An Experimental-Numerical Procedure to Characterize Compressor Performance under Cycling Operating Conditions of Refrigerators

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ABSTRACT

The design of reciprocating compressors is commonly based on steady-state operating conditions. However, most household refrigerators operate under transient and periodic regimes, characterized by alternate periods in which the compressor is either operating (on) or not operating (off). The result is a decoupled design approach since stabilized conditions not necessarily represent the actual operating conditions of refrigerators. This paper presents a strategy in which a virtual refrigeration system is developed and coupled to an experimental facility in order to test compressors under on-off conditions typical of household refrigerators. The virtual refrigeration system simulates the dynamic behavior of a household refrigerator, except for the compressor, and provides the instantaneous operating condition to the compressor in the test bench. The developed procedure was used to emulate a refrigerator operating at ambient temperature of 32°C and freezer cut-off temperature of -16°C. The results were in good agreement with the experimental data, with the system energy consumption and the compressor run time predicted with a maximum deviation of 1.3%. The experimental facility is particularly useful to evaluate the effect that changes in the compressor design may have on the refrigerator energy consumption.

1. INTRODUCTION

Most household refrigerators control the cooling capacity by employing a thermostat that defines if the compressor is either operating (on) or not operating (off). Such a strategy, usually called on-off cycling, induces transient operating conditions, especially after the compressor start-up, when the redistribution of refrigerant through the system components gives rise to pressure transients in the evaporator and condenser (Hermes and Melo, 2008). The on-off cycling influences the refrigerator energy consumption, not only due to the variation of the electrical power consumed by the compressor throughout the on-period, but also due to compressor runtime ratio.

The transient conditions depend on aspects such as refrigerant charge, thermostat settings (cabinet temperatures), geometry of each component and thermodynamic interaction between them. Since user habits also play an important role, the actual transient conditions of a household refrigerator are not easy to predict. Therefore, the performance of household refrigerators is normally evaluated through energy consumption tests performed in climate chambers with air temperature and humidity strictly controlled (ISO 15502, 2007).

Despite the transient operating conditions that occur in household refrigerators, reciprocating compressors are traditionally designed for steady-state conditions defined by standards. This procedure results in a decoupled design since calorimeter conditions do not necessarily represent the actual operating conditions of the compressor in a refrigerator. In fact, the interaction between the components of the refrigerator represents a challenge for the integrated refrigerator-compressor design.

This paper presents an experimental-numerical procedure developed to analyze the performance of reciprocating compressors in refrigeration systems. The procedure consists in the coupling of a virtual refrigeration system to an experimental facility capable of testing compressors under on-off conditions typical of household refrigerators. The virtual refrigeration system simulates the dynamic behavior of a household refrigerator, except for the compressor, and provides the instantaneous operating condition to the test bench while interacting with the compressor being tested (hardware-in-the-loop).

2. EXPERIMENTAL FACILITY

The experimental test bench developed in this work consists of a calorimeter capable of testing compressors in transient operating conditions similar to those occurring in household refrigerators, including on-off cycling conditions. As shown in the diagram of Figure 1, the experimental setup is composed by a compressor (C), control valves (CVs), pressure transducers (PTs), thermocouple (TC1), heat exchanger (HX), electrical resistance (EH), mass flow meter (FM) and power meter (PM). This arrangement allows the control of the compressor operating condition and measurements of mass flow rate, electrical power consumption and compressor temperatures. The calorimeter is designed in such a way that the refrigerant can flow through the high and low-pressure lines in the superheated state, as indicated in the pressure–enthalpy diagram of Figure 2.

The operating condition of the compressor (upstream and downstream pressures and suction line temperature) is accomplished through micrometric expansion valves (processes a-b and c-d) and a heat exchanger (d-1). The control system of the calorimeter was developed to make it possible to perform tests with suction and discharge pressures varying with time. This is accomplished with the use of stepper motors coupled to the micrometric valves and a PID algorithm that compares setpoint values with those measured with pressure transducers.

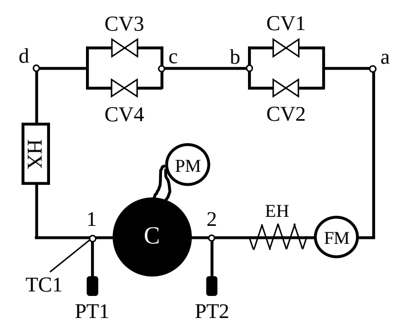


Figure 1: Schematic of the calorimeter facility.

