A SYSTEMATIC APPROACH TO ANALYZE VOLUMETRIC INEFFICIENCIES IN REFRIGERATION COMPRESSORS

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A very important parameter adopted to characterize the performance of compressors is the volumetric efficiency, which is the ratio between the actual mass of gas pumped as compared to the theoretical maximum. A number of effects act to reduce the mass flow rate in a compressor and this paper presents an approach to quantify the influence of each one of them, based on the efficiency detachment procedure. To this extent, a small 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 inefficiencies 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.

***Keywords**: refrigeration, compressor, volumetric efficiency, efficiency detachment.*

1. INTRODUCTION

A standard refrigeration system is basically composed by 5 components: evaporator, condenser, expansion device, compressor and fluid refrigerant. Heat that should be extracted from the cold environment is received in the evaporator by the fluid refrigerant at lower pressure and then is rejected to the hotter environment at higher pressure. The compressor is the equipment responsible for setting the pressure difference between the heat exchangers. Moreover, together with the expansion device, the compressor also establishes the refrigerant mass flow rate, and therefore, the system refrigeration capacity.

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 mainly influenced by the evaporation pressure as well as the suction temperature. The gas temperature is increased after the entrance of the compressor since the flow enters in contact with hot components before reaching the compression chamber. However, the effect of superheating is not the only effect that reduces the overall mass flow rate of the compressor. Other sources of inefficiency that take place are the clearance volume in the compression chamber, backflow in valves and internal heat exchange between gas and the cylinder walls.

The volumetric efficiency is a parameter adopted to express the mass flow rate reduction, by relating the actual and ideal mass flow rates of a given compressor rate (Gosney, 1983):

$$\eta_v = \frac{\dot{m}_{real}}{\dot{m}_{ideal}} \tag{1}$$

Nevertheless, the volumetric efficiency as written in Eq. (1) is not adequate to quantify the individual contributions of each one of thermodynamic irreversibilities, which is useful during the design and analysis of a given compressor. The present paper aims to apply an efficiency detachment procedure as proposed by Pèrez-Segarra (2005) in order to determine the impact of different phenomena in the volumetric efficiency. Initially, a brief overview of the theoretical formulation of the compression chamber is given. Then the efficiency detachment procedure is applied to the analysis of a small reciprocating compressor.

2. THEORETICAL MODELING OF THE COMPRESSION CHAMBER

The compression chamber of a reciprocating compressor comprehends the volume formed by the piston, cylinder walls and the valve plate (Fig. 1). The piston moves alternatively along the cylinder axis, from the bottom dead center to the top dead center close to the valve plate. A crank-shaft mechanism is used to convert the rotational movement of the electric motor into the alternative axial movement of the piston. The volume of the compression chamber can be described according to some geometrical parameters (Ussyk, 1984):

$$\forall_{CIL}(t) = \pi R_{CIL}^2 \left\{ C_{PMS} - \left[-e\cos(2\pi f_n t) + \left(C_b^2 - \left(e\sin(2\pi f t) - d_m^2 \right)^2 \right)^{1/2} \right] \right\} + \forall_{TDC} \tag{1}$$

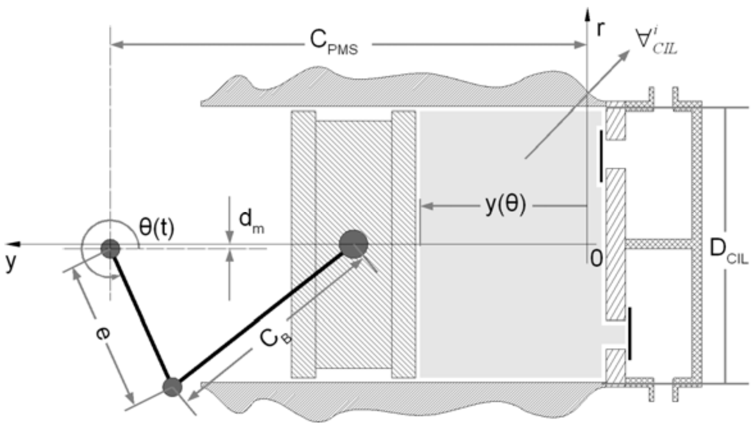


Figure 1. Schematic view of the compression chamber.

