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ABSTRACT

The helical suspension spring system has a significant importance on the operation of hermetic compressors, considering the aspects of noise and vibration. Another important point to consider is that, in the moments of start/stop, the internal components (mainly crankcase) must not hit against the housing. In this case, the interaction with the stoppers is very important. Normally a balance between vibration and displacement is aimed through the stiffness of the spring. An increase of the spring stiffness will reduce the kit displacement, but it will increase the transmitted vibration, and vice-versa.

Other than the functional aspect mentioned above, the suspension springs must be designed for reliability. The major loading on the springs happens during the compressor start/stop time causing an oscillatory displacement. In this way, the springs must be designed to support the fatigue process where there is a mean load due the kit weight, and an alternative load due to the lateral/torsional motion of the kit.

Compressors suspension are usually evaluated at 500,000 cycles of start/stop. It is estimated that this number of cycles represents about 10 or 15 years of the compressor life. The approval start/stop tests are performed in special devices, where each cycle needs some time to reflect on the operating conditions of the compressor. These tests can take more than two months to be done, and it is necessary to have a minimum number of samples to estimate the reliability with an acceptable confidence level. In short, the start/stop approval test is expensive and time consuming.

Nowadays, with the necessity of launching products as early as possible (shorter time for project development), it is very important to improve the methodology of designing the suspension system. The objective of this work is to present an improved methodology, based on numerical and experimental analysis, to evaluate the fatigue life of the spring suspension system.
1. INTRODUCTION

Figure 1 shows a schematic view of a reciprocating compressor where it is possible to see the suspension system. Usually the suspension system is composed of four helical cylindrical springs, and one stopper for each spring end. The functions of these stoppers are: to allow the assembly of the mechanical kit (the pump) with the housing; to keep the springs in the correct position; to avoid the contact between the coils; and to reduce the kit lateral movement during the instants of starting and stopping the compressor.

![Figure 1: Schematic view of a reciprocating compressor](image)

The helical suspension spring system has a significant importance on the operation of hermetic compressors, considering the aspects of noise and vibration. Another important point to consider is that in the moments of start/stop, the internal components (mainly crankcase) must not hit against the housing. In this case, the interaction with the stoppers is very important. Normally a balance between vibration and displacement is aimed through the stiffness of the spring. An increase of the spring stiffness will reduce the kit displacement, but it will increase the transmitted vibration, and vice-versa.

Other than the functional aspect mentioned above, the suspension springs must be designed for reliability. An excellent article about this subject is presented by Zaccone (2001), where the main points related to the failure of the spring are showed. The major loading on the springs happens during the compressor start/stop time, causing an oscillatory displacement. In this way, the springs must be designed to support the fatigue process, where there is a mean load due the kit weight, and an alternative load due the lateral/torsional motion of the kit.

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Nowadays, with the necessity of launching products as early as possible (shorter time for project developments), it is very important to improve the methodology for designing suspension systems. The objective of this work is to present an improved methodology, based on numerical and experimental analysis, to evaluate the fatigue life of the spring suspension system.

2. FATIGUE ANALYSIS

“Fatigue is the process where repeated variations in loading cause failure even when the nominal stresses are below the material yield strength; and is made up of crack initiation and subsequent crack growth as a result of cyclic plastic deformation.” (MSC/Institute of Technology, 1999)

Independent of the project, when the fatigue phenomenon is present, the flow shown in figure 2 indicates all the steps necessary to develop the project considering this aspect. The process requires several inputs, such as geometry, load history, environment, design criteria, material properties and process effects. With these inputs, fatigue design
is performed through synthesis, analysis and testing. The process from design, analysis and test is highly interactive and iterative. The number of loops is directly related to the quality of the inputs, and the accuracy to predict the life of the component or system.

In this work, the focus will be on just one part of the flow, which is related to stress analysis, fatigue life and cumulative damage models, and life prediction.

![Fatigue design flow chart](adapted from Stephens et al. (2001))

**2.1. Stress Analysis (Analytical Approach)**

Norton (2000) showed that the maximum shear stress for a helical spring is calculated from:

$$\tau_{\text{max}} = \frac{8F}{\pi d^3} K_w$$

(1)

The $K_w$ (Wahl correction factor) is calculated from:

$$K_w = \frac{4C - 1}{4C - 4} + \frac{0.615}{C}$$

(2)

The spring index $C$ is calculated from:

$$C = \frac{D}{d}$$

(3)

The maximum shear stress is a function of the spring geometry and the load on it. The load is related to the pump weight (internal components), where the load of each spring is approximately the total pump weight divided by the number of springs.

