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A MODELICA MODEL TO SIMULATE DOMESTIC REFRIGERATION COMPRESSORS UNDER TRANSIENT CONDITIONS

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Abstract. *Simulation models are important in the concept and design of complex thermal systems. Low-level programming languages are commonly employed to develop such models, but in recent years high-level programming languages, such as Modelica, have shown to be a more convenient platform for the simulation of complex systems. This paper presents a transient model developed with Dymola, a development environment for Modelica, to simulate the refrigeration cycle of a household refrigerator under on-off cycling conditions. The model was developed using the ThermoCycle library and includes the description of refrigerant flow and heat transfer in the system components (compressor, heat exchangers and expansion device). After being adjusted, predictions were compared to experimental data. Good agreement was observed for the compressor model when fixed boundary conditions were prescribed. When the complete model was adopted in the simulation, greater deviations were observed due to underestimation of refrigerant charge.*

Keywords: *compression refrigeration cycle, dymola, modelica, household refrigerator*

1. INTRODUCTION

Household refrigerators are fundamental in the modern society, being present in 96% of the Brazilian residences (PROCEL, 2005). Between 1997 and 1999, refrigerators represented about 40% of the residential energy consumption (Fedrigo *et al.*, 2009), which dropped to around 30% (Cardoso and Nogueira, 2007) in 2005. According to Cardoso and Nogueira (2007), between 1990 and 2005, the average annual energy consumption of household refrigerators went from 491.3 to 270.4 [kWh/year], mainly as a result of the research and development in the field. However, despite the low individual energy consumption, the huge number of refrigerators in operation still makes it very important to study these appliances in order improve their efficiency.

The majority of these appliances employ a vapor compression refrigeration cycle with a reciprocating compressor, a condenser, an evaporator, a capillary tube and a capillary tube-suction line internal heat exchanger (SL-HX). The performance of household refrigerators is usually evaluated through the so-called energy consumption tests, which are conducted according to standards such as ISO 15502 (2005) or IEC 62552 (2013), as indicated by Hermes *et al.* (2012). However, such procedures are expensive and time consuming.

The first simulation models for refrigeration systems were developed in the 70's, initially for steady state regime. Later, dynamic models started to appear, but with emphasis in heat pumps and air conditioning systems. Simulation models for vapor compression cycles employed in household refrigerators only appeared in the late 80's (Hermes, 2006). Since then, many models have been proposed based on different simplifications and considerations. Janssen *et al.* (1992) reported one of the first simulation models of a household refrigerator under *on-off* cycling conditions. The compressor and expansion device were modelled following empirical approaches and the results were compared with experimental data. A comprehensive model was presented by Hermes and Melo (2008), in which both *pull down* and *on - off* conditions were simulated. The models for the evaporator and the condenser considered air flow in the outer part and a void fraction formulation to predict two-phase flow inside the heat exchangers. The compressor model took into account heat transfer in the compressor shell and the different processes of the compression cycle. The different time scales of the problem and the high complexity of the physical models required a laborious numerical scheme, which was implemented in Fortran.

The dynamical simulation of vapor compression refrigeration cycles includes the solution of differential and algebraic equations, giving rise to the so-called differential-algebraic system of equations (DAE). Several high-level programming languages have been developed to assist engineers in the concept and simulation of such complex systems. *Modelica*, one of these languages, is a declarative open-source modeling framework based on object oriented principles, which includes built-in DAEs *Solvers*, such as the implicit *Dassl* method. The concept behind the development of models in this language improves code re-usability and reduces errors in the implementation of numerical procedures, thus allowing the engineer to focus on the physical model rather than on the mathematical solution of DAEs.

This paper presents a model developed with Dymola (Dassault, 2017) - a development environment for *Modelica* - to predict the refrigeration cycle of a household refrigerator. The model was developed using the open-source ThermoCycle library and focuses on the reciprocating compressor. The model is presented and then validated against experimental data.

