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FINITE ELEMENT MODELING OF A REFRIGERATION COMPRESSOR
FOR SOUND PREDICTION PURPOSES

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ABSTRACT

This study involves the development of a finite element model of a hermetic reciprocating compressor for noise prediction applications. Inherent difficulties in developing the finite element model of a complicated structure are discussed and appropriate modeling strategies are evolved. The development of the complete compressor finite element model is carried out in two stages - modeling of the compressor housing and the assembly of components into the compressor assembly.

The compressor housing is isolated for detailed modeling. Geometry complexity, secondary masses, spring mounts, lap-joint and manufacturing variations pose challenges in developing a reliable model. Frequent comparisons are made with experimental mobility scans to obtain insights into the actual behavior of the modeled structure. When possible, weaknesses are located in the finite element model and corrected. After sufficient revisions, 23 natural frequencies (excluding the rigid body modes) are found for the compressor housing in the low frequency range (below 2000 Hz) of analysis. Forced response calculations are also used to correlate the analytical model and test data, with a maximum of 5% disagreement for the 14 natural frequencies that could be correlated.

Compressor assembly modeling involves detailed solid modeling of internal components for inertia properties, developing reduced-degrees-of-freedom models of mounting springs and modeling of the shockloop. Finally, the components are assembled and the compressor assembly is solved for its natural frequencies by the component mode synthesis method. Eighty-seven natural frequencies below 2000 Hz (excluding the rigid body modes) are found for the compressor assembly model. This model can be used to predict velocity responses on the surface of the housing, with the internally generated forces as excitations. Velocity response data are directly used in sound prediction.

INTRODUCTION

The object of this study is Bristol Compressors, Incorporated hermetically sealed reciprocating compressor, used in industrial air-conditioning and refrigeration systems. The study is a co-ordinated effort of several investigators working on different areas of the project. The study uses results from experimental modal analysis in the development of the analytical compressor model. Correlation between the analytical model and experimental analysis instills confidence in the finite element (FE) model. When successfully developed, the FE model would provide velocity responses at chosen points on the compressor housing, from which acoustic behavior could be predicted, so that, a reasonably accurate FE model is essential for predicting acoustic properties fairly accurately. Acoustic measurements made on the housing could also be used to compare with the results obtained analytically. If good correlation is achieved, then the modeling techniques used could be independently applied to analyze alternative models or redesigns and develop or design compressors with better acoustic performance.

Review of Analytical Compressor Sound Prediction Techniques

The problem of sound reduction in hermetic compressors has been studied by several investigators over the last two decades. Technology for developing lighter compressors became well-developed in the 1970's; however, the compressors were noisier. Hence, focus shifted to developing compressors that were quieter. In 1972, Ingalls [1] made an introductory study on the noise generated in hermetic compressors and categorized noise into two types. A group of investigators from Japan [2] identified two noise transmission paths - the first, the airborne path and the second, the suspension system path. They used finite element analysis (FEA) in modeling the dynamics of the compressor and suggest the use of semi-spherical shell designs vis-a-vis cylindrical shell geometries for better acoustic properties. Soedel [3] used circular and oval cylindrical models of the compressor shell and found two similar modes of equal number of node lines to exist at two different frequencies. He also identified controlling factors that affect this kind of behavior. Tavakoli and Singh [4] used state-space mathematics to model compressor shell dynamics and considered sundry features like local masses and the welded joint.
Smith, Kiel and Hurst [5] studied sound intensities in a Freon compressor and suggested modification of the shell-top to a domed form, for reducing radiated noise. Kelly and Knight [6] used FEM to construct a dynamic model of a reciprocating hermetic compressor to predict velocity responses at the housing surface, for acoustic intensity predictions. Their work was co-ordinated with experimental modal analysis. Limited modes were calculated and forcing functions were not used to determine operational sound levels. The present work follows similar lines with enhanced computational facilities enabling better model development and carrying out revisions within reasonable time.

**FINITE ELEMENT MODELING AND ANALYSIS**

Finite element analysis was performed using CAES® versions 4.1 and 4.2 by the Structural Dynamics Research Corporation (SDRC), on an IBM RISC 6000 workstation. Frequent references to the software manuals were required to understand element formulation and the theory on which the software code is based. At some points, test cases were run to confirm correct working of the code. Sometimes, alternative procedures or means of solution had to be adopted, as required options were not available with the software.

