Modeling of the Bottom Cover Dynamics of a Scroll Compressor

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Modeling of the bottom cover dynamics of a scroll compressor

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Abstract
This study is based on the procedure used to determine whether a new cover design will be as immune to vibration problems as a current production design. In particular, the focus is on the development of a finite element model for the proposed design whose results are then compared to both experimental and computational results of the production design. Issues associated in modeling realistic boundary conditions using combinations of classical boundary conditions are covered in detail. Using such a procedure, a maximum of 4% discrepancy in the calculated modes of interest was achieved.

Introduction
If the resonances of the compressor mounting feet are not properly adjusted, structure-borne sound transmission may then become the dominant noise path particularly in light of continued reductions in fan and compressor gas noise [1]. This was the case with some first generation scroll compressor designs, as resonances of the bottom cover were excited leading to vibration issues. However subsequent designs have been able to avoid these issues through a careful analysis of the cover dynamics using finite element and experimental modal methods. Such a methodology requires the ability to reproduce the boundary conditions represented by the connection between the shell and cover. Success in modeling of the boundary conditions in the finite element analysis is then judged by whether the predominant foot bending modes match the modal analysis results. To demonstrate the method, it is applied to both a known and a proposed cover design. Unlike more elaborate efforts [2,3] which require a finite element model of the entire compressor to assist in airborne noise calculations, we are only interested the more local behavior of the compressor feet in structure-borne noise transmission [1].

Summary of Results
A model of each cover design was created using plate elements (Figure 1). The finite element analysis was performed using MSC/NASTRAN [4] with the MSC/XL pre-processor on an IBM RISC 6000 workstation. To accurately recreate the welded interface between the compressor shell and the bottom cover, a variety of boundary conditions were tried in an attempt to hopefully bracket the actual conditions. These boundary conditions entailed either placing classical boundary conditions (i.e. clamped or simply supported) along the weld location or modeling the shell up to the bottom of the stator (which may be treated as a fixed connection). A summary of the models with different boundary conditions is as follows (Figures 2 and 3):

- **BC1**: bottom cover only, simply supported at shell location (inner edge of weld fillet)
- **BC2**: bottom cover with compressor shell clamped at bottom of stator
- **BC3**: bottom cover only, clamped at shell location (inner edge of weld fillet)
- **BC4**: bottom cover only, clamped at outer edge of weld fillet
- **BC5**: bottom cover only, simply supported at inner and outer edge of weld fillet
The reasoning behind the particular boundary conditions will be discussed later. An eigen-solution analysis was then conducted to obtain the mode shapes and frequencies. For this study we limited our attention to the lower order modes which involved bending of the feet, since our prior experience has shown these to be the modes associated with earlier resonance problems. Next, to test which boundary condition is the most accurate, the actual modes of the production cover were obtained experimentally. The experimental modal analysis was based on impact testing in conjunction with SMS STAR Modal software [5].

The results of the experimental and finite element analyses are given in Tables 1 and 2. For the production cover (Table 1), the natural frequencies are too low for the first boundary condition and hence the interface is too compliant since it does not constrain rotations. The second boundary condition was anticipated to be the most realistic, but again the frequencies appear much lower than the experimental values. The third boundary condition set produces results nearly identical to those of the second set, and therefore the interface is still too compliant. Moving the fixed boundary conditions (where both translations and rotations are constrained) from the inner fillet radius (as in BC3) to the outer radius (as in BC4) does stiffen the structure in terms of the foot bending modes, however now the frequencies are too high. From the results of BC3 and BC4, we can deduce that the effective constraint location appears to lie between the inner and outer weld fillet radius, and from the results of BC1 we realize that constraining rotations is also important. Therefore we choose another boundary condition set (BC5) that is simply supported on both the inner and outer weld fillet radii, which then effectively constrains rotations at an intermediate radius. The frequencies for this set were indeed bracketed by those of BC3 and BC4 and furthermore yielded the best match when compared to experiment. In fact, for the modes of interest, there was less than a 4% discrepancy between finite element using BC5 and experiment. In the absence of experimental data for the proposed design, boundary condition set 5 will be considered the most accurate for both models though the complete set of boundary condition were tried for both designs. Note that while only one of the four primary bending modes was in the 1250 Hz band with the current design (Table 1), all four lie in that band with the proposed design (Table 2), which suggests a stiffer design.

