A THEORETICAL ASSESSMENT OF TWO TYPES OF POSITIVE DISPLACEMENT COMPRESSORS FOR HOUSEHOLD REFRIGERATION

Alberto R. Gomes
Programa de Pós-Graduação em Engenharia Mecânica
Universidade Federal de Santa Catarina
88040-900, Florianópolis, SC, Brazil
alberto@polo.ufsc.br

Cesar J. Deschamps
Departamento de Engenharia Mecânica
Universidade Federal de Santa Catarina
88040-900, Florianópolis, SC, Brazil
deschamps@polo.ufsc.br

Abstract. *A comparative analysis of thermodynamic performance between reciprocating and rolling piston compressors is carried out for household refrigeration. The models adopted to simulate each compressor are based on an integral formulation, resulting in a set of ordinary differential equations that are solved using a time explicit Euler method. The numerical predictions were validated through comparisons with experimental data for each compressor, obtained in a calorimeter experimental facility. Another important aspect of the work was the optimization of each compressor according to the refrigeration conditions chosen for the analysis. Results for valve dynamics, refrigerant leakages, pressure and temperature along the compression process are used to assess energy losses in the suction, compression and discharge processes, allowing estimates of isentropic efficiency and volumetric efficiency for each compressor. The reciprocating compressor was seen to return the best isentropic efficiency due to low levels of energy losses. The performance of the rolling piston compressor was seen to be affected to a great extent by excessive gas leakage, which drastically reduces its isentropic and volumetric efficiencies.*

Keywords*: household refrigeration, reciprocating compressor, rolling piston compressor.*

1. INTRODUCTION

The role of the compressor in a refrigeration system is to supply the required mass flow rate of refrigerant and to establish the pressure difference between the condensing and evaporating lines. Compressors can be divided into two main groups: roto-dynamic (centrifugal, axial, radial, etc) and positive displacement (reciprocating, rotary, scroll, screw, etc). A roto-dynamic compressor subjects a steadily flow of gas to forces that result in a continuous rise in pressure. Positive displacement compressors take in an amount of gas and trap it inside a volume that is diminished by deformation. In this process, the pressure rises and when it reaches a certain value the gas is pushed out against the pressure in the discharge chamber. Hence, opposite to what happens in a roto-dynamic compressor, the flow in a positive displacement compressor occurs in pulses. An interesting review about the stage of development of different types of compressors was presented by Pearson (1999).

Several parameters have to be considered in the design of a competitive compressor for refrigeration purpose, such as energy consumption, environmental impact and manufacturing cost. Reciprocating compressors have been a common choice for a wide range of refrigerating applications, but recent development in manufacturing techniques and materials are allowing the production of other types of compressors for situations not considered before. This paper presents a comparative analysis between reciprocating and rolling piston compressors, considering their thermodynamic performance for household refrigeration.

Very few investigations have considered comparisons between different types of refrigeration compressors. Ozu and Itami (1981) investigated, both theoretically and experimentally, the rolling piston compressor and the reciprocating compressor applied to air conditioning. They found that despite its higher mechanical losses, the rolling piston compressor offers the best overall performance because of its higher volumetric efficiency and lower thermodynamic losses in valves, the latter made possible by longer time intervals for the suction and discharge processes. Collings *et al.* (2002) carried out a theoretical analysis of the performance of three types of compressor (scroll, rolling piston and reciprocating) for CO_2 as the refrigerant fluid. Concerning the reciprocating compressor, the authors observed a great potential for leakage blockage but also a very high torque needed for the compression cycle. None of the mechanisms could offer the best performance in all aspects (energy consumption, leakage blockage and torque), implying that further development is still required to allow an optimum compressor design for CO_2.

The contribution of this paper is to provide a thermodynamic analysis of reciprocating and rolling piston compressors applied to domestic refrigeration, in the capacity range between 60 and 250 W. The emphasis of the study is to identify the main advantages and drawbacks of each compressor, through a detailed investigation of their thermodynamic losses.

2. MATHEMATICAL MODELS

Soedel (1974) describes a generalized integral formulation to simulate positive displacement compressors, demonstrating that the main phenomena during the compressor operation may be modeled by coupling four systems of equations: (i) equations for variations of geometric parameters as a function of the motor shaft angle; (ii) equations to evaluate the refrigerant thermodynamic properties, such as pressure and temperature, in the compression chamber; (iii) equations to estimate mass flow rate in different parts of the compressor, including leakage through clearances and (iv) equations to describe the dynamics of valve systems. Results of such equations for pressure, temperature, mass flow rate through suction and discharge systems and leakages are used then to characterize the compressor performance.

