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A Procedure to Optimize Reed Type Valves Considering Efficiency and Bending Fatigue

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ABSTRACT

Reed valves are the most common type of valves used in small compressors due to their simplicity and low cost. Such valves greatly affect the compressor volumetric and isentropic efficiencies and their suitable design is also crucial for reliability, since they may be subjected to severe bending and impact stresses. This paper reports an optimization procedure for reed type valves considering efficiency and bending fatigue, which is attained by coupling a thermodynamic model for the compression cycle and a finite element model for the valve dynamics. Then, a genetic algorithm is adopted to find different valve geometries that give the best compromise between efficiency and reliability. As an example, the method is applied to optimize the suction valve of a small reciprocating compressor.

1. INTRODUCTION

Refrigeration systems are employed in a wide variety of applications that includes food storage and cooling of electronic devices, being responsible for a significant fraction of worldwide energy consumption. In fact, recent studies show that 8.9% of the residential electric energy consumption in the United States is associated with refrigerators and freezers (U.S. Energy Information Administration, 2010). For this reason, the continuous development of higher-efficiency refrigeration systems is mandatory.

An important amount of energy consumption of vapor compression systems is related to electrical, mechanical and thermodynamic irreversibilities. Ribas et al. (2008) pointed out that reciprocating compressors typically used for household refrigeration have electric efficiency between 87 and 88%, mechanical efficiency around 92%, and thermodynamic efficiency of approximately 80%. The same authors indicated that in a compressor with cooling capacity of 900BTU/h (ASHRAE/LBP) operating with R134a the main sources of thermodynamic losses are related to superheating (49%) and suction and discharge processes (47%). In fact much effort has been directed to improve the efficiency of suction and discharge processes with special attention to valve reliability since valve failure is the most common cause of compressor shutdowns (Hatch and Woollatt, 2002).

In the literature a variety of publications on valve optimization can be found. MacLaren et al. (1976), for example, applied the COMPLEX method in a coupled manner with a thermodynamic model to optimize ring valves of air compressors. The valve behavior was modeled via a one-degree-of-freedom method and the objective function of the optimization procedure was defined in terms of volumetric efficiency, valve losses, and isentropic work of
compression. The authors also adopted a constraint for the maximum impact velocity of the valve. Later, Shu and Tramschek (1984) considered the use of the SIMPLEX method and reformulated the objective function only in terms of volumetric efficiency, $\eta_v$, and isentropic efficiency, $\eta_s$, which can be defined as follows:

$$
\eta_v = \frac{\dot{m}}{\dot{m}_w} \quad \eta_s = \frac{\dot{m}(h_{2,s} - h_1)}{W_{ind}}
$$

where $\dot{m}$ is the actual mass flow rate, $\dot{m}_w$ is the ideal mass flow rate, obtained when the swept volume is completely filled with gas in the same state as in the entrance of the compressor, $h_1$ is the specific enthalpy of the gas in the suction of the compressor, $h_{2,s}$ is the specific enthalpy with discharge pressure and suction entropy, and $W_{ind}$ is the indicated power.

Prins et al. (1996) applied a genetic algorithm in conjunction with a thermodynamic model of the compressor to optimize the ring type suction valve of a reciprocating compressor with high cooling capacity, with the valve being modeled as a one-degree-of-freedom system. An objective function was defined in terms of the volumetric efficiency, isentropic efficiency, and the maximum impact velocity of the valve. The authors noticed that valves that present lower impact velocity in operation, and consequently higher life-time, also present lower volumetric and isentropic efficiencies. More recently, Yaroğlu and Kara (2008) analyzed the influence of the geometry of a discharge reed type valve on the cooling capacity and EER of a household hermetic compressor. In order to predict the valve behavior the authors applied the modal analysis technique in conjunction with a thermodynamic model to predict the compressor performance. Improvements of 3.5% in EER and 7% in cooling capacity were obtained by varying the first natural frequency of the valve between 312Hz and 520Hz.

There are several other studies in the literature related to the optimization of compressor valves but most of them consider only the thermodynamic efficiency aspect. Optimization procedures considering efficiency and bending stress are scarce. The present paper reports the development of an optimization procedure for reed type valves, which takes into account both efficiency and reliability for bending fatigue. In order to predict compressor efficiency, a simplified model is adopted, which considers the compression process, the valve behavior, and the flow through suction and discharge mufflers.

Since the valve bending stress levels must be known, a finite element model is used in the optimization procedure. Also a genetic algorithm is adopted so as to identify the optimal valve geometries. To illustrate the use of this procedure, the suction valve of a reciprocating compressor used in household appliances has been defined for analysis. Experimental data and predictions of fluid-structure interaction are used to validate the results obtained with the optimization procedure.

