2010

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ABSTRACT

This paper considers a systematic analysis of volumetric inefficiency sources in reciprocating compressors. The procedure is based on the efficiency detachment approach discussed by Pèrez-Segarra et al. (2005), but includes a more detailed evaluation of inefficiencies associated with the compression cycle. A small-capacity reciprocating compressor adopted for household refrigeration is simulated through a numerical methodology capable of evaluating all the main parameters concerning the compressor operation. An assessment of several volumetric inefficiency sources is carried out with reference to results for the thermodynamic process inside the cylinder, fluid flow through valves, piston-cylinder clearance leakage, gas pulsation in mufflers and refrigerant thermophysical properties.

1. INTRODUCTION

A compressor is a positive displacement machine, in which the density of the gas entering the compression chamber has a major influence on the mass flow rate. The gas density at the suction chamber is influenced by the evaporating pressure as well as the suction temperature, which is increased by heat transfer in the suction system. Besides suction superheating, other well-known effects reduce the overall compressor mass flow rate, such as the clearance residual mass, in-cylinder superheating and leakage through the gap between cylinder and piston.

The volumetric efficiency $\eta_v (= \dot{m}/\dot{m}_{ideal})$ is a parameter frequently adopted to relate the actual mass flow rate of a compressor, $\dot{m}$, with its theoretical maximum, $\dot{m}_{ideal}$. This parameter plays a major role in compressor design since it defines the required compressor swept volume for a certain application. Several thermodynamic phenomena that take place inside the compressor affect directly the volumetric efficiency and the aforementioned definition does not discriminate each one of them.

Stouffs *et al.* (2001) proposed dimensionless parameters to model some of the aforementioned effects, such as suction superheating due to cylinder wall heat transfer and valve pressure drop, and successfully established the volumetric effectiveness of an air compressor. However, the calibration of the dimensionless parameters is of great importance in their approach and, therefore, an experimental characterization of the compressor is always needed.

Pèrez-Segarra *et al.* (2005) proposed an efficiency detachment procedure to analyze the impact of different phenomena on the isentropic, mechanical, electrical and volumetric efficiencies. The volumetric losses were divided into two main contributions: before the cylinder and inside the cylinder. For the latter contribution, the authors made a further division and considered the compression cycle to be formed by four major sub-processes (suction, compression, discharge and expansion) with a local efficiency for each one of them. As identified by Pèrez-Segarra *et al.* (2005), such a procedure assumes that different phenomena affect each sub-process and, therefore, the efficiency of each sub-process is a combination of two or more physical effects.

The present study follows the efficiency detachment procedure proposed of Pèrez-Segarra *et al.* (2005), but further discriminates the in-cylinder thermodynamic processes in order to provide a more detailed understanding of the compressor volumetric inefficiencies.

2. THEORETICAL MODELING OF THE COMPRESSION CHAMBER

The compression chamber of a reciprocating compressor encompasses the volume formed by the piston, cylinder walls and the valve plate (Figure 1). The piston moves alternatively along the cylinder axis, between the bottom dead center and the top dead center close to the valve plate. A crankshaft mechanism is used to convert the rotational movement of the electric motor into the alternative movement of the piston. The volume of the compression chamber can be described according to some geometrical parameters (Ussyk, 1984). During the compression cycle, mass enters or leaves the cylinder through the suction and discharge valves. Small reciprocating compressors adopt reed type valves that open and close automatically, due to the pressure difference between the cylinder and the suction/discharge chamber.

The compression process can be represented by a pressure-volume diagram, as shown in Figure 2. Initially, the piston is in a certain position *A* and the mass of gas inside the cylinder is M^t_A. As the piston moves downwards, the pressure inside the cylinder is decreased. At a certain position between *A* and *b*, the piston reaches a position where the suction valve opens and low pressure vapor is drawn into the cylinder 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 as the piston moves towards the bottom dead center (point *b*). At point *b*, the piston inverts the direction of its motion, therefore increasing the gas pressure. From *b* to *B*, mass can still enter the cylinder because of flow inertia or may exit due to backflow. The total mass that enters the cylinder after a complete suction process is denoted by m_{suc} and the amount that eventually leaves the cylinder is denoted by $m_{suc,r}$.

