29	23.44
Proportional	1.21	43.71	36.05	25.96	47.61	20.17
Time-based	1.31	43.77	33.30	23.97	57.39	20.33

Using the thermal model, the gains through the application of variable speed compressor associated to control strategies were evaluated in several conditions: two ambient temperatures (22 and 32°C) and three different preset of thermostat (min.: 3.0 to 4.5°C, med.: 5.0 to 6.5°C and max.: 7.0 to 8.5°C). The application of variable speed strategies have shown gains up to 31% on consumption reduction, see the results displayed in table 20. Nevertheless, it is important to highlight that the control adjustments to attend different requirements on a practical case may decrease the obtained gains. In the presented simulations the proportional strategy brings more advantages as the operational conditions become more severe.

Again, as presented on the mathematical modeling section, the sub-cooling and superheating degrees are fixed as inputs for the thermal model. These values were acquired experimentally for the five rotations tested (pull-down) and interpolated for the intermediate ones.

Table 20: Performance comparison at several conditions: changing thermostat, ambient temperature and compressor control strategy.

		Ambient: 22 °C			Ambient: 32 °C			
		Control	On/off	Prop.	t _{on}	On/off	Prop.	t _{on}
Thermostat (3 to 4.5 °C)	COP	1.01	1.24	1.34	0.9	1.24	1.17	
	Consumption (kWh/month)	25.66	20.44	18.81	41.38	29.62	31.24	
	Consumption Reduction (%)	-	20.34	26.69	-	28.42	24.50	
	$\dot{Q}_e$ (W)	36.11	35.31	34.99	51.59	51.09	50.64	
	Time on (min)	19.00	35.08	40.83	34.75	92.50	60.75	
	Time off (min)	30.58	26.27	26.58	19.67	16.5	17.25	
Thermostat (5 to 6.5 °C)	COP	1.04	1.28	1.32	0.90	1.20	1.21	
	Consumption (kWh/month)	22.80	17.62	17.06	38.59	28.63	28.03	
	Consumption Reduction (%)	-	22.72	25.18	-	25.81	27.36	
	$\dot{Q}_e$ (W)	32.92	31.39	31.38	48.03	47.56	47.05	
	Time on (min)	16.67	29.58	33.67	28.42	62.42	60.75	
	Time off (min)	34.75	30.25	30.17	21.58	18.33	18.75	
Thermostat (7 to 8.5 °C)	COP	1.10	1.33	1.37	0.89	1.20	1.26	
	Consumption (kWh/month)	18.97	15.09	14.62	36.33	26.26	25.08	
	Consumption Reduction (%)	-	20.45	22.93	-	27.72	30.97	
	$\dot{Q}_e$ (W)	29.02	27.79	27.99	44.79	43.75	43.91	
	Time on (min)	14.75	25.42	28.42	23.83	47.92	60.00	
	Time off (min)	39.75	34.83	34.67	23.67	20.33	20.33	

4.2.3 Control strategies comparison - influence of refrigerant charge

It is also interesting to evaluate the behavior of the refrigeration system with variable speed control strategies and different refrigerant charges. Next, figures 65 to 69 bring the results for the simulation of refrigerator with the proportional logic control with 70 and 130g of R134a, using the capacitive model.

The reduction of the cooling capacity causes considerably longer cycles on the 70g case. The compressor electric power and the rotation follows the same expected behavior as presented in figures 66 and 67. Concerning to the refrigerant mass distribution, with more refrigerant amount, the simulations show that the mass content in all components, as well as the mass difference between the heat exchangers, increase, as expected, see figure 68. This fact leads to the augmentation of cycle pressure differences, causing the observed higher compressor electric power in figure 66.

Figure 65: Temperatures of components for proportional control simulations: (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

Figure 66: Compressor electric power for proportional control simulations: (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

Figure 67: Compressor rotation for proportional control simulations: (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

Figure 68: Mass of refrigerant on heat exchangers and compressor for proportional control simulations: (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

Figure 69: Sub-cooling and superheating degrees with proportional control simulations: (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

With more refrigerant content, the sub-cooling degree is enhanced and the superheating degree, reduced as displayed in figure 69.