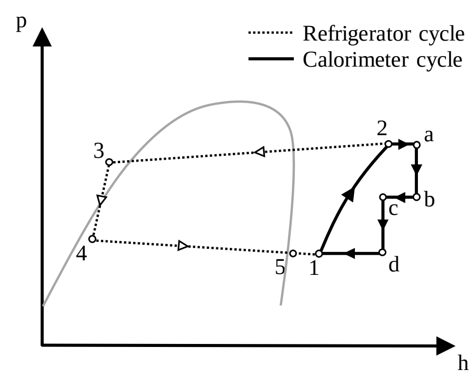


Figure 2: p-h diagram of calorimeter and refrigerator cycle.

3. REFRIGERATOR MODEL

The virtual refrigeration system coupled to the experimental facility simulates the dynamic behavior of a household refrigerator. However, a sub-model for the compressor is not required since the output data from this component are obtained from measurements. By interacting with the compressor being tested, the model simulates the refrigerator components, including the thermostat, thus providing to the experimental facility the instantaneous operating condition of the system and the duration of the on-off periods.

A single-door 300-L frost-free refrigerator was selected for the present study. The refrigerator uses isobutane as the refrigerant, a fixed speed (50 Hz) 5.96 cm^3/rev reciprocating compressor, a natural draft wire-and-tube condenser, a tube-fin "no-frost" evaporator with forced ventilation, a capillary tube as the expansion device, which is connected to the compressor suction line to form a capillary tube-suction line heat exchanger, herein referred to as internal heat exchanger (IHX). The refrigerator has two compartments: freezer and fresh-food.

The simulation model was developed in the Modelica modeling language by using basic components available in two commercial libraries, ThermoFluidPro and AirConditioning, both available in the Dymola platform (Dassault Systèmes, 2017). The complete refrigerator model is composed by sub-models that characterize each one of its components. These models are detailed next.

3.1 Heat exchangers: condenser and evaporator

The models of the condenser and evaporator provide the overall heat transfer rate, the refrigerant pressure, and the thermodynamic state of the refrigerant at the coil outlet. The heat exchangers were divided into three domains: the internal flow of refrigerant, finned-walls and external air flow.

Refrigerant flow inside the tube was modeled based on equations for mass conservation and thermal energy. The following simplifying assumptions were adopted: (i) one-dimensional flow, neglecting variations in radial and circumferential directions; (ii) straight, horizontal and constant cross-sectional tube; and (iii) negligible heat diffusion in the axial direction. The solution domain was divided into *n* control volumes and the governing differential equations were integrated in each of the control volumes using the finite volume method. An upwind scheme was used to interpolate fluid properties, and two-phase flow was considered homogeneous. For this reason, an alternative strategy was necessary to improve the calculation of refrigerant density (and thus total charge of refrigerant inside the heat exchanger) since homogeneous models are known to underestimate the refrigerant density in the two-phase flow region (Kaern *et al.*, 2011). The strategy consists in adding virtual volumes in the inlet and outlet of the heat exchanger. These virtual volumes are adiabatic and isobaric and are used only to estimate the extra charge of refrigerant in the component as a function of the thermodynamic state of the fluid at the inlet and outlet of the heat exchanger.

The heat exchanger (the tube wall and fins) was solved by applying a one-dimensional thermal energy balance which neglects axial conduction. The heat transfer rate between the heat exchanger and the surrounding air was estimated using heat transfer coefficients obtained from empirical correlations available in the literature, which require geometric parameters of the heat exchangers and air thermodynamic properties. For the evaporator, the correlation proposed by Barbosa *et al*. (2008), which can be used for finned-tube heat exchangers, was employed. The correlation of LeFevre and Ede (1956) for natural convection in long vertical cylinders was adopted for the compressor discharge line, whereas the correlation of Cyphers *et al*. (1958), multiplied by an adjustment factor, was used for the condenser finned-walls. Heat exchange due to thermal radiation was also considered in the condenser.