The equations 1, 2 and 3 are from the classic approach for the helical springs, where the load is just in axial direction. Different from the common use, for the compressor, the behavior of suspension springs is different. When the compressor undergoes start/stop conditions, the kit rotates inside the housing, as shown in figure 3. There is evidence that this rotation is significant, and must be considered in the stress analysis. The addition of this lateral motion superimposes an additional shear stress on the spring as the compressor rotates in the shell. Zaccone (2001) presents a methodology to calculate analytically the additional stress caused by the lateral motion. The final result is a factor that must be multiplied by the maximum stress calculated by equation 1. This factor is a function of the
lateral and axial stiffness, and the height of the spring. Table 1 presents an example of the helical spring with the main characteristics and calculus. It is important to observe that the lateral motion is responsible for a significant increase in the value of the torsional stress. Since this example was taken of a real compressor spring, Zaccone (2001) showed that it is very important to include the lateral motion in the stress analysis.

In spite of the previous consideration of lateral motion shown above, the analytical approach has some limitations, which are:

a. There is not an interaction between the spring and the stoppers. It is known that the stoppers have an influence on the kit motion, mainly for large displacement values.
b. The spring motion is the same for all the springs.
c. The spring real motion is not just in two directions. The motion has a tri dimensional behavior.
d. The theory is just for helical cylindrical springs.
e. The analysis is linear. It is not true for large displacement of the kit.

In this way, the suggestion is the application of Finite Element Method (FEM) to improve the stress analysis precision.

Figure 3: Schematic diagram showing the movement of the compressor assembly inside the shell during start/stop conditions and the spring displacement (adapted from Zaccone (2001))

Table 1: Helical spring example

<table>
<thead>
<tr>
<th>Characteristics of the helical spring</th>
<th>values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wire diameter</td>
<td>1.5 mm</td>
</tr>
<tr>
<td>Free length</td>
<td>29.55 mm</td>
</tr>
<tr>
<td>Number of dead coils</td>
<td>2 upper / 2 lowers</td>
</tr>
<tr>
<td>Active coils</td>
<td>7</td>
</tr>
<tr>
<td>Outside diameter</td>
<td>14.8 mm</td>
</tr>
<tr>
<td>Shear modulus</td>
<td>79.300 MPa</td>
</tr>
<tr>
<td>Axial stiffness</td>
<td>2.91 N/mm</td>
</tr>
<tr>
<td>Axial displacement</td>
<td>4.55 mm</td>
</tr>
<tr>
<td>Maximum torsional stress without lateral motion</td>
<td>161.7 MPa</td>
</tr>
<tr>
<td>Maximum torsional stress with lateral motion of 9 mm</td>
<td>535.2 MPa</td>
</tr>
<tr>
<td>Maximum torsional stress with lateral motion of 5 mm</td>
<td>393.0 MPa</td>
</tr>
</tbody>
</table>

2.2. Finite Element Analysis

Bortoli (2009) showed the application of FEM to calculate the shear stress for a helical spring. The figure 4 shows the stress intensity of a spring from table 1, with an axial displacement of 4.55 mm, and without lateral motion. The FEM analysis was done with beam (figure 4-a) and solid elements (figure 4-b). The shear stress is half of stress intensity. Considering the results of solid elements as reference, the error in the solution is about 7% for analytical analysis, and 9% for beam elements. Although the precision of the results with solid elements, the application of this element to solve a complete suspension system (spring and stopper) with a dynamic load (start/stop event) is unfeasible due to high CPU time to perform the analysis. In this way, the use of beam elements is a good option to solve the problem (the precision is smaller, but much faster).
2.3. Fatigue Life Model