Figures 70 to 74 present the results for the simulations of the refrigerator under the time-based control, with 70 and 130g of refrigerant. The controller with this strategy set the compressor rotation in greater values, related to the minimum of 1600 *rpm*, with 70g. This is performed to avoid excessively long functioning periods (according to

the target time), which occur due to reduced instantaneous cooling capacity values. So, the compressor electric power is increased and the duration of on/off cycles is reduced, compared to the proportional case.

Figure 70: Temperatures of components for time-based control simulations: (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

Figure 71: Compressor electric power for time-based control simulations: (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

Figure 72: Compressor rotation for time-based control simulations: (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

Figure 73: Mass of refrigerant on heat exchangers and compressor for time-based control simulations: (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

Figure 74: Sub-cooling and superheating degrees for time-based control simulations: (a) 70g and (b) 130g of R134a.

(a)

(b)

Source: created by the author.

Table 21 shows some average performance parameters for all the above studies. All these results show that the proportional control is better in more severe conditions (70g of refrigerant), which is expected due the faster system reaction provided by this logic. The better COP is obtained in this case for the time-based control strategy with 115g of R134a. These simulations are performed for empty compartments.

Table 21: Average performance parameters for simulations using control strategies with different refrigerant charges.

Control strategy	COP	$\dot{Q}_{e,avg}$ [W]	$\dot{W}_{avg}$ [W]	Consumption [kWh/month]	t_{on} [min]	t_{off} [min]
Proportional 70g	1.16	43.00	34.04	26.67	202.89	17.31
Time-based 70g	1.05	42.68	40.51	29.16	61.96	18.94
Proportional 95g	1.18	43.47	36.95	26.61	52.62	19.78
Time-based 95g	1.28	43.27	33.74	24.30	60.20	20.10
Proportional 105g	1.21	43.71	36.05	25.96	47.61	20.17
Time-based 105g	1.31	43.77	33.30	23.97	57.39	20.33
Proportional 115g	1.20	43.74	36.43	26.23	47.74	20.20
Time-based 115g	1.32	43.70	32.96	23.73	56.61	20.39
Proportional 130g	1.15	43.91	38.14	27.46	49.20	20.13
Time-based 130g	1.27	43.76	34.38	24.75	58.17	20.31

4.2.4 Extra thermal load (goods) and variable speed strategies

The following simulations are performed with the capacitive model to study the system behavior with variable speed strategies, $T_{amb} = 32^\circ C$, 105g of R134a and the presence of goods inside the cabinet. In figure 75, it is shown how the temperatures increase at minute 200, when the goods are inserted into the compartments (6kg in the fresh-food compartment and 3kg in the freezer).

The graphs displayed in figures 76 and 77, clearly show how the proportional control, with electronic thermostat and reading the temperature on fresh-food compartment, is faster than the time-based control to detect the situation inside the cabinet, and this fact directly reflects on the system behavior. When the food is put inside the compartments, the temperature inside them raises, so, the electronic thermostat turns on the compressor. As the calculated speed for proportional control is based on the error between the preset inferior limit temperature and the one on the fresh-food compartment, the speed (and the electric power) increases until the limit of 4500 rpm. This way the cooling capacity is maximized and the temperature in the compartments decreases quickly.

With the time-based control, when the temperature on fresh-food compartment reaches the thermostat superior limit, the system is turned on and maintains 1600 rpm up to the target time, so, a new rotation is calculated and this process is repeated until the thermostat inferior limit is established and the system turns off. Then, in the next cycles, the rotation is decreased and becomes 1600 rpm again.

The mass flow rate, shown in figure 78, has the same behavior of the compressor rotation. When the mass flow rate is increased, due the rotation augmentation, the refrigerant mass on condenser increases and on evaporator decreases and the opposite also occurs, which is coherent, see figure 79.

Figure 75: Temperatures of components and compartments air with the variable speed control strategies and goods inside compartments: (a) proportional, (b) time-based.

(a)

(b)

Source: created by the author.

Figure 76: Compressor electric power with the variable speed control strategies and goods inside compartments: (a) proportional, (b) time-based.

(a)

(b)

Source: created by the author.

Figure 77: Compressor rotation with the variable speed control strategies and goods inside compartments: (a) proportional, (b) time-based.

(a)

(b)

Source: created by the author.