3.2 Internal heat exchanger

Refrigerant flow through the capillary tube was solved by integrating the equations for mass conservation and thermal energy. Since the momentum equation is not solved, pressure drop in each of the control volumes was calculated based on the Darcy friction factor determined using the correlation proposed by Colebrook (1939), which requires the Reynolds number and relative surface roughness (e/D) of the pipe. The capillary tube model was previously validated by comparing predictions with experimental results of mass flow in adiabatic capillary tubes operating with isobutane (Melo *et al.*, 1999; Schenk and Oellrich, 2014). The model was able to predict the mass flow rate through the capillary tube with deviations of less than 10% when compared to the experimental data.

The model for the compressor suction line estimates the enthalpy of the refrigerant at the compressor inlet. The heat exchange in the suction line was determined by prescribing a constant heat transfer coefficient between the tube wall and the external environment. Part of the compressor suction line is in contact with the capillary tube, forming the internal heat exchanger (IHX), whose function is to decrease the refrigerant quality at the evaporator inlet, increasing the system cooling capacity. This effect was incorporated to the capillary tube model by correcting the specific enthalpy of the refrigerant at the capillary tube outlet. The heat transfer rate removed from the capillary tube was estimated based on an overall heat transfer coefficient (UA_{IHX}).

3.3 Refrigerated compartments

Figure 3 provides a schematic representation of the thermal interactions in the refrigerator. The model of the refrigerated cabinet predicts the air temperature inside the freezer and fresh-food compartments, as a function of the system cooling capacity and heat load. The freezer and fresh-food compartments were modeled by applying mass and energy balances following a lumped formulation. The heat transfer rate between the evaporator and the surrounding air ($\dot{Q}_e$) is determined by the evaporator model and removed from the freezer control volume. The exchange of heat between the refrigerated compartments and the external environment ($\dot{Q}_{f,w}$ and $\dot{Q}_{ff,w}$), and between the freezer and the freshfood ($\dot{Q}_{f-ff,w}$), occurs by conduction through the cabinet walls. The heat conduction through the walls was considered one-dimensional and only the thermal resistance of the insulating layer was taken into account. It was also assumed that the cabinet is hermetically sealed.

A fan is used to establish air flow within the refrigerated compartments. The fan heat generation was modeled as a source term in the energy balance of the freezer ($\dot{W}_{fan}$). Part of the air flow rate supplied by the fan flows directly to the freezer while the other part is directed, through a channel, to the fresh-food compartment ($\dot{m}_{air,d}$). The air flow to the fresh-food compartment is controlled by the action of a mechanical damper.

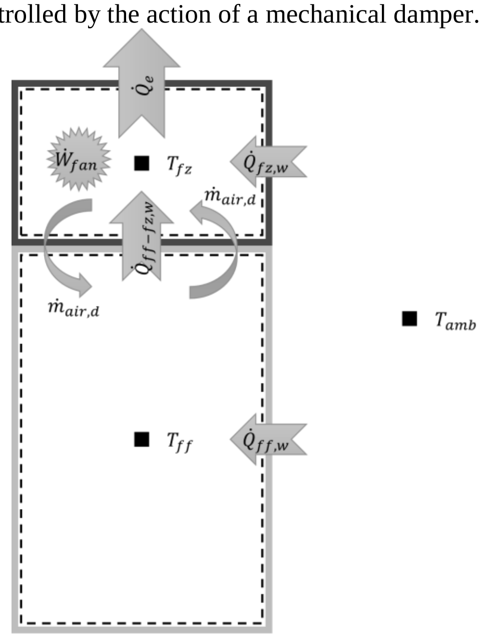


Figure 3: Thermal interaction of the refrigerated compartments model.

3.4 Model adjustments

Some semi-empirical parameters are required to close the system of equations that form the refrigerator model. These parameters were determined from measurements carried out in the refrigerator (Diniz *et al.*, 2018) in a single operating condition with ambient temperature of 32 °C and freezer cut-off temperature of -16 °C. The semi-empirical parameters of the model are: adjustment factor for the correlation of the heat transfer rate in the condenser ($C_{A,c} = 1.1875$), additional thermal capacity within the refrigerated compartments ($C_{T,fz} = 3 \times 10^3$ and $C_{T,ff} = 2.45 \times 10^4$ J/K), mass flow rate of air that circulates from the freezer to the fresh-food through the damper ($\dot{m}_{air,da} = 2.9$ x 10^{-3} kg/s), overall heat transfer coefficient in the internal heat exchanger ($UA_{IHX} = 1.0$ W/K) and external heat transfer coefficient of the suction line ($h_{sl} = 150$ W/(m^{2}K)).

