A parametric optimization procedure for the suction system of reciprocating compressors

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Abstract. The design of the suction system of compressors is of fundamental importance for efficiency and reliability. This paper reports a method developed to optimize the suction system of a reciprocating compressor, by using the genetic algorithm NSGA-II. The isentropic and volumetric efficiencies are used as objective functions, while the bending fatigue stress is used as a constraint to meet valve reliability. A simulation model of the compression cycle was coupled to the optimization procedure, with correlations for flow and force effective areas in terms of geometric parameters of the suction valve. Valve dynamics was numerically solved via the finite element method. The proposed optimization procedure was applied to a reciprocating compressor adopted for household refrigeration, identifying suction system geometries more efficient than the original design.

1. Introduction

There is a continuous effort to increase the efficiency of reciprocating compressors adopted in household refrigeration. According to [1], investments related to the development of such compressors in the period between 1980 and 2002 were responsible for an average increase of 60% in the COP. However, the performance of refrigeration systems is far from the theoretical limit and improvements in the compressor are still possible.

The performance of reciprocating compressors is commonly quantified through the isentropic efficiency, $\eta_s$, and volumetric efficiency, $\eta_v$, defined as

$$\eta_s = \frac{\dot{W}_s}{\dot{W}_i}$$
$$\eta_v = \frac{\dot{m}}{\dot{m}_{stw}}$$

where $\dot{W}_s$ and $\dot{W}_i$ are the isentropic and indicated power, and $\dot{m}$ and $\dot{m}_{stw}$ are the measured and theoretical mass flow rates, respectively.

The isentropic efficiency is appropriate to analyze the efficiency of the compression cycle and is affected by inefficiencies in the processes of suction, discharge, compression, and expansion of the gas, not accounting for electrical and mechanical inefficiencies. The volumetric efficiency is reduced by gas re-expansion in the compression chamber, gas superheating, leakage, backflow in valves, and viscous friction in the flow through the suction and discharge systems.

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Much effort has been directed to improve the efficiency of the suction and discharge processes in compressors, with special attention to valve reliability because valve failure is a common cause of compressor shutdowns [2]. For instance, parametric optimization procedures have been used to identify optimum geometric configurations of compressor valves [3, 4], adopting the impact velocity of the valves against their seats as an optimization constraint. A similar procedure was adopted by [5], but using a genetic optimization algorithm and the impact velocity as part of the objective function rather than a constraint. Recently, Silva et al. [6] developed a genetic optimization algorithm for the optimization of valves in which the valve dynamics was solved via the finite element method in order to assess valve reliability for bending fatigue.

This paper presents a geometric optimization procedure for the suction system of reciprocating compressors. Correlations for effective area coefficients required to characterize valve performance were obtained in terms of geometric parameters of the suction system. The genetic algorithm NSGA-II for multi-objective optimization was employed and the isentropic and volumetric efficiencies of the compressor were set as independent objective functions. The bending fatigue stress was used as a constraint to meet valve reliability.

2. Compressor thermodynamic model

The numerical model used to predict the compressor performance is based on equations for the thermodynamic state of the gas in the compression chamber, valve dynamics, and flow in the suction and discharge mufflers [6].

The thermodynamic state of the gas is determined by applying the equations of conservation of mass and energy in the compression chamber in conjunction with the specification of an equation of state for the gas. The mass flow rates flowing in and out of the compression chamber are necessary in the equation of conservation of mass. Particularly, the mass flow rates through the suction and discharge valves are obtained with reference to the isentropic flow in a convergent nozzle and the introduction of the effective flow area coefficient, \( C_{ep} \):

\[
\dot{m} = C_{ep} A_o p_{up} \frac{2k}{R T_{up} (k-1)} \sqrt{\frac{k}{k-1} (\Pi^2 - 1)^{\frac{k}{k-1}}}
\]

(2)

where \( A_o \) is the area of the orifice, \( p_{up} \) is the upstream pressure, \( k \) is the ratio of specific heats, \( R \) is the specific gas constant, \( T_{up} \) is the upstream temperature, and \( \Pi \) is the pressure ratio. The leakage of gas through the piston-cylinder gap is obtained by assuming a fully developed Couette-Poiseuille laminar flow.

The dynamics of the suction valve was solved with the finite element model based on beam elements, since bending stress has to be evaluated in the optimization procedure. On the other hand, the dynamics of discharge valve is described as a rigid body moving perpendicularly to the seat, via a single degree-of-freedom model. In both cases, the resulting force acting on the valve may include the oil stiction force and preload stress imposed on the valve. The force due to the gas pressure load, \( F_o \), is evaluated through the concept of effective force area coefficient, \( C_{ef} \):

\[
F_o = C_{ef} A_o \Delta p
\]

(3)

where \( \Delta p \) is the pressure difference between both sides of the valve.

The pulsating compressible flow through the suction and discharge mufflers is modeled via the conservation equations for mass, energy, and momentum following a one-dimensional formulation. Such equations are solved using the finite volume method and taking into account heat transfer and viscous effects. Details of this modeling approach can be obtained in [7].
3. Correlations for effective flow and force areas

3.1. Geometric parametrization

The coefficients for effective flow area and effective force area are functions of the compressor geometry. Since the suction system is to be defined via an optimization procedure, correlations for these coefficients are required in terms of geometric parameters. Therefore, the geometry of the suction system was parametrized and CFD simulations were performed in order to obtain these correlations.