Figure 78: Mass flow rate through compressor and capillary tube with the variable speed control strategies and goods inside compartments: (a) proportional, (b) time-based.

(a)

(b)

Source: created by the author.

Figure 79: Mass of refrigerant on compressor, condenser and evaporator with the variable speed control strategies and goods inside compartments: (a) proportional, (b) time-based.

(a)

(b)

Source: created by the author.

Figure 80: Pressure on condenser and on evaporator with the variable speed control strategies and goods inside compartments: (a) proportional, (b) time-based.

(a)

(b)

Source: created by the author.

The pressures behavior shown in figure 80 is also coherent according to mass flow rate variations. When the mass flow rate is increased, the pressure on condenser raises and on evaporator decreases; when it is reduced, the inverse occurs.

Finally, with higher mass flow rate, the evaporator is emptier and the condenser is more filled, so, the superheating degree increases, as well the sub-cooling degree. On the other hand, with lower mass flow rate, the evaporator, with more refrigerant content,

presents lower superheating degree and the condenser, with less refrigerant, lower sub-cooling degree. This behavior is displayed in figure 81.

Figure 81: Sub-cooling and superheating degrees with the variable speed control strategies and goods inside compartments: (a) proportional, (b) time-based.

(a)

(b)

Source: created by the author.

Table 22 presents some performance parameters of the system operation for the whole operation time simulated. The gain on COP and consumption is evident with variable speed applications and there are the data of the goods cooling time, t_c , for the cycle when the goods are put inside compartments. It can be noted the fact that

the proportional control provided faster reading of the fresh-food compartment situation caused higher COP and less energy consumption.

Table 22: Average performance parameters of simulations using the control strategies with goods in compartments.

Control strategy	COP	$\dot{Q}_{e,avg}$ [W]	$\dot{W}_{avg}$ [W]	Consumption [kWh/month]	t_c [min]
On/off	0.96	58.46	60.77	43.76	232.79
Proportional	1.25	57.84	46.28	33.32	388.57
Time-based	1.19	57.44	48.16	34.67	318.67

Table 23: Performance comparison at several conditions: changing thermostat, ambient temperature and compressor control strategy, with goods in compartments.

		Ambient: 22 °C			Ambient: 32 °C			
		Control	On/off	Prop.	t _{on}	On/off	Prop.	t _{on}
Thermostat (3 to 4,5 °C)	COP	1.15	1.39	1.42	1.04	1.31	1.15	
	Consumption (kWh/month)	31.79	25.97	25.11	45.61	36.42	42.28	
	Consumption Reduction (%)	-	18.31	21.01	-	20.15	7.30	
	$\dot{Q}_e$ (W)	50.88	50.12	49.59	65.79	65.96	67.32	
	Cooling Time	205.00	295.75	286.33	372.17	633.25	385.08	
Thermostat (5 to 6,5 °C)	COP	1.16	1.42	1.43	1.01	1.30	1.17	
	Consumption (kWh/month)	29.26	23.40	22.82	44.44	34.51	37.90	
	Consumption Reduction (%)	-	20.02	22.01	-	22.34	14.72	
	$\dot{Q}_e$ (W)	47.07	46.15	45.27	62.32	62.19	61.36	
	Cooling Time	162.50	251.92	247.75	282.75	496.67	337.67	
Thermostat (7 to 8,5 °C)	COP	1.19	1.45	1.47	1.00	1.28	1.22	
	Consumption (kWh/month)	25.90	20.96	20.10	42.18	32.68	34.24	
	Consumption Reduction (%)	-	19.07	22.39	-	22.52	18.82	
	$\dot{Q}_e$ (W)	42.86	42.11	41.15	58.62	57.88	58.34	
	Cooling Time	125.92	193.83	217.42	234.92	354.08	316.5	

The thermal model was used to evaluate several performance parameters, varying the thermostat set and the ambient temperature, now with the addition of the goods inside compartments. The results are displayed in table 23. The control strategies presented better results when compared to the on/off operation. The simulated results are coherent,

as the consumption, the time of goods cooling on the first cycle and the cooling capacity increase on more severe conditions. The time-based control produced the highest system COP at ambient temperature of $22^\circ C$ for all thermostat settings. In the case of $T_{amb} = 32^\circ C$ the proportional control provided the highest COP values. This reinforces the fact that this control is preferred in more disfavoured working conditions.