4. COMBINED EXPERIMENTAL-NUMERICAL PROCEDURE

In order to test the compressor, the experimental facility requires setpoints of suction pressure, discharge pressure and compressor inlet temperature. Such data is calculated by the refrigerator model, which in turn uses the mass flow and compressor outlet temperature as its input data. The exchange of information between the test bench and the refrigerator model, illustrated in Figure 4, is conducted every 10 seconds throughout the experiment. At each communication event, the testbench updates the compressor operating condition, calculated by the model, and provides average values of mass flow rate and compressor outlet temperature measured in the previous interval. In this way, the refrigerator model is always a step of time ahead with respect to the operating condition being submitted to the compressor. The refrigerator model also controls the start-up and shutdown of the compressor based on the prediction of the freezer temperature, in line with the thermostat control adopted in refrigerators.

5. RESULTS

The experimental-numerical procedure developed in this study was used to test the compressor in the test bench under the cyclic operating condition associated with the refrigerator. The refrigerator selected for this analysis is the same used by Diniz *et al.* (2018) for measurements in a climate chamber with controlled air temperature and humidity. The present study reproduced the on-off cycling of the refrigerator operating at an environment of 32 °C and with the thermostat controlling the freezer temperature between -13°C and -16°C.

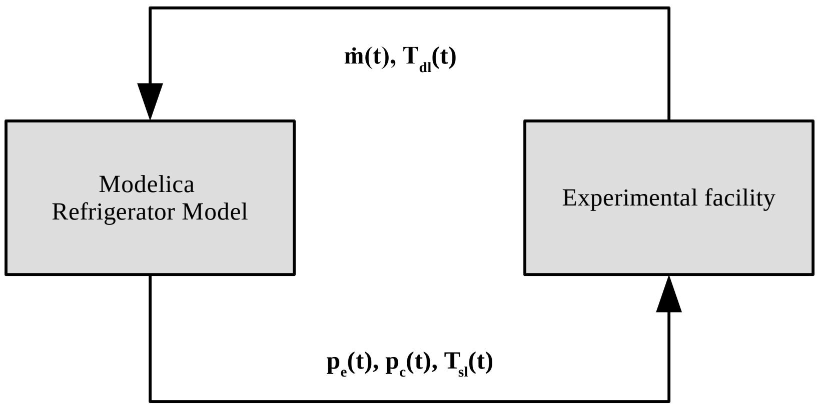


Figure 4: Diagram of the combined experimental-numerical procedure.

The experimental procedure in the calorimeter consists in performing repeated on-off cycles until a periodic transient regime is established. The criterion used to characterize cyclic behavior was the compressor thermal profile between two consecutive on-off, which should have a deviation of less than 0.5 °C. The on-off cycles were analyzed to calculate the refrigerator energy consumption, runtime ratio and on and off-period lengths, as shown in Table 1. As can be seen, the coupled experimental-numerical procedure presents good repeatability, with variations of less than 0.8% for the results between the 5 cycles presented.

Table 1: Model repeatability for energy consumption and runtime ratio.

Cycle	Energy Consumption (kWh/month)		Runtime ratio (dimensionless)		On - period (s)		Off - period (s)	
	Estimated	Difference	Estimated	Difference	Estimated	Difference	Estimated	Difference
1	25.4	0.6%	0.758	0.4%	1911	0.2%	610	-1.3%
2	25.4	0.5%	0.757	0.2%	1931	1.3%	620	0.3%
3	25.2	0.0%	0.753	-0.3%	1890	-0.9%	621	0.5%
4	25.2	-0.3%	0.755	0.0%	1910	0.2%	620	0.3%
5	25.0	-0.8%	0.753	-0.3%	1891	-0.8%	620	0.3%
Mean	25.2	---	0.755	---	1907	---	618	---

Figure 5 presents suction and discharge pressure during a full on-off cycle. The dotted black curves represent the setpoints of suction and discharge pressures calculated by the refrigerator model while the solid lines represent the pressures measured by pressure transmitters in the experimental facility. It is worth noting that during the compressor off period the pressures in the test bench cannot be controlled, since the equalizing pressure depends on the charge of refrigerant and room temperature.