During the compression cycle, mass enters or leaves the cylinder through the suction and discharge valves. Reciprocating compressors usually adopt reed type valves made of steel that open and close 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 Fig. 2. Initially, the piston is in a given position *A* and the mass of the 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 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 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 the inertia of the flow or may exit as a 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 *B*-*C*, the suction valve is closed and the vapor trapped in the cylinder has its pressure raised as the cylinder volume decreases. Eventually, during *C*-*t*, the pressure in the cylinder is such that discharge valve is forced to open. Mass then enters the cylinder until the top dead center *t*. From there on, the piston changes again its movement direction, reducing the volume. During the interval *t*-*D*, the discharge valve closes, and in the same manner as in 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 $m_{disr,}$ represents the mass associated with backflow.

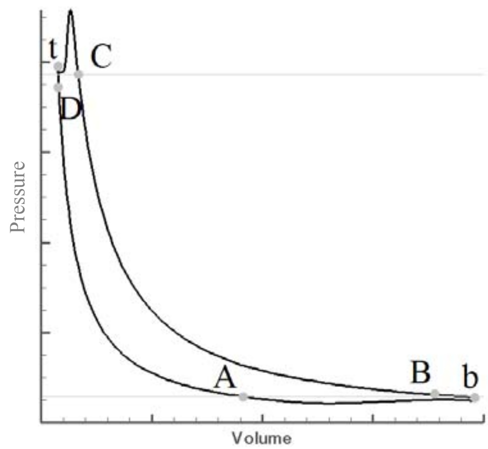


Figure 2. Pressure x Volume diagram

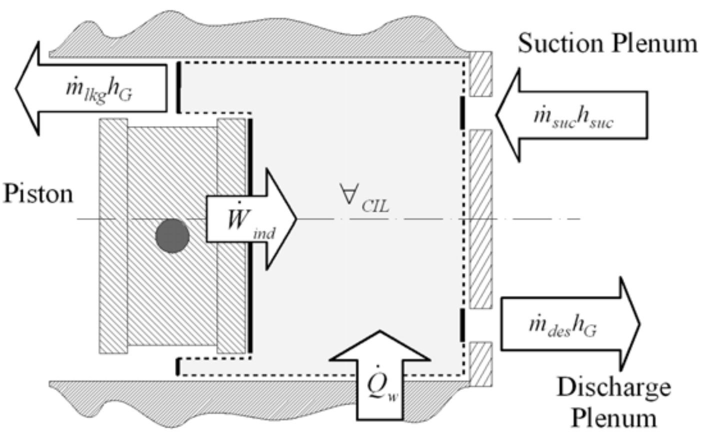


Figure 3. Control volume inside the compression chamber.

In addition to the flow through valves, during the entire compression cycle mass enters and leaves the compression chamber through the gap between the piston and the cylinder walls. The net value of mass that leaves the compression chamber through such 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:

$$\left(M_A^{t+\Delta t} - M_A^t\right) + \left(m_{suc} - m_{suc,r} - m_{dis} + m_{dis,r} - m_{lkg}\right) = 0 \tag{3}$$

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 and, therefore:

$$m_{suc} - m_{suc,r} - m_{dis} + m_{dis,r} - m_{lkg} = 0 \tag{4}$$

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

$$\frac{dT_G}{dt} = A_T - B_T T_G \tag{5}$$

where

$$A_T = \frac{1}{m_G c_{v,G}}\left(H_w A_w T_w - h_G \frac{dm_G}{dt} - \sum \dot{m} h\right) \tag{6}$$

$$B_T = \frac{1}{m_G c_{v,G}}\left(H_w A_w + \left.\frac{\partial p_G}{\partial T_G}\right|_v \frac{d\forall_{CIL}}{dt} - \left.\frac{\partial p_G}{\partial T_G}\right|_v v_G \frac{dm_G}{dt}\right) \tag{7}$$

In the relations above, 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.

