THE EFFECT OF AMBIENT TEMPERATURE ON THE PERFORMANCE OF A HERMETIC RECIPROCATING COMPRESSOR IN A SMALL HOUSEHOLD REFRIGERATOR WORKING WITH R600A

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The work presents the effect of ambient temperature on the performance of a hermetic reciprocating compressor in a small household refrigerator working with R600a. The small household refrigerator has a capacity of 65 L and it is equipped with a hermetic reciprocating compressor (HRC). The HRC has been dismounted and the dead volume coefficient has been evaluated at 1.15 %. Several NTC sensors have been placed inside the compressor in order to measure the temperature of the shell, the temperature of the refrigerant right before the suction port, and the temperature of the refrigerant as close as possible to the discharge. The instrumentation process of the HRC is described. Experiments have been conducted using refrigerant R600a at different ambient temperatures. For each experiment, the energy consumption and time have been registered to enable the determination of the HRC power absorbed from the grid. After, the experimental data has been used as input data in a compressor model in order to estimate the refrigerant mass flow rate. The results are presented graphically depending on the ambient temperature. The results are commented and conclusions are drawn. Also, some technical difficulties are pointed out. Future work is discussed.

Keywords: Hermetic reciprocating compressor, household refrigerator, R600a, experimental work

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1. Introduction

Since the beginning of the 20th century, hermetic reciprocating compressors (HRC) have been and are still extensively used in vapor compression refrigeration or air conditioning systems due to their simplicity and reliability when operating in a variety of environments [1], [2]. According to [3] 17% of the world's energy demand in the last decade was consumed by refrigeration and air conditioning equipment and 85% of this energy is used in households and industrial applications. The compressor is the main energy consumer in vapor compression refrigeration systems (VCRS). Today the compressor effectiveness can be evaluated using computer-aided methods, which can help develop more effective compressors[4]. Investigators still continue to utilize fundamental theory to describe the link between the thermodynamic characteristics of HRC in order to make further progress in compressor performance optimization. Multiple papers focusing on simulation and mathematical modeling of compressors have been published, as presented below. The authors of [5], and [4] reported thermodynamic simulation models and experimental methodologies for different hermetic reciprocating compressors. Some research is focused on improving the energy consumption of HRCs [6], [7], [8]. Other researchers, focused on the mechanical performance characteristics such as speed, crank, and oil [9], [10], [11] Generally, refrigerant fluid dynamics, heat transfer mechanisms, pressure drops, and the thermodynamic characteristics of the refrigerant are all taken into account while simulating the operation of reciprocating compressors. The theoretical results are compared with experimental data and calibrated [4], [12], [13] Another important parameter in the simulation of reciprocating compressors is the temperature of the refrigerant at the suction and discharge. The real values of the suction and discharge temperatures are not so easily obtained during operation [14]. In paper [15] the authors have developed simulation software for a household refrigerator operating in transient conditions. The results have been validated with experimental work and a good agreement has been reached. Authors [16], and [17] investigated the relationship between HRC efficiency and the refrigerant type. The work conducted in [18] and [19] shows experimental tests on various parts of the compressor. The results are used to calibrate a mathematical model. The authors of [8] conducted experimental work to evaluate the HRC performance of a refrigerator. They noticed that the capillary tube partial blockage reduces the performance by reducing the mass flow rate of the compressor. Because of this, the refrigerator did not reach a steady-state operation.

As presented before, the number of papers containing both experimental work and mathematical modeling of HRCs is rather small. This low number could be explained by the difficulties encountered during experimental work. In this context, the paper aims to point out the effect of ambient temperature on the performance of a hermetic reciprocating compressor mounted on a small household refrigerator working with R600a. The mass flow rate of refrigerants is evaluated based on a straightforward mathematical model presented in previous work. The energy consumption and the power absorbed from the grid of the HRC are also experimentally evaluated. Sensors are placed inside the HRC shell and in the muffler discharge and near the suction port in order to measure the corresponding temperatures. Some technical difficulties are discussed [20].

2. Experimental setup

A schematic of the experimental setup is given in figure 1. Basically, the experimental setup is a small household refrigerator comprised of an evaporator, HRC, a hot wall condenser, and a capillary tube which is partially mounted in a suction line heat exchanger (SLHX). The dimension of the household refrigerator is 50 cm x 50 cm x 60 cm, having a net volume of 65 L, energy class A+, with a power supply of 220-240 V and 50 HZ.