**2. COMPRESSOR MODEL**

The simulation model used in this work to predict the compressor performance is an improved version of that proposed by Ussyk (1984). It contains equations that describe the thermodynamic state of the gas in the compression chamber, the valve dynamics, and the flow in the suction and discharge mufflers.

**2.1 Compression Chamber**

The equations that describe the thermodynamic state of the gas in the compression chamber are mass conservation, energy conservation, and an equation of state. Mass and energy equations can be obtained applying a volume of control in the compression chamber and assuming spatially uniform thermodynamic properties. As a result, the mass conservation equation can be written as:

$$
\frac{dm}{dt} = \dot{m}_{suc} - \dot{m}_{dis} - \dot{m}_{lea} - \dot{m}_{hout} + \dot{m}_{hdis}
$$

where $m$ is the mass in the compression chamber, $\dot{m}_{suc}$ is the mass flow rate entering the compression chamber through the suction valve, $\dot{m}_{dis}$ is the mass flow rate leaving the compression chamber through the discharge valve,
and \( \dot{m}_{bds} \) is the mass flow rate through the piston-cylinder gap. The terms \( \dot{m}_{bsuc} \) and \( \dot{m}_{bds} \) represent backflow through suction and discharge valves, respectively. In addition, the energy conservation equation is expressed as follows:

\[
\frac{dT_i}{dt} = \frac{1}{m_i c_i} \left[ H_i A_i T_i + \dot{m}_{suc} (h_{suc} - h_i) + \dot{m}_{dis} (h_{dis} - h_i) \right] - \frac{1}{m_i c_i} \left[ H_i A_i \frac{\partial p_i}{\partial T_i} \frac{d\psi_i}{dt} - \frac{\partial p_i}{\partial T_i} v_i (\dot{m}_{suc} - \dot{m}_{dis} - \dot{m}_{lea} - \dot{m}_{h suc} + \dot{m}_{bds}) \right] T_i
\]

where \( H_i \) is the convective heat transfer coefficient between the gas and the cylinder walls, \( A_i \) is the instantaneous heat transfer area between gas and cylinder, \( T_i \) is the cylinder temperature, \( h_{suc} \) is the gas specific enthalpy in the suction chamber, and \( h_{dis} \) is the gas specific enthalpy in the discharge chamber. \( T_i, m_i, c_i, h_i, p_i, \forall_i \) and \( v_i \) are the temperature, mass, specific heat at constant volume, specific enthalpy, pressure, volume, and specific volume related to the gas in the compression chamber.

Equations for mass flow rate through the valves and piston-cylinder gap are also necessary. The former is obtained considering an isentropic flow through a nozzle and correcting it with effective flow area coefficients. The second is expressed in terms of the instantaneous average flow velocity in the gap, which can be obtained via integration of the velocity profile considering a one-dimensional laminar flow. The relation between the compression chamber volume, \( \forall_i \), and the compressor shaft angle, \( \theta \), is dependent on the compression mechanism.

### 2.2 Valve Dynamics

In most simulation models it is a common practice to describe valves as rigid bodies moving perpendicularly to the valve seat, with a single degree of freedom. From the thermodynamic point of view this consideration is very satisfactory since valve motion is ruled basically by the first mode of vibration of the valve. However, this model is not able to predict stress levels on valves.

In this work the valve dynamics for the discharge valve is described by a one-degree-of-freedom model since it is not considered in the optimization procedure. As a result, its equation can be described as:

\[
F(t) - k_{eq} x(t) - c_{eq} \ddot{x}(t) = m_{eq} \dddot{x}(t)
\]

where \( x \) is the valve displacement, \( m_{eq}, k_{eq} \) and \( c_{eq} \) are, respectively, the equivalent mass, stiffness and damping of the valve and \( F \) accounts for the external forces applied to the valve.

As the bending stress in the suction valve must be considered in optimization procedure, this valve is described as a clamped beam and its equation is solved via the finite element method. In this manner, the valve geometry can be described by the thickness and width of the beam elements used in the valve discretization. The suction valve equation, considering an explicit approach for time discretization, can be described as:

\[
\left\{ \frac{I}{\Delta t^2} [M] + \frac{I}{2 \Delta t} [C] \right\} [D]_{r+1} = [F] - [K] [D]_r + \frac{I}{\Delta t^2} [M] [2 [D]_r - [D]_{r-1}] + \frac{I}{2 \Delta t} [C] [D]_{r-1}
\]

where \([M], [K], \) and \([C]\) are the mass, stiffness, and damping matrices, respectively. \([F]\) is the loading vector, \([D]_r\) is the displacement vector for time step \( r \), and \( \Delta t \) is the time increment adopted. Also the penalty method was used to prevent valve penetration into its seat.