By using an equation of state for the gas (Eq. 8) and the energy transport equation, all thermodynamic states the gas can undergoes during the compression cycle can be described.

$$p_G = p\left(T_G, v_G\right) \tag{8}$$

3. VOLUMETRIC EFFICIENCY

As shown in Eq. (1), to quantify the volumetric efficiency of a refrigeration compressor, it is necessary to define 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 inlet of the compressor. Additionally, an ideal compressor also 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:

$$\dot{m}_{id} = \rho_1 \forall_{sw}^{c=0} f_n \tag{9}$$

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

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

$$\eta_v = \frac{f_r}{f_n} \frac{\forall_{sw}}{\forall_{sw}^{c=0}} \frac{m_{evap}}{\rho_1 \forall_{sw}} \tag{10}$$

The first term on the right side of Eq. (10) represents the effects of electric motor slippage, which reduces the compressor nominal frequency of operation. The second term, called theoretical volumetric efficiency, considers the effect of the clearance volume, which reduces the total swept volume of the compressor and gives rise to a reexpansion of the remaining gas in the cylinder after the discharge process. The third term is defined as a secondary volumetric efficiency, which takes into account all other irreversible phenomena that occurs inside the compression chamber. Pèrez-Segarra *et al.* (2005) break the compression cycle into 4 stages (suction, compression, discharge and expansion) and define a volumetric efficiency for each one of them through the efficiency detachment methodology. This procedure may cause some misleading conclusions in the analysis, since there are several processes in each stage. The present paper follows a different approach, by distinguishing all physical phenomena that takes place inside the compression chamber. The first step is to define the mass that flows through the evaporator during one cycle of compression, m_{evap}. Based on the control volume shown in Fig. 4, m_{evap} can be evaluated from:

$$m_{evap} = m_{suc} - m_{suc,r} - \left(m_{lkg} + m_{lkg,r}\right) = m_{suc} - m_{suc,r} - m_{lkg} \tag{11}$$

The substitution of this expression into Eq. (10) results:

$$\eta_v = \frac{f_r}{f_n} \frac{m_{evap}}{\rho_1 \forall_{sw}^{c=0}} = \frac{f_r}{f_n}\left(1 - \frac{\rho_1 \forall_{sw}^{c=0} - m_{suc}}{\rho_1 \forall_{sw}^{c=0}} - \frac{m_{suc,r}}{\rho_1 \forall_{sw}^{c=0}} - \frac{m_{lkg}}{\rho_1 \forall_{sw}^{c=0}}\right) = \eta_{v,f}\eta_{v,m} \tag{12}$$

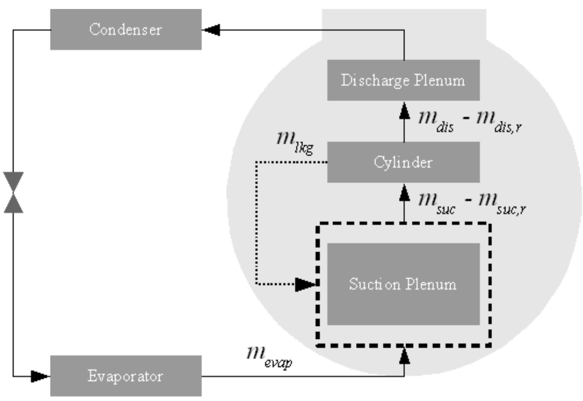


Figure 4. Control volume outside the suction plenum.