During the time interval between points *B* and *C*, the suction valve is closed and the vapor trapped in the cylinder has its pressure raised as the cylinder volume is decreased. Eventually, during the period *C-t*, the pressure in the cylinder is such that the discharge valve is forced to open. Mass then exits the cylinder until the top dead center *t*. From there on, the piston changes again its motion direction, increasing the volume. During the interval *t-D*, the discharge valve closes, and in the same manner as for the suction process, mass may enter or leave the cylinder. The total mass that has left the cylinder through the discharge valve is denoted by m_{dis}, whereas m_{disr} is the amount of mass associated with backflow.

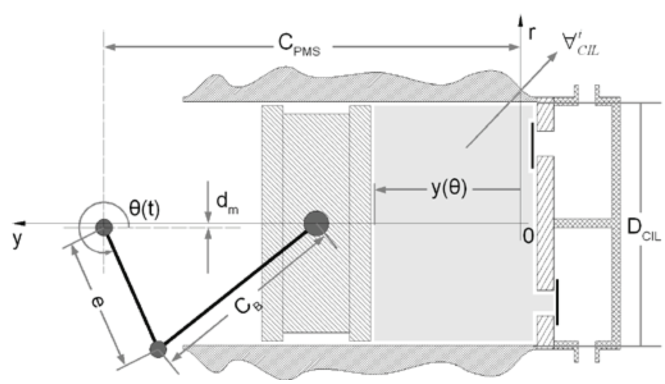

Figure 1. Schematic view of the compression chamber.

In addition to the flow through valves, mass enters and leaves the compression chamber through the gap between the piston and the cylinder walls during the entire compression cycle. The net amount of mass that leaves the compression chamber through the gap is denoted by m_{lkg}. When the piston is back to the position *A*, the mass inside the cylinder is $M_A^{t+\Delta t}$. Applying mass conservation to the compression chamber along the entire cycle results:

$$\left(M_A^{t+1/f} - M_A^t\right) + \left(m_{suc} - m_{suc,r} - m_{dis} + m_{dis,r} - m_{lkg}\right) = 0 \tag{1}$$

After the compressor reaches a steady cyclic operation, the mass at any corresponding points of the p-$\forall$ diagram of different cycles is always the same; thereafter the first term may be neglected.

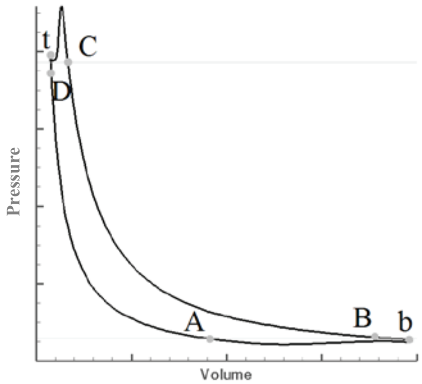

Figure 2. Pressure x Volume diagram.

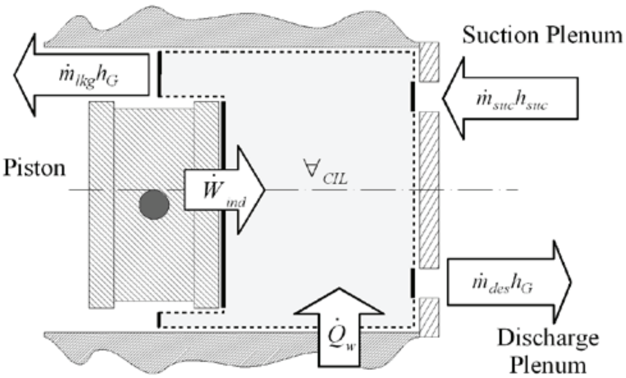

Figure 3. Control volume over the compression chamber.

