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THERMAL AND FLUID DYNAMIC CHARACTERIZATION OF HERMETIC RECIPROCATING COMPRESSORS.

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ABSTRACT

A review of the different parameters which characterize the thermal and fluid dynamic behaviour of hermetic reciprocating compressors is presented. Some new parameters are introduced to get a better understanding of the complex physical processes involved in this kind of compressors. All the parameters have been tested on the basis of the numerical solutions obtained by means of an advanced simulation code developed by the authors and considering a wide range of working conditions, refrigerant fluids and compressor capacities. Attention is focused on predicting the performance of hermetic reciprocating compressors under different working conditions and using the above mentioned non-dimensional parameters.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>A</td>
<td>heat transfer area ($m^2$)</td>
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<tr>
<td>COP</td>
<td>Coefficient of Performance</td>
</tr>
<tr>
<td>$c$</td>
<td>clearance ratio</td>
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<td>specific heat capacity ($J kg^{-1} K^{-1}$)</td>
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<td>friction factor</td>
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<td>$f_m$</td>
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<td>$\rho$</td>
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1 INTRODUCTION

Thermal and fluid dynamic behaviour of hermetic reciprocating compressors are characterized by complex heat transfer and fluid flow phenomena: three-dimensional, turbulent, compressible flow, fast transient processes (pulsatory), complex geometries, moving surfaces, etc. At the beginning of the 1970’s, the energy crisis obliges to improve efficiencies. The Montreal 1987 and Kyoto 1997 Protocols [1,2] establish the use of non-contaminant refrigerants. The challenge of new refrigerants, the need for high efficiencies and the necessity of noise reductions are strong incentives to develop general and accurate prediction methodologies. Thus, the necessity to optimize, improve and develop better compressors becomes stronger than ever.

The idea of this paper is to review and present different physically meaningful parameters that characterize the reciprocating compressor behaviour, their influence detachment and evolution under different working conditions.

All the test cases presented have been numerically obtained using an advanced numerical simulation model of the thermal and fluid dynamic behaviour of hermetic reciprocating compressors [3-5].
Different authors present several parameters to define compressor behaviour. Villadsen and Boldvig [6] base compressor efficiency on volumetric and isentropic efficiencies and COP. They study experimentally the influence on compressor ratio, speed, and length of stroke for different refrigerant fluids. Scalabrin and Bianco [7] work on experimental thermodynamic analysis of the mechanical and thermal performance of compressors as a function of compressor speed. They study torque, mechanical power, volumetric efficiency and COP considering a wide range of working conditions. MacGovern and Harte [8] present an overall exergy analysis for compressor performance, which take into account exergy destruction rates, using a global numerical thermodynamic model. Tassou and Quershi [9] analyse experimentally the performance of compressors for different fluids on the basis of volumetric efficiency, isentropic efficiency, and COP. Stoufs et al. [10] present a detailed group of physical parameters for compressor performance and their prediction (volumetric effectiveness, indicated efficiency, reversible heat transfer, etc.) on the basis of a global thermodynamic model considering polytropic coefficients to evaluate compression and expansion processes.

In this study, the most important compressor parameters to define compressor performance are considered: volumetric efficiency, isentropic efficiency, heat transfer efficiency, and mechanical and electrical motor efficiency. Some of these parameters have been split to get a better understanding of the physical phenomena. Other parameters that help to compressor behaviour analysis are: pressure pulsation due to noise reduction and exergy analysis due to available energy improvements and the coefficient of performance COP, which characterizes compressor influence under the whole refrigerating system.

2 IDEAL THERMODYNAMIC COMPRESSOR BEHAVIOUR

The majority of the non-dimensional compressor parameters which define thermal and fluid dynamic compressor behaviour are related with different values corresponding with an hypothetical and ideal thermodynamic compressor model.

The main hypothesis of this ideal model are as follows: steady state cycle (periodical conditions), negligible kinetic and potential variations, isentropic process and ideal gas. The compressor scheme is based on three zones: one suction plenum chamber, one compression chamber and one discharge plenum chamber. Thus, inlet compressor conditions are directly inlet suction plenum conditions.