The first term, $\eta_{v,f}$, is the volumetric efficiency associated with the operation frequency, as defined by Pèrez-Segarra (2005). The second term, $\eta_{v,m}$, is a associated mass volumetric efficiency and takes into account the losses that occur during the suction process, due to the total mass leakage through the piston/cylinder gap and due to the backflow during suction. Thus, the expression for the mass associated volumetric efficiency can be rewritten in a different manner:

$$\eta_{v,m} = 1 - P_m^{suc} - P_m^{suc,r} - P_m^{lkg} \tag{13}$$

According to the efficiency detachment procedure, an associated efficiency can be written as a function of the summation of losses due to n sub-processes k. Hence, the associated efficiency for the suction process can be defined as:

$$\eta_{v,suc} = 1 - P_m^{suc} = \frac{m_{suc}}{\rho_1 \forall_{sw}^{c=0}} \tag{14}$$

During the gas path from the suction line to the compression chamber, different sources of irreversibility occur. The first one is the superheating, which modifies the 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 hot components of the compressor, such as the discharge plenum or the crankcase. Other irreversibilities are associated with heat transfer that takes place when the gas enters the compression chamber and gets in contact with hot walls in the suction port and the cylinder itself. The thermodynamic state of the gas changes this time from ϕ_{suc} to $\phi_G = \phi\,(p_G, T_G)$, which is the instantaneous state of the gas in the compression chamber during the suction process. In order to quantify such effects, Eq. (14) is modified to the following expression:

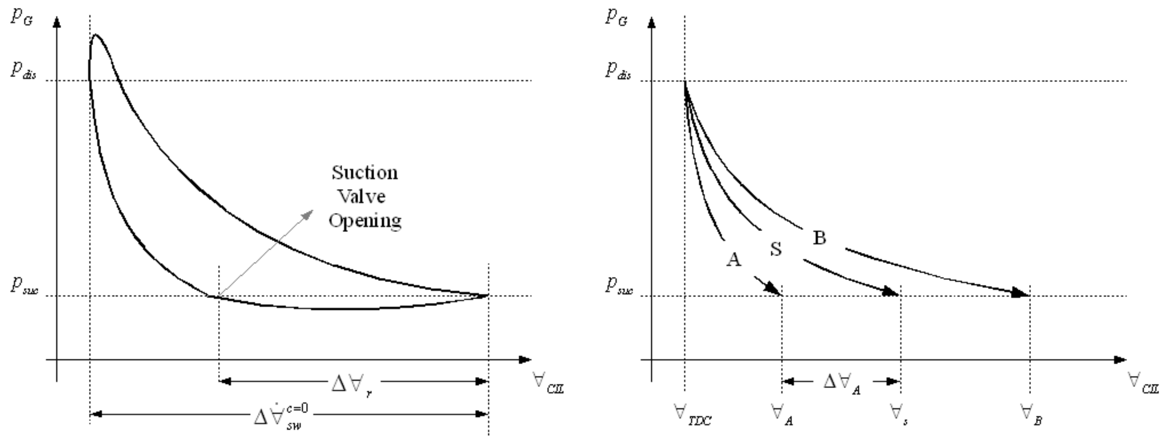
$$\eta_{v,suc} = \frac{m_{suc}}{\rho_1 \Delta\forall_{sw}^{c=0}} \frac{\Delta\forall_r}{\Delta\forall_r} \frac{\rho_{suc}}{\rho_{suc}} = \frac{\rho_{suc}}{\rho_1} \frac{m_{suc}}{\rho_{suc} \Delta\forall_r} \frac{\Delta\forall_r}{\Delta\forall_{sw}^{c=0}} = \eta_{v,suc}^{sc} \eta_{v,suc}^{cc} \eta_{v,v} \tag{15}$$

The first term on the right hand side of Eq. (15) 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:

$$\eta_{v,suc}^{cc} = \frac{1}{\rho_{suc}} \frac{m_{suc}}{\Delta\forall_r} = \frac{\rho_{suc}^{*}}{\rho_{suc}} \tag{16}$$

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 Fig. 5a. The ratio between the mass that actually enters the cylinder and the mass that could enter if the entire compression chamber volume were swept provides an apparent gas density for the suction process. 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. (15) is the secondary volumetric efficiency $\eta^{sc}_{v,v}$ and represents the volume displaced by the piston that is actually used to suction the gas. The reduction of volumetric efficiency is directly caused by in-cylinder thermodynamic phenomena during the expansion process. A detailed analysis of these irreversibilities can be carried out by examining three generic expansion processes as shown in Fig. 5b.