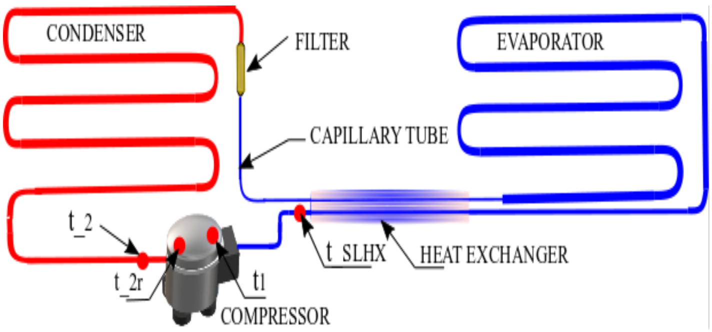

Fig 1. Schematic of the experimental setup

The experimental setup operates using R600a as a refrigerant. The main thermodynamic properties of the refrigerant R600a are presented in Table 1.

Table 1

Thermodynamic properties of the refrigerants R600a [6],[21], [22], [23], [24].

properties	
Numerical designation	Isobutane Methyl-propane
Chemical formula	$CH(CH_3)_2CH_3$
Normal boiling point	(-13)-(-9) °C, 260-264 K
Critical Temperature	134.7 °C , 407.81 K
Critical pressure	3.65 MPa
Vapour pressure	204.8 kPa at 21 °C

Molar mass	58.12 g·mol^{-1}
Density-vapour	7.73 kg/m^3
Density- liquid	557.5 kg/m^3
Boiling point	-11.7 °C, 261 K
ODP	0
GWP	3
Latent heat of evaporation	362.6 kJ/kg
Class	A3

The temperatures on the experimental setup are determined using NTC thermistors. The evaporating (p_{evap}) and condensing (p_{cond}) pressures are determined using pressure transducers. For the present work the main temperature measuring points, as presented in figure 1, are described in table 2.

Table 2

Main temperature measuring points.

	Descriptions	Positions
t_1	Suction muffler orifice	inside the compressor
t_{2r}	Muffler discharge	inside the compressor
t_2	Discharge pipe	outside the compressor
$t_{\_SLHX}$	SLHX outlet	at the exit of SLHX
$t_{\_shell}$	Compressor shell Temp.	inside compressor shell

In order to simulate the heat demand, an adjustable electric heater is placed inside the refrigerated enclosure. The electric heater has a power rating of 80 W and is connected to the power regulator. The static heater's power may be adjusted between 0 and 100 %. Using the adjustable electric heater, a quasi-steady-state operation of the experimental setup can be achieved as explained in figure 3. The temperature inside the refrigerated enclosure is measured using an NTC sensor. When the temperature inside the refrigerated enclosure reaches 8 °C, the adjustable electric heater starts operating. When the temperature inside the refrigerated enclosure reaches 10 °C, the electric heater turns off. Concerning the compressor, it starts operating at a temperature inside the refrigerated enclosure of 12 °C and turns off at 5 °C.

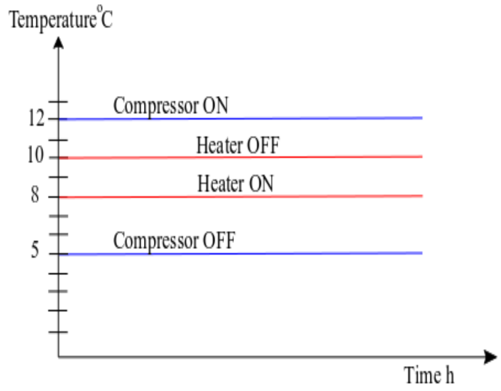

Fig. 2. Quasi-steady-state operation of the experimental setup

As mentioned above, the aim of the paper is to point out the effect of ambient temperature on the performance of the HRC. In order to do so, the HRC has been instrumented as describer next. The upper shell of the HRC has been removed using a cutting machine thus gaining access to the muffler discharge. Also, the compressor and the electric motor have been extracted. The cover of the muffler discharge was removed and a 2 mm circular hole was drilled. Inside, an NTC-type thermistor has been installed to monitor the temperature of the refrigerant as close as possible to the discharge chamber. A detail of the muffler discharge with the NTC sensor placed inside is presented in figure 3.

