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EXERGY ANALYSIS OF HERMETIC RECIPROCATING COMPRESSORS ADOPTED IN HOUSEHOLD REFRIGERATION SYSTEMS

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Abstract. *Hermetic reciprocating compressors are prevalent in small domestic and light commercial refrigeration systems. Due to the growing demand for energy-efficient products, these compressors are currently highly optimized, and their further improvement requires suitable methods to identify their main sources of inefficiency. The second law of thermodynamics deals directly with irreversibilities and hence allows the unambiguous assessment of these inefficiencies. The present paper reports a method to identify and quantify inefficiencies based on the second law of thermodynamics. This method is then used to make the inventory of energy losses, expressed as exergy destruction, of a reciprocating compressor used in household refrigeration systems. The inventory of losses allowed a clear distinction between the main causes of inefficiency, by dividing the total loss into partial losses associated with different phenomena. The results for two operating conditions show that heat transfer contributes considerably to the exergy destruction in the compressor.*

Keywords: *refrigeration, reciprocating compressor, efficiency, energy, exergy*

1. INTRODUCTION

The compressor is one of the four basic components of a vapor compression refrigeration cycle, being responsible for providing the mass flow of refrigerant required by the system and, along with the expansion device, establishing the pressure difference between the evaporator and the condenser. Reciprocating compressors are positive displacement machines that increase the pressure of the working fluid by means of the action of a piston that reduces the volume of gas trapped in a cylinder. Most reciprocating compressors use a crank-rod mechanism to convert the rotation of the motor shaft into the reciprocating movement of the piston. This type of compressor is used in a wide range of engineering applications due to its relatively simple operating principle, low cost and suitability for different pressure ratios. Hermetic reciprocating compressors are commonly adopted in small capacity systems, especially in domestic refrigerators. This type of compressor adopts simple reed-type valves that open and close automatically in response to the imposed pressure difference (Schreiner *et al*., 2010). Fig. 1 shows a schematic drawing showing some of the basic components of this type of compressor.

The compression cycle of a reciprocating compressor can be represented by a pressure-volume (p-V) diagram, as shown in Fig. 2a, in which the pressure of the gas inside the cylinder (compression chamber) is indicated as a function of the cylinder volume. The pressures p_d and p_s correspond to the pressures in the discharge and suction lines (outlet and inlet of the compressor, respectively), while the area enclosed by the diagram curve represents the compression work performed by the piston in one cycle (indicated work). As shown in Fig. 2, the compression cycle consists of four processes: compression, discharge, expansion and suction. During the compression process (1-2) the valves are closed, and the piston moves upwards, reducing the cylinder volume and increasing the gas pressure until the discharge valve is opened. During the discharge process (2-3) the compressed gas reaches the discharge system through the discharge valve. During the expansion process (3-4), the piston moves downwards, increasing the volume of the cylinder, and the residual mass of gas in the clearance volume between the piston and the valve plate is expanded until the suction valve opens. During the suction process (4-1) a new charge of gas is drawn into the cylinder through the suction valve.