2. THERMOCYCLE LIBRARY

The open source *ThermoCycle* Library (Quoilin *et al.*, 2014) was chosen to model the refrigeration cycle. The library combines thermal-fluid models with the External Media Library based on *CoolProp* (Casella and Richter, 2008), which allows for the evaluation of fluid thermodynamic properties.

The basic fluid model unit (Cell1Dim), illustrated in Fig.1a, follows the finite volume approach as described by Quolin (2011), which employs enthalpy and pressure as state variables. The model is based on dynamic mass and energy balances integrated along the cell. The velocity is considered uniform on the cross section (1-D lumped parameter model), constant pressure is assumed in the cell and axial thermal energy transfer is neglected. The composition of basic units in series is used to increase the numbers of the control volumes (Fig.1b).

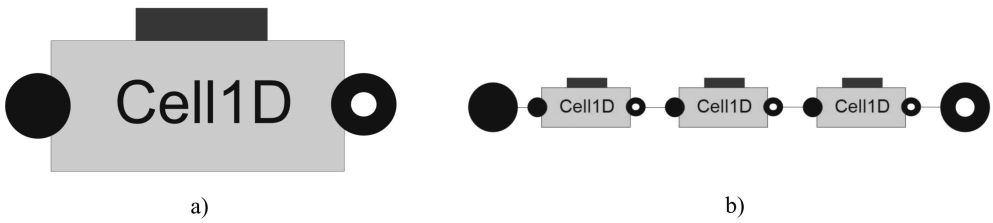

Figure 1. Basic Thermal-Fluid Model (Cell1Dim)

The basic fluid model is composed by two flow connectors and a lumped thermal port connector. The value of the property at the center of the control volume can be interpolated either by a central difference or by an upwind scheme.

The mass and energy balances are given by:

$$\dot{m}_{in} - \dot{m}_{out} = \frac{dm}{dt} = V \cdot \frac{d\rho}{dt} = V \cdot (\frac{\partial \rho}{\partial h} \cdot \frac{dh}{dt} + \frac{\partial \rho}{\partial p} \cdot \frac{dp}{dt}) \tag{1}$$

and

$$V \cdot \rho \cdot \frac{dh}{dt} = \dot{m}_{in} \cdot (h_{in} - h) - \dot{m}_{out} \cdot (h_{out} - h) + \dot{Q} + V \cdot \frac{dp}{dt} \tag{2}$$

In the above equations, $\dot{m}$ is the mass flow rate, m the mass, V the volume, ρ the density, h the specific enthalpy, p the pressure, t the time and $\dot{Q}$ the heat transfer. The subscript *in* represents the inlet of the control volume, the subscript *out* the outlet and when no subscript is used it refers to the center of the control volume.

3. REFRIGERATION CYCLE MODEL

The basic finite volume unit was used to model fluid flow in the system components, except in the capillary tube. The following sections describe the model for each component.

3.1 Compressor

The compressor was divided into several non-overlapping control volumes, similarly to what was performed by Diniz and Deschamps (2016). The thermal network that represents the compressor is depicted in Fig. 2. The basic fluid model was adopted for the following compressor components: Internal Environment (ie), Suction Muffler (sm), Discharge Chamber (dc), Discharge Muffler (dm) and Discharge Tube (dt). Except of the Internal Environment, fluid flow was considered quasi-steady and, therefore, there is no mass accumulation. The cylinder wall, lubricant oil, motor and shell were modeled using a lumped formulation for the energy equation. A split component separates the fluid at the compressor inlet to simulate a semi direct suction. The thermal resistances between the control volumes were modeled using the concept of global heat conductances, which were obtained through available experimental data.