### 2.3 Suction and Discharge Mufflers

The 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 at the walls. Details of this modeling approach can be obtained in Deschamps et al. (2002).
3. OPTIMIZATION PROCEDURE

The compressor thermodynamic efficiency can be characterized by means of two global parameters: volumetric efficiency, \( \eta_v \), and isentropic efficiency, \( \eta_s \). Therefore, the optimization problem considered here can be described as a multi-objective optimization (MOO) problem and the objective function, \( f \), defined as:

\[
f(\vec{x}) = -A\eta_v(\vec{x}) - B\eta_s(\vec{x})
\]

where \( \vec{x} \) is the array of variables to be optimized and \( A \) and \( B \) are coefficients of the objective function. The variables to be optimized are: the width of the beam elements used in the valve discretization, the thickness of the valve, and the length and diameter of the tube that connects the suction muffler with the suction chamber. In fact, pressure pulsation in the suction muffler affects the valve performance and, therefore, geometric parameters of the suction muffler must be included in the optimization process.

In order to guarantee valve reliability regarding bending fatigue, a stress constraint is also adopted in the optimization problem. In the present work, the Soderberg criterion is adopted:

\[
\sigma_a \leq \sigma_f \left( 1 - \frac{\sigma_m}{\sigma_y} \right)
\]

where \( \sigma_a \) and \( \sigma_m \) are the alternating and mean component of the stress acting on the valve, which are related to valve loading, and \( \sigma_f \) and \( \sigma_y \) are the fatigue limit for null mean stress and the yield stress of the valve material, respectively.

In the optimization procedure proposed, each valve configuration must be analyzed for efficiency and reliability. In this respect, two operating conditions, A and B, are adopted in the simulations. The first one is related to a standard steady-state operating condition of refrigeration systems, whereas the second represents a critical condition under which the compressor must be able to operate. Hence, condition A is used to evaluate the compressor volumetric and isentropic efficiencies for each valve configuration and condition B is adopted to verify a possible valve failure. After the simulations of conditions A and B, the objective function associated to each valve configuration can be evaluated. If certain valve geometry violates the stress constraint, it is simply discarded.

![Figure 1: Schematic of the optimization procedure.](image-url)
methods and this can result in high computational cost for many optimization variables; (ii) there is no guarantee that the objective function is convex, which can lead deterministic algorithms to get stuck in local minima. A schematic view of the optimization algorithm is presented in Figure 1.

The most expensive operation of the genetic algorithm is certainly the evaluation of the objective function, since two operating conditions must be run for each valve configuration. Such operations are also characterized by its repetitiveness. With the purpose of reducing computational cost, this solution procedure was parallelized using the application programming interface (API) OpenMP. Consequently, the computational time associated to the optimization procedure was reduced approximately by four. A processing time of approximately 16h on a computer with an Intel Core i7 870 @ 2.93 GHz processor was required for a typical optimization.

4. RESULTS AND DISCUSSION

The optimization procedure proposed was applied to the optimization of the suction valve of a 60Hz reciprocating compressor operating with R134a, being submitted to the operating conditions A and B explained in section 3. Four different valve thicknesses were considered in the process. Also the length of the suction muffler tube was allowed to vary between 50% and 120% of its original value, while its diameter was allowed to vary between 90% and 110%. Finally, the width of the beams used to model the valve was constrained by the cylinder dimensions. The optimization algorithm adopted the following parameters: i) population size: 40; ii) number of generations: 50; iii) mutation rate: 30%.

![Figure 2: Efficiencies of compressor with different valves](image_url)

Figure 2 presents the volumetric and isentropic efficiencies related to valve configurations that satisfied the stress constraint. In this figure, each mark represents one configuration and the different mark types represent different valve thicknesses. The thicknesses of the valves were made dimensionless by dividing its values by the original valve thickness. It is clear from Figure 2 that valves with smaller thickness provide better thermodynamic performance for the compressor, because they are less restrictive to the flow, reducing energy losses during valve

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opening and, as a consequence, raising the isentropic efficiency. Therefore, the suction power related to the different valve thickness #1 (1.00), #2 (1.33), #3 (1.67), and #4 (2.00) are equal to 3.56 W, 4.04 W (+13.5%), 5.80 W (+62.9%) and 6.56 W (+84.3%), respectively. The comparison between the openings of valves #1, #2, #3, and #4 presented in Figure 3a helps to explain why the increase of valve thickness end up reducing the volumetric efficiency.

Another important aspect that can be noticed from Figure 2 is that even when efficiencies of valves of the same thickness are compared, they can present significant differences. For instance, valves #1 and #5 have a difference of almost 2% in the volumetric efficiency. When the motion of these two valves are compared (Figure 3b), one can notice that valve #5 presents a higher opening amplitude, which should result in higher volumetric efficiency. However, as can also be seen from Figure 3b, valve #5 closes quite late, when the piston is already in its ascending stroke (0°-180°). This brings about a strong backflow through the valve, reducing the compressor volumetric efficiency. This backflow, characterized by negative mass flow rate, can be observed in Figure 4 for both valves.