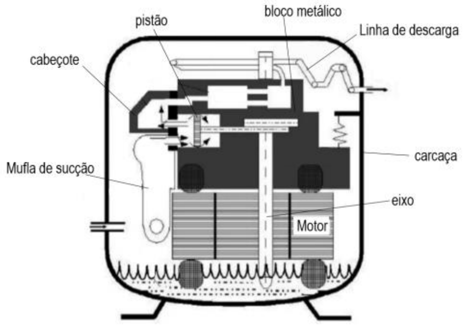
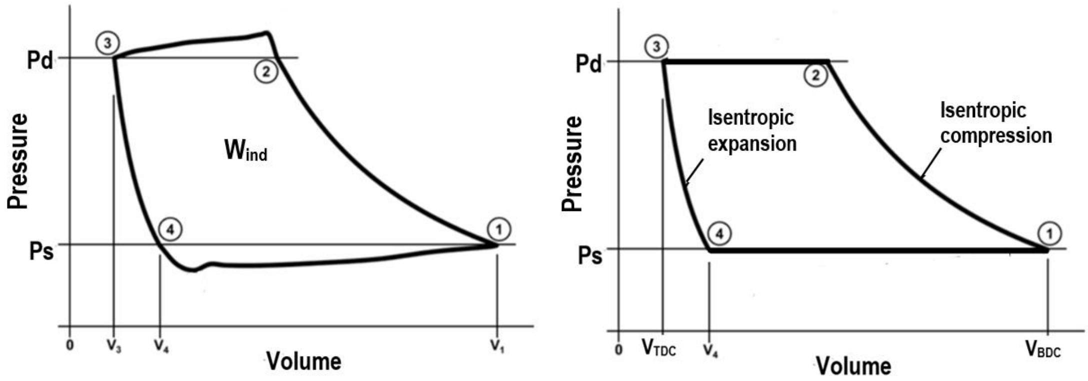


Figure 1. Basic components of a hermetic reciprocating compressor.



(a) (b)

Figure 2. Compressor p-V diagram: (a) real compression cycle; (b) ideal compression cycle.

Compressors consume most of the electrical power supplied to the refrigeration system, and consequently, their performance directly affect the energy efficiency of the refrigeration system as a whole. According to the Energy Research Office (EPE, 2017), the residential sector is responsible for 28.8% of the electricity consumption in Brazil. Within this sector, approximately a third of the consumption is attributed to refrigerators and freezers. These data show the significance of domestic refrigeration systems in the electricity consumption, as well as the need to increase their efficiency.

The efficiency of a hermetic compressor is usually assessed by comparing its actual energy consumption with that of an ideal compressor operating under the same conditions. The compression cycle of an ideal compressor (Fig. 2b) consists of isentropic expansion and compression processes and isobaric suction and discharge processes. Accordingly, the gas is drawn into the compression chamber during the suction process at the same thermodynamic state of the compressor inlet. Mechanical losses in bearings and electrical losses in the motor are assumed to be non-existent. The isentropic efficiency is defined as the ratio between the compression work associated with an isentropic process and the actual compression work. In the case of hermetic compressors, it is convenient to define an overall efficiency as the ratio between the power required for an isentropic process considering the actual mass flow rate and the electrical power consumed by the motor.

In order to increase the compressor efficiency, its main inefficiencies have to be identified and quantified. Most studies (Stouffs *et al.*, 2001; Pérez-Segarra *et al*., 2005; Phillipi, 2016) adopt the p-V diagram (Fig. 2a) so as to identify the losses with reference to the ideal compression cycle (Fig. 2b). Despite allowing the design of highly optimized compressors, these traditional approaches cannot identify some inefficiencies from the p-V diagram. On the other, the assessment of inefficiencies based on the second law of thermodynamics brings about the possibility of more accurate analyses, since it deals directly with irreversibilities, source of inefficiencies, required for a complete loss inventory. Prakash and Singh (1974) argued that the second law efficiency should be used to assess the performance of compressors. McGovern (1988) pointed out that the isentropic efficiency is not always the best parameter to optimize the compressor.

McGovern and Harte (1995) proposed an exergetic analysis for a positive displacement compressor, with the purpose of identifying and quantifying its inefficiencies. A case study was carried out for an open type refrigeration compressor working with R12 refrigerant, taking into account inefficiencies due to friction in bearings, heat transfer, pressure drop and fluid mixing. These inefficiencies were expresses in the form of exergy destruction rates. The present paper reports a method based on the proposal McGovern and Harte (1995) to determine the energy loss inventory of a hermetic reciprocating compressor working under two operating conditions typical of domestic refrigeration.