In order to apply the energy equation to the gas volume inside the compression chamber, a control volume is defined as shown in Figure 3. As can be seen, energy can be transported through the valves and the piston/cylinder gap. On the other hand, heat is transferred through the cylinder walls at the same time as work is exchanged between the gas and the piston. Based on the energy conservation, the following equation can derived to calculate the time variation of the in-cylinder gas temperature (Todescat *et al.*, 1990):

$$\frac{dT_G}{dt} = A_T - B_T T_G \tag{2}$$

$$A_T = \frac{1}{m_G c_{v,G}}\left(H_w A_w T_w - h_G \frac{dm_G}{dt} - \sum \dot{m}h \right) \tag{3}$$

$$B_T = \frac{1}{m_G c_{v,G}}\left(H_w A_w + \left.\frac{\partial p_G}{\partial T_G}\right|_v \frac{d\forall_{CIL}}{dt} - \left.\frac{\partial p_G}{\partial T_G}\right|_v v_G \frac{dm_G}{dt} \right) \tag{4}$$

In Eqs. (2) to (4), p_G, T_G, v_G and h_G are, respectively, the instantaneous pressure, temperature, specific volume and specific enthalpy of the gas inside the cylinder. Furthermore, H_w is the convective heat transfer coefficient, calculated according to the correlation proposed by Annand (1963), A_w is the heat transfer area between gas and compression chamber walls and $\forall_{CIL}$ is the volume of the cylinder. Energy and mass conservation equations, along with an adequate equation of state, can describe the thermodynamic process the gas undergoes during the entire compression cycle.

3. VOLUMETRIC EFFICIENCY

The concept of volumetric efficiency requires the definition of an ideal compressor. According to Gosney (1982), an ideal compressor executes an isentropic compression, in which the gas entering the cylinder has the same thermodynamic state ϕ_1 of the suction line at the compressor inlet. Additionally, an ideal compressor neglects the presence of a clearance volume when the piston is at the top dead center. Therefore, the mass flow rate of this theoretical compressor is:

$$\dot{m}_{id} = \rho_1 \forall_{sw}^{c=0} f_n \tag{5}$$

where ρ_1 is the gas density at the compressor inlet, $\forall_{sw}^{c=0}$ is the total swept volume of the compression chamber and f_n is the nominal operating frequency of the compressor.

For a standard refrigeration system, the compressor delivers the same quantity of mass that passes through the evaporator during one cycle of compression, m_{evap}, at the actual operating frequency f_r. Consequently, the volumetric efficiency can be defined as (Pèrez-Segarra *et al.*, 2005):

$$\eta_v = \frac{f_r}{f_n}\frac{\forall_{sw}}{\forall_{sw}^{c=0}}\frac{m_{evap}}{\rho_1 \forall_{sw}} \tag{6}$$

The first term on the right side of Equation (6) represents the effect of electric motor slippage, which reduces the compressor nominal frequency of operation. The second term, the theoretical volumetric efficiency, considers the effect of the clearance volume, which reduces the compressor volume displacement because the remaining gas in the cylinder after the discharge process is re-expanded. The third term is defined as a secondary volumetric efficiency, which takes into account all other irreversible phenomena that occur inside the compression chamber. Pèrez-Segarra *et al.* (2005) divide the compression cycle into 4 stages (suction, compression, discharge and expansion) and define a volumetric efficiency for each one of them by means of the efficiency detachment procedure.

Due to large number of processes in each stage of the compression cycle, the present study follows a more detailed approach, by distinguishing all physical phenomena taking place inside the compression chamber. The first step is to define the mass that flows through the evaporator during one compression cycle, m_{evap}. Based on the control volume shown in Figure 4, m_{evap} can be evaluated from:

$$m_{evap} = m_{suc} - m_{suc,r} - \left(m_{lkg} + m_{lkg,r}\right) = m_{suc} - m_{suc,r} - m_{lkg} \tag{7}$$

