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Virginia Polytechnic Institute and State University

C. E. Knight
Virginia Polytechnic Institute and State University

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DYNAMIC FINITE ELEMENT MODELING AND ANALYSIS OF A HERMETIC RECIPROCATING COMPRESSOR

A. D. Kelly and C. E. Knight
Virginia Polytechnic Institute and State University
Dept. of Mechanical Engineering
Blacksburg, VA 24061

ABSTRACT

Finite element modeling of a refrigeration compressor was investigated as part of a noise prediction study. Structural modes and the corresponding natural frequencies were calculated for major components of the compressor. The components were later combined to form a model of the compressor assembly which was subsequently solved for its dynamic properties. Model development included coordination with test data for verification and revision to improve model prediction accuracy. A history of numerous modeling updates, made to improve analytical-experimental agreement, established many critical factors necessary to generate a good model.

Considerable efforts were made to accurately represent the hermetic shell which presents several inherent modeling difficulties. The importance of physical simplifications such as geometry representation, thickness variation, attachments, the welded seam, and residual stresses were established. In addition, theoretical limitations of the finite element method were addressed as a cause for analysis-test discrepancies. Analytical normal modes were visually compared to test data with good but incomplete correlation. Forced vibration response calculations became necessary to simulate the experimental resonance dwell excitation procedure. This improved the analysis-test agreement by showing mode shape combination of closely spaced modes.

The assembled compressor model included a rigid mass compressor mechanism, suspension springs, and discharge tube. Analysis showed that interactions with the internal components, particularly resonances within the suspension springs, are important for a valid representation of the compressor assembly. Resonances within the internal suspension components more than double or nearly triple the number of resonance frequencies in the compressor assembly.

INTRODUCTION

Noise emission from refrigeration equipment continues to be a concern in the compressor industry. Over the years, structural dynamics analysis, both analytical and experimental, has provided an understanding of compressor vibration and the related noise emission characteristics. Dynamic modeling has been of particular interest, because it promises to give engineers the ability to efficiently evaluate design changes or predict the characteristics of new compressor designs prior to construction. During the past two decades, compressor modeling efforts have been approached by classical shell theory, the transfer matrix method, and the finite element method. This work has led to improved designs, however, few have addressed the question of modeling accuracy and reasons for deviation from test data. This paper seeks to clarify the area of compressor model development and experimental correlation.

Recently, Bristol Compressors, Inc. initiated a research project with the long-term goal of developing or adapting the technologies necessary for designing quieter compressors. The project was approached by the coordinated efforts of acoustics research, experimental dynamics analysis, and the finite element method (FEM). In regard to the finite element efforts, this study seeks to develop an FEM model for use in noise prediction applications.
The model has three specific roles; (1) to predict the modal properties (eigenvalues/eigenvectors) of the major compressor components, (2) to combine the components into a system representing the compressor assembly, and (3) to calculate the forced vibration response of the compressor assembly to mechanical and acoustic forcing functions. Experimental dynamics analysis verifies and guides development of the FEM model. Ultimately, acoustic modeling will calculate sound intensities from the FEM calculated velocities of radiating surfaces.

This paper will focus on the FEM modeling aspects of this project. Development of a compressor housing model is discussed first, since obtaining good performance from the housing is essential before adding assembly components. In addition, the housing plays a significant role in noise radiation so it warrants considerable attention. A history of model updates, made to improve agreement between the analytical and test data, are summarized to aide future modeling efforts. Next, a compressor assembly model is presented. Changes which occur upon "loading" of the housing are emphasized in the discussion. Selected results which include modal properties and forced vibration response are reported for both the housing and assembly models.

HOUSING FINITE ELEMENT MODEL

Hermetic compressors possess several features which complicate analytical model development. Most designs involve top and bottom sections that are welded together with a single lap joint, referred to as the girth, which is difficult to represent analytically. Next, service connections may add significant additional mass to the compressor and require local geometry modifications to incorporate. Finally, since the housing sections are formed in a deep-drawing process, the housing may have regions with local yielding, thickness changes, and geometric variations different from the design specifications and from unit to unit. Even with the capabilities of FEM, features such as this are troublesome to include and simplifications must often be made.

Figure 1a shows the initial housing model of this study. Linear shell elements form smooth and symmetric surfaces with a uniform average thickness which is less than stock material, however, it does not include the regional variation known to exist in the housing thickness. A band of thick elements was used to represent the girth. The bottom mesh conforms to the surface recesses which incorporate the mounting feet. Nodes in the bottom are located at points corresponding to the spot welds used to attach the feet. No other accessories or geometric attributes were included.