2.1. Reciprocating compressor

A schematic view of a reciprocating compressor is given in Fig. 1. This type of compressor possesses a compression chamber defined by a reciprocating motion of a piston inside a cylinder. When the piston moves towards the bottom dead center (BDC), it reaches a position where low pressure vapor is drawn in through the suction valve, which is opened automatically by the pressure difference between the cylinder and the suction chamber. The vapor keeps flowing in during the suction stroke, filling the cylinder volume with vapor at suction pressure. After reaching the BDC, the piston starts to move in the opposite direction, the suction valve is closed, the vapor is trapped inside the cylinder and its pressure rises as the cylinder volume decreases. Eventually, the pressure reaches the pressure in the discharge chamber and the discharge valve is forced to open. Even after the opening of the discharge valve, the piston keeps moving towards the top dead center (TDC). The suction and discharge processes do not take place at constant pressure, due to the valve dynamics and flow restriction imposed by the valve passage areas.

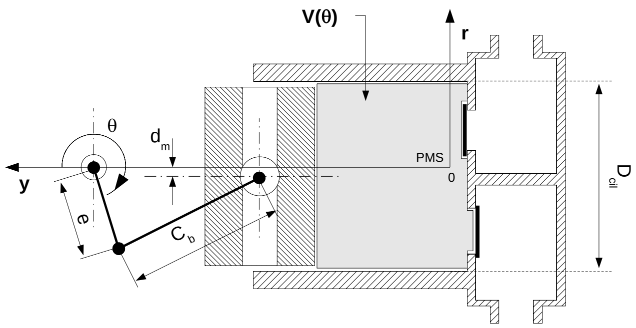


Figure 1. Schematic of a reciprocating compressor.

The integral formulation adopted in this study to simulate the compressor is a version of the code proposed by Ussyk (1984). The code accounts for piston displacement as a function of crankshaft angle, the thermodynamic process inside the cylinder, mass flow rate through the valves, valve dynamics, gas pulsation inside the mufflers and refrigerant thermodynamic properties. Several parameters are calculated during the compressor cycle, such as instantaneous pressure throughout the compressor, mass flow rate, valve dynamics, energy and mass losses, refrigerating capacity, etc.

The compression chamber volume for a certain crankshaft angle, $V(\theta)$, is given by the instantaneous position of the piston, $y(\theta)$, the cylinder diameter, D_{cil}, and the dead volume, V_m:

$$V(\theta) = (\pi D_{cil}^2 / 4)\ y(\theta) + V_m \tag{1}$$

Based on the energy conservation, the following equation can be written for the energy inside the cylinder:

$$\frac{d}{dt}(m\,e) = \dot{m}_{in}(e + p/\rho)_{in} - \dot{m}_{out}(e + p/\rho)_{out} - \dot{m}_{cl}(e + p/\rho)_{cl} + \dot{W} + \dot{Q} \tag{2}$$

where e and m are, respectively the specific energy and mass of the refrigerant inside the cylinder. On the other hand, $\dot{W}$ represents the work between piston and gas. In order to evaluate Eq. (2), mass flow rates through valves, $\dot{m}_{in}$ and $\dot{m}_{out}$, and across clearances, $\dot{m}_{cl}$, must be evaluated. The heat transfer between the gas and the compression chamber walls, $\dot{Q}$, is expressed as:

$$\dot{Q} = h_c \; A_t (T_w - T) \tag{3}$$

where h_c is the convective heat transfer coefficient, A_t is the instantaneous heat transfer area, T_w is the chamber wall temperature and T is the gas temperature. Prata *et al.* (1992) verified that the most appropriate correlation to evaluate the heat transfer coefficient H_c in the compression chamber of reciprocating compressors is that proposed by Annand (1963).

By manipulating equation (2) and assuming that inside the cylinder only the internal energy is relevant, the following equation can be derived for the gas temperature variation:

$$\frac{dT}{dt} = \frac{1}{m\, c_v}\left[h_c A_t T_w - h\frac{dm}{dt} - \dot{m}_j h_j \right] - \frac{T}{m\, c_v}\left[h_c A_t + \left.\frac{\partial p}{\partial T}\right|_v \frac{d\forall}{dt} - \left.\frac{\partial p}{\partial T}\right|_v v\frac{dm}{dt} \right] \tag{4}$$

The mass flow rates, $\dot{m}_j$, presented in the above equation are associated with leakage through the clearance between the piston and cylinder and with flow through the valves. The former is evaluated by assuming the presence of pure refrigerant in the clearance, according to a model presented by Lilie and Ferreira (1984). The mass flow rate through valves is evaluated from data for effective flow area, A_{ee}. For a certain pressure drop, A_{ee} expresses the ratio between the actual mass flow rate, $\dot{m}$, and the theoretical value given by an isentropic flow condition. Therefore,

$$\dot{m} = A_{ee} \; p_{up} \sqrt{2k/[(k-1)RT_{up}]} \; \sqrt{r_s^{2/k} - r_s^{(k+1)/k}} \tag{5}$$

where p_{up} and T_{up} denote pressure and temperature of the flow upstream, k is the specific heat ratio, R is the refrigerant gas constant and r_s is the pressure ratio between the cylinder and discharge/suction chamber. Naturally, values for p_{up}, T_{up} and r_s vary according to the flow condition through the valve (i.e., subsonic or sonic) and also with the flow direction.