Both pressure-volume and temperature-entropy evolution in the compression chamber are shown in Fig. 1. The ideal thermodynamic operating cycle presents four processes: i) isentropic compression process of the gas contained in \( V_a \) at the inlet temperature and pressure conditions to outlet temperature and pressure conditions in \( V_b \); ii) isobaric and isothermal discharge of the gas contained in \( V_b \) minus the remaining gas in \( V_c \); iii) isentropic expansion process of the gas contained in \( V_c \) at the outlet temperature and pressure conditions to inlet temperature and pressure conditions in \( V_d \); iv) isobaric and isothermal suction of gas from \( V_d \) to \( V_a \).

![Fig. 1: pV and Ts diagram inside compressor chamber volume (ideal and real processes).](image-url)
With all these assumptions the minimum input parameters required to define the compressor behaviour are: displacement volume, dead volume, inlet gas pressure, outlet gas pressure, nominal frequency, inlet gas temperature and thermodynamic fluid properties (suction gas density and isentropic coefficient).

Based on the input data explained, and considering the hypothesis above mentioned, ideal mass flow rate, ideal outlet temperature and the ideal or isentropic compression work can be obtained as follows, where $\Pi$ is the compression ratio between discharge and suction pressure, and $V_{\text{max}}$ is the maximum cylinder capacity:

$$\dot{m}_{\text{ideal}} = \rho_{\text{in}} V_{cl} f_n \left[1 + c \left(1 - \Pi \right)\right]$$  \hspace{1cm} (1)

$$T_{\text{out}} = T_{\text{in}} \left[1 + \Pi^{(\gamma - 1)/\gamma} \right]$$  \hspace{1cm} (2)

$$\dot{W}_{\text{ideal}} = \dot{W}_{\text{ISO}} = \rho_{\text{in}} V_{\text{max}} \left[1 - \frac{c}{1 + c} \Pi^{(1/\gamma)} \right] \bar{f}_n = \frac{\dot{m}_{\text{ideal}} p_{\text{in}}}{\rho_{\text{in}}} \frac{\gamma}{\gamma - 1} \left[\Pi^{(\gamma - 1)/\gamma} - 1\right]$$  \hspace{1cm} (3)

The ideal or isentropic work can be split into the four parts which compose the compression cycle i.e. compression (ab), discharge (bc), expansion (cd) and suction (da) as can be seen in Fig. 1a. Then, $\dot{W}_{\text{ideal}} = \dot{W}_{\text{ab}} + \dot{W}_{\text{bc}} + \dot{W}_{\text{cd}} + \dot{W}_{\text{da}}$, where:

$$\dot{W}_{\text{ab}} = \frac{p_{\text{in}} V_{\text{max}}}{1 - \gamma} \left[\Pi^{(\gamma - 1)/\gamma} - 1\right] \hspace{1cm} \dot{W}_{\text{bc}} = -p_{\text{in}} V_{\text{max}} \left[\frac{c}{1 + c} \Pi - \Pi^{(\gamma - 1)/\gamma} \right]$$  \hspace{1cm} (4)

$$\dot{W}_{\text{cd}} = \frac{p_{\text{in}} V_{\text{max}}}{1 - \gamma} \frac{c}{1 + c} \left[\Pi^{(1/\gamma)} - \Pi \right] \hspace{1cm} \dot{W}_{\text{da}} = -p_{\text{in}} V_{\text{max}} \left[1 - \frac{c}{1 + c} \Pi^{(1/\gamma)} \right]$$  \hspace{1cm} (5)

### 3 COMPRESSOR BEHAVIOUR PARAMETERS

The reciprocating compressor behaviour can be described by means of a compressor parameter set. A selection of a minimum of non-dimensional parameters needed to describe the compressor performance has been carried out. Volumetric, isentropic, heat transfer, mechanical and electrical motor efficiencies define all thermal and fluid dynamic behaviour of compressors. COP facilitates the interaction between compressor and the rest of refrigerating equipment. The rest of the parameters, pressure pulsation and exergy balances help to improve compressor behaviour in noise reduction, maximum available efficiencies, etc.

#### 3.1 Volumetric efficiency

One of the most important parameters of the global behaviour of hermetic reciprocating compressors is the volumetric efficiency, which is defined as the relation between the actual mean mass flow rate and the maximum one: $\eta_v = \dot{m} / (\rho_{\text{in}} V_{cl} f_n)$, where $\rho_{\text{in}}$ refers to the gas density entering into the shell.