The solution domain of these simulations is presented in Figure 1. The parameters that describe the suction system geometry are: the valve opening, $s$, the valve opening angle, $\theta_v$, the width of the orifice, $w_p$, the valve width, $w_v$, and the height and width of the seat, $h_s$ and $w_s$, respectively (Figure 2). Some relations of dependence between these parameters can be specified. For instance, by assuming that no bending exists when the valve opens, $s$ and $\theta_v$ can be easily related. Also, the valve width can be defined as

$$w_v = w_p + 2w_s + 2d_v$$  \hspace{1cm} (4)

where $d_v$ is the length of the valve that extends beyond the edge of the seat. If $d_v$ is held constant, then $C_{ep}$ and $C_{ef}$ can be described in terms of four parameters: $s$, $w_p (= R_e - R_i)$, $w_s$, and $h_s$.

The orifice geometry is completely described in terms of these parameters. For instance, the area of the suction orifice, $A_p$, is the sum of the areas $A_1$ and $A_2$ (Figure 3a), i.e.:

$$A_p = A_1 + A_2$$  \hspace{1cm} (5)

where,

$$A_1 = \frac{\pi}{8} (R_e - R_i)^2$$  \hspace{1cm} (6)

$$A_2 = \frac{\theta_v}{2} (R_e^2 - R_i^2)$$  \hspace{1cm} (7)
Figure 3. Compressor geometry: (a) suction orifice area, (b) suction orifice and cylinder profiles, and (c) trigonometric relation between geometric parameters.

In these equations, \( R_i \) and \( R_e \) are, respectively, the radius of the internal and external arcs used in the generation of the orifice profile and \( \theta_0 \) is their aperture angle.

From Figure 3b, the following relation can be established:

\[
R_e = R_{cyl} + d_s - d_1
\]

where \( R_{cyl} \) is the cylinder radius, which has a fixed value for a particular compressor, \( d_s \) is the eccentricity between the axis of the cylinder and the origin of the arcs that define the orifice profile, and \( d_1 \) is the maximum distance between the external arc defining the orifice profile and the cylinder wall.

Also, from Figure 3c, a trigonometric relation can be obtained:

\[
R_{cyl}^2 = d_s^2 + (R_e + d_2)^2 - 2d_s(R_e + d_2)\cos\theta_0
\]

where \( d_2 \) stands for the minimum distance between the external arc defining the orifice profile and the cylinder wall. Therefore, if \( A_p \), \( d_3 \), and \( d_2 \) are fixed, the orifice geometry is completely described solely by the parameter \( w_p \). The seat and valve geometries are obtained directly from the orifice geometry and the parameters \( w_s \) and \( d_v \).

3.2. Numerical simulation

The flow of gas in the suction system was solved with the commercial code ANSYS CFX 14.5. A steady solution was provided for each valve position of interest, considering the effect of gas compressibility. Based on the results and recommendations of [8, 9], the RNG k-\( \varepsilon \) turbulence model was adopted.
The boundary conditions of the solution domain (Figure 1) were based on pressure and temperature measurements carried out in the compressor. At the inlet section, pressure, temperature, and parameters related to the turbulence model were specified, while at the outlet only pressure was prescribed. The no-slip and temperature boundary conditions were adopted at the walls. A condition of flow symmetry was also adopted to reduce computational time, as indicated in Figure 1.

An unstructured computational mesh was adopted in the simulations, with prismatic elements near the boundaries and tetrahedral elements in the remainder of the domain. A mesh with approximately 700,000 elements was found adequate for the analyses. In order to reduce computational cost the isothermal condition was adopted, since the comparison of the effective area coefficients shows that they do not change significantly in comparison to the non-isothermal solution (Figure 4). This caused a reduction of almost 20% in the computational cost.

4. Optimization procedure
The variables of the optimization problem are the geometric parameters of the suction system, composed by the suction valve, valve plate, and suction muffler. As a result of the finite element method applied to predict the suction valve dynamics, the valve is represented by a group of beam elements. Therefore, the valve geometry is defined based on the valve thickness and the width of each beam element. In possession of correlations for the effective area coefficients, the orifice width, \( w_p \), the height of the seat, \( h_s \), and the width of the seat, \( w_e \), are adopted as project variables related to the valve plate geometry. The diameter and length of the tube of the suction muffler connected to the suction chamber affect significantly the valve dynamics [10] and are included as variables of the optimization procedure.

As proposed by [6], the objective functions, \( \eta_s \) and \( \eta_v \), are evaluated for the operating condition in which the highest performance is required. On the other hand, valve reliability is verified in the operating condition that result the highest stress levels, as in conditions where higher mass flow rates exist. Therefore, the compressor model must be run twice for each configuration to be evaluated. The Soderberg criterion provides a way to verify whether the failure limit is violated, being a constraint of the problem, together with the range limits of each of the variables considered. If this constraint is not satisfied, null values are set to the isentropic and volumetric efficiencies and the configuration is discarded from the optimization procedure.