2. METHODOLOGY

2.1 Compression cycle

A computational model was used to simulate the operation of the compressor under analysis. The code solves the compression cycle based on models for calculating the piston position as a function of the crank angle, thermodynamic processes in the compression chamber, flow through valves, valve dynamics, leakage, gas pulsation in mufflers, motor dynamics and bearing losses.

The compression chamber is modeled through balances of mass and energy. Valve dynamics is modeled as a single degree of freedom mass-spring system and the mass flow rate through the valves is calculated with reference to isentropic flow in nozzles. The concept of effective force area is used to estimate the flow induced force acting on the valve, whereas the effective flow area is adopted to correct the mass flow rate given by the isentropic flow so as to account for viscous effects. The pulsating compressible flow in the suction and discharge systems is simulated by solving the one-dimensional conservation equations with the finite volume method.

Figure 4 shows the procedure for solving a compression cycle. Temperatures in different regions of the compressor, used as input data in the solution of the compression cycle, are calculated by using a thermal simulation model that is solved in a coupled manner with the compression cycle. At each time step of the solution procedure, the instantaneous values for temperature, pressure, volume, mass flow, valve displacement, etc., are calculated. At the end of each compression cycle, average values of mass flow rate, discharge temperature and power are computed and used as input data for the thermal simulation model. The temperatures inside the compressor are calculated via steady-state energy balances in nine control volume: suction system, motor, cylinder, discharge chamber, discharge muffler, discharge tube, oil, compressor shell and internal environment. The heat transfer between the control volumes is calculated with the use of experimentally calibrated global conductances. The program runs several cycles until reaching the fully cyclical regime.

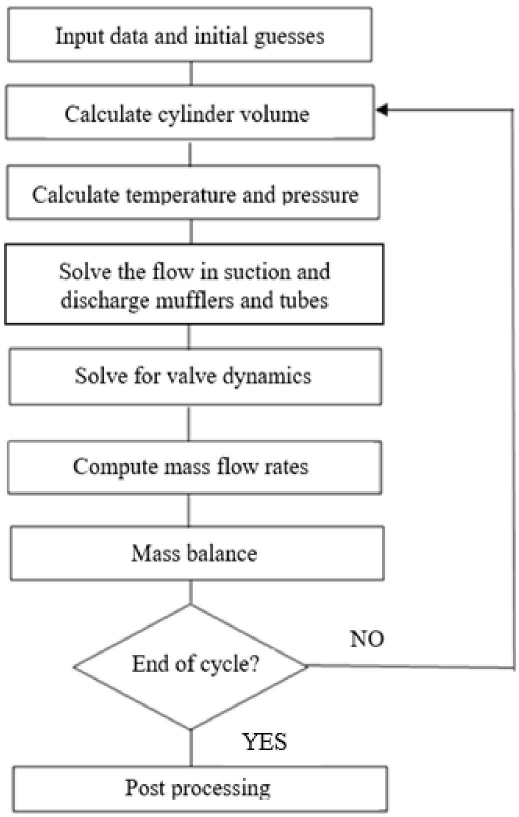


Figure 3. Solution procedure of the compression cycle.

2.2 Loss inventory

In addition to global efficiencies to characterize the compressor performance as a whole, the compressor optimization requires the main sources of inefficiency in the form of a loss inventory, in which a given global loss is divided into partial losses associated with specific phenomena. The breakdown of the main sources of inefficiency allows the identification of the most suitable modifications in design to increase the compressor efficiency.