In terms of vibration control, Table 1 demonstrates why the current cover has satisfactory vibrational behavior -- the primary foot bending modes (Figure 4) lie below the 1600 Hz band which is associated with mechanical excitations in the compressor. Similarly for the proposed cover design, Table 2 shows that the primary bending modes (with BC5) again lie below the 1600 Hz band. As a result, with no bending modes lying in the 1600 Hz band this cover appears to avoid vibration problems. However, the third distinct bending mode (1390 Hz), appears extremely close (within 2%) to the lower bound of the 1600 Hz band (1414 Hz) and may actually fall in this band given the uncertainties in the finite element model and boundary conditions. Therefore, this proposed cover should undergo a minor re-design to ensure that the bending modes remain outside this band.

Conclusions

The net finding of these studies is that the foot bending modes of both designs lie below the band of mechanical excitation. This is significant since foot bending modes appear to be favored under operating conditions and have been responsible for resonance problems in prior designs. A modal analysis of the current design corroborates the finite element results by showing that the foot bending modes of that cover indeed lie below the mechanical excitation frequency. Therefore based on these
findings, the new design may avoid foot bending resonances as does the current design. However, experimental modal tests on an actual cover of the proposed design are needed to verify this finding, since one of these bending modes appears to be very close to the band of excitation.

References

![Figure 1. Comparison of two bottom cover designs. a. production b. proposed.](image-url)
simply supported (BC1,BC5) 
shell interface  (BC2) 
clamped (BC3) 
free (BC4) 

Figure 2. Schematic of boundary condition sets. See Figure 3 for BC2.

free (BC1,BC2,BC3) 
clamped (BC4) 
simply supported (BC5)

weld fillet

Figure 3. Finite element model of bottom cover with shell (BC2).
Table 1. Natural frequencies for current production bottom cover.

<table>
<thead>
<tr>
<th>Octave band</th>
<th>Modal (exper.)</th>
<th>BC1</th>
<th>BC2</th>
<th>BC3</th>
<th>BC4</th>
<th>BC5</th>
</tr>
</thead>
<tbody>
<tr>
<td>630</td>
<td>699</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>800</td>
<td>978</td>
<td>812(×2)</td>
<td>785</td>
<td>794</td>
<td></td>
<td>1017</td>
</tr>
<tr>
<td>1000</td>
<td>1077(×2)</td>
<td>980</td>
<td>903(×2)</td>
<td>910(×2)</td>
<td></td>
<td>1103(×2)</td>
</tr>
<tr>
<td>1250</td>
<td>1264</td>
<td></td>
<td>1067</td>
<td>1083</td>
<td>1096</td>
<td>1231</td>
</tr>
</tbody>
</table>

Table 2. Natural frequencies for proposed bottom cover.

<table>
<thead>
<tr>
<th>Octave band</th>
<th>Modal</th>
<th>FEM (see notes for boundary conditions)</th>
</tr>
</thead>
<tbody>
<tr>
<td>630</td>
<td>BC1</td>
<td>869</td>
</tr>
<tr>
<td>800</td>
<td>BC2</td>
<td>865</td>
</tr>
<tr>
<td>1000</td>
<td>BC3</td>
<td>964(×2) 978(×2) 938</td>
</tr>
<tr>
<td>1250</td>
<td>BC4</td>
<td>1074 1098 1064(×2) 1203</td>
</tr>
</tbody>
</table>

Note:
- (×2) denotes repeated natural frequencies due to symmetry
- 1600 Hz band highlighted since contains excitation frequency

Boundary conditions (see Figures 2 and 3)
- BC1. bottom cover only, simply supported at shell location
- BC2. bottom cover with compressor shell clamped at bottom of stator
- BC3. bottom cover only, clamped at shell location
- BC4. bottom cover only, clamped at outer edge of weld fillet
- BC5. bottom cover only, simply supported at inner and outer edge of weld fillet
Figure 4. Comparison of mode shapes and frequencies from experiment and finite element (BC5)