**Initial Housing Model**

The modeling approach involved the development of an initial housing FE model, which was successively refined based on engineering judgement, to obtain good correlation with experimental results. Model development of the housing posed serious difficulties in developing and meshing mesh-areas on a surface as complicated as the housing. The initial models were solved for natural frequencies and mode shapes. A free-free eigensolution was obtained with six kinematic degrees of freedom to prevent rigid-body modes. This simulated the experimental determination of response shapes, in which the feet were mounted on rubber pads. A consistent mass matrix was used and 11 modes were sought. It was observed that the frequency agreement between analysis and experiment was quite satisfactory; however two major differences between analysis and experiment were observed by comparing experimental laser scans with analytical mode shapes. The experimental response shapes showed a marked coupling between the housing top and side motion. However, the FE model showed one mode with independent top motion. Another effect was a restraining action by the lap-joint on the side motion, observed experimentally. This was not seen in the analytical mode shapes. At higher frequencies, experimental deflection shapes were very complex and made direct visual comparison difficult.

**Final Housing Model**

The initial housing model underwent a sequence of updating schemes. The number of elements in the model was thought to be adequate, by observing the mode shape wave patterns and so mesh refinement was considered unnecessary, at least, initially. Adjustments were made to the initial model based on physical reasoning. Chief model revisions included top-geometry refinement, addition of secondary masses, improved discretizations for side spring mounts and girth-joint modeling. These revisions resulted in a final model, shown in Figure 1, which was solved for its modal properties. The model was final, in the sense that agreement with natural frequencies and response shapes were adequate to proceed to the assembly model. The final housing model had the following features.

- Top geometry refinement with the flattened area for the top spring mount at the compressor top.
- A lumped mass element representing the top spring mount mass and inertia.
- Shell and lumped mass element combination for the electrical box.
- Girth-joint representation consistent with the welded lap joint.
- Shell and lumped mass element combination for the side mounts.
- Elements for the housing feet with their mass zeroed to smother feet resonances.

A summary of the natural frequencies up to 2000 Hz are listed in Table 1 along with the experimental frequencies for comparison. Figure 2 shows a sample mode shape of the housing at 651 Hz. After all the revisions to the housing model, it can be seen that there is good correlation for about 14 natural frequencies. Response shapes for some frequencies were correlated after forced response calculations.

On comparing the analytical mode shapes with the experimental response shapes, it was found that the modes corresponding to 1109 Hz and 1164 Hz were not found by experimental testing. Also, the independent top mode at 1193 Hz was not found experimentally. Only the mode shape at the bottom half of the housing matched for the 975 Hz mode.
Table 1. Final model frequency comparison

<table>
<thead>
<tr>
<th>Analytical frequency (Hz)</th>
<th>Experimental frequency (Hz)</th>
<th>Difference (%)</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>651</td>
<td>656</td>
<td>-0.08</td>
<td>Experimental mode on another housing</td>
</tr>
<tr>
<td>717</td>
<td>744</td>
<td>-3.62</td>
<td></td>
</tr>
<tr>
<td>740</td>
<td>748</td>
<td>-1.07</td>
<td></td>
</tr>
<tr>
<td>889</td>
<td>918</td>
<td>-3.16</td>
<td>Feet out of phase</td>
</tr>
<tr>
<td>975</td>
<td>1024</td>
<td>-4.79</td>
<td>Bottom matches</td>
</tr>
<tr>
<td>1109</td>
<td></td>
<td></td>
<td>Not found</td>
</tr>
<tr>
<td>1164</td>
<td></td>
<td></td>
<td>No match</td>
</tr>
<tr>
<td>1193</td>
<td>1229</td>
<td>-2.93</td>
<td>Rough match after forced response</td>
</tr>
<tr>
<td>1220</td>
<td>1280</td>
<td>-4.69</td>
<td>Rough match after forced response</td>
</tr>
<tr>
<td>1357</td>
<td>1315</td>
<td>3.19</td>
<td></td>
</tr>
<tr>
<td>1382</td>
<td>1333</td>
<td>3.68</td>
<td></td>
</tr>
<tr>
<td>1390</td>
<td>1408</td>
<td>-1.28</td>
<td></td>
</tr>
<tr>
<td>1467</td>
<td>1464</td>
<td>0.89</td>
<td>Error mode</td>
</tr>
<tr>
<td>1558</td>
<td></td>
<td></td>
<td>Rough match</td>
</tr>
<tr>
<td>1616</td>
<td>1609</td>
<td>0.44</td>
<td>General match after forced response</td>
</tr>
<tr>
<td>1644</td>
<td></td>
<td></td>
<td>No match - complex</td>
</tr>
<tr>
<td>1683</td>
<td>1674</td>
<td>0.54</td>
<td>General match after forced response</td>
</tr>
<tr>
<td>1757</td>
<td>1708</td>
<td>2.87</td>
<td>General match after forced response</td>
</tr>
<tr>
<td>1744</td>
<td></td>
<td></td>
<td>Complex</td>
</tr>
<tr>
<td>1812</td>
<td>1854</td>
<td></td>
<td>Complex</td>
</tr>
<tr>
<td>1908</td>
<td></td>
<td></td>
<td>Complex</td>
</tr>
<tr>
<td>1943</td>
<td></td>
<td></td>
<td>Complex</td>
</tr>
<tr>
<td>1966</td>
<td></td>
<td></td>
<td>Complex</td>
</tr>
<tr>
<td>1987</td>
<td></td>
<td></td>
<td>Complex</td>
</tr>
</tbody>
</table>