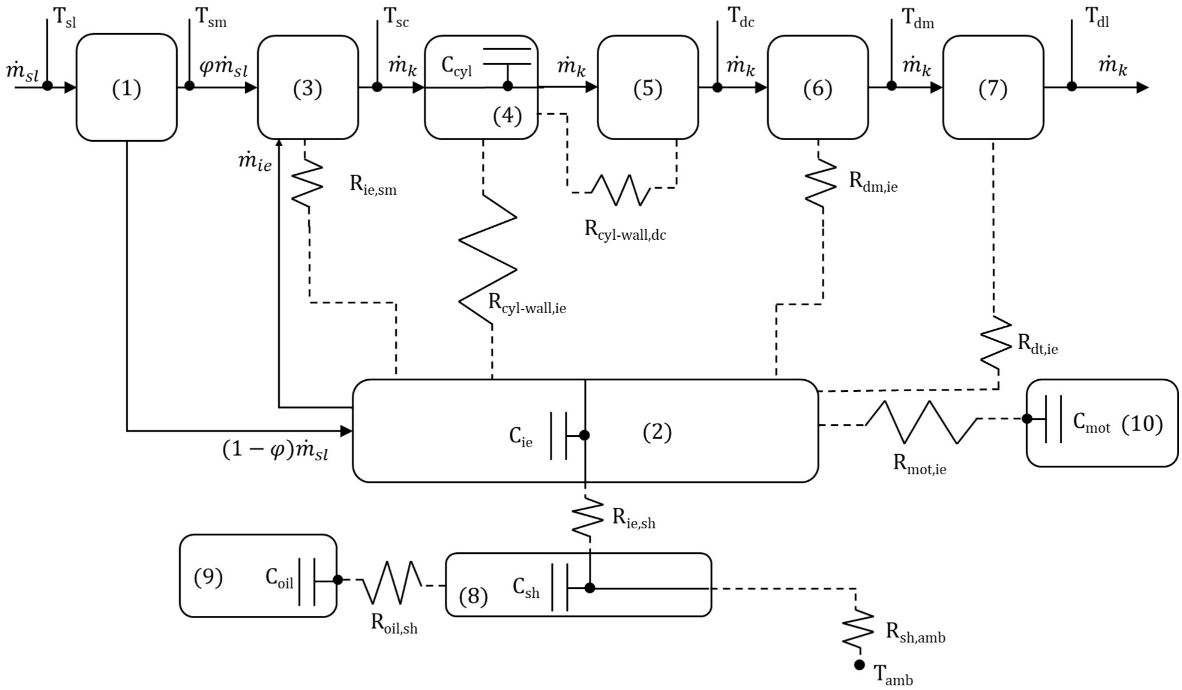

Figure 2. Thermal-Fluid compressor diagram

Table 1. Compressor control volumes

Control Volume (number)	Component (name)
1	Split
2	Internal Environment (*IE*)
3	Suction Muffler (*sm*)
4	Compression Chamber (*CC*)
5	Discharge Chamber (*dc*)
6	Discharge Muffler (*dm*)
7	Discharge Tube (*dt*)
8	Shell (*sh*)
9	Oil
10	Motor

The compression chamber model calculates the specific enthalpy ($h_{cc,out}$) at the chamber outlet, the mass flow rate ($\dot{m}_k$), indicated power ($\dot{W}_{ind}$) and the heat flow leaving the cylinder ($\dot{Q}_{cc}$). To obtain the specific enthalpy of the refrigerant leaving the compression chamber, a quasi-steady energy balance is performed:

$$\dot{m}_k \cdot h_{sm,out} + \dot{W}_{ind} = \dot{m}_k \cdot h_{cc,out} + \dot{Q}_{cc} \tag{3}$$

The mass flow rate, indicated power and heat flow are calculated using dimensionless efficiencies (Eqs. 4 to 6).

$$\dot{m}_k = \eta_v \cdot \rho_{sl} \cdot V_{sw} \cdot f \tag{4}$$

$$\dot{W}_{ind} = \dot{m}_k \cdot (h_{cc,out}^{s=sl} - h_{sl})/\eta_t \tag{5}$$

$$\dot{Q}_{cc} = \eta_c \cdot \dot{W}_{ind} \tag{6}$$

where η_v is volumetric efficiency, ρ_{sl} the density of the refrigerant at the suction line, V_{sw} the displacement volume of the compression chamber and f is the compressor operating frequency. $h_{cc,out}^{s=sl}$ is the specific enthalpy at the discharge port calculated using suction line entropy and discharge pressure, h_{sl} is the specific enthalpy at the suction line and the η_t the thermodinamic efficiency. η_c is a parameter to calculate the heat transfer during the compression cycle. These efficiencies are obtained from an external compression cycle model called *Recip* model (Todescat *et al.*, 1992), being updated throughout the transient simulation exercise.