The dynamics of reed valves adopted in reciprocating compressors can be expressed in a simplified way, by using a one-degree-of-freedom model:

$$F_v - K_v x - C_v \dot{x} = m_v \ddot{x} \tag{6}$$

where x, $\dot{x}$ and $\ddot{x}$ are, respectively, the lift, velocity and acceleration of the reed valve. On the other hand, F_v is the resulting force on the valve, originated by the flow induced pressure distribution the reed valve, oil stiction and valve pre-load. In the present investigation, the latter two contributions were not included. The valve stiffness and damping coefficients, K_v and C_v, respectively, as well as the valve mass, m_v, are determined experimentally. The force, F_v, resulting from the pressure distribution on the reed surface is obtained with reference to the effective force area, A_{ef}, which is determined from the pressure difference across the valve, Δ_{pv}, according to $A_{ef} = F_v/\Delta_{pv}$. The effective force area can be understood as a parameter related to how efficiently the pressure difference Δ_{pv} opens the valve.

2.2. Rolling piston compressor

The rolling piston compressor is composed by a compression chamber and a suction chamber operating simultaneously, separated from each other by a rolling piston and a contact vane, as illustrated in Fig.2. During the suction part of the cycle, refrigerant gas is drawn through a suction port into the suction chamber as its volume is increased. At the same time, the compression and discharge processes take place in the decreasing volume on the opposite side.

Padhy and Dwivedi (1994) presented a simulation methodology for rolling piston compressors, based on mass and energy conservation equations, with a correlation for estimating the heat transfer between the refrigerating fluid and solid walls in different parts of the compressor.

Equations for the instantaneous volumes of the compression and suction chambers are required to simulate the rolling piston compressor. For instance, the compression chamber volume can be determined from the following relationship:

$$V_{cc}(\theta) = V_t - V_{sc}(\theta) - V_b(\theta) \tag{7}$$

In the above equation, V_t, V_{sc} and V_b are, respectively, the total volume of gas inside the compressor, the suction chamber volume and the volume occupied by the vane. Such volumes can be expressed through the following relationships:

$$V_t = \pi H_c (R_c^2 - R_p^2) + H_{od}\ (\pi D_{od}^2\ /\ 4) + V_4 \tag{8}$$

$$V_{cs}(\theta) = (R_c H_c \theta / 2) - \{R_p^2 H_c [\theta + \text{arcsen}\ (ER\ \ \text{sen}\ \ \theta)]/2\} - (e\ \text{sen}\ \theta\ H_c C / 2) - V_b(\theta)/2 \tag{9}$$

$$V_b(\theta) = H_c B_b D - 2V_4 \tag{10}$$

where R_c and H_c are the radius and the height of the fixed cylinder. On the other hand, R_{pi} is the internal radius of the rolling piston, R_b is the radius of the vane tip and B_b is the vane thickness. Finally, D_{od} and H_{od} are the diameter and height of the discharge orifice, whereas e is the eccentricity and θ is the shaft angular position. The volume V_4 (= A_4 H_c) appearing in Eqs. (8) and (10) corresponds to the volume formed between the contact vane and the rolling piston, as identified in Fig. 2.

The rate of volume variation can be obtained from:

$$\dot{V}_{cc}(\theta) = \frac{dV_{cc}}{dt} = \frac{\partial V_{cc}}{\partial \theta}\frac{d\theta}{dt} \tag{11}$$

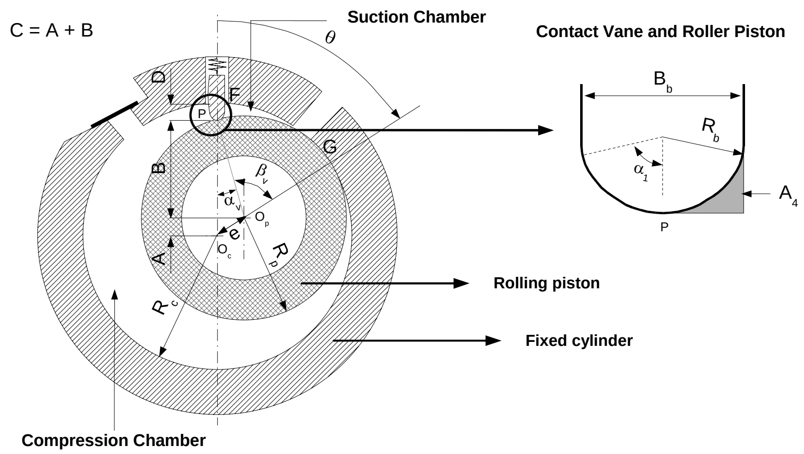


Figure 2. Schematic view of a rolling piston compressor.