Figure 6 (a) and (b) present in detail the suction and discharge pressures, respectively, during the first 300 seconds after the compressor start-up. It is possible to observe the control acting on the expansion valves so that the pressures in the testbench approach the setpoint. As can be seen, the suction and discharge pressures submitted to the compressor in the first moments of the on-period are difficult to control, due to the different equalizing pressures between the refrigerator and the calorimeter and to the fact that no phase change occurs in the calorimeter. For this reason, approximately 100 seconds are needed to ensure that the pressures approach an error band of ±5% with respect to the setpoints. After this initial period, the experimental facility controls the pressures within a range of ±1% of the setpoint value. Figure 6 also shows that the setpoint curves vary smoothly between two points of communication, indicating that the exchange of information between the model and the testbench every 10 seconds is adequate.

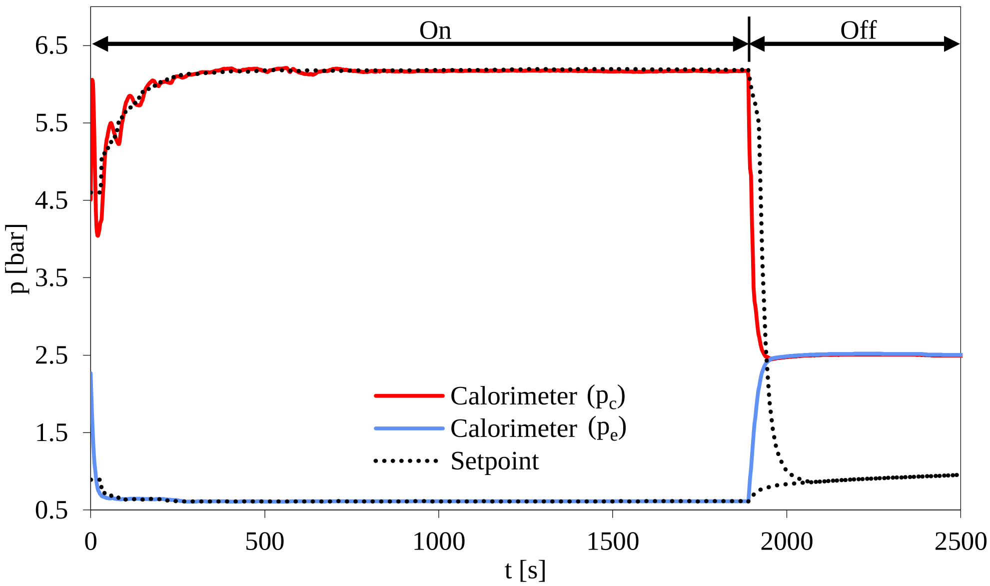
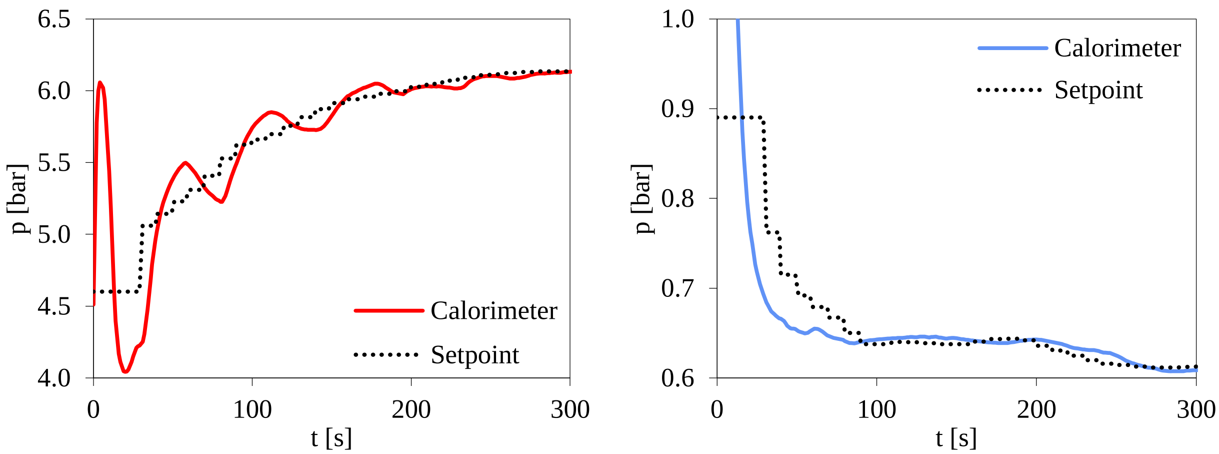


Figure 5: Suction and discharge pressures in the cycling transient.