Simulation results with the capacitive model for the refrigerator with goods in cabinet and refrigerant charge variation are shown in Table 24. The proportional control resulted in the best COP values for all the refrigerant charges. The system performance increases for the charges about 95 and 105g.

Table 24: Average performance parameters of simulations using the control strategies with goods in compartments and different refrigerant charges.

Control strategy	COP	$\dot{Q}_{e,avg}$ [W]	$\dot{W}_{avg}$ [W]	Consumption [kWh/month]	t_c [min]
Proportional 70g	1.17	55.62	47.37	34.10	901.20
Time-based 70g	1.02	56.01	54.79	39.45	365.75
Proportional 95g	1.31	61.74	47.11	33.92	525.18
Time-based 95g	1.17	57.01	48.88	35.19	322.39
Proportional 105g	1.25	57.84	46.28	33.32	388.57
Time-based 105g	1.19	57.44	48.16	34.67	318.67
Proportional 115g	1.22	57.82	47.35	34.09	385.94
Time-based 115g	1.15	57.43	49.84	35.89	320.73
Proportional 130g	1.19	58.26	48.99	35.27	405.81
Time-based 130g	1.08	57.23	52.93	38.11	326.8

4.3 R290 refrigerator

4.3.1 Pull-down simulations with thermal and capacitive models

The pull-down experimental tests were performed for ambient temperatures of $25^\circ C$ and a compressor rotation of 4500 *rpm*. These conditions are used for the numerical simulations. In figures 82 and 83 are presented the obtained results, allowing to a comparison between simulation results and experimental data. The solid symbols refer to the simulation results with the thermal and capacitive models, while the hollow symbols refer to the experimental data for the R290 refrigerator with variable speed compressor.

On the next figures it can be observed that the transient and steady-state simulated temperatures are close to the experimental measurements. The highest discrepancies are observed in the transient value of the cabinet temperature, see figures 82a and 83a. In the stationary state the simulation results match very well the experimental ones. This includes the refrigeration cycles shown in the Pxh diagrams, see figures 82b and 83b. Through these Pxh diagrams, it is clear that the capacitive model presented better results,

mainly regarding the accuracy for computing the system pressures. The good agreement between the simulated and experimental temperature values indicates that the thermal conductances (UA) and the thermal capacities (C) were correctly determined.

Figure 82: Simulation (thermal model) and experimental results on pull-down test: (a) temperatures; (b) P_{xh} diagram in steady state.

(a)

(b)

Source: created by the author.

Figure 83: Simulation (capacitive model) and experimental results on pull-down test: (a) temperatures; (b) P-h diagram in steady state.

(a)

(b)

Source: created by the author.

In terms of entropy generation, the table 25 presents the results in steady-state, with a second law analysis, $N_{rpm} = 4500 \text{ rpm}$ and $T_{amb} = 25^\circ\text{C}$ for compressor, condenser, capillary tube, evaporator and total.

Again, the entropy generation on compressor is more intense, due the greater amount of irreversibilities, followed by the one on capillary tube.

Table 25: Entropy generation of components in steady-state.

$\dot{S}_{com}$ (W/K)	$\dot{S}_c$ (W/K)	$\dot{S}_{cap}$ (W/K)	$\dot{S}_e$ (W/K)	$\dot{S}_{tot}$ (W/K)
0.4211	0.0614	0.1209	0.0807	0.6842

4.3.2 On/off operation (comparison between thermal and capacitive models)

Figure 84: Simulation and experimental results for the temperatures on on/off operation: (a) thermal model; (b) capacitive model.

(b)

Source: created by the author.

The simulations and experimental results with the R290 refrigeration system in the

on/off working regime were obtained for the following operational conditions: compressor velocity of $N_{rpm} = 4500 \text{ rpm}$, ambient temperature equal to $T_{amb} = 25^\circ\text{C}$, refrigerant charge of 103g of R290 and thermostat limits of -22.3 to -19.5°C . In figures 84 and 85 are presented the component temperatures (simulated and measured) and the electric power consumption (simulated and measured). The simulation results were obtained with the two models in the on/off operation condition.