The rolling piston requires a discharge valve only, since the suction and discharge chambers are separated from each other. As a consequence, a valveless orifice is the only restriction to the refrigerant intake. The same concept of effective flow area described for reciprocating compressor is adopted to determine the mass flow rate in the discharge valve, whereas a classical relationship for orifice is used for the suction process. The discharge valve dynamics is solved accordingly with the procedure already explained for the reciprocating compressor.

Internal gas leakage is a very important aspect that must be considered in the analysis of rolling piston compressors, due to the presence of many clearances in the compressor. Such leakages reduce the mass flow rate of refrigerant supplied by the compressor to the system, deteriorating the refrigerating capacity. The main contributions to the gas leakage can be summarized as follows: i) leakage through the minimum radial clearance between rolling piston and cylinder; ii) leakage through the clearance between vane and cylinder; iii) leakage across the lateral clearance between rolling piston and bearings; iv) leakage between the discharge and suction chambers through the vane lateral clearance.

The leakage through the minimum clearance in rolling piston compressor is the main contribution for the internal mass flow loss, may accounting for almost 70% of the total internal gas leakage. In the present analysis the model proposed by Ferreira *et al.* (1992) for the oil leakage through the minimum clearance has been adopted:

$$\dot{m}_{\text{omc}} = 0.0162\mu\ \delta_{mc}[\Delta P\ \rho_{oleo}(\delta_{mc}/\mu)^2][H_c/\delta_{mc}]^{0.504} \tag{12}$$

where δ_{mc} is the minimum clearance between the rolling piston and the cylinder. Then, based on data for absorption of refrigerant vapor in the lubricant oil, w, an estimate for the refrigerant leakage is obtained as:

$$\dot{m}_{gmc} = w\ \dot{m}_{omc} \tag{13}$$

Leakage through the slot between the vane and cylinder surfaces is evaluated by considering a steady laminar flow between two parallel plates:

$$\dot{m}_{ovc} = \frac{H_c U \delta_{vc}}{2} - \frac{H_c \delta_{vc}^3}{12\mu}\frac{\Delta P}{L} \tag{14}$$

where δ_{vc}, U, L and μ are, respectively, the clearance between the vane and the cylinder, the vane velocity, the vane length and the oil viscosity. The pressure drop ΔP corresponds to the difference between the discharge pressure, P_d, and the pressure inside the suction or discharge chambers, P_c, depending on which chamber is being considered. Again, based on the absorption of refrigerant vapor in the lubricant oil, w, an estimate for the gas leakage through the clearance can be evaluated.

The oil leakage between the piston and bearings is determined by considering a steady incompressible laminar flow between two parallel disks (Krueger, 1988), resulting in the following relationships for the oil mass flow rate to the suction and compression chambers:

$$\dot{m}_{ops} = (\delta_p^3/4)\frac{(P_d - P_{sc})}{24\mu\pi \ln(R_{pe}/R_{pi})}\int_0^{2\pi}\theta\, d\theta \quad ; \quad \dot{m}_{opd} = (\delta_p^3/4)\frac{(P_{dc} - P_d)}{24\mu\pi \ln(R_{pe}/R_{pi})}\int_0^{2\pi}(2\pi - \theta)d\theta \tag{15}$$

where δ_p is the axial clearance between the vane and the cylinder. The refrigerant mass flow rate is determined by using data for the absorption of refrigerant vapor in the lubricant oil, w.

Finally, the leakage across the clearance between vane and bearings is estimated through the hypothesis of isentropic compressible flow of a perfect gas in a convergent-divergent nozzle. The assumption of absence of oil in that region has been confirmed by experimental observation. Therefore, an expression similar to Eq. (5) can be used to find the refrigerant leakage between the compression and suction chambers. No leakage is considered through the contact line formed between the vane and the rolling piston, since both surfaces are in permanent contact throughout the whole compressor cycle.

With known values for mass flow rate through valves and due to leakages, the gas mass variation in each chamber can be evaluated. Then, in a similar approach adopted for the reciprocating compressor, the energy equation is used to evaluate the refrigerant thermodynamic properties in the suction and discharge chambers. Following Liu and Soedel (1992), wall heat transfer is evaluated from distinct relationships for each chamber.

3. NUMERICAL SOLUTION METHODOLOGY

The differential equations required to describe the thermodynamic processes in the compressors are numerically solved through an explicit Euler method. The solution procedure is repeated for a number of compression cycles and convergence is assessed by examining whether the compressor operation conditions are cyclically repeated.

For a meaningful comparison of different compressors it is of fundamental importance that they are optimized for each refrigerating capacity analyzed. In this regard, an optimization procedure for each compressor was developed using the commercial software modeFRONTIER. The structure of this procedure can be divided into four parts: i) input data for compressor geometric parameters, main clearances, valve characteristics, refrigerating capacity, refrigerant fluid and lubricating oil; ii) compressor simulation models, as detailed in the previous section; iii) optimization algorithm; iv) output data on variables needed for the analysis, such as the isentropic efficiency. The genetic algorithm adopted for the compressor optimization selects and combines parameters that offer the best energy efficiency for a certain refrigerating capacity, respecting a number of restrictions imposed by the compressor design.