(a) (b)

Figure 5. (a) p-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 a quantity of mass m_{TDC} in a confined volume $\forall_{TDC}$ and the process ends when the pressure reaches p_{suc} at $\forall_s$. A second process A is at same initial thermodynamic condition, however a given quantity of mass leaves the compression chamber during the expansion and thus, the final volume in which the pressure reaches p_{suc}, $\forall_A$, is smaller then $\forall_s$. 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:

$$\forall_B = \forall_s + \Delta\forall_B \tag{17}$$

The actual expansion process in a compression chamber occurs in the presence of the aforementioned effects and, therefore, can be thought as the summation of multiple processes. In other words, any isentropic expansion of the same gas mass that initially was at $\forall_{TDC}$ would end at $\forall_s$ for a specified pressure p_{suc}. However, heat or mass transfer during the expansion will increase (+) or decrease (-) the final volume of the expansion process, as shown in Tab. 1.

As indicated in Eqs. (5)-(7), the energy transport equation for the gas inside the compression chamber can be written in terms of the gas instantaneous temperature, T_G, which is affected by aforementioned phenomena. However, each sub-process k (heat transfer, discharge backflow, etc.) can be defined by its characteristic thermodynamic state $\phi_G^k=\phi\,(v_G^k,T_G^k)$. A simplification of the energy and mass conservation equations allows one to represent the direct effect of each phenomenon in terms of a corresponding T_G. The general transport equations for mass and energy are, respectively:

$$\frac{dm_G^k}{dt} = C_2^k \tag{18}$$

$$\frac{dT_G^k}{dt} = \frac{1}{m_G^k c_{v,G}^k}\left\{C_1^k - T_G^k\left[\left.\frac{\partial p_G^k}{\partial T_G^k}\right|_v\left(\frac{\partial\forall_{CIL}}{\partial t}\right) - v_G^k C_2^k\right]\right\} \tag{19}$$

The constants C_1^k and C_2^k are expressed according to Tab 1. Firstly, Eqs. (18) and (19) are solved to determine the values of specific volume and temperature for each phenomenon. Then, the 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 is reached.

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

Phenomenon	Effect	C_1^k	C_2^k
Cylinder walls 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	**(-)**	$\dot{m}_{lkg}\left(h_{ie}-h_G\right)$	$-\dot{m}_{lkg}$

Now it is possible to define the swept volume during the suction process as:

$$\Delta\forall_r = \Delta\forall_{sw}^{c=0} - \Delta\forall_s - \sum\Delta\forall_k - \Delta\forall_a \tag{20}$$

where, $\Delta\forall_s$, $\Delta\forall_k$ and $\Delta\forall_a$ represent, respectively, the swept volume of the isentropic expansion, the swept volume of the k sub-processes and an additional volume due to delays in the valve opening. This last term is a consequence of valve dynamics, 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. (20) into the expression for $\eta_{v,v}$, one finds:

$$\eta_{v,v} = \frac{\Delta\forall_r}{\Delta\forall_{sw}^{c=0}} = 1 - \frac{\Delta\forall_s}{\Delta\forall_{sw}^{c=0}} - \sum\frac{\Delta\forall_k}{\Delta\forall_{sw}^{c=0}} - \frac{\Delta\forall_a}{\Delta\forall_{sw}^{c=0}} = 1 - P_{v,v}^{c=0} - \sum P_{v,v}^{irr} - P_{v,v}^{a} \tag{21}$$

In Eq. (21), the first source of volumetric loss represents the effect of an isentropic expansion of the gas left in the cylinder clearance volume. The second source takes into account all irreversibilities that occur during the expansion process and, finally, the third reduction is related to a delay in the suction valve opening.