By substituting Eq. (7) into Eq. (6), it follows that:

$$\eta_v = \frac{f_r}{f_n}\frac{m_{evap}}{\rho_1 \forall_{sw}^{c=0}} = \frac{f_r}{f_n}\left(1 - \frac{\rho_1 \forall_{sw}^{c=0} - m_{suc}}{\rho_1 \forall_{sw}^{c=0}} - \frac{m_{suc,r}}{\rho_1 \forall_{sw}^{c=0}} - \frac{m_{lkg}}{\rho_1 \forall_{sw}^{c=0}}\right) = \eta_{v,f}\eta_{v,m} \tag{8}$$

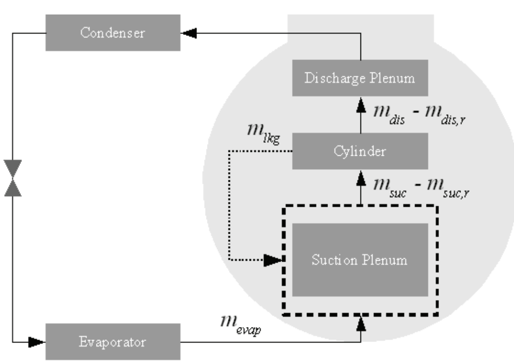

Figure 4. Control volume outside the suction plenum.

The first term on the rightmost side of Eq. (8), $\eta_{v,f}$, is the volumetric efficiency associated with the operation frequency, as defined by Pèrez-Segarra (2005). The second term, $\eta_{v,m}$, is the volumetric efficiency related to losses that occur during the suction process, P_m^{suc},, backflow, $P_m^{suc,r}$, and leakage through the piston/cylinder gap, P_m^{lkg}. Thus, the expression for this volumetric efficiency can be rewritten in a different manner:

$$\eta_{v,m} = 1 - P_m^{suc} - P_m^{suc,r} - P_m^{lkg} \tag{9}$$

According to the efficiency detachment procedure, an associated efficiency can be written by adding the losses due to sub-processes k. Hence, the associated efficiency for the suction process can be defined as:

$$\eta_{v,suc} = 1 - P_m^{suc} = \frac{m_{suc}}{\rho_1 \forall_{sw}^{c=0}} \tag{10}$$

During the gas path from the suction line to the compression chamber, different sources of irreversibility occur. The first one is due to gas superheating, which modifies its thermodynamic state from $\phi_l=\phi\ (p_{evap},T_{evap})$ to $\phi_{suc}=\phi\ (p_{evap},T_{suc})$ due to heat transfer between the gas and warmer surfaces inside the compressor, such as the discharge plenum or the crankcase. Other irreversibilities are related to heat transfer that takes place when the gas enters the compression chamber and gets into contact with the walls of both the suction port and the cylinder. The gas thermodynamic state then changes from ϕ_{suc} to $\phi_G=\phi\ (p_G,T_G)$, which is the instantaneous gas state in the compression chamber during the suction process. In order to quantify such effects, Eq. (10) is modified to read the following expression:

$$\eta_{v,suc} = \frac{m_{suc}}{\rho_1 \Delta\forall_{sw}^{c=0}} \frac{\Delta\forall_r}{\Delta\forall_r} \frac{\rho_{suc}}{\rho_{suc}} = \frac{\rho_{suc}}{\rho_1} \frac{m_{suc}}{\rho_{suc}\Delta\forall_r} \frac{\Delta\forall_r}{\Delta\forall_{sw}^{c=0}} = \eta_{v,suc}^{sc}\eta_{v,suc}^{cc}\eta_{v,v} \tag{11}$$

The first term on the rightmost side of Eq. (11) is the volumetric efficiency associated with the suction superheating, $\eta^{sc}{}_{v,suc}$. The second term is defined as the efficiency due to the in-cylinder superheating, $\eta^{cc}{}_{v,suc}$, and can be rewritten according to:

$$\eta_{v,suc}^{cc} = \frac{1}{\rho_{suc}} \frac{m_{suc}}{\Delta\forall_r} = \frac{\rho_{suc}^*}{\rho_{suc}} \tag{12}$$

The term $\Delta\forall_r$ considers the swept volume between the position in which the suction valve opens and the position in which the piston reaches the bottom dead center (BDC), as shown in Figure 5a. The relation between the mass that actually enters the cylinder and the volume $\Delta\forall_r$ provides an apparent gas density for the suction process, ρ_{suc}^*. This parameter conveniently represents the irreversibilities that occur when the gas enters the cylinder, due to the heat transfer between the gas and the cylinder walls, as well as the losses caused by the flow restriction in the suction valve.