An important simplification in the present study was to assume the refrigerant temperature at the suction chamber to be 57.8 ^{o}C for all compressors. Hence, the gas superheating throughout the suction muffler is not evaluated. Additionally, the pressure pulsation in the suction chamber caused by the propagation of pressure waves was also neglected. Therefore, the analysis is directed to thermodynamic losses associated with the compression process itself, represented by losses due to refrigerant leakage, valve restriction and residual mass at the end of each compression cycle.

4. RESULTS

A first step in the present analysis was to validate the numerical simulation methodology, through comparisons of predictions with experimental data. For this purpose, two commercially available compressors were selected: i) reciprocating compressor ($\dot{Q}_e$= 155 W, LBP condition, R600a, 50 Hz); ii) rolling piston compressor ($\dot{Q}_e$= 2,052 W, HBP condition, R22, 60 Hz). All tests were conducted in a calorimeter facility, in which the uncertainty associated with the measurements are ± 2 % for refrigerating capacity, mass flow rate and power consumption. On the other hand, measurements of thermodynamic losses have an uncertainty of ± 2 %.

As Tab. 1 and 2 show, there is a good agreement between experimental and numerical results for the reciprocating and rolling piston compressors. Differences are seen to be in the range of ± 3% for global quantities, such as the refrigerating capacity, $\dot{Q}_e$, and compression power, $\dot{W}_{pV}$, and ± 10% for energy losses in the suction and discharge processes, $\dot{W}_s$ and $\dot{W}_d$.

Table 1 - Experimental data and numerical prediction for reciprocating compressor performance; CECOMAF condition. ($T_{evap.}$ = -25°C, $T_{cond.}$ = 55 °C, $T_{sup.}$ = 32.2 °C and $T_{sub.}$ = 55 °C).

	$\dot{Q}_e$	$\dot{W}_{pV}$	$\dot{W}_d$	$\dot{W}_s$
Exp. data	154	88.4	4.0	5.4
Prediction	155 (0.6%)	85.3 (-3.5%)	3.8 (-5.0%)	4.6 (-14.8%)

Table 2 - Experimental data and numerical prediction for rolling piston compressor performance; ASHRAE condition. ($T_{evap.}$ = 7.2 °C, $T_{cond.}$ = 54.4°C, $T_{sup.}$ = 32.2°C and $T_{sub.}$ = 32.2°C).

	$\dot{Q}_e$	$\dot{W}_{pV}$	$\dot{W}_d$	$\dot{W}_s$
Exp. data	2,148	508	8.2	12.9
Prediction	2,082 (-3.1%)	506 (-0.4%)	8.2 (0.0%)	14.3 (10.9%)

After validating the simulation procedure, the work was directed to the comparative analysis of the compressors for the following LBP condition: evaporating and condensing temperatures (T_{evap} and T_{cond}) of -23.3°C and 54.4°C, respectively, with subcooling and superheating of 32.2 °C. The compressors were analyzed for three refrigerating capacities (60, 150 e 250 W), with a fixed rotation speed of 50 Hz and the R600a as the refrigerant fluid.

In all simulations of the rolling piston compressor, temperatures at the solid walls and lubricating oil temperature were prescribed with reference to measurements carried out by Puff (1990). In the case of the reciprocating compressor, experimental data for temperature at different components of the compressor were supplied by EMBRACO.

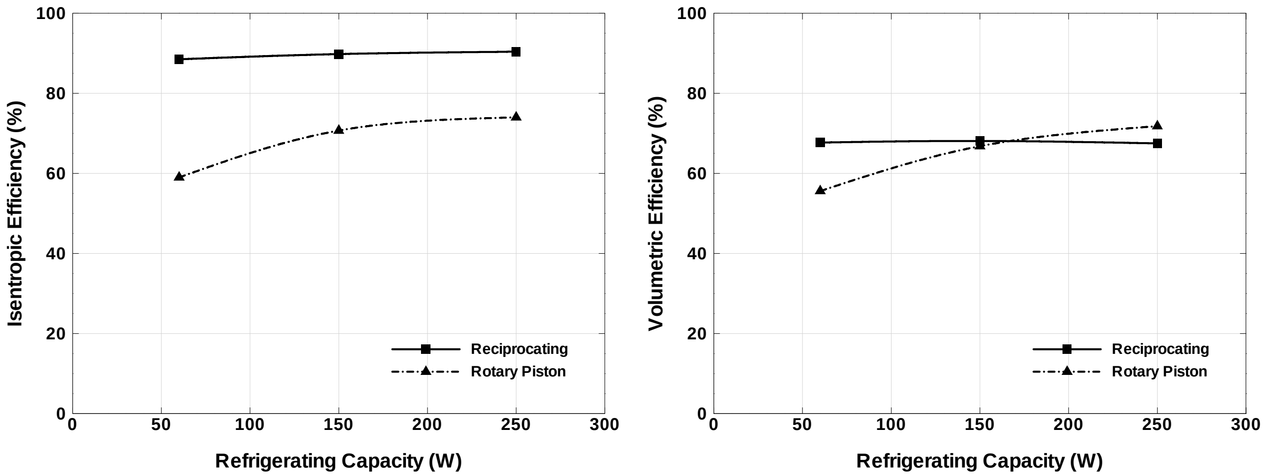
Figure 3(a) shows the isentropic efficiencies of both compressors as a function of the refrigerating capacity. As can be seen, the reciprocating compressor offers the best performance in all capacities, with the isentropic efficiency being 22% to 50% higher than that of the rolling piston. A similar characteristic verified for the two compressors is the increase of the isentropic efficiency with the refrigerating capacity, particularly for the rolling piston compressor when the capacity is varied from 60 W to 150 W.