The analysis of such non-dimensional parameters is very useful to identify the main sources of reduction of the compressor volumetric efficiency, allowing a complete assessment of impacts originated by new designs through parametric studies.

4. APPLICATION OF NON-DIMENSIONAL PARAMETERS FOR THE ANALYSIS OF A SMALL RECIPROCATING COMPRESSOR

Each phenomenon during the expansion process is adequately identified by an associated efficiency, but can also be represented in terms of a dimensional loss of refrigeration capacity. The refrigeration capacity, $\dot{Q}_{evap}$ is defined as:

$$\dot{Q}_{evap} = \dot{m}_{evap}\left(h_3 - h_1\right) \tag{22}$$

where h_3 is the specific enthalpy at the end of the condenser. The system operating conditions are defined in Table 2.

Table 2. Refrigeration system and compressor operation condition.

Gas refrigerant	R600a
Operation frequency	60Hz
Evaporation temperature (T_{evap})	-25ºC
Condensation temperature (T_{cond})	55ºC
Subcooling temperature (T_3)	32.2ºC
Superheating temperature (T_1)	32.2ºC
Suction temperature (T_{suc})	71.9 ºC
Piston swept volume ($\Delta\forall_{sw}$)	9.48×10^{-6}m^3
Cylinder clearance volume ($\forall_{TDC}$)	7.48×10^{-7}m^3

The methodology adopted here to simulate the compressor is an enhanced version of that developed by Ussyk (1984), mathematically describing each one of the compressor components. 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. Thermodynamic properties for the refrigerant were evaluated through a program link to the REFPROP 7.0 (Lemmon *et al.*, 2002). For the valve dynamics, it was implemented a one-degree of freedom model, as described in Deschamps *et al.* (2002). Gas leakage through the clearance between piston and cylinder was as a permanent, viscous and incompressible (Lilie e Ferreira, 1984).

Table 3 presents numerical results for the reciprocating compressor characterized in Table 2. It is possible to notice that superheating and the gas reexpansion after discharge are the two main aspects that reduce the system refrigeration capacity. The cylinder clearance volume is responsible for about 60.7% of the total loss, followed by the 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%.

Table 3. Refrigeration capacity losses (W).

Ideal Refrigeration Capacity	**257,6**
Total Loss	**-147,8**
Superheating (suction line to suction chamber)	-33,8
In-cylinder Superheating	-12,2
Backflow in the suction valve	-0,1
Leakage through the piston/cylinder gap	-4,0
Isentropic expansion of cylinder clearance volume	-89,7
Summation of irreversibilities during expansion	-3,2
Backflow in the discharge valve	*-4,6*
Heat transfer at the cylinder walls	*-0,0*
Leakage through the piston/cylinder gap	*+1,4*
Inertial Direct Discharge flow	*0,0*
Opening delay of the suction valve	-4,8
Actual Refrigeration Capacity	**109,8**

The irreversibilities inside the compression chamber bring about only 2.1% of loss. It should be noticed that gas leakage during the expansion process may offer a benefit to the refrigeration capacity, since it can anticipate the opening of the suction valve. On the hand, the overall effect of mass leakage is negative because gas is lost during the entire compression cycle.

5. CONCLUSIONS

The volumetric efficiency is an important non-dimensional parameter commonly adopted to characterize compressors. In this paper, a detachment procedure was presented to break up the reduction in the volumetric efficiency according with sub-processes that take place along the gas path from the suction line to the compression chamber. This methodology is more appropriate for numerical simulation of compressors, since some data, such as the apparent suction density, are very difficult, or even impossible, to be experimentally assessed.

For the compressor analyzed in this study, the cylinder clearance volume is the most important source of reduction in the volumetric efficiency, representing approximately 60% of the total reduction in the mass flow rate. In this respect, the irreversibilities during the expansion process are virtually of no importance when compared to the aforementioned effect.

6. ACKNOWLEDGEMENTS

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