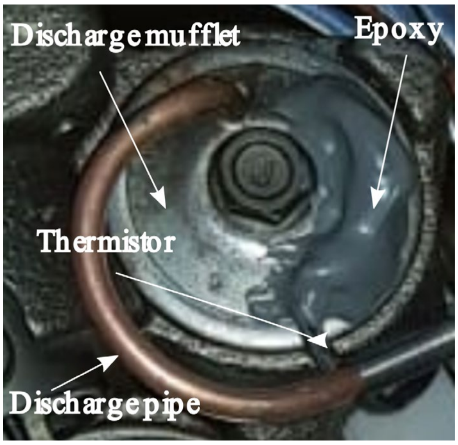

Fig 3. Detail of muffler discharge.

Afterward, an electric drill was used to drill 2 holes of 4 mm diameter. The first hole is near the suction pipe and the second is on the other side of the lower section of the compressor shell. The HRC has been instrumented inside the shell using four NTC sensors, as shown in figure 4.

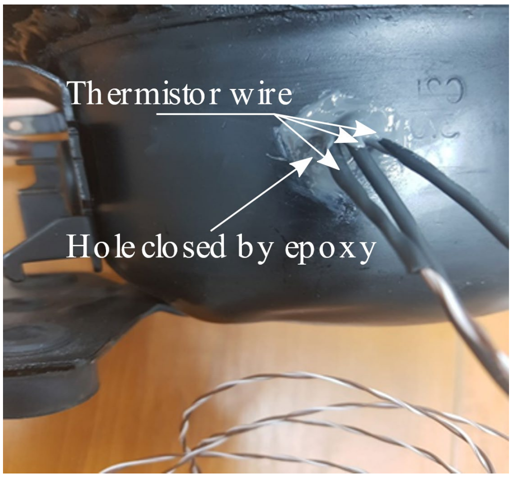

Fig. 4. Thermistors installed inside the HRC shell.

To close back the HRC, a rectangular plate of iron measuring 17 cm in width and 40 cm long with a thickness of 5 mm was used for both upper and lower shells. The plate of iron was divided into two parts (17 cm and 20 cm). The ring's thickness of 5 mm is utilized to prevent denting and distortion during welding the ring over the shell. Before welding, the two rings were smoothed and cleaned to suit the top and lower outer shells of the compressor. After they were welded to the compressor body the two shell sections and the rings were cleaned and smoothed again. On the two rings, 22 holes of 6 mm diameter were drilled with an electric drill. After finishing the hole in the cover of the muffler discharge, the high-pressure discharge pipe was reinstalled and welded. After the internal parts of the compressor were installed and before the upper shell was closed, an initial test was performed to confirm that the sensors function properly. To avoid any leakage in the discharge pressure muffler, a charging valve was welded to the suction and discharge compressor pipes. The compressor was tested at a pressure of 20 bar. To guarantee full sealing, 22 screws are used to secure the upper and lower section of the compressor shell. Thermal silicon that bears a temperature of 300°C has been used between the upper and lower shell sections. A pressure gauge has been installed to the suction line and a vacuum was created to check that there were no leaks in the compressor's exterior casing. The system was maintained in a vacuum for more than 24 hours.

3. Refrigerant mass flow rate and compressor power evaluation

The refrigerant mass flow rate is evaluated based on a straightforward mathematical model described in previous work [12] The main characteristics of the HRC needed to evaluate the mass flow rate are given in table 3.

Table 3

Input Fixed Data.

Descriptions	Details
HRC diameter bore (D)	0.021 m
HRC stroke (S)	0.0165 m
Piston rotation (n_r)	3000 rot/min
Number of cylinders (i)	1
Number of compressors (z)	1
Dead volume coefficient (ε)	0.115

According to [12], the refrigerant mass flow rate is highly dependent on the refrigerant properties at the suction port. For the present work, the properties of R600a needed to evaluate the mass flow rate are determined considering the temperature measured inside HRC at the entrance in the suction port (t_1). The mathematical model described in [12] has been implemented in Engineering Equation Solver [25]. The compressor power has been determined using the energy meter available on the experimental setup and the operation time for each individual test.