The fatigue life model adopted for the analysis is the stress-life (S-N) theory. The S-N approach estimates total life without distinguishing crack initiation from crack propagation. The S-N approach uses the (assumed elastic) nominal stress range as a measure of the severity of fatigue loading. Tests at several levels of stress range characterize the S-N curve. Zaccone (2001) presents a modified Goodman diagram for 1.5 mm music wire, and the ultimate shear strength is defined as a function of the tensile strength (equal $1/3 \sqrt{2\sigma_f}$). For loads above the fatigue limit, a linear approximation of constant-life will be used to estimate the finite (see figure 6, Zahavi (1996)). From the example of table 1, the life for the spring with mean stress of 162 MPa, and with amplitude of 373 MPa (spring with axial displacement of 4.55 mm and 9 mm of lateral deflection) is 800,000 cycles.
2.4. Cumulative damage model
The basic S-N curve shows the simplest situation, in which a uniform cyclic load is applied. In the more general practical case, with randomly distributed stresses, and with some peaks of stress, above the infinite life limit, it is necessary to quantify the damage caused by each peak. In this work, the Palmgren-Miner method will be used to analyze the cumulative damage. The Palmgren-Miner's Law is expressed as (Zahavi (1996), O'Connor (2008)):

$$\sum_{i=1}^{k} n_i / N_i = n_1 / N_1 + n_2 / N_2 + n_3 / N_3 + \ldots + n_k / N_k$$  \hspace{1cm} (4)

where $n_i$ is the number of cycles at a specific stress level, above the fatigue limit, and $N_i$ is the average number of cycles to failure at that level, as shown on the S-N curve. Failure occurs when:

$$\sum_{i=1}^{k} n_i / N_i = 1$$  \hspace{1cm} (5)

For the spring from table 1, 100,000 cycles with axial displacement of 4.5 mm and lateral motion of 9 mm accumulates a damage of 12.5%.

![Figure 6: Linear approximation of a constant-life diagram](image)

2.5. Life prediction
Figure 7 summarizes all the process to evaluate the component life. The process begins with the load on the component, extraction of the peaks and valleys, a counting process (racetrack counting), a damage counting, a damage histogram and finally the life.

![Figure 7: Overview of the life prediction analysis (adapted from MSC/Institute of Technology (1999))](image)
3. APPLICATION

A real application of the methodology proposed by this work is presented below. The whole process is based on experimental data of the suspension spring motion (displacement). Figure 8-a presents the motion of one spring during an occurrence of start/stop. The compressor analyzed has four springs (figure 8-b), and only the suspension spring system is considered flexible in the Finite Element Analysis. The other components (the kit) are treated as rigid (figure 8-c). In this picture it can be seen that each spring has its own motion, but they are related with each other.

![Figure 8: The compressor suspension system analysis](image)

Figure 9 presents the stress level for each spring, during the start and stop process. As already mentioned earlier, there is a different load (motion) for each spring, and the loads are more critical for the stop process. An important point to take into account is that one occurrence of compressor start/stop produces about 20 cycles that cause damage to the spring. This fact must be observed in accelerated life tests to evaluate the spring reliability.

![Figure 9: The stress on the springs](image)

With all the information available, the last step is to estimate the life of the springs, following the procedure specified on item 2.5.

The next step of this work is to compare the expected life with real information to confirm the method effectiveness.
6. CONCLUSION

- The lateral motion of the suspension springs during the compressor start/stop has an important influence on the springs fatigue life.
- The Finite Element Method brings important improvements to the suspension spring analysis. With FEM it is possible to include the interaction between springs and stoppers, and to apply the real loads on the springs. These factors allow a better precision of the stress analysis.
- The use of beam elements makes the analysis of the complete suspension system feasible (considering the CPU time).
- The experimental data for the motion of the kit has fundamental importance on the fatigue analysis of the spring system.
- For each start/stop occurrence, more cycles could take place causing damage to the spring life.

NOMENCLATURE

\( \tau_{\text{max}} \) — maximum shear stress
\( F \) — load applied
\( D \) — mean spring diameter
\( d \) — wire diameter
\( K_w \) — Wahl correction factor
\( C \) — spring index
\( n_i \) — the number of cycles at a specific stress level
\( N_i \) — the average number of cycles to failure at stress level
\( \sigma_f \) — tensile strength

REFERENCES


MSC/Institute of Technology, 1999, Durability and Fatigue Life Analysis Using MSC/Fatigue.


Zahavi, Eliahu, 1996, Fatigue Design - Life Expectancy of Machine Parts, CRC Press, USA

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