(a) (b)

Figure 6: (a) Discharge and (b) suction pressures.

The results obtained with the experimental-numerical procedure were compared with measurements carried out in the refrigerator. Figure 7 shows the curves of discharge pressure (a) and suction pressure (b) established in the experimental-numerical procedure and measured in the refrigerator during the on-period. In the same figure, curves characterizing the deviation between both results are also provided. It can be observed that the operating condition established in the experimental-numerical procedure is in good agreement with that measured in the refrigerator. Close examination of the curves in Figure 7 indicates a deviation of less than ±5% during the entire on-period, except in the first 100 seconds after the compressor start-up, when limitations of the model and control of the expansion valves do not allow good agreement. Figure 8 shows the temperature variation in the refrigerated compartments throughout the on-off cycle. The model is seen to predict the temperatures in the freezer and fresh-food compartments with a difference of less than ±1 °C during the entire cycle.

Considering the good agreement between the measurements and predictions for the refrigerator operating condition, the compressor performance obtained with the combined experimental-numerical procedure was also in agreement with the data measured in the refrigerator, as shown in Figure 9 and Figure 10, for mass flow rate and electrical power consumption. It should be mentioned that after approximately 1000 seconds of the compressor on-period, the experimental-numerical procedure predicts a slightly higher suction pressure than that measured in the refrigerator.

Consequently, the mass flow rate and the electrical power measured in the test bench are also higher than in the refrigerator.

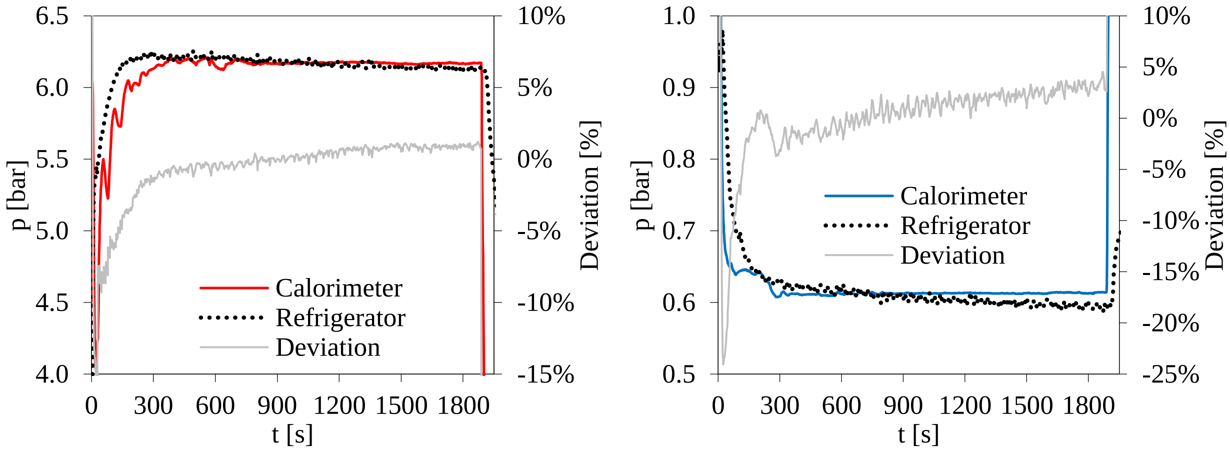


Figure 7: Model validation in the cycling transient. (a) Discharge pressure. (b) Suction pressure.