The solution procedure that involves the coupling between the two compressor sub models (thermal and *Recip*) is depicted in Fig. 3. It is interesting to note that by using a quasi-steady model for the compression chamber with dimensionless efficiencies that are regularly updated, the risk of numerical issues due to the stiffness of the equations is considerably reduced. During the on period, the interval between two consecutive calls to the *Recip* can be adapted. It has to be smaller in the beginning of the on cycle, when the transients are stronger, but can be larger when the operation becomes more steady. In this way, the algorithm keeps the efficiencies variation small and avoids calling the compression model, computationally more expensive, when it is not necessary.

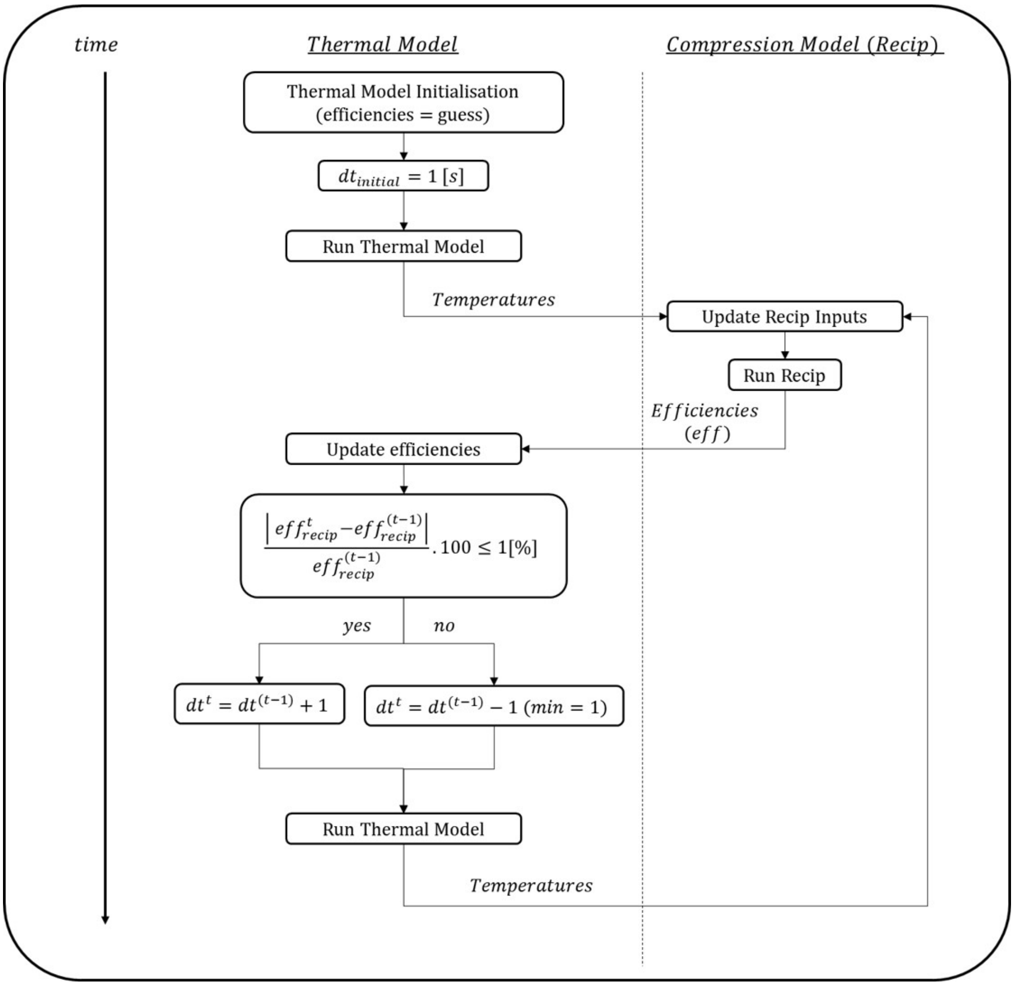

Figure 3. Thermal-Fluid model and $Recip$ diagram

3.2 Compressor Discharge Line

The compressor discharge line was modeled as a tube without pressure drop. The heat transfer is calculated using a NTU method which considers an infinite heat capacity for the external environment ($C_r = 0$). A correlation proposed by Lefevre and Ede (1956) was employed to calculate the heat transfer coefficient with a correction factor (C_{dl}).

3.3 Condenser

The refrigerant flow in the condenser was modeled using ten (10) finite volumes. The model assumes mass accumulation in all the volumes, neglects pressure drop and considers homogeneous two phase flow. The temperature of the condenser wall is calculated by applying an energy balance neglecting axial conduction. The heat exchange between the condenser and the surrounding environment was calculated using a correlation developed specifically for wire-and-tube condensers, which considers natural convection and radiation (Hermes and Melo, 2009).

3.4 Evaporator

The refrigerant flow in the evaporator was also modeled using ten (10) finite volumes. Mass accumulation in all the volumes, no pressure drop and a homogeneous two phase flow were assumed. To model the heat exchange with the refrigerated compartment, a forced convection correlation proposed by Barbosa *et al.* (2009) was used, which was developed for fin and tube heat exchangers.

3.5 Refrigerated Compartment