In order to have a better understanding of this parameter, a quick look on the compression cycle shown in Fig. 1 is useful. Firstly, after the suction valve is closed at point 1, the piston compresses the gas inside the compression chamber until the discharge valve begins to open at point 2. The gas is discharged from point 2 to 3, where the discharge valve closes. Discharge valve closes slightly after piston reached the TDC at point 3. Then the gas left inside the cylinder is expanded until its pressure is low enough to open the suction valve at point 4. Finally, a new fresh charge enters the compression chamber from point 4 to 1’ slightly after the BDC is reached of point 1. Valves are opened or closed depending on different aspects such as: gas conditions inside the compression chamber, pressures in suction and discharge manifolds ( $p_s$ and $p_d$ respectively), pressure drop through the valves, etc.

From the definition of the volumetric efficiency, and taking into account the different processes described above, Escanes [11] proposed a decomposition of this parameter into different parts using the following equations (see Fig. 1a):

$$\eta_v = \frac{\dot{m}}{\rho_{\text{in}} V_{cl} f_n} = \frac{m_{13}}{\rho_{\text{in}} V_{cl} f_n} \bar{f} \left[1 + \frac{m_{1} - m_{3}}{m_{13}}\right]$$  \hspace{1cm} (6)
In this expression, the mass of the gas contained in the compressor chamber in a process from point 'i' to point 'j' is represented by \( m_{ij} \), which is the difference of the instantaneous mass inside the cylinder at these points, i.e., \( m_{ij} = m_j - m_i \). For convenience, equation (6) is approximately defined as:

\[
\eta_c \approx \frac{m_{\text{max}}}{\rho_{\text{in}}V_{cl}} \cdot \frac{m_{13}}{m_{\text{max}}} \cdot \frac{f}{f_n} \cdot \left[ 1 + \frac{m_{13}}{m_{13}} \right] \cdot \left[ 1 - \frac{m_{33}}{m_{13}} \right] = \eta_{id} \cdot \eta_{\text{irr}} \cdot \eta_{\omega} \cdot \eta_{\text{suc}} \cdot \eta_{\text{dis}} \tag{7}
\]

The first volumetric efficiency \( \eta_{id} \) corresponds to an ideal compression process; this efficiency only takes into account the gas expansion in the clearance volume, and can be obtained from eq. (1). The second efficiency term, \( \eta_{\text{irr}} \), takes into account the irreversibilities due to friction, heat transfer, leakages, etc. The third term, \( \eta_{\omega} \), is a consequence of the difference between the nominal frequency and the real one, which is basically due to mechanical and electrical motor losses. The delay in closing the suction valve, after the BDC is reached, is considered in the fourth term, \( \eta_{\text{suc}} \), which is usually greater than 1 (super-charging process). The last term, \( \eta_{\text{dis}} \), represents the delay in closing the discharge valve after the TDC is reached.

3.2 Isentropic Efficiency

The second meaningful non-dimensional parameter that defines the thermodynamic behaviour of compressors is the isentropic efficiency, which is defined as the relation between the isentropic compression work, eq. (3), and the real compression work: \( \eta_{\text{iso}} = \dot{W}_{\text{iso}}/\dot{w}_{cp} \), or in terms of the specific work, \( \eta_{\text{iso}} = w_{\text{iso}}/w_{cp} \). Both works can be split into the different compression steps (compression, discharge, expansion and suction). It is interesting to remark the differences due to real and ideal compression work at each of the four compressor processes inside compression chamber which are able to be evaluated as:

\[
\eta_{\text{iso}} = w_{\text{iso}}/w_{cp} = \frac{w_{ab} + w_{bc} + w_{cd} + w_{da}}{w_{12} + w_{23} + w_{34} + w_{41}'}
\]

Or using algebra:

\[
\eta_{\text{iso}} = 1 + \frac{w_{ab} - w_{12}}{w_{cp}} + \frac{w_{bc} - w_{23}}{w_{cp}} + \frac{w_{cd} - w_{34}}{w_{cp}} + \frac{w_{da} - w_{41}'}{w_{cp}} \tag{9}
\]