4. Results

The refrigerator is placed inside a laboratory where the ambient temperature can be controlled via a regular air conditioning split unit. For the present work, tests have been conducted for an ambient temperature ranging from 22 °C to 29 °C with a step of 1 °C. The experimental data used in the EES program to evaluate the mass flow rate are presented in table 4.

Table 4

Experimental for R600a input data

Input parameter	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6	Test 7	Test 8
p_{cond} [bar]	7.179	7.506	7.438	7.597	7.768	7.646	7.828	7.793
p_{evap} [bar]	0.514	0.533	0.539	0.533	0.560	0.514	0.551	0.514
$t_{\_amb}$ [°C]	22	23	24	25	26	27	28	29
$t_{\_2r}$ [°C]	87.8	88.8	89.9	90.4	90.3	92.2	92.7	95.3
$t_{\_2}$ [°C]	59.8	60.6	61.8	62.5	62.9	63.8	64.9	66.4
$t_{\_1}$ [°C]	50.5	51.8	52.9	53.7	53.7	55.9	56.7	59.7
$t_{\_shell}$ [°C]	58.3	59.1	59.9	60.8	60.6	62.7	62.9	64.9
t_{SLHX} [°C]	5.7	6.9	9.7	10.9	11.4	12.2	12.5	14.6

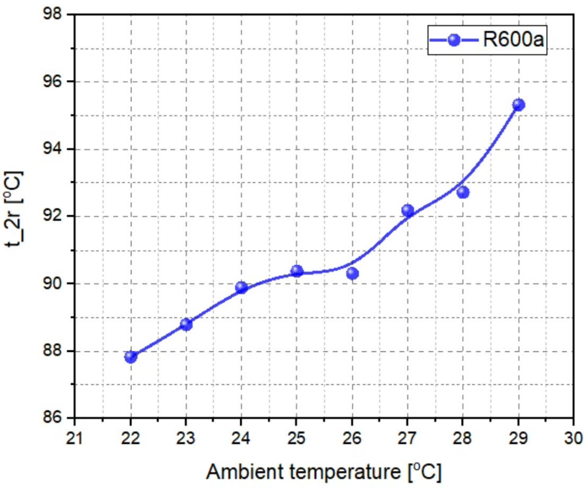

Fig. 5. Influence of ambient temperature on the muffler discharge temperature

From table 4 and figure 5 one can notice that as the ambient temperature increases, the temperature of the refrigerant in the muffler discharge also increases. The temperature of the refrigerant in the muffler discharge is not the discharge temperature. The muffler discharge is the closest measuring point easily reachable to the HRC discharge. However, considering the influence of ambient temperature on the discharge temperature one can assume that it is similar to the influence on the muffler discharge temperature. Also, from figure 5, one can notice the high muffler discharge temperatures at ambient temperatures of 28 $^{\circ}$C and 29 $^{\circ}$C. At an ambient temperature increase of 30% the compressor muffler discharge temperature increases by roughly 8 %.

Figure 6 presents the influence of ambient temperature on the refrigerant mass flow rate. As the ambient temperature increases the mass flow rate decreases. At an increase of ambient temperature of 30% the mass flow rate decreases roughly by 1.34%.