The last term of Eq. (11) is the secondary volumetric efficiency $\eta_{v,v}$ and represents the volume displaced by the piston that is actually used to take in the gas. This effect can be better understood by observing the processes that occur during the expansion stage of the compression cycle. A suitable analysis of such irreversibility can be carried out by examining three generic expansion processes as shown in Figure 5b.

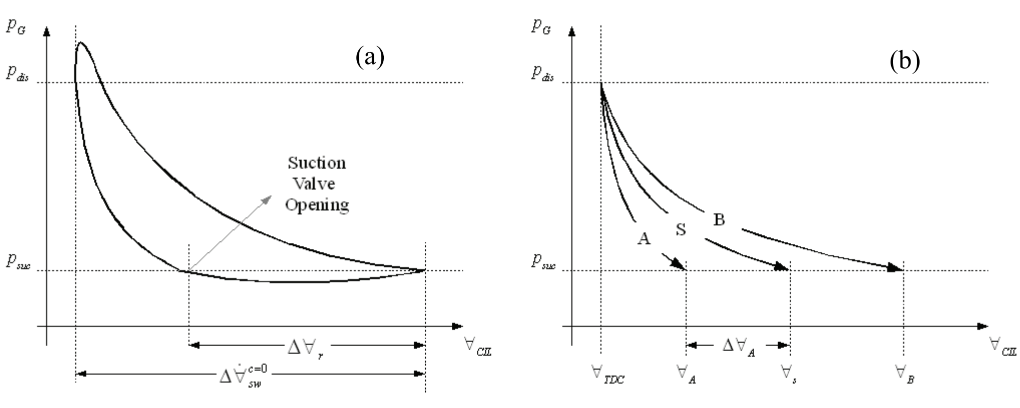

Figure 5. (a) p x V Diagram showing the total swept volume and the volume swept during suction.
(b) Three generic expansion processes.

Process *S* represents an isentropic expansion between pressures p_{dis} and p_{suc}. Initially, there is an amount of mass m_{TDC} in the clearance volume $\forall_{TDC}$ and the process ends when the pressure reaches p_{suc} at $\forall_s$. A second process *A* is at the same initial thermodynamic condition, however, some mass leaves the compression chamber during the expansion and the final volume at which the pressure reaches p_{suc}, $\forall_A$, is smaller than $\forall_s$. Finally, a third expansion can occur with heat being transferred to the gas, resulting in a final volume $\forall_B$ that can be expressed as:

$$\forall_B = \forall_s + \Delta\forall_B \tag{13}$$

The actual expansion process in a compression chamber occurs in the presence of the aforementioned effects and, therefore, can be thought as a combination of multiple processes. Therefore, although an isentropic expansion of $\forall_{TDC}$ would reach the specified pressure p_{suc} at $\forall_s$, heat and mass transfer during the expansion will increase (+) or decrease (-) the value of the final volume, as shown in Tab. 1.