Figure 3(b) presents results for volumetric efficiency. The reciprocating compressor is always affected by the presence of an amount of refrigerant at discharge pressure left inside the cylinder, since a gap between the piston and cylinder head is necessary. The dead volume resulting from such a clearance may account for up to 5 % of the total cylinder volume and is the main source of volumetric inefficiency of the reciprocating compressor. Therefore, as the piston moves downwards, the refrigerant in the clearance is initially re-expanded, causing a delay in the opening of the suction valve. Hence, instead of taking in a volume of vapor equal to the cylinder volume, a smaller volume of gas is drawn from the suction chamber, reducing the compressor volumetric efficiency.

In the rolling piston compressor, a residual mass is also present due to a remaining volume in the compression chamber, added to fixed contributions of the discharge orifice and of the volume V_4 formed between the vane tip and the piston. At the end of the compression process, the residual mass is expanded to the suction chamber, reducing the volume of refrigerating fluid that can be draw in, which affects the compressor volumetric efficiency. Figure 3(b) shows that the volumetric efficiency of the rolling piston compressor increases considerably with the refrigerating capacity and is even greater than that of the reciprocating compressor for $\dot{Q}_e$= 250 W.

The effects caused by the dead volume are quite different on the reciprocating and rolling piston compressors. Whereas in the former the main consequence is a decrease in the volumetric efficiency, in the rolling piston compressor there is also a great impact on the isentropic efficiency. This can be explained by remembering that the energy supplied to the residual mass during the compression process cannot be returned to the shaft; in fact, the energy is completely lost through leakages. This inconvenient aspect of the rolling piston compressor will be explored in further details shortly.

It should be mentioned that other aspects also affect the compressor efficiency, such as heat transfer between the refrigerant and the cylinder wall, throttling in valve ports, gas leakage through the clearance between the piston and the cylinder, backflow in the valves, etc.



(a) (b)

Figure 3. Results for isentropic and volumetric efficiencies.

Table 3 presents the main mass losses for the rolling piston and reciprocating compressors, and their effect on the on the volumetric efficiency. The highest level of leakage is found for the rolling piston compressor, with an average value corresponding to approximately 27% of the ideal mass flow rate. Such levels of leakage require a volume displacement considerably greater than that theoretically needed for a certain refrigerating capacity. The main leakage path is that across the minimum radial clearance between the rolling piston and the cylinder, being responsible for half of the total leakage. For the reciprocating compressor the leakage is very small (~ 1%). On the other hand, the dead volume is responsible for a loss of 23% of the mass the reciprocating compressor could theoretically provide to the system. No dead volume loss is indicated in Tab. 3 for the rolling piston compressor because any residual mass in the compression chamber is lost through leakage.

The leakage mass flow rate is affected by the pressure difference throughout the clearance and also by the flow passage area created by the clearance. For a given refrigeration condition, the aforementioned pressure difference will be constant, regardless of the compressor capacity. Therefore, as the compressor size decreases the same should occur with the leakage, since the flow passage area becomes smaller. Nevertheless, the area available for leakage does not decrease proportionally with the compressor capacity and the effect on the volumetric efficiency is even worse for small refrigerating capacities. It should be mentioned that other parameters that affect the refrigerant leakage, such as oil viscosity and absorption of refrigerant vapor in the oil, were kept constant in the present analysis.

Table 3. Results for mass loss and volumetric efficiency.