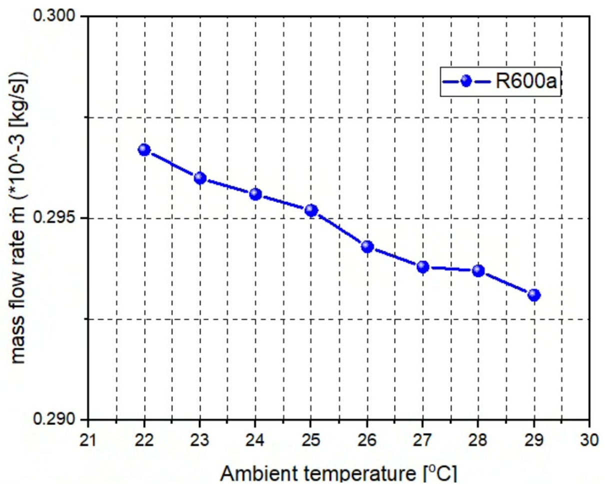

Fig. 6. Influence of ambient temperature on the refrigerant mass flow rate

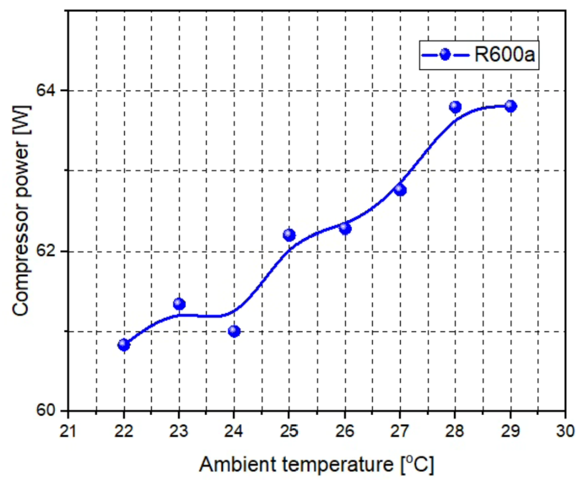

Fig. 7**.** Influence of ambient temperature on the compressor power.

Figure 7 presents the influence of ambient temperature on the compressor power. As the ambient temperature increases the compressor power increases also. At an ambient increase of 30%, the compressor power increase by approximately 4.5 %.

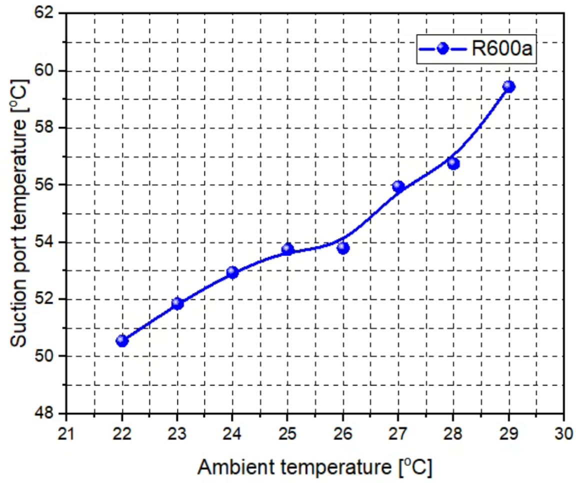

Fig. 8. Influence of the ambient temperature on the refrigerant temperature at the suction port

Figure 8 presents the influence of the ambient temperature on the refrigerant temperature at HRC at the suction port. One can notice that as the ambient temperature increases, the temperature of the refrigerant at the HRC suction port increases also. At an ambient increase of 30%, the refrigerant temperature at the HRC suction port increases by approximately 13%. The temperature registered by the sensor mounted at the HRC suction port is high. This indicates a high superheating degree and consequently a high discharge temperature. Also, from figure 8 and figure 5 one can notice that the HRC operates at high-temperature levels.

5. Conclusions

The paper presents experimental work conducted on a small household refrigerator equipped with a hermetic reciprocating compressor working with R600a. The paper aims to point out the influence of ambient temperature on the performance of the hermetic reciprocating compressor. A brief description of the experimental setup is given. Details about the work related to the hermetic reciprocating compressor instrumentation are also provided. The mass flow rate of the hermetic reciprocating compressor has been evaluated based on a mathematical model presented in previous work. The power of the hermetic reciprocating compressor has been evaluated experimentally.

Eight experimental tests have been conducted for an ambient temperature ranging from 22 $^{\circ}$C to 29 $^{\circ}$C with a step of 1 $^{\circ}$C. As the ambient temperature increases, the temperature of the refrigerant in the muffler discharge increases also. At an ambient temperature increase of 30% the compressor muffler discharge temperature increases by roughly 8 %. As the ambient temperature increases the mass flow rate decreases. At an increase of ambient temperature of 30% °C the mass flow rate decreases roughly by 1.34% As the ambient temperature increases the compressor power increases also. At an ambient increase of 30%, the compressor power increase by approximately 4.5 %. If the ambient temperature increases, the temperature of the refrigerant at the hermetic reciprocating compressor suction port increases also. At an ambient increase of 30%, the refrigerant temperature at the hermetic reciprocating compressor suction port increases by approximately 13%. The highest influence of the ambient temperature is on the refrigerant temperature at the hermetic reciprocating compressor suction port.

Future work will be conducted to extend the instrumentation of the hermetic reciprocating compressor so that more data about the temperature distribution inside the capsule can be obtained. Also, future experiments can be conducted using the same compressor and different refrigerants. Results could enable the real COP comparison at different ambient temperatures.

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