The second law of thermodynamics allows the evaluation of inefficiencies based on the concept of exergy destruction (irreversibility). The exergy ψ is a co-property of the system and the reference environment surrounding all the systems under analysis, which is conceived as being infinite and in thermodynamic equilibrium at a well-defined state, called the dead state. The exergy of a system corresponds to the maximum useful work that could be obtained by allowing the system to come into equilibrium with the reference environment (dead state). The second law of thermodynamics establishes that exergy is preserved in ideal processes and destroyed in real processes, which determines the directionality of spontaneous processes in nature. Like energy, exergy can be transferred between systems through interactions in the form of heat, work and mass transport. The well-known Guoy-Stodola relationship states that the destruction of exergy ($\dot{I}$) is directly proportional to the entropy generation ($\dot{S}_{gr}$) by a factor equal to the dead state temperature (T_0):

$$\dot{I} = T_0 \dot{S}_{gr} \tag{5}$$

With reference to Fig. 4, the overall exergy destruction rate, or total loss, in the compressor and its immediate vicinity, which can be obtained from the following steady-state balance of exergy:

$$\dot{I}_{total} = \dot{W}_{el} - \dot{m}_c \left(\psi_2 - \psi_1 \right) \tag{6}$$

The system boundary is extended in such a way that $T_{boundary} = T_{surroundings}$, as suggested by the dashed line in Fig. 4. The temperature T_0 is considered to be the temperature of the local environment in which the compressor operates ($T_0 = T_{surroundings}$).

In order to break down the total destruction of exergy, McGovern and Harte (1995) first identified the different irreversibility mechanisms associated with compressor operation. Then, quasi-steady models were adopted to calculate the exergy destruction (entropy generation) associated with each mechanism. During the operation of a hermetic reciprocating compressor, the following irreversibility mechanisms are expected to take place: (i) friction in bearings; (ii) energy dissipation in the motor; (iii) fluid mixture; (iv) heat transfer; (V) viscous flow friction. The approaches used to estimate the exergy destruction associated with these mechanisms are presented next.

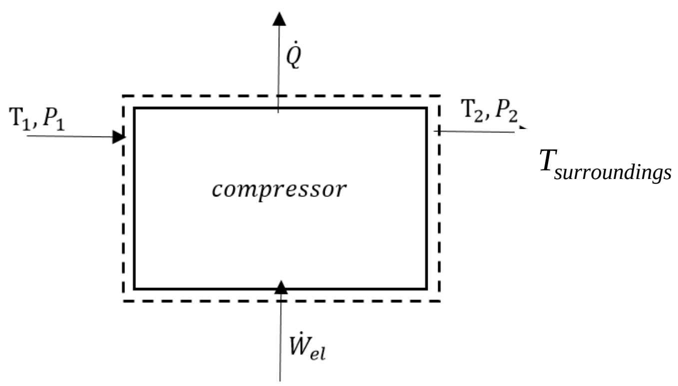


Figure 4. Control volume for balance of exergy.

2.2.1 Friction in bearings and dissipation of electrical energy in the motor

Friction in bearings and dissipation in the motor give rise to conversion of useful work into thermal energy (heat). Thermal energy has a lower exergetic potential than energy in the form of work. If the dissipated power is transferred to a mass at temperature T_i, the exergy destruction rate associated with these two mechanisms can be calculated by:

$$\dot{I} = \dot{W}_{dis} \left(1 - \frac{T_i - T_0}{T_i} \right) \tag{7}$$

If $T_i = T_0$, the exergy destruction rate equals the dissipated power $\dot{I} = W_{dis}$. If $T_i > T_0$, the exergy destruction rate is less than the dissipated power because part of the generated heat corresponds to exergy gained by the hot mass.

2.2.2 Heat transfer through a finite temperature difference

Any spontaneous heat transfer between two bodies represents wasted work potential and, therefore, a destruction of exergy. Considering a process in which heat is transferred through a thermal resistance at a rate $\dot{Q}$ between two bodies at temperatures $T_1 > T_2$, the associated exergy destruction rate is