Defining the different contributions (compression, discharge, expansion and suction) to the isentropic efficiency as: \( \eta_{\text{sec}} = e^{(w_{ab} - w_{12})/w_{cp}}, \eta_{wd} = e^{(w_{bc} - w_{23})/w_{cp}}, \eta_{we} = e^{(w_{cd} - w_{34})/w_{cp}} \) and \( \eta_{\text{us}} = e^{(w_{da} - w_{41}')/w_{cp}} \), the following final expression is obtained:

\[
\eta_{\text{iso}} = 1 + \ln (\eta_{\text{sec}} \cdot \eta_{wd} \cdot \eta_{we} \cdot \eta_{\text{us}}) \tag{10}
\]

3.3 Heat Transfer Efficiency

A non-dimensional parameter that defines the behaviour of compressors is the heat transfer efficiency, which is defined as the relation between the total energy transfer to the fluid (compression work + heat transfer gains) and the real compressor work:

\[
\eta_q = (w_{cp} + q_{\text{fluid}})/w_{cp} \tag{11}
\]

The heat transferred gains \( q_{\text{fluid}} \) are defined as the heat exchanged during all compressor process, i.e. from the moment the fluid enters through the shell until the fluid goes out along the discharge tube. This heat transfer process can be detached into three compressor zones: suction line \( (q_{sl}) \), compression chamber \( (q_{cc}) \) and discharge line \( (q_{dl}) \). Evaluating heat transfer efficiency in a similar manner as the isentropic efficiency, eq. (10), the following heat transfer efficiencies for the three zones (suction, compression and discharge) can be defined as: \( \eta_{qsl} = q_{sl}/w_{cp}; \eta_{qcc} = q_{cc}/w_{cp} \) and \( \eta_{qdl} = q_{dl}/w_{cp} \). Then heat transfer efficiency can be expressed in a similar way:

\[
\eta_q = 1 + \ln(\eta_{qsl} \cdot \eta_{qcc} \cdot \eta_{qdl}) \tag{12}
\]
3.4 Mechanical, electrical and heat losses efficiencies

The influence of the electrical motor, mechanical transmission and external heat losses from the shell to the environment should be considered in a specific way. Thus, mechanical efficiency ($\eta_m$) and electrical efficiency ($\eta_e$) are defined as:

$$\eta_m = \frac{w_{cp}}{w_m} \quad \eta_e = \frac{w_m}{w_e}$$

(13)

where $w_m$ and $w_e$ represent mechanical and electrical works.

Based on an energy balance in all the hermetic reciprocating compressor, and over a complete cycle (under periodical conditions), the total energy transfer to the fluid is equal to the total energy transfer between compressor shell and environment (specific electrical consumption + specific heat transfer losses from the shell to the environment), $w_{cp} + q_{fluid} = w_e - q_{shell}$. Then,

$$\eta_{shell} = \frac{w_e - q_{shell}}{w_e} = \eta_q \cdot \eta_m \cdot \eta_e$$

(14)

Eq. (14) shows a direct relationship between heat losses efficiency from shell to environment and mechanical, electrical and heat transfer efficiencies inside compressor.

3.5 Coefficient of Performance

Considering the whole refrigerating system, the non-dimensional parameter that defines its global thermodynamic behaviour in the whole refrigeration process is the COP (coefficient of performance), which relates evaporator refrigerating capacity with electrical power consumption, or alternatively compression work. Coefficient of performance is easily detached as a function of different efficiencies presented above:

$$COP = \frac{\dot{Q}_{ev}}{\dot{W}_e} = \frac{\Delta h_{evap}}{w_{iso}} \cdot \eta_m \cdot \eta_e \cdot \eta_{iso} = K \cdot \eta_m \cdot \eta_e \cdot \eta_{iso}$$

(15)

where $K$ is the non-dimensional ratio between evaporator heat rate per mass unit and isentropic work.

3.6 Pressure pulsations

Instantaneous refrigerant pressure at each time-step or crank angle degree at different compressor locations, allows to have a periodical function of local pressure vs. time. Post processing Fast Fourier Transform analysis allows to obtain amplitude pressure maps vs. frequency. Pressure amplitude helps to identify critical frequencies of noise due to fluid pressure pulsations. After pressure pulsations amplitude are obtained, different acoustic parameters are available to study critical points, or noise reduction [12].