This model calculates the temperature of the refrigerated compartment by considering an equivalent thermal capacitance in a lumped energy balance:

$$C_{rc} \cdot \frac{dT_{rc}}{dt} = \dot{Q}_{load} - \dot{Q}_{evap} \tag{7}$$

where C_{rc} is the equivalent thermal capacitance, T_{rc} the refrigerated compartment temperature, $\dot{Q}_{load}$ the heat transfer rate from the external environment and $\dot{Q}_{evap}$ the heat transfer rate to the evaporator. The heat transfer rates are estimated using thermal conductances.

3.6 Expansion device

The expansion device of household refrigerators is usually a capillary tube. In the present model, the flow through this component was assumed adiabatic, incompressible and a quadratic relationship between the pressure drop and the mass flow rate is assumed (turbulent flow). The following expression is used:

$$\dot{m}_{cap} = A_{cap} \cdot X_{open} \cdot \sqrt{2 \cdot \rho_{cap,in} \cdot \Delta p} \tag{8}$$

where A_{full} is the cross section area, X_{open} is a calibration factor (between 0 and 1), $\rho_{cap,in}$ the density of the fluid at the entrance of the capillary tube and Δp the difference between the condenser and evaporator pressures.

3.7 SL-HX and Compressor Suction Line

The suction line was divided in two parts: the first where heat transfer with the capillary tube (SL-HX) occurs and the second where the heat exchange takes place with the external environment (compressor suction line). In both cases, the flow was also modeled using a several finite volumes and a global conductance model was used.

4. RESULTS

The model was calibrated in three different steps. First considering only the compressor model, followed by including the heat exchangers and capillary tube and finally the refrigerated compartment. Then, the results were obtained for two cases: (i) only simulating the compressor model; (ii) considering the complete system at 32 °C of ambient temperature.

4.1 Calibration

The compressor was calibrated in a steady-state condition with 32 °C ambient temperature, where the suction pressure, suction line temperature and discharge pressure were used as a boundary conditions. The compressor global heat conductances are presented in the Tab.2.