Figure 85: Simulation and experimental results for the electric power on on/off operation: (a) thermal model; (b) capacitive model.

(a)

(b)

Source: created by the author.

The capacitive model presents better results, for both, component temperatures and electric power. Comparing the results in figures 84a and 84b, the simulated compressor housing temperature using the capacitive model fits better to the experimental data. The influence of the mass distribution dynamics on the most unstable periods (when the system starts and turns off) is well described by the capacitive model. Due to this fact the simulation of condenser and evaporator temperatures is also improved. The condenser wall temperature values are those with higher errors regarding the experimental data. But, it should be noted that are compared the average experimental values, which were locally measured in different regions, with one mean value obtained by the numerical simulations.

The simulated and experimental values of the compressor electric power, shown in figures 85a and 85b, demonstrate that the capacitive model produces a better behavior of the electric power. In these figures it is also clear that the duration of on and off periods are more adjusted to the experiment when it is used the capacitive model. It should be noted that the off periods are relatively short. This is explained by the low temperature levels at which the refrigerator works and also due to its great size and due to the operation of the defrost electric resistor. The graphs shown in figure 85 combine the electric power of compressor and fan for the on periods and of the defrost resistor for the off periods, this way, the power is not zero when the system turns off.

In table 26 are presented important average performance parameters obtained from the simulations and the experimental test. Comparing the simulated results and the experimental data, it can be affirmed that the capacitive model leads to better numerical results.

Table 26: Average performance parameters obtained by simulations and experimental tests.

	COP	Consumption [kWh/month]	$\dot{Q}_{e,avg}$ [W]	$\dot{W}_{avg}$ [W]
Thermal	0.71	197.01	196.09	273.62
Capacitive	0.63	216.08	188.72	300.11
Experimental	0.65	222.50	201.01	309.03

Next, are presented simulation results related to the refrigerant distribution inside the system components. These results are obtained with the capacitive model for the on/off operation. The results for mass flow rate on compressor and capillary tube, condensing and evaporating pressures, mass of refrigerant on heat exchangers and sub-cooling and superheating degrees are presented in figures 86 to 89. Experimental results are also shown for comparison purpose.

The analysis to be performed is analogous to the one for the R134a refrigerator with the constant speed compressor (section 4.1.2). In fact the obtained results are very

Figure 86: Mass flow rate on compressor and on capillary tube.

Source: created by the author.

Figure 87: Pressures on condenser and on evaporator.

Source: created by the author.

similar. Follow a briefly explanation.

Figures 86 and 87 show the transient behavior of the mass flow rate on compressor and capillary tube, and the fluid pressures transient behavior, respectively. As in the previous case, it is noted a higher mass flow rate on compressor when the system starts, filling up the condenser and increasing its pressure. This take place in the few first seconds or minute. With the fast augmentation of system pressure difference, the capillary tube mass

Figure 88: Mass of refrigerant on condenser, evaporator and compressor

Source: created by the author.

Figure 89: Sub-cooling and superheating degree.

Source: created by the author.

flow rate increases and attains the compressor mass flow rate values after this first some small period of time. When this occurs the pressures in condenser and evaporator become more stable, coinciding with the experimental measured values. When the system turns off, the mass flow through compressor ceases, so the refrigerant migrates to evaporator, which is at lower pressure, through capillary tube till the system “pressure equalization”. As a consequence the mass of content on the heat exchangers vary according to figure 88.

Finally figure 89 shows the behavior of sub-cooling and superheating degrees on the heat exchangers. These results are compared with the experimental ones, showing a reasonable agreement. The highest differences between experimental and numerical results are obtained for the superheating degree, but the transient profiles of both is qualitatively very similar. In the case of sub-cooling, degree the numerical and experimental results show a very good agreement. From the results it is possible to establish that the condenser has three zones regarding the refrigerant thermodynamic state (superheated, two-phase and sub-cooled) and that the evaporator has two zones (two-phase and superheated).

4.3.3 Effect of extra thermal load - goods (simulation vs. experiment)

In this section it is evaluated the influence of the goods presence inside the cabinet of the R290 refrigeration system. In order to perform this study, the capacitive model was used to simulate the refrigerator with 400kg of meat, $N_{rpm} = 4500\text{ rpm}$, $T_{amb} = 25^\circ\text{C}$ and 103g of R290.