The experimental frequencies corresponding to the FE frequencies at 740 Hz and 889 Hz were observed as twin peaks with the same mode shape on the housing surface but differing feet motion. This is due to the zeroing of the feet-element masses in the analytical model. Also observed at 1467 Hz is an error mode with erratic displacements of some elements. With increasing frequencies, correlation became increasingly difficult because of the complex mode shapes. This raised the question of modal combination in the experimental mobility scans. Forced response analysis was performed on the housing and resulted in better correlation as seen in Table 1. Other factors are also responsible for errors, viz., element formulation, mass and inertia properties of secondary masses, mass and stiffness matrix formulation, boundary conditions, local thickness and material property changes, approximations to structure-geometry, assumptions made about the nature of damping, etc. A significant manufacturing variation was housing thickness changes resulting from the drawing process. Kelly [7] conducted detailed studies on this variation, with thickness measurements at several points on the surface, and found some increase in certain frequencies.

**EXPERIMENTAL RESULTS**

Experimental modal analysis included impact testing for the determination of natural frequencies and laser scanning for obtaining visual response shapes. Modal testing was performed on the compressor housing. FRFs were determined by impact testing with a hammer, the acceleration response being measured by an accelerometer. Natural frequencies were
identified. Next, response shapes were determined experimentally, by shaking the structure with a harmonic excitation force generated by a shaker. Velocity response was recorded by scanning the structure with a laser doppler vibrometer which captured the mobility response of the structure. These mobility scans were obtained at all natural frequencies (below 2000 Hz) for the housing. These mobility scans do not represent the pure mode shapes, but only the operating shapes which are closer to the true mode shapes if coupling due to adjacent modes is negligible.

Results of experimental testing on the housing revealed 27 natural frequencies in the 0-2000 Hz range. Impact testing on five housings revealed an overall standard deviation of 14.8 Hz in the natural frequencies; mobility scans were obtained on a housing whose natural frequencies were statistically close to the average group of frequencies. However, the experimental response shapes were not averaged over the five samples due to the time involved in obtaining the mobility scans. Therefore, the average group of experimental frequencies and the response shapes from one housing whose natural frequencies were close to the average group of frequencies formed the database with which the analytical results would be compared. Response shapes of the housing showed three prominent characteristics
- a pronounced constraining effect by the girth joint, observed at all frequencies; more so at higher frequencies. These are seen in Figure 2 as a dip in the response amplitudes near the middle of the housing sides.
- visible coupling between housing top and side motions at isolated groups of frequencies and
- complex housing response at higher frequencies, particularly at the lower half of the housing, near the location of the side mounts.

MODELING OF THE COMPRESSOR ASSEMBLY

With the housing model fully developed, the compressor assembly model was then constructed with additional components, viz., the shockloop, mounting springs and the compressor, and solved by component mode synthesis (CMS). Individual components solved for natural frequencies in the 0-2000 Hz range were assembled to form the full compressor. Frequencies over 2000 Hz were represented by a residual flexibility matrix. With the CMS model, strain energy contributions from individual components could be calculated.