Meaningful non-dimensional acoustic parameters considered are Sound Pressure Level and the Transmission Losses:

$$SPL = 20 \log_{10} \frac{p}{p_{ref}} \quad TL = 20 \log_{10} \frac{p_i}{p_o}$$

(16)

where $p_{ref}$ is the sound pressure level on the air (20 $\mu$Pa), and $p_i$ and $p_o$ are the inlet and outlet pressure in the selected zone to be studied.

Sound pressure level in different strategic compressor locations helps to know the maximum amplitude in the main frequencies, which is useful to design compressor shell and reduce resonances. Transmission losses from any inlet areas to any outlet areas gives a value of noise reductions or amplifications.

3.7 Exergy analysis

Together with energy and entropy thermodynamic balances, it is advisable to carry out overall exergy analysis in order to quantify irreversibilities [13]. Combining the first law of thermodynamics and the second law of thermodynamics based on entropy, the second law of thermodynamics based on exergy or available energy (maximum work that could be derived if the system were allowed to come to equilibrium with environment) can be written. Considering periodical conditions, exergy equation is as follows:

$$\dot{i} = - \left( \sum \dot{m}_o \xi_o - \sum \dot{m}_i \xi_i \right) + \sum \int \delta Q_{fluid} \left( 1 - \frac{T_o}{T_w} \right) + W_{cp}$$

(17)
Thus, exergy destruction or irreversibilities are determined by: convective exergy flux \( \dot{\varepsilon}_c = \sum \dot{m}_i \dot{\varepsilon}_{ki} \), exergy associated to heat transfer gains \( \dot{\varepsilon}_Q = \sum \left( \delta Q_{fluid}(1 - T_o/T_w) \right) \) and exergy associated to input compressor shaft work \( \dot{W}_{cp} \).

\[
\dot{I} = -\dot{\varepsilon}_c + \dot{\varepsilon}_Q + \dot{W}_{cp} \tag{18}
\]

The exergy efficiency of hermetic reciprocating compressor can be defined as the ratio between exergy flux and exergy associated to input compressor work. This ratio allows to compare the well-used available energy and the lost available energy.

\[
\eta \xi = \frac{\dot{\varepsilon}_c}{\dot{W}_{cp}} \tag{19}
\]

## 4 NUMERICAL SIMULATION MODEL

All numerical results have been obtained by means of a detailed numerical simulation model [3-5]. In this model, the whole compressor domain (tubes, chambers, shell, crankcase, oil, etc.) is divided into strategically control volumes. For each CV a set of algebraic equations is obtained by means of the integration of the conservation governing equations (continuity, momentum, energy and state), in a one-dimensional and transient form.

\[
\frac{\partial m}{\partial t} + \sum \dot{m}_o - \sum \dot{m}_i = 0 \tag{20}
\]

\[
\frac{\partial m \dot{\bar{v}}}{\partial t} + \sum \dot{m}_o \bar{v}_o - \sum \dot{m}_i \bar{v}_i = (p_i - p_o) S_k - f \frac{|\dot{m}| \dot{\bar{v}}}{2 S_s} P \Delta z \tag{21}
\]

\[
\frac{\partial m \dot{h}_k}{\partial t} - V \frac{\partial \dot{\bar{p}}}{\partial t} + \sum \dot{m}_o \dot{h}_t o - \sum \dot{m}_i \dot{h}_t i = \sum \dot{Q}_{fluid} \tag{22}
\]

\[
\frac{\partial m s_t}{\partial t} + \sum \dot{m}_o s_t o - \sum \dot{m}_i s_t i - \sum \int \frac{\delta Q_{fluid}}{T_w} = \dot{S}_{gen} \tag{23}
\]

\[
f (p, p, T) = 0 \tag{24}
\]

Momentum equation (21) must be specifically characterized when the fluid refrigerant is flowing through a singularity (contract coefficient) or through a valve (effective force and flow areas) [14].