In figures 90 to 95 and are shown the components temperatures, compressor electric power, mass flow rate through compressor and capillary tube, condensing and evaporating pressures, refrigerant mass in the heat exchangers and the sub-cooling and superheating degrees.

Figure 90: Components surface and goods temperatures.

Source: created by the author.

Figure 91: Refrigerator electric power consumption.

Source: created by the author.

Figure 92: Mass flow rate on compressor and capillary tube.

Source: created by the author.

Figure 93: Pressures on condenser and evaporator.

Source: created by the author.

Figure 94: Mass on the heat exchangers.

Source: created by the author.

Figure 95: Sub-cooling and superheating degrees.

Source: created by the author.

The presence of goods in the refrigerator provoked an augmentation of condensing and evaporating pressures, of the sub-cooling and superheating degrees, as well as, of the operating time of on phase. Overall, the simulated results show a very good agreement with the experimental data, allowing to conclude that the capacitive model provides trustworthy results.

In table 27 are shown the average performance parameters obtained from simulations and those obtained with the available experimental data.

Table 27: Average performance parameters on simulations and experiment.

	COP	Consumption [kWh/month]	$\dot{Q}_{e,avg}$ [W]	$\dot{W}_{avg}$ [W]	t_{on} [min]	t_{off} [min]
Simulation	0.70	225.06	219.63	312.59	330.37	8.40
Experimental	0.72	238.46	237.07	331.20	329.0	10.0

4.3.4 Control strategies comparison

Several simulations of the refrigerator operating with the three control strategies, presented in section 3.2.4, were carried out. Figures 96 to 99 present the simulation results for the R290 refrigerator with the proportional and time-based control strategies with the capacitive model operating for $T_{amb} = 25^\circ C$ and with a charge of 103g of R290.

Figure 96: Simulation results for (a) electric power and (b) compressor rotation with the proportional strategy.

(a)

(b)

Source: created by the author.

Figure 97: Simulation results for (a) electric power and (b) compressor rotation with the time-based strategy.

(a)

(b)

Source: created by the author.

Figure 98: Simulation of the operation with proportional control strategy: (a) temperatures; (b) mass flow rate, (c) mass of refrigerant on heat exchangers and compressor; (d) pressures; (e) sub-cooling and superheating degrees.

Source: created by the author.

Figure 99: Simulation of the operation with time-based control strategy: (a) temperatures; (b) mass flow rate, (c) mass of refrigerant on heat exchangers and compressor; (d) pressures; (e) sub-cooling and superheating degrees.

Source: created by the author.

In table 28 are presented average performance parameters acquired through the simulations with the two variable speed control strategies. In the present case the time-based control strategy produced the better performance results.

Table 28: Average performance parameters on simulations of control strategies.

	COP	Consumption [kWh/month]	$\dot{Q}_{e,avg}$ [W]	$\dot{W}_{avg}$ [W]
On/off	0.63	216.08	188.72	300.11
Proportional	0.71	191.84	189.69	266.45
Time-based	0.74	182.27	186.27	253.15

The rotation limits for the simulated compressor, according to the manufacturer, are 2500 and 4500 *rpm*. However, the inferior limit adopted was 3000 *rpm*, because with 2500 *rpm*, the refrigerator is maintained always on and the temperature does not reach the inferior limit of thermostat. The target time on time-based control was increased to 150 *min*, to allow the compressor average speed becomes lower; with reduced times, the rotation is enhanced to higher values, providing reduced gains.

Again, in comparison to the on/off case, the temperature variations are lower and slower, the instantaneous electric power consumption is reduced and the operating time on each cycle is amplified due the decreasing on the cooling capacity because of lower rotations and, then, the mass flow rate. Also the sub-cooling and superheating degrees are decreased, while the condensing and evaporating pressures changes are small. All these events lead to the increase of the system *COP*.

5 Conclusions

Two approaches for modeling and simulation of vapor compression refrigeration systems were developed, using the Python language and based on the work of Jakobsen (1995). Besides, an experimental procedure with data post-processing was presented to provide some important parameters to the models and to compare the simulation results to the experimental ones.