		$\dot{Q}_e$ = 60 W		$\dot{Q}_e$ = 150 W		$\dot{Q}_e$ = 250 W	
		Recip.	R.Piston	Recip.	R.Piston	Recip.	R.Piston
Dead volume mass loss (DVL)	$\dot{m}_{dv}/\dot{m}_{ideal}$	23.1	---	23.0	---	22.0	---
Leakage mass loss (LL)	$\dot{m}_{leak}/\dot{m}_{ideal}$	1.6	36.1	1.0	25.1	0.8	20.1
Remaining mass losses (RL)	$\dot{m}_{others}/\dot{m}_{ideal}$	2.2	0.2	2.4	0.0	4.6	0.0
Vol. efficiency with DVL	$\eta_{v,dvl}$ [%]	76.9	---	77.0	---	78.0	---
Vol. efficiency with LL and RL	$\eta_{v,llrl}$ [%]	88.1	55.6	88.5	66.8	86.5	71.8
Overall volumetric efficiency	η_v [%]	67.7	55.6	68.1	66.8	67.5	71.8

Table 4 shows thermodynamic losses associated with the suction, compression and discharge processes of both compressors. The results show that a much larger energy loss occurs in the compression process of the rolling piston compressor. It will be shown that this is due to high levels of gas leakage, which dissipate the energy supplied during the gas compression. Therefore, leakage in the rolling piston affects both the volumetric efficiency and the isentropic efficiency, making it difficult to design a competitive compressor for small refrigerating capacities.

The energy losses in the suction and discharge systems also affect the compressor efficiency and vary according with the compressor type. Because no suction valve is used in the rolling piston, the energy loss is caused exclusively by the flow restriction of the suction orifice, which only becomes considerable for much larger capacities. For the reciprocating compressor there are two additional effects that act to increase the energy loss in the suction process: i) the use of a reed valve and ii) the short time interval available for the suction process. In addition to the fact of not having a suction valve, the rolling piston compressor has a longer time interval for the suction process, resulting in negligible energy loss during the gas intake.

The reciprocating compressor and the rolling piston compressor analyzed in this study require a discharge valve and, therefore, the valve dynamics is a common parameter that affects their discharge energy loss. Any difference in the valve performance of both types of compressor should be explained by the following aspects: i) discharge orifice flow passage area; ii) valve constructive parameters, such as mass and stiffness, and iii) time available for the discharge process. Naturally, the larger is the valve passage area the smaller will be the energy loss, if the valve dynamics is satisfactory. On the other hand, for a certain capacity, the velocity levels in the valve and, as consequence, the energy losses are reduced when the time available for the discharge process is increased. The discharge valve system optimized for the reciprocating compressor in this study adopts multiple orifices, with a flow passage area between fifteen and thirty times larger than that which could be designed for the rolling piston compressor. As a consequence, even though the time interval for the discharge process in the rolling piston compressor is twice that of the reciprocating compressor, it is sufficient to offset the negative effect of its small valve area, as cab seen in Tab. 4.

Table 4. Results for energy losses and isentropic efficiency.

		$\dot{Q}_e$ = 60 W		$\dot{Q}_e$ = 150 W		$\dot{Q}_e$ = 250 W	
		Recip.	R.Piston	Recip.	R.Piston	Recip.	R.Piston
Compression energy loss	$(W_{ef} - W_s)/W_{ind}$	10.2	39.1	8.9	26.7	8.4	23.3
Suction energy loss	W_{suc}/W_{ind}	0.8	0.0	0.9	0.0	0.8	0.1
Discharge energy loss	W_{des}/W_{ind}	0.4	1.9	0.4	2.5	0.4	2.6
Isentropic efficiency	η_s	88.5	59.0	89.8	70.7	90.4	74.0

Table 5 was prepared to help explain the influence of leakage on the rolling piston compressor performance, by providing results of volumetric and isentropic efficiencies for the cases with and without leakage. As can be observed, in the presence of leakage the compression loss corresponds to approximately 39% of the total energy consumption, considering a refrigerating capacity of 60 W. However, if no leakage was present, the energy loss would drop to just 4.2%. Therefore, the reduction of gas leakage is a crucial measure to improve not only the volumetric efficiency of rolling piston compressors, but also their isentropic efficiency. As suggested by results in Fig. 3, the relative importance of leakage in the rolling piston compressor decreases as the refrigerating capacity becomes larger.

Table 6 shows results for the reciprocating compressor performance, with and without the presence of dead volume. It can clearly be seen that, despite having a great impact on the volumetric efficiency, the dead volume has virtually no influence on the isentropic efficiency, η_s. The reason is that during the gas re-expansion in the reciprocating compressor, nearly all the energy used to compress the residual mass is returned to the crankshaft mechanism through the work of a pressure load on the piston. Naturally, this does not occur in the rolling piston compressors because any residual mass is lost through leakage and the energy supplied in its compression cannot be returned to the shaft. The small difference observed in Tab. 6 for η_s can be attributed to the valve dynamics, which may change with the dead volume, affecting energy losses in the suction and discharge processes.