Table 2. Compressor calibration result

Parameter	Value
UA_{ie-sm}	0.37
UA_{cyl-ie}	1.00
UA_{dm-ie}	0.31
UA_{dt-ie}	0.44
UA_{ie-sh}	5.45
UA_{dc-cyl}	0.19
UA_{oil-sh}	0.15
UA_{mot-ie}	0.83
UA_{sh-amb}	1.86

The heat exchangers, the capillary tube and the refrigerated compartment were calibrated in periodic transient condition with 32 °C ambient temperature. The parameters were tuned in order to match the experimental measurements for the on and off cycle periods, which were 1908.8 and 580.4 seconds respectively. The results are presented in the Tab. 3.

Table 3. Compressor calibration result

Parameter	Value
UA_{sl-hx} [W/K]	6
UA_{sl} [W/K]	6
C_{dl} [-]	0.36
A_{cap} [m^2]	2.28e-8
X_{open} [-]	0.47
m_{system} [g]	20.5
UA_{rc} [W/K]	1.81
C_{rc} [J/K]	11833.9

4.2 Results

The results predicted by the present model were compared to the experimental data obtained by Diniz *et al.* (2018). In this way, a data set obtained for the ambient temperature of 32 °C has been selected. The refrigerated compartment temperatures when the compressor switches off (T_{off}) and switches on (T_{on}) are, respectively, -16 °C and -13.2 °C. Initially, only the compressor model was simulated. Then, the complete model was simulated.

The compressor simulation accuracy was verified by prescribing experimental curves of suction and discharge pressure and suction temperature as boundary conditions. Figure 4a compares model predictions (red) with experimental data (black dashed) of mass flow rate in the primary axis and of electrical and indicated powers in the secondary axis. Deviations smaller than 2% for the electrical power and 5% for the mass flow rate were observed after 0.1 normalized time. It can be observed that both electrical and indicated power have a similar deviation from time 0 to 0.2. Such behavior can be explained by some deviations in the prediction of the thermodynamic efficiency. In Fig 4b, the suction muffler and internal environment gas temperatures are presented in the primary axis and the suction chamber gas temperature is shown the secondary axis.

Figure 5a show comparisons of measurements and predictions of temperature for the gas in the discharge chamber (primary axis) and in discharge muffler and discharge line (secondary axis). In the Fig 5b, temperatures of the cylinder wall and lubricant oil are presented in the primary axis and of the motor and shell in the secondary axis. The largest deviations for the gas temperature were observed in the suction chamber (Fig 4b) and discharge chamber (Fig. 5a), reaching values up to 10 °C. The compressor shell was where the largest deviation (2.5 °C) was found for the temperatures of solid components.

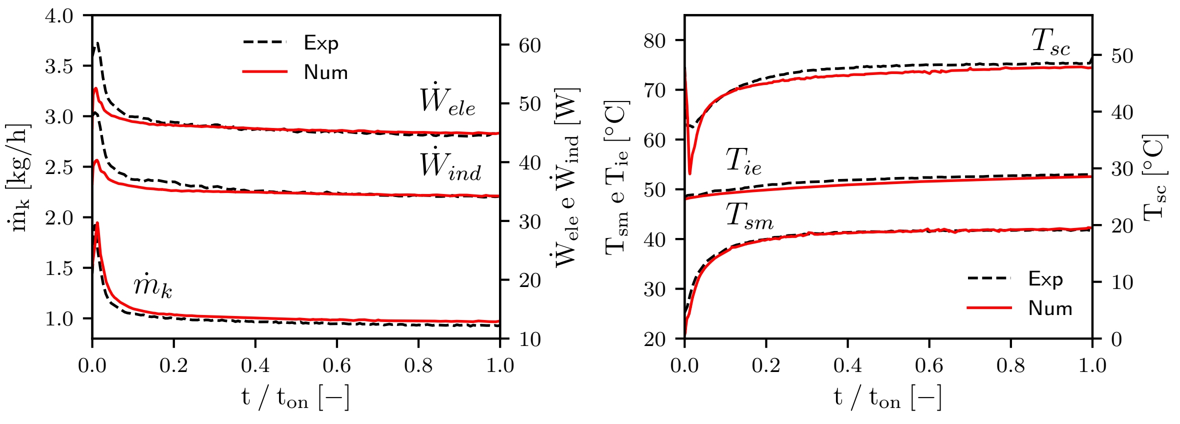

Figure 4. Numerical and experimental mass flow rate and electric and cycle powers (a) and suction muffler, suction chamber and internal environment temperatures (b) for the compressor at 32 °C ambient