The solid thermal behaviour is evaluated by means of heat global balances, at each solid elements, considering convection between the solid element considered \( k \) and the surrounding fluid \( i \), and considering conduction and radiation between the different solid elements. Then, for a given solid element \( k \) [4]:

\[
\frac{p_k c_p k (T_k - T_{k,0}) V_k}{\Delta t} = \sum_{solid_j} \frac{T_k - T_j}{R_{kj}} A_{kj} + \sum_{fluid_{ij}} \alpha_{conv_{ki}} (T_k - T_i) A_{ki} + \sum_{solid_j} \alpha_{rad_{kj}} (T_k - T_j) A_{kj} \tag{25}
\]

The model also carries out, at each time-step, the force balances in the connecting rod and crankshaft mechanism which allows to evaluate the instantaneous angle, crank velocity and motor torque. The simulation incorporates a multidimensional model for the valve dynamic based on frequency modal analysis. Thermodynamic and transport properties for different refrigerant fluids and mixtures are evaluated at the different local conditions using REFPROP properties program [15].

The governing equations of the flow are discretised by means of an implicit control volume formulation and using staggered grids to compute velocity field. The convective terms are numerically approximated using the first order upwind numerical scheme, while transient terms allow numerical approximation using second and third order numerical schemes. A SIMPLE-like algorithm, extended to compressible flow, has been used for velocity-pressure coupling. The complete set of of discretised momentum, energy and continuity (or in its pressure correction form) equations in the whole compressor domain are solved at each time step using the line by line TDMA (Tri-Diagonal Matrix Algorithm) method. Parallel circuit and extra elements (such as double orifices, resonators, etc.) are considered in the formulation. The motor torque equation system is linearly independent, thus is solved directly by means of inverse matrix system LU resolution. Instantaneous crank
angle position is obtained from crank angle acceleration by means of Heun method. Finally, macro-volumes energy balances are also directly solved by means of inverse matrix system LU resolution.

All this formulation needs some common relations to evaluate local fluid friction and heat transfer together with pressure drop through singularities and friction forces among solids.

5 NUMERICAL RESULTS

Based on the code developed, and after a careful verification of computational errors and physical errors, all the different numerical parameters which define hermetic reciprocating compressor performance are presented in order to evaluate their influence and their detachments under periodical conditions.

All cases presented have been simulated in a PC cluster with 48 AMD-K7 processors working at 900 MHz with a RAM memory of 512 MB. Software code language is FORTRAN 95 and all variables type are double precision 8 bites.

The results correspond to the Unidad Hermética commercial hermetic reciprocating compressor with a 7.5 cm³ cylinder capacity, working with R134a and a nominal frequency of 50 Hz. The following single stage vapour compression low pressure cycle has been considered: both inlet compressor temperature and outlet condenser temperature of 32 °C, condensation temperature of 55 °C and evaporation temperature of -23.3 °C. Ambient temperature is fixed at 32 °C. Table 1 shows numerical illustrative results for the different evaporation temperatures considered.

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<th>$q_{fluid}$ (kJ/kg)</th>
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<th>$\eta_id$ (%)</th>
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<th>$\eta_m$ (%)</th>
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5.1 Volumetric efficiency detachment

Table 1 shows volumetric efficiency and ideal volumetric efficiency for the studied cases. Both group of values present same tendency although shifted. Figs. 2a and 2b show suction and discharge valve efficiency respectively. Suction valve efficiency gives values greater than 100%, which indicates supercharging effects. Suction valve efficiency has almost a maximum -20 °C of evaporation temperature, while discharge valve efficiency always has an approximate value of under 100%. Fig. 2c shows irreversibilities efficiency increasing when evaporation temperature increases, and Fig. 2d shows frequency efficiency decreasing when evaporation temperature increases.