Some interesting aspects were observed during the optimization procedure: i) the fundamental natural frequency and the equivalent stiffness of the optimal valves after each generation varied slightly (219.3Hz to 235.4Hz and 216.8N/m to 243.8N/m, respectively); ii) during the whole optimization procedure 76.6% of the valve configurations attended the fatigue criterion while for the first and last generation these rates were 75.0% and 85.0%, respectively, showing that the algorithm looks for reliable configurations along the optimization procedure.
The results provided by the optimization algorithm for one of the configurations were compared to the results obtained with a CFD model that considers fluid-structure interaction (FSI). This model was previously used and validated by Shiomi (2011). Table 1 presents mass flow rate, $\dot{m}$, indicated power, $W_{ind}$, suction power, and $W_{suc}$, obtained with both models. Figure 5 also compares the valve motion predicted by both models. As can be seen, there is a good agreement between the results predicted by both models, which is an indication of accuracy of the proposed optimization procedure. Although the FSI model is capable of providing more detailed information about the flow and bending stress, it takes about 20 hours to simulate a single compression cycle, becoming prohibitive for optimization purposes. On the other hand, the simplified model used in this work takes only about 2 minutes to simulate five compression cycles.

Table 1: Comparison between predictions of FSI code and the present optimization procedure.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Optimization Procedure</th>
<th>FSI code</th>
<th>Relative Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{m}$ (kg/h)</td>
<td>5.04</td>
<td>5.34</td>
<td>3.4%</td>
</tr>
<tr>
<td>$W_{ind}$ (W)</td>
<td>94.7</td>
<td>98.1</td>
<td>6.7%</td>
</tr>
<tr>
<td>$W_{suc}$ (W)</td>
<td>4.2</td>
<td>3.9</td>
<td>5.6%</td>
</tr>
</tbody>
</table>

Figure 5: Valve motion predicted with the FSI code and the one-degree-of-freedom model.

Configurations obtained from the optimization procedure, which lay on the Pareto frontier, were selected and their parameters used to prototype new valves. Such valves were tested and their performances compared with that of the original valve. Results for cooling capacity, $Q_{evap}$, electric power consumed, $W_{elec}$, indicated power, $W_{ind}$, suction power, $W_{suc}$, discharge power, $W_{dis}$, mass flow rate, $\dot{m}$, volumetric efficiency, $\eta_v$, isentropic efficiency, $\eta_s$, and COP are shown in Table 2.

As can be seen, the performances of the new optimized valves are quite close to the performance of the original valve, which has been predicted with the optimization procedure. This is a consequence of the fact that the original valve is already an optimized valve adopted in a compressor commercially available. The results provide further validation of the proposed procedure and its application can help reducing the time required for designing valves for new compressor designs or different operating conditions.

Table 2: Experimental results of performance for different valve designs.
Parameter | Original Valve | Valve A | Valve B | Valve C | Valve D
---|---|---|---|---|---
$\dot{Q}_{\text{evap}}/\dot{Q}_{\text{evap original}}$ | 100.0% | 100.6% | 99.9% | 99.2% | 100.0%
$W_{\text{ele}}$ (W) | 126.3 | 126.9 | 126.1 | 125.9 | 126.0
$W_{\text{ind}}$ (W) | 102.9 | 103.9 | 103.0 | 102.7 | 102.5
$W_{\text{vac}}$ (W) | 3.8 | 3.8 | 3.7 | 3.7 | 3.8
$W_{\text{diss}}$ (W) | 6.5 | 6.8 | 6.6 | 6.7 | 6.6
$m$ (kg/h) | 4.98 | 5.01 | 4.97 | 4.93 | 4.98
$\eta_s$ | 77.7% | 77.4% | 77.6% | 77.2% | 78.0%
$\eta_v$ | 76.7% | 77.1% | 76.6% | 76.0% | 76.7%
$\text{COP}_{\text{original}}$ | 100.0% | 100.0% | 100.0% | 99.5% | 100.5%

5. CONCLUSIONS

This paper reported the development of an optimization procedure for reed type valves used in refrigeration compressors which considers both efficiency and reliability aspects. The procedure was validated with reference to comparisons with predictions obtained by a FSI model and experimental data. The procedure was applied to the optimization of the suction valve of a household hermetic reciprocating compressor. A marginal improvement has been observed with the new valve configurations because the reference valve was already an optimized valve of a commercially available compressor. The application of the proposed procedure can result in important gains in terms of time for designing high performance valves for new compressor designs or operating conditions.

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