Table 5. Gas leakage effect on the rolling piston compressor performance ($\dot{Q}_e$ = 60 W).

		with gas leakage	**without gas leakage**
Volumetric efficiency [%]	η_v	55.6	85.7
Compression loss [%]	$(W_{ef} - W_s)/W_{ind}$	39.1	4.2
Isentropic efficiency [%]	η_s	59.0	95.1

Table 6. Dead volume effect on the reciprocating compressor performance ($\dot{Q}_e$ = 60 W).

		with dead volume	**without dead volume**
Volumetric efficiency [%]	η_v	67.7	89.4
Compression loss [%]	$(W_{ef} - W_s)/W_{ind}$	10.2	9.1
Isentropic efficiency [%]	η_s	88.5	89.8

5. CONCLUSIONS

This paper presented a thermodynamic analysis of two types of positive displacement compressors (reciprocating and rolling piston) applied to domestic refrigeration. The numerical predictions were validated through comparisons with experimental data for each compressor, obtained in a calorimeter experimental facility. The comparison was made possible through an optimization of each type of compressor for the refrigeration conditions chosen for the analysis.

The rolling piston compressor was found to have the lowest energy loss in the suction process because its passage area and time interval for the gas intake are greater than those associated with the reciprocating compressor. Although the discharge process in the rolling piston compressor occurs also in a longer time interval longer, the smaller valve passage area returns the highest energy loss.

The study identified the dead volume as the main responsible for mass loss in reciprocating compressors. However, it could be noted that such mass loss does not contribute to energy loss. For this reason, the reciprocating compressor was found to return the highest isentropic efficiency in all refrigerating capacities investigated. The performance of the rolling piston compressor was seen to be affected to a great extent by excessive gas leakage, which drastically reduces its volumetric efficiency at small refrigerating capacities. Moreover, the residual mass considerably decreases its isentropic efficiency, because the energy supplied to the gas during the compression process is not returned to the shaft, as it occurs with the reciprocating compressor.

6. ACKNOWLEDGEMENTS

This study forms part of a joint technical-scientific program of the Federal University of Santa Catarina and EMBRACO. Support from FINEP (Federal Agency of Research and Projects Financing) and CAPES (Coordination for the Improvement of High Level Personnel) is also acknowledged.

7. REFERENCES

Annand, W. D., 1963, “Heat Transfer in the Cylinders of Reciprocating Internal Combustion Engines”, Proc. Institution of Mechanical Engineers, Vol. 177, pp. 973.

Collings, D. A., Yap, Z. K. and Haller, D. K., 2002, “Compressor Mechanism Comparison for R744 Application”, Proc. Int. Compressor Engineering Conference at Purdue, paper C9-1.

Ferreira, R.T.S., Lilie, D.E.B., Gasche, J.L. and Prata, A.T. 1992, “Bicylindrical Coordinate Formulation for the Leakage Flow through the Minimal Clearance in a Rolling Piston Compressor”, Proc. Int. Compressor Engineering Conf. at Purdue, pp. 393-401.

Krueger, M., 1988, “Theoretical Simulation and Experimental Evaluation of an Hermetic Rolling Piston Rotary Compressor”, M.Sc. Thesis, School of Mechanical Engineering, Purdue University, West Lafayette, USA.

Lilie, D.E.B. and Ferreira, R.T.S., 1984, “Evaluation of the Leakage through the Clearance Between Piston and Cylinder in Hermetic Compressors”. Proc. Int. Compressor Engineering Conf. at Purdue, pp. 1-6.

Liu, Z. and Soedel, W., 1992, “Modeling Temperatures in High Speed Compressor for the Purpose of Gas Pulsation and Valve Modeling”. Proc. Int. Compressor Engineering Conf. at Purdue, pp. 1375-1384.

ModeFRONTIER, ESTECO, Version 3.1.0, 2005.

Ozu, M. and Itami, T., 1981, “Efficiency Analysis of Power Consumption in Small Hermetic Refrigerant Rotary Compressors”, 1981, Int. J. Refrigeration, Vol. 4, pp. 265 – 270.

Padhy, S. K. and Dwivedi, S. N., 1994, “Heat Transfer Analysis of a Rolling-Piston Rotary Compressor”, 1994, Int. J. Refrigeration, Vol.17, pp 400 – 410.

Pearson, S.F., 1999, “Natural Selection – The Struggle for Survival in the Competitive World of Refrigeration Equipment”, Proc. 20th Int. Congress of Refrigeration, paper 730.

Prata, A.T., Ferreira, R.T.S., Todescat, M.L. and Fagotti, F., 1992, “Thermal Energy Analysis in Reciprocating Hermetic Compressors”. Proc. Int. Compressor Engineering Conf. at Purdue, pp. 1419-1428.

REFPROP, NIST Standard Reference Database 23, Version 7.0, 2002.

Soedel, W., 1974, “Introduction to Computer Simulation of Positive Displacement Type Compressors”. Ray W. Herrick Laboratories, Purdue University.

Ussyk MS. 1984, “Numerical Simulation of the Performance of Hermetic Reciprocating Compressors”. M.Sc. Thesis, Dept. Mech. Eng., Federal University of Santa Catarina, Brazil.

8. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.