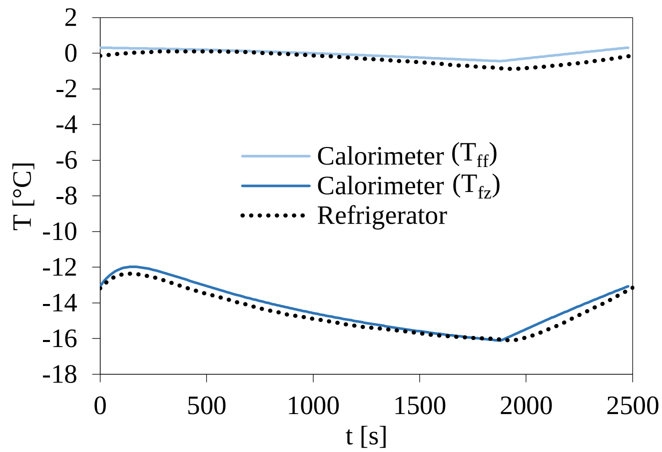


Figure 8: Model validation in the cycling transient: compartment's temperature.

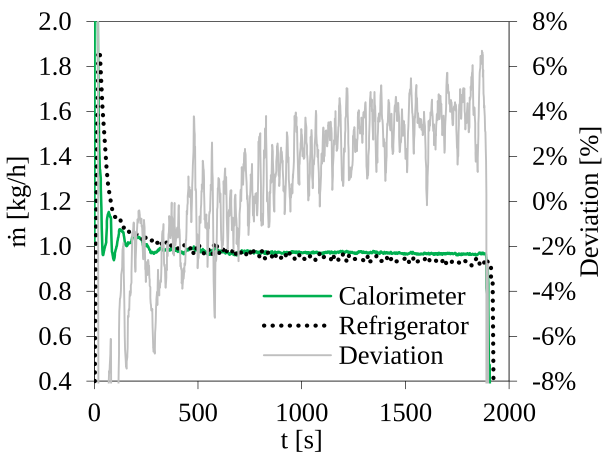


Figure 9: Mass flow rate.

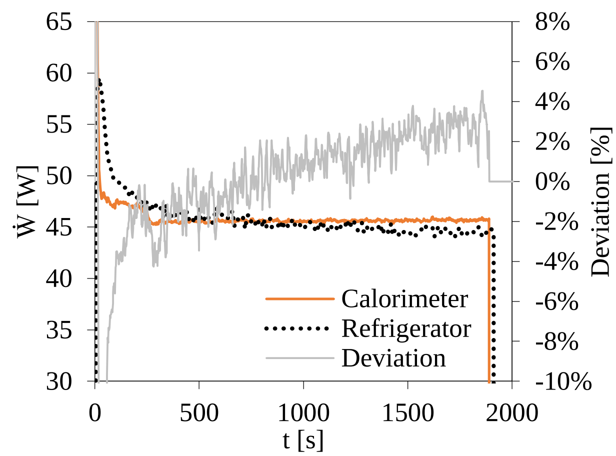


Figure 10: Electrical power consumption.

For this operating condition, the refrigerator operates with a runtime ratio of 0.765 (On-Off: 1910/580 seconds) and energy consumption of 25.3 kWh/month. The experimental-numerical procedure predicted a runtime ratio of 0.755 and energy consumption of 25.2 kWh/month, with average deviation of ±0.4% and ±1.3%, respectively, in respect to the actual refrigerator.

6. CONCLUSIONS

The present study developed a novel experimental-numerical procedure to assist the design of reciprocating compressors under typical operating conditions of household refrigerators. A simulation model of a household refrigerator was coupled to a testbench capable of submitting the compressor to transient operating conditions. This combined procedure takes into account the interaction between the compressor and other components of the refrigeration system. The coupling strategy between the refrigerator simulation model and the compressor in the experimental facility was found to be adequate. The results showed that the experimental-numerical procedure developed in this study was able to predict the energy consumption and the runtime ratio of the refrigerator with a deviation smaller than ±1.3%. The new procedure can be used to assess the effectiveness of changes in the compressor design aimed at reducing energy consumption or manufacturing cost.

NOMENCLATURE

General symbols

C_A	Heat transfer adjust constant	[-]	p	Pressure	[Pa]
C_T	Heat capacity	[W/K]	$\dot{Q}$	Heat transfer rate	[W]
D	Diameter	[m]	T	Temperature	[ºC]
e	Absolute pipe roughness	[m]	UA	Overall heat conductance	[W/K]
$\dot{m}$	Mass flow rate	[kg/s]	$\dot{W}$	Electrical power	[W]

Subscripts

c	Condensing	ff	Fresh-food
$comp$	Compressor	fz	Freezer
d	Discharge	IHX	Internal heat exchanger
da	Damper	s	Suction
dl	Discharge line		

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