$$\dot{I} = T_0 \dot{Q} \left(\frac{1}{T_2} - \frac{1}{T_1} \right) \tag{8}$$

2.2.3 Irreversible fluid mixture

The irreversible mixing mechanism is defined as a process in which a stream of fluid at temperature T_i enters a volume and mixes with the bulk fluid in the volume at a temperature $T \neq Ti$. The mechanism is modeled as heat transfer through finite temperature differences. The incoming stream of fluid loses (or receives) heat over a temperature range from T_i to T, while the bulk fluid in the volume receives (or loses) heat at temperature T. The corresponding exergy destruction rate is given by:

$$\dot{I} = T_0 \dot{m}_i \left[\left(s - s_i \right) - \frac{\left(h - h_i \right)}{T} \right] \tag{9}$$

where s_i and h_i are the specific entropy and specific enthalpy of the incoming stream of fluid, whereas s and h are the specific entropy and specific enthalpy of the bulk fluid in the volume.

2.2.3 Viscous flow friction

Viscous flow friction converts mechanical energy into thermal energy. Therefore, the destruction of exergy attached to the heat generated by the viscous friction in the fluid flow can be calculated by the general expression established for friction, that is:

$$\dot{I} = \dot{W}_{visc} \left(1 - \frac{T_i - T_0}{T_i} \right) \tag{10}$$

where T_i is the bulk fluid temperature and $\dot{W}_{visc}$ is the viscous friction power.

For the flow through the valves and piston-cylinder clearance, the exergy destruction rate is estimated from the entropy variation in an adiabatic process (h = constant):

$$\dot{I} = T_0 \dot{m} \left(s_2^{'} - s_1 \right) \tag{11}$$

where $\dot{m}$ is the mass flow rate through the restriction, s_1 is specific entropy of the fluid in the upstream chamber and $s_2^{'}$ is the specific entropy after the restriction (valve or piston-cylinder clearance):

$$s_2^{'} = s\left(P_2, h \right) \tag{12}$$

2.2.4 Overall exergy destruction

The overall exergy destruction rate was divided into several partial destruction rates associated with different phenomena, each linked to one or more irreversibility mechanisms in different regions inside the compressor. For instance, viscous friction and irreversible fluid mixing occur in the flow through the piston-cylinder clearance, suction valve, discharge valve, between suction line and suction muffler, in tubes and mufflers of the suction system and in tubes and mufflers of the discharge system. Moreover, heat transfer through finite temperature takes place between the compressor shell and external environment, internal environment and compressor shell, engine and internal environment, cylinder and internal environment, internal environment and suction muffler, discharge muffler and internal environment, discharge chamber and cylinder, discharge tube and internal environment, and gas in the compression chamber and cylinder. Finally, exergy destruction occurs due to friction in bearings and heat dissipation in the electrical motor.

The exergy destruction associated with the flow through the valves and the piston-cylinder clearance is brought about by viscous flow friction, Eq. (11), and irreversible fluid mixture, Eq. (9). The destruction of exergy associated with viscous friction in fluid flow in the tubes and mufflers of the suction and discharge systems is calculated based on Eq. (10), using friction factors and local loss coefficients. The exergy destruction associated with the partial mixture of the fluid from the suction line and the internal environment is calculated based on Eq. (9) and an experimentally calibrated semi-direct suction factor. The exergy destruction rates associated with heat transfer are calculated based on global conductances and temperatures, the latter predicted with a thermal simulation model.

3. RESULTS

The compressor was simulated under the operating conditions LBP (Low Back Pressure) and MBP (Medium Back Pressure) defined by the ASHRAE 23.1 standard to assess the performance of domestic refrigeration compressors. These conditions are characterized by the evaporating temperature, T_v, and the condensing temperature, T_c, of the refrigeration system: LBP (T_v = -23.3 °C; T_c = 54.4 °C; MBP (T_v = -6.7 °C; T_c = 54,4 °C.