In general, most of hermetic reciprocating compressors tested follow same tendencies indicated in these figures, except suction and discharge valve efficiencies (Figs. 2c and 2d) which strongly depends on valve stop position, orifice diameter, valve properties (mass, stiffness, damping, etc.).
5.2 Isentropic efficiency detachment

Table 1 also shows real compression work and isentropic efficiency respectively for the studied cases, while Fig. 3 show isentropic efficiency ratios detached. Real compression work decreases almost linearly when evaporation temperature increases. However, isentropic efficiency is almost unaffected and close to 73% by the evaporation temperatures studied. Figs. 3a and 3b show compression and expansion work efficiency respectively. Compression work increases when evaporation temperature decreases and always below 100%, while expansion work decreases when evaporation temperature increases always over 100%. Figs. 3c and 3d show suction and discharge work efficiency. The first one has a maximum value at approximately -20 °C of evaporation temperature, while the second one decreases when evaporation temperature increases.

\[ \eta_{we} \]  \[ \eta_{we} \]  \[ \eta_{wd} \]  \[ \eta_{we} \]

Fig. 3: Isentropic efficiency detached. (a) \( \eta_{we} \); (b) \( \eta_{we} \); (c) \( \eta_{wd} \); (d) \( \eta_{we} \).

Compression and expansion work per mass flow unit increases when evaporation temperature increases, i.e. both work necessary to compress and work obtained in the expansion are greater when evaporation temperature increases. However, suction and discharge work per mass unit are very similar in all cases.

5.3 Heat transfer efficiency detachment

Fig. 4 shows the results of the heat transfer efficiency and its detachment for the numerical cases tested. Fig. 4a shows that suction heat efficiency increases when evaporation temperature increases although tending to a value. Fig. 4b shows that compression heat efficiency increases almost linearly when evaporation temperature increases. Fig. 4c shows that discharge heat efficiency is almost constant, increasing slightly when evaporation temperature increases. Finally, Fig. 4d clearly shows that global heat transfer efficiency increases linearly when evaporation temperature increases.

\[ \eta_{hc} \]  \[ \eta_{hc} \]  \[ \eta_{hd} \]  \[ \eta_{hc} \]

Fig. 4: Heat transfer efficiency detached. (a) \( \eta_{hc} \); (b) \( \eta_{hc} \); (c) \( \eta_{hd} \); (d) \( \eta_{hc} \).

5.4 Pressure pulsations ratios

Fig. 5 shows instantaneous local mean pressure values along different strategic compressor points during a periodical cycle. The results presented are amplitude pressure pulsation vs. frequency (from 0 to 2500 Hz).

Sound Pressure Level is obtained at different compression locations: inlet compressor section Fig. 5a, suction compression chamber Fig. 5b, discharge compression chamber Fig. 5d and outlet compressor section Fig. 5e. As can be seen, pressure pulsations at inlet compressor cross section decrease from 50 Hz to 1000 Hz, similar to the outlet compressor cross section, although with less amplitude.
Inlet and outlet compression chamber section present pressure pulsations along all the frequency range with amplitudes between 125 and 175 dB. Figs. 5c and 5f present Transmission Losses due to Sound Pressure Level at suction and discharge lines respectively. Transmission Losses increase both cases from 50 Hz to 500 Hz and then maintain the amplitude with some oscillation until 2500 Hz. Transmission Losses amplitude is higher along the suction line than along the discharge line.

5.5 Exergy efficiency ratio

The different contributions to the exergy balances are shown in Fig. 6 as function of the evaporation temperatures, or compression ratios: convective exergy flux $\dot{\Xi}_c$ (Fig. 6a), exergy associated to heat transfer gains $\dot{\Xi}_Q$ (Fig. 6b) and irreversibilities $\dot{I}$ (Fig. 6c). Exergy associated to compressor shaft work is equal to compression work values in Table 1. $\dot{W}_c$ and $\dot{\Xi}_c$ almost increase linearly when evaporation temperature decreases. $\dot{\Xi}_Q$ is almost constant for the different compressor ratios, while irreversibilities increase between 20W for low evaporation temperatures (-35°C) to 50W for high evaporation temperatures (-5°C).

Finally, Fig. 6d shows exergy efficiency for this kind of compressors, which is around 72% with a maximum at -20°C of evaporation temperature.
6 CONCLUSIONS

A review of different hermetic reciprocating compressors parameters performances has been carried out. The parameters selection, their detachment and physical meaning have been presented. Illustrative numerical results obtained from an advanced numerical simulation model have been used to carry out a study of the different compressor parameters performances presented in the paper.

The results have shown the importance of these parameters in compressor behaviour. Their detachment has demonstrated how performance can be improved on the basis of these parameter studies.

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8 REFERENCES