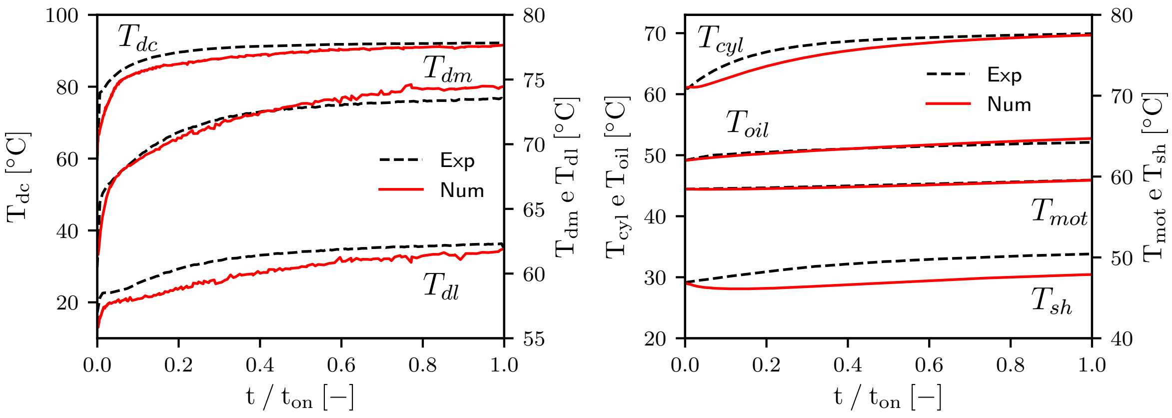

Figure 5. Numerical and experimental discharge muffler, chamber and line temperatures (a) and cylinder, oil, motor and shell temperatures (b) for the compressor at 32 °C ambient

Figures 6 and 7 present the results obtained with the complete simulation model. In this case, a thermostat was used to define whether the compressor is operating (on) or not operating (off), based on the temperature of the refrigerated compartment. Figure 6a compares measurements and predictions of condensing and evaporating pressures and Fig 6b the refrigerated compartment temperature. The deviation observed is smaller than 10% in the on period after normalized time of 0.1 for the evaporating pressure and after 0.05 for condensing pressure. The deviations verified in the predictions of the pressures are associated with the limitations of the homogeneous model to predict the correct refrigerant charge in the system, underestimating it in about 15 g (40%). The bow observed in the measurement of the refrigerated compartment temperature is, probably, caused by the copper mass that involves the thermocouple, which increases thermal inertia (Borges, 2013).

Figure 7a compares experimental data and numerical results of electrical power. Figure 7b presents similar results for mass flow rate. The deviation for the power is smaller than 10% after normalized time of 0.1 and below 5% after normalized time of 0.2 for the mass flow rate. The initial deviation in both power and mass flow rate can be explained by the fact that model underestimates evaporating pressure in the first few minutes of the on period (Fig 6a), which is caused by the absence of appropriate void fraction models in the library used in the model.