The developed models are able to provide important performance parameters such as COP , energy consumption, system and components entropy generation, surfaces and internal compartments temperatures (thermal model), pressures, mass distribution on compressor and heat exchangers, mass flow rate on compressor and capillary tube and sub-cooling and superheating degrees (capacitive model). Also, the mass of refrigerant dissolved on compressor oil and verification of choked flow on capillary tube, were implemented.

Several parametric analysis were carried out, evaluating the effects of ambient temperature and refrigerant charge changing and the introduction of goods inside compartments. Also were analysed the advantages of using variable speed compressors, associated to rotation control strategies.

Two different refrigeration systems (two compartments and one compartment) with two different working fluids (R134a and R290) were tested and simulated. The comparison of these both results showed that the simulations are consistent and coherent to the expectations and literature works. For example, the numerical results obtained for the analysis of refrigerant charge influence coincide with the experimental findings of De Rossi *et al.* (2011) and Yusof *et al.* (2018). These results are very interesting for the project of domestic refrigeration systems.

Among the results obtained, the main conclusions are:

- The ambient temperature increasing promotes the reduction of COP , the increasing of all system components temperatures, electric power consumption and pressure difference between condenser and evaporator.
- The steady-state entropy generation on compressor is greater due its massive presence of irreversibilities, followed by the capillary tube. The total entropy generation follows the ambient temperature trends, i.e., it is higher for higher ambient temperatures.
- The fluid mass distribution has a great influence on the system behavior, mainly

on the most unstable conditions (when turns on or off). This fact conducts fast temperature variations especially on condenser.

- The capacitive model proved to me more accurate than the thermal, since it considers aspects of the refrigerant mass variation in the system components, which was clear on the results.
- The thermal model proved to be a great feature for a first project and some parametrical analysis, considering its higher simplicity, fast processing and mean results coherence.
- The capacitive model, produced more complex results computing phenomena such as: mass flow rate on compressor and capillary tube, pressures on condenser and evaporator, mass content on heat exchangers and compressor, and sub-cooling and superheating degrees. This model proved to be a great feature for more detailed project of the system, for its optimization and for understanding and predicting some other processes.
- The refrigerant charge evaluation showed that there is an optimum value where the COP and the second law efficiency are maximized. The cooling capacity has the same behavior and the electric power consumption, the mass flow rate, the sub-cooling, the pressures and the masses on heat exchangers and compressor grows with more amount of refrigerant. The heat transfer through compressor housing and the superheating after the internal heat exchanger reach a peak and, then, decreases. The useful superheating (on evaporator) decreases. The refrigerant distribution among the components was also evaluated and this result, as well as those mentioned before are in good agreement with literature, for example, the experimental works of De Rossi *et al.* (2011) and Yusof *et al.* (2018).
- The evaluation of the influence of goods inside the refrigerator and the application of variable speed compressor were also studied and the models performed well. With greater rotations, the system presents higher temperatures on the high temperature regions and electric power consumption as well as lower temperatures on low temperature regions. This way, more rotation implies on greater pressure difference, cooling capacity and mass flow rate. Therefore, both sub-cooling and superheating are decreased.
- The application of variable speed compressor associated to speed control strategies promotes great gains in several conditions simulated. Energy consumption reductions up to 31% were obtained.

- The proportional logic proved to be more efficient than the time-based one, as, in more severe operation, the controller with this strategy is able to read the situation inside the compartment faster and take the best decision.
- For the R290 refrigerator, the same conclusions about the performance of the models were obtained and, here, the presence of goods was simulated and compared to experimental data, showing good agreement.

5.1 Future work suggestions

Some future work suggestions for the vapor compression refrigeration systems modeling through thermal and capacitive models are presented below.

- Discrete modeling for the heat exchangers, where more effects can be described.
- More detailed modeling of the compressor, analysing the fluid behavior inside the compartments, the transient behavior and the interactions between the mechanical elements and the fluid.
- Modeling of the temperature distribution of air inside the cabinet.
- Addition of door opening effects, frost formation and some pressure losses which were neglected in the present approach.