Table 1 shows the results of the simulation for the electrical power consumed by the motor, indicated power, mass flow rate, gas temperature at compressor outlet, and overall exergy destruction rate calculated with Eq. (6). Due to the higher pressure ratio associated with the LBP condition, the resulting mass flow rate is considerably lower whereas the specific compression work is higher. However, the higher mass flow rate of the MBP condition gives rise to greater power consumption. The overall exergy destruction is 43,7% and 35,3% of the electrical power in the LBP and MBP conditions, of which 19,44% and 17,14% is due to friction in bearings and motor losses, respectively. As can be seen, the overall exergy destruction is higher in the LBP condition, meaning the compressor performs better in the MBP condition.

Tables 2 and 3 contain, respectively, the inventory of inefficiencies (losses) attributed to fluid flow and heat transfer phenomena, evaluated for the LBP and MBP conditions, obtained from the procedure indicated in section 2 for the exergy destruction breakdown. All losses are shown relatively to the electrical power ($\% \dot{W}_{el}$). As can be seen, the irreversible heat transfer mechanism is significant in the overall destruction of exergy, amounting to 17,7% of the electrical power in the LBP condition and 10,9% in the MBP condition. The greater inefficiency due to the leakage in the piston-cylinder clearance in the LBP condition can be explained by the greater pressure difference that takes place in this condition. As expected, the higher mass flow rate delivered by the compressor in the MBP condition increases the losses associated with flow through valves, tubes and mufflers. In both test conditions, the exergy destruction associated with the discharge valve is larger than the destruction in the suction valve because the diameter of the discharge valve is considerably smaller.

Table 1. Results for global performance parameters.

Parameter	LBP	MBP
Electrical power $\dot{W}_{el}$ (W)	45.3	75.5
Indicated power $\dot{W}_{ind}$ (W)	35.6	61.1
Mass flow rate (kgh^{-1})	0.88	2.21
Outlet temperature (°C)	67.1	81.4
Isentropic power ($\dot{W}_s$)	28.2	50.0
Overall isentropic efficiency (%)	62.2	48.9
Exergy destruction ($\% \dot{W}_{el}$)	43.7	35.3

Table 2. Inefficiencies associated with the fluid flow under LBP and MBP test conditions.

Parameter	LBP	MBP
Overall inefficiency	**6.47%**	**7.20%**
Piston-cylinder clearance	1.96%	1.17%
Discharge valve	1.61%	1.83%
Discharge tubes and mufflers	0.34%	1.42%
Suction valve	1.25%	1.08%
Suction tubes and mufflers	0.68%	0.87%
Suction line-internal environment mixture	0.62%	0.83%

Table 3. Inefficiencies associated with heat transfer under LBP and MBP test conditions.

Parameter	LBP	MBP
Overall inefficiency	**17.71%**	**10.91%**
shell – external environment	4.77%	2.57%
internal environment - shell	0.95%	0.52%
motor - internal environment	0.31%	0.55%
cylinder - internal environment	1.99%	0.83%
internal environment – suction muffler	0.15%	0.12%
discharge muffler - internal environment	1.82%	1.28%
discharge chamber - cylinder	1.14%	0.73%
discharge tube - internal environment	0.64%	0.99%
oil - shell	0.04%	0.02%
gas in compression chamber - cylinder	5.88%	3.30%

4. CONCLUSIONS

The identification of the main causes of inefficiency is indispensable for the optimization of a compressor's energetic performance. In this paper, a small compressor used in household refrigeration was simulated under two typical sets of operating conditions (ASHRAE LBP and MBP). With respect to the simulation results, a second law methodology was employed to obtain an inventory of energy losses (inefficiencies) in the form of exergy destruction rates, for each test condition. In the inventory of losses, the inefficiencies were attributed to several phenomena associated with the compressor operation. The results show a better overall performance in the MBP condition, with an exergetic inefficiency of 35.3%. Heat transfer phenomena were shown to have a significant part in the overall exergy destruction, taking a toll of 17.7% and 10,9% on the electrical power in the LBP and MBP conditions, respectively.

5. ACKNOWLEDGEMENTS

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