The compressor is assumed to be rigid in comparison with the housing. Hence, the compressor was modeled as a lumped mass, whose mass and inertia properties were determined from a detailed solid model of the compressor internal components, and a subsequent assembly of the individual components to create an assembled unit [8].

The shockloop is a discharge tube that connects the muffler to the discharge outlet on the housing. Using beam elements of appropriate cross-section to model the shockloop, ten natural frequencies were found with free-free boundary conditions in the 0-2000 Hz range.

Forces generated by the internal mechanism are transmitted by the top and side springs to the housing. It was not sufficient to represent these springs by single spring elements in the compressor assembly model, for a quick calculation revealed that both the top and side springs had about four surge frequencies in the 0-2000 Hz range. Torsional and bending modes could also be present. Considerable effort was made to minimize the dofs, so that the final assembly would have considerably less dofs than would otherwise be the case, thus avoiding long solution times, while ensuring conservative use of solution space.

In component assembly, individual models like the housing, the shockloop and the springs, which were already solved for natural frequencies with free-free boundary conditions, were assembled into the full compressor model, shown in Figure 3. All six dofs had to be defined at the connection nodes of the components to ensure that all the modes of the full assembly were extracted. The internal compressor unit was represented by a rigid-body component with mass and inertia properties determined from the solid model.

Normal Modes of the Compressor Assembly

Normal modes of the compressor assembly were calculated by a similarity transformation method. The interaction of all the components resulted in an approximate quadrupling of modal density. Eighty seven natural frequencies were found below 2000 Hz. The compressor assembly modes were classified by Kelly [7] as compressor suspension modes, modified housing modes, internal surge modes and coupled modes.
A compressor assembly mode shape at 897 Hz is shown in Figure 4. The first six modes, discounting the rigid-body modes, were the suspension modes of the compressor mechanism on the mounting springs and ranged from 7.7 Hz to 29.3 Hz. They were not pure translational or rotational modes of the internal unit, but a combination in which one of the motions is dominant. Internal surge modes represent the motion of the mounting springs and the shockloop, with little or no activity in the housing. Some surge modes were seen in only one of the mounting springs, with almost negligible housing motion. There were other modes, in which a spring surge had a local effect on the housing deformation. For example, there were modes in which top spring surge was coupled to the housing-top motion and side spring bending modes were accompanied by housing-side motion. Other spring surge modes were coupled to one or more housing modes. Strictly speaking, any internal surge mode is also accompanied by some housing motion. Strain energy distributions were used to decide the primary location and nature of modal response. At low frequencies, some internal surge modes coupled only to one housing mode. However, at higher frequencies, several housing modes coupled to several internal surge modes, resulting in new mode patterns. This was probably due to the density of the housing natural frequencies above 1000 Hz. A third set of natural frequencies showed mode shapes which were similar to those of the housing mode shapes, but occurred at slightly different frequencies compared to the corresponding housing frequencies. No general shift pattern was observed; frequencies shifted above and below the corresponding housing frequencies. Some housing mode shapes were either not found in the assembly model, or were weakly coupled to deflections of mounting components, and hence were not detected. At higher frequencies, there were new mode shapes due to the increased modal density of the individual components. Such mode shapes were complex, and showed that the mounting spring components severely affected housing motion in these high frequencies.

CONCLUSIONS

Finally, it may be worthwhile to note the following points regarding the development of the compressor model for sound prediction.

(i) For noise prediction applications, satisfactory correlation of natural frequencies and mode shapes for the housing model was obtained.
(ii) Guidelines for solid modeling of internal components were established.
(iii) Procedures for reducing degrees of freedom in mounting spring components were established.
(iv) The components were assembled into the compressor assembly whose natural frequencies and mode shapes were obtained by CMS.
(v) Verification of assembly frequencies and mode shapes with experiment will give confidence in the assembly model's modal properties.
(vi) Forcing functions can be used as excitations for the assembly model and velocity response at the surface of the housing can be evaluated.
(vii) Velocity response data are directly used in sound prediction by the boundary element method.
(viii) Subsequent redesigns can be tested and investigated using the compressor assembly model.

ACKNOWLEDGEMENTS

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REFERENCES


Figure 1. Final housing model

Figure 2. Housing mode shape at 651 Hz

Figure 3. Compressor assembly model

Figure 4. Compressor assembly mode shape at 897 Hz