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Appendix

APPENDIX A – Experimental uncertainties calculation procedure

The uncertainties are calculated by the Taylor and Kuyatt (1994) procedure. The heat transfer rates uncertainties are calculated by Eq. A1, as they depend on the mass flow rate, on the inlet and outlet enthalpy values and on compressor electric power consumption:

$$\delta_{\dot{Q}} = \sqrt{\left(\frac{d\dot{Q}}{d\dot{m}_{com}}\right)^2 \delta_{\dot{m}_{com}}^2 + \left(\frac{d\dot{Q}}{dh_{in}}\right)^2 \delta_{h_{in}}^2 + \left(\frac{d\dot{Q}}{dh_{out}}\right)^2 \delta_{h_{out}}^2 + \left(\frac{d\dot{Q}}{d\dot{W}}\right)^2 \delta_{\dot{W}}^2} \quad (\text{A.1})$$

In Eq. A1 $\delta_{\dot{Q}}$, $\delta_{\dot{m}_{com}}$, $\delta_{h_{in}}$, $\delta_{h_{out}}$ and $\delta_{\dot{W}}$ are the uncertainties of heat transfer, mass flow rates, enthalpies at inlet and outlet sections, and electric power, respectively.

The uncertainties for thermal conductances and thermal capacity are calculated by Eqs. A2 and A3, respectively:

$$\delta_{UA} = \sqrt{\left(\frac{d(UA)}{d\dot{Q}}\right)^2 \delta_{\dot{Q}}^2 + \left(\frac{d(UA)}{dT_s}\right)^2 \delta_{T_s}^2 + \left(\frac{d(UA)}{dT_{surr}}\right)^2 \delta_{T_{surr}}^2} \quad (\text{A.2})$$

$$\delta_C = \sqrt{\left(\frac{dC}{d(UA)}\right)^2 \delta_{UA}^2 + \left(\frac{dC}{d\tau}\right)^2 \delta_{\tau}^2} \quad (\text{A.3})$$

In Eqs. A2 and A3 δ_{UA} and δ_C stands for the conductance and capacity uncertainties. The symbols δ_{T_s} , $\delta_{T_{surr}}$, and δ_{τ} represent the uncertainties for the component surface temperature, T_s , the component surrounding air temperature (which could be the external ambient, the fresh food compartment and the freezer ones), T_{surr} , and its time constant, τ , respectively.

The parameter $f_{\Delta P}$ is the fraction of pressure loss that occurs on the adiabatic part of the capillary tube (before the heat exchanger with the suction line component) and it is determined as follows:

$$f_{\Delta P} = \frac{P_{cond} - P_{3i}}{P_{cond} - P_{evap}} \quad (\text{A.4})$$

So, the uncertainty is:

$$\delta_{f_{\Delta P}} = \sqrt{\left(\frac{df_{\Delta P}}{dP_{cond}}\right)^2 \delta_{P_{cond}}^2 + \left(\frac{df_{\Delta P}}{dP_{evap}}\right)^2 \delta_{P_{evap}}^2 + \left(\frac{df_{\Delta P}}{dP_{3i}}\right)^2 \delta_{P_{3i}}^2} \quad (\text{A.5})$$

The pressure P_{3i} is calculated through the temperature measured at the inlet of internal heat exchanger on capillary tube side (T_{3i}). With T_{3i} and the enthalpy on this point, h_{3i} , which is equals to the enthalpy on the condenser outlet/capillary tube inlet, it can be verified if there is two-phase fluid or not and calculate this pressure. The uncertainty for this pressure is calculated by Eq. A6:

$$\delta_{P_{3i}} = \sqrt{\left(\frac{dP_{3i}}{dT_{3i}}\right)^2 \delta_{T_{3i}}^2 + \left(\frac{dP_{3i}}{dh_{3i}}\right)^2 \delta_{h_{3i}}^2} \quad (\text{A.6})$$

In Eq. A6 δP_{3i} , δT_{3i} and δh_{3i} are the uncertainties of P_{3i} , T_{3i} and h_{3i} , respectively. Finally the uncertainties for enthalpies are calculated by Eq. A7:

$$\delta_h = \sqrt{\left(\frac{dh}{dT}\right)^2 \delta_T^2 + \left(\frac{dh}{dP}\right)^2 \delta_P^2} \quad (\text{A.7})$$

Where δ_h , δ_T and δ_P are the uncertainties of, enthalpy, temperature and pressure.