As indicated by Eq. (2), the energy transport equation for the gas inside the compression chamber can be written in terms of its instantaneous temperature, T_G, which is affected by the aforementioned phenomena. Alternatively, the present formulation allows the analysis of each sub-process k (heat transfer, discharge backflow, etc.) in terms of its characteristic end thermodynamic state $\phi_G^k=\phi\,(v_G^k,T_G^k)$. Consequently, the energy and mass conservation equations can be used to represent the direct effect of each phenomenon in terms of a corresponding T_G. The generalized balance equations for mass and energy are given by:

$$\frac{dm_G^k}{dt} = C_2^k \tag{14}$$

$$\frac{dT_G^k}{dt} = \frac{1}{m_G^k c_{v,G}^k}\left\{C_1^k - T_G^k\left[\left.\frac{\partial p_G^k}{\partial T_G^k}\right|_v \left(\frac{\partial \forall_{CIL}}{\partial t}\right) - v_G^k C_2^k\right]\right\} \tag{15}$$

The constants C_1^k and C_2^k are expressed according to Tab 1. Firstly, Eqs. (14) and (15) are solved to determine the values of specific volume and temperature for each phenomenon. Then, an equation of state is used to find the corresponding pressure value. Once the pressure has reached the suction pressure, p_{suc}, the final volume $\forall_k$ for the sub-process k can be determined.

Table 1. Thermodynamic sub-processes present during the expansion process of the compression cycle.

Phenomenon	Effect	C_1^k	C_2^k
Cylinder wall heat transfer	**(+) / (-)**	$\dot{Q}_w$	0
Direct Flow through discharge after TDC	**(-)**	0	$-\dot{m}_{des}$
Discharge valve backflow	**(+)**	0	$+\dot{m}_{des,r}$
Piston/Cylinder gap leakage	**(-)**	$\dot{m}_{lkg}\left(h_{ie}-h_G\right)$	$-\dot{m}_{lkg}$

It is now possible to define the swept volume during the suction process as follows:

$$\Delta\forall_r = \Delta\forall_{sw}^{c=0} - \Delta\forall_s - \sum \Delta\forall_k - \Delta\forall_a \tag{16}$$

In Eq. (16), $\Delta\forall_s$, $\Delta\forall_k$ and $\Delta\forall_a$ represent, respectively, the swept volume connected to the isentropic expansion, the swept volume due to the k sub-processes and an additional volume due to delays in the valve opening. The latter term is a consequence of valve dynamics, which happens when the pressure difference between the suction chamber and the cylinder is not sufficient to open the valve, reducing the time available for the suction process.

By substituting Eq. (16) into the expression for $\eta_{v,v}$, one finds:

$$\eta_{v,v} = \frac{\Delta\forall_r}{\Delta\forall_{sw}^{c=0}} = 1 - \frac{\Delta\forall_s}{\Delta\forall_{sw}^{c=0}} - \sum\frac{\Delta\forall_k}{\Delta\forall_{sw}^{c=0}} - \frac{\Delta\forall_a}{\Delta\forall_{sw}^{c=0}} = 1 - P_{v,v}^{c=0} - \sum P_{v,v}^{irr} - P_{v,v}^{a} \tag{17}$$

The first source of volumetric loss represents the isentropic expansion of the gas left in the cylinder clearance volume. The second source is the sum of all irreversibilities that occur during the expansion process and, finally, the third source is related to a delay in the opening of the suction valve. From this example, it can be seen the present approach is very useful to discriminate the main sources of volumetric inefficiency, allowing a complete assessment of new compressor designs through parametric studies.

4. RESULTS

The method presented in this paper was applied to assess volumetric inefficiencies in a small reciprocating compressor. To this extent, an enhanced version of the simulation code developed by Ussyk (1984) was employed, which mathematically describes each compressor component. The code accounts for piston displacement as a function of crankshaft angle, the thermodynamic process inside the cylinder, mass flow rate through valves, valve dynamics, gas pulsation inside mufflers and refrigerant thermodynamic properties. Several parameters are calculated during the compressor cycle, such as the instantaneous pressure in different regions inside the compressor, mass flow rate, energy and mass losses, refrigerating capacity, etc. Thermodynamic properties for the refrigerant were evaluated through a program link to REFPROP 7.0 (Lemmon *et al.*, 2002). For the valve dynamics, a one-degree of freedom model was adopted. A steady-state, incompressible, fully developed flow was assumed for estimates of gas leakage through the piston/cylinder clearance (Lilie e Ferreira, 1984).