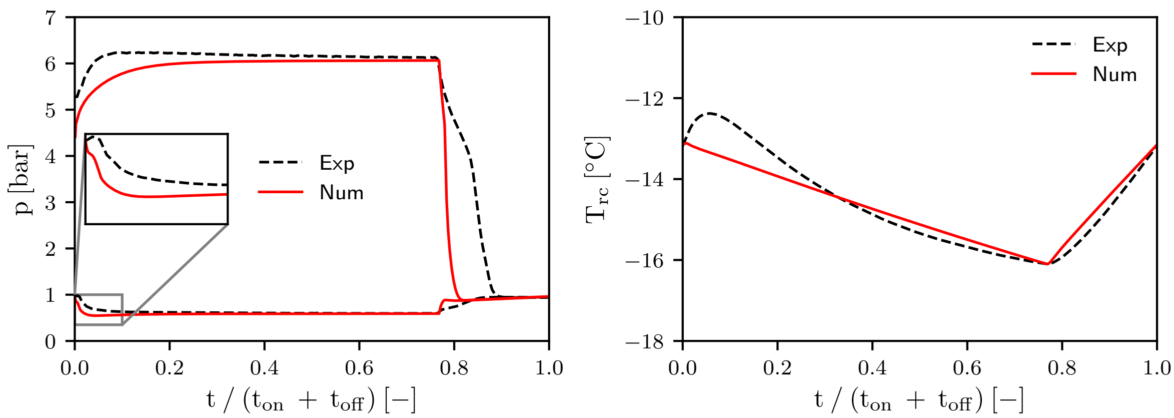

Figure 6. Numerical and experimental suction/discharge pressures (a) and refrigerated compartment temperature (b) for the system at 32 °C ambient

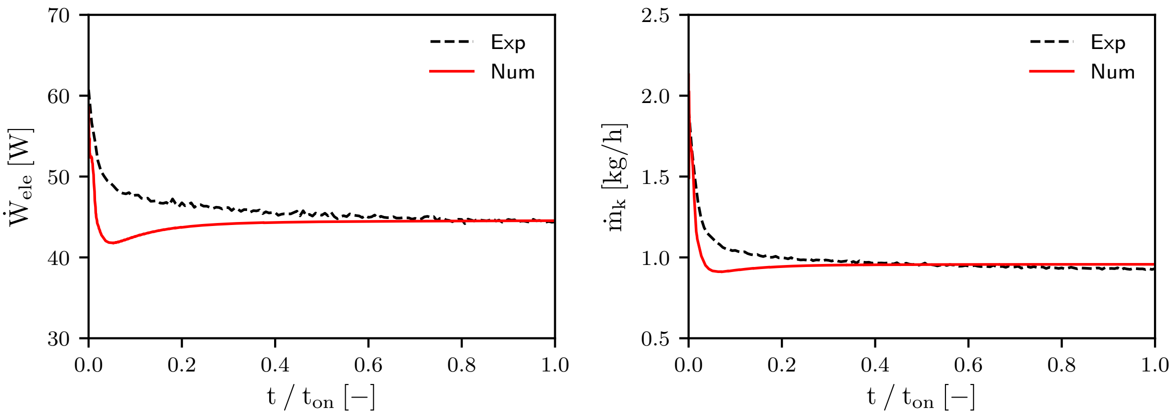

Figure 7. Numerical and experimental compressor power (a) and mass flow rate (b) for the system at 32 °C ambient

As the on and off period lengths were used to calibrate the model, the numerical and experimental compressor runtime ratio (RTR) are the same. The numerical and experimental monthly energy consumption are 24.44 and 25.94 kWh/month, respectively. Although a deviation of almost 6% is observed, due to differences in the first few minutes of the on-period, the model is capable of predicting the main performance parameters with good accuracy.

5. CONCLUSIONS

This paper presented a transient model to simulate vapor compression refrigeration cycles, with the *Modelica* programming language, and focusing on the reciprocating compressor. The compressor model was based on several control volumes in which mass and energy balances were applied. The mass flow rate, discharge temperature and indicated power were calculated using dimensionless efficiencies, which were regularly updated by an external compression cycle model. The remaining components of the refrigeration system were modeled using components in the open-souce *ThermoCycle* library. After adjusting the model, some predictions were compared to experimental data. Good agreement was observed for the compressor model when measured boundary conditions were prescribed. However, the system model was unable to predict properly the system pressures, due to unsatisfactory prediction of refrigerant charge, giving rise to under predictions of the compressor power and mass flow rate.

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