The genetic algorithm NSGA-II (Elitist Non-Dominated Sorting Genetic Algorithm) developed by [11] was selected because it is commonly used in the solution of optimization problems with multiple objectives and real-valued variables. In this algorithm the ranking of the feasible solutions is based on the fitness and density of solutions in the objective space. In this case, there is no necessity to combine the efficiencies in a unique equation by specifying weighting factors, as described in [6]. The adopted stopping criterion is based on the number of generations of the algorithm. A simplified flowchart of the procedure is presented in Figure 5.

![Figure 4. Comparison of isothermal and non-isothermal solutions.](image-url)
5. Results and Discussion

5.1. Correlations for effective flow areas

Correlations for coefficients of effective flow area and effective force area were obtained from CFD simulations, considering different geometric configurations of the suction system characterized by the variables $h_s$, $w_s$, $w_p$, and $s$. Three values were considered for each variable, except for the valve opening, $s$, which had eight levels, since it is considered the most influential parameter. Therefore 216 CFD simulations were run for a full factorial design of experiments.

Correlations for $C_{ep}$ and $C_{ef}$ were obtained, using the code Eureqa. From these results:

$$C_{ep} = C_1 \sin(C_2 s^*) + \frac{C_3 s^* C_4 \cos(C_5 h_s^* + C_6 s^* + C_7 h_s^*)}{C_8 w_p^*} \quad (10)$$

$$C_{ef} = C_9 + C_{10} w_s^* w_p^* s^* - s^* C_{11} C_{12} s^* e^{C_{13} (w_s^* - h_s^*)} + C_{14} s^* + C_{15} s^* \sin(C_{17} s^*) \quad (11)$$

where the superscript “*” indicated non-dimensional variables, obtained by dividing the original variable by the equivalent diameter of the suction orifice. The values of the coefficients in these correlations are presented in Table 1.

| Coefficient | Value       | Coefficient | Value       | Coefficient | Value       |
|-------------|-------------|-------------|-------------|-------------|-------------|
| $C_1$       | 7.33x10^{-1}| $C_7$       | 4.77x10^{-1}| $C_{13}$    | 8.09        |
| $C_2$       | 3.48        | $C_8$       | 8.1         | $C_{14}$    | 8.09        |
| $C_3$       | 7.76x10^{-1}| $C_9$       | 1           | $C_{15}$    | 3.30        |
| $C_4$       | 4.31x10^{-1}| $C_{10}$    | 1.44x10^{1} | $C_{16}$    | 5.19x10^{-1}|
| $C_5$       | 8.1         | $C_{11}$    | 8.09        | $C_{17}$    | 8.09        |
| $C_6$       | 8.1         | $C_{12}$    | 2.42x10^{-3}|             |             |

The values of $C_{ep}$ and $C_{ef}$ given by the correlations and CFD simulations were compared for two geometric configurations of the suction system not included to generate the correlations. The results in
Figure 6 show that the correlations give results in close agreement with the simulations, with a small deviation in the value for $C_{ef}$ at large valve lifts $s^*$. 

5.2. Optimization of the suction system
The optimization procedure was applied to redesign the suction system of an existing compressor for household refrigeration application. Two different methods were used to define the valve shape. In the first method (A), the width of each beam element in the finite element model was treated as an isolated variable of the optimization problem. In the second method (B), the widths of only a few elements were treated as isolated variables, and the others were obtained from a spline interpolation. Figure 7 presents the results given by both methods in terms of the valve thickness $t_v$ commercially available. The results show that there is a relation between the region of the objective space explored by the algorithm and the valve thickness, with a trend of increasing the compressor efficiency as the valve thickness is reduced. However, thin valves are associated to higher stress levels, and hence are more susceptible to fatigue failure. Thus, the reliability constraint limits the search for more efficient valves.

![Figure 6. Comparative analysis of the correlations with CFD results.](image)

![Figure 7. Solutions of the optimization procedure: (a) method A, and (b) method B.](image)
Figure 8 presents the isentropic and volumetric efficiencies of the optimal configurations, i.e., the configurations located in the Pareto frontier found by both methods (A and B). The efficiencies predicted for the existing compressor design are presented for reference. It is clear that both methods found new design alternatives with greater performance. As expected, the reduced number of variables defining the valve geometry in method B produces fewer optimum solutions. However, method B simplifies the development of new valve concepts, since it avoids discontinuities between the widths of neighbor elements.

![Efficiencies of configurations in the Pareto frontier.](image)

6. Conclusions
This paper presented an optimization procedure for the suction system of reciprocating compressors. The isentropic and volumetric efficiencies were treated as individual objective functions to be maximized by the NSGA-II algorithm. A reliability constraint was used to prevent valve failure due to bending fatigue. Since the simulation model uses coefficients of effective flow area and effective force area to characterize valve performance, correlations for these coefficients were obtained in terms of geometric parameters of the suction system. The optimization procedure was applied to redesign the suction system of an existing compressor and found more efficient and yet reliable alternatives.

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