The R-600a compressor had a swept volume, $\Delta\forall_{sw}$, of 9.5×10^{-6} m^3 and a clearance volume fraction, $\forall_{TDC}/\Delta\forall_{sw}$, equal to 7.5%. The operating condition was represented by the evaporating and condensing temperatures, which were equal to -25ºC and 55ºC, respectively. Subcooling and superheating were kept constant at 32ºC. Under the above listed conditions, the temperature in the suction chamber reached 72 ºC.

The relative importance of each parameter that affects the refrigeration capacity can be assessed by examining Table 3. As can be seen, gas superheating and isentropic expansion are the two main aspects that reduce the refrigerating capacity. The cylinder clearance volume is responsible for about 60.7% of the total loss, followed by gas superheating between the suction line and the suction chamber, which corresponds to 22.8%. The amount of superheating that occurs when the gas enters the compression chamber contributes to an extra loss of 8.3%. The irreversibilities inside the compression chamber bring about only 2.1% of loss.

Table 3. Refrigeration capacity losses (W).

	Ideal Refrigeration Capacity	**257.6**
Superheating (suction line to suction chamber)	$= \Delta\forall_{sw}^{c=0}\left(\rho_{suc} - \rho_1\right) f \Delta h_{evap}$	-33.8
In-cylinder Superheating	$= \Delta\forall_{sw}^{c=0}\left(\rho_{suc}^{*} - \rho_{suc}\right) f \Delta h_{evap}$	-12.2
Backflow in the suction valve	$= \dot{m}_{suc,r}\Delta h_{evap}$	-0.1
Leakage through the piston/cylinder gap	$= \dot{m}_{lkg}\Delta h_{evap}$	-4.0
Isentropic expansion of cylinder clearance volume	$= P_{v,v}^{c=0}\left(\Delta\forall_{sw}^{c=0}\rho_{suc}^{*} f \Delta h_{evap}\right)$	-89.7
Backflow in the discharge valve	$= P_{v,v}^{dis,r}\left(\Delta\forall_{sw}^{c=0}\rho_{suc}^{*} f \Delta h_{evap}\right)$	*-4.6*
Heat transfer at the cylinder walls	$= P_{v,v}^{qw}\left(\Delta\forall_{sw}^{c=0}\rho_{suc}^{*} f \Delta h_{evap}\right)$	*0.0*
Leakage through the piston/cylinder gap	$= P_{v,v}^{lkg}\left(\Delta\forall_{sw}^{c=0}\rho_{suc}^{*} f \Delta h_{evap}\right)$	*+1.4*
Inertial Direct Discharge flow	$= P_{v,v}^{dis}\left(\Delta\forall_{sw}^{c=0}\rho_{suc}^{*} f \Delta h_{evap}\right)$	*0.0*
Opening delay of the suction valve	$= P_{v,v}^{a}\left(\Delta\forall_{sw}^{c=0}\rho_{suc}^{*} f \Delta h_{evap}\right)$	-4.8
	Actual Refrigeration Capacity	**109.8**

5. CONCLUSIONS

The volumetric efficiency is an important parameter commonly adopted to quantify the performance of compressors. In this paper, the detachment approach presented by Pèrez-Segarra (2005) was extended to further divide the volumetric inefficiencies due to sub-processes taking place inside the compression chamber. The approach developed here is more appropriate for numerical simulation, since data like the apparent suction density, are very difficult to be experimentally assessed. For the small reciprocating compressor analyzed in this study, the cylinder clearance volume is the most important source of volumetric inefficiency, representing approximately 60% of the total reduction in the mass flow rate. On the other hand, irreversibilities during the expansion process are seen to be negligible.

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ACKNOWLEDGEMENTS

This study is part of a technical-scientific program between the Federal University of Santa Catarina and EMBRACO, partially funded by CNPq (Brazilian Research Council) through Grant No. 573581/2008-8 (National Institute of Science and Technology in Refrigeration and Thermophysics). Support from FINEP (Federal Agency of Research and Projects Financing) and CAPES (Coordination for the Improvement of High Level Personnel) are also acknowledged.