![Figure 1. Housing Finite Element Model Comparison](image_url)
Results of the initial model disagree with the test data somewhat, as shown in Table 1. Natural frequency differences range from 45 to 532 Hz (4.6 to 56 %) of test values. The model also predicts an independent top mode rather than the opposite phase top-side coupling as found experimentally. In addition, the experimental housing shape found when driving the housing at 1069 Hz was not predicted.

Table 1. Housing Model Normal Mode Comparison

<table>
<thead>
<tr>
<th>Mode</th>
<th>Initial Housing</th>
<th>Updated Housing</th>
<th>Test</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Freq., Hz</td>
<td>Freq., Hz</td>
<td>Freq., Hz</td>
</tr>
<tr>
<td>M8(3,1)</td>
<td>737 (+7.0 %)</td>
<td>662 (-3.9 %)</td>
<td>689</td>
</tr>
<tr>
<td>M9(3,1)</td>
<td>802 (+9.0 %)</td>
<td>724 (-1.6 %)</td>
<td>736</td>
</tr>
<tr>
<td>M10(2,1)</td>
<td>843 (+6.6 %)</td>
<td>772 (-2.4 %)</td>
<td>791</td>
</tr>
<tr>
<td>M11(4,1)</td>
<td>1029 (+4, +9 %)</td>
<td>874 (-7.4, -11 %)</td>
<td>944 (top in) 984 (top out)</td>
</tr>
<tr>
<td>M12(0,0,top)</td>
<td>1476 (+56,+50 %)</td>
<td>967 (+2.4, -1.7 %)</td>
<td>944 (top in) 984 (top out)</td>
</tr>
<tr>
<td>M13(4,1)</td>
<td>1119 (+8.4 %)</td>
<td>971 (-5.9 %)</td>
<td>1032</td>
</tr>
<tr>
<td>M14(2,1,top)</td>
<td>1135</td>
<td>1102</td>
<td>1069 shapes differ</td>
</tr>
</tbody>
</table>

1 Modes are described by an (n,m) type descriptor, where n corresponds to the circumferential deflection pattern and m corresponds to the axial deflection pattern.
Updajjnq Finite Element Models

To guide model development of this study, an experimental database was available for the housing and the compressor assembly [1]. It included frequency response functions (FRFs) between various points on the housing and laser-based resonance deflection shapes for visual comparison. Adjustments made to the model follow (1) a philosophy which emphasized physical insights to justify changes, and (2) a desire to generalize updates for applicability to other modeling ventures. For example, a visual comparison of modes may indicate a portion of the housing that is too stiff. Rather than arbitrarily reduce the elastic modulus in the stiff region to force agreement, a physical cause such as excessive geometric curvature, local mass loading, or a too coarse mesh was investigated. The results of each update was accepted. Updates are classified into two areas: those targeting physical modeling differences and those targeting theoretical limitations of FEM.

Physical Updating

Modeling errors are differences between the true physical system and the analytical representation. Troublesome areas in compressor modeling were found to be (1) geometric variations, (2) the girth joint representation, (3) local mass loading, (4) material property difference, and (5) realizing loads and restraints. Table 2 summarizes the physical updates and shows the natural frequency sensitivity to each change.

All of the physical differences were not resolved by this study, but some insights were found that are worth sharing. Geometric variations are felt to be most important but a note of caution should be added: correction by arbitrary adjustment of mesh geometry may cause errors related to non-planar nodes in quadrilateral elements. Next, the girth joint is also thought to be critical and better representations should be investigated. For instance, one may consider using an effective modulus in the girth which would be determined from separate detailed modeling of the joint. Test data showed a significant restraint by the girth, similar to a "waistline". This could be a constraining effect by the weld bead. Loads are obviously not applied to the eigenvalue model, but restraints are a concern. This study considered mostly free-free boundary conditions which resulted in a large number of mounting feet resonances. These became a computational burden so they were eliminated from the frequency range of interest by decreasing the value of their mass density. Finally, material property values were not verified by tests. It is possible that local residual stresses create regions which no longer display a linear elastic modulus but respond with a reduced, tangent modulus. However, in this study, we did not find evidence of this happening.

Table 2. Model Updating Significance Summary

<table>
<thead>
<tr>
<th>Parameter</th>
<th>$\Delta \omega_n$</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geometric variations</td>
<td>-24 to -36 %</td>
<td>• Local flattened spot on top</td>
</tr>
<tr>
<td></td>
<td>-1.9 to -5.4 %</td>
<td>• Less curvature of actual top due to</td>
</tr>
<tr>
<td></td>
<td></td>
<td>manufacturing variance</td>
</tr>
<tr>
<td>Coincident elements at</td>
<td>-17 to -8.8 %</td>
<td>• Improves test-analysis agreement</td>
</tr>
<tr>
<td>girth</td>
<td></td>
<td>• Double thick elements too stiff</td>
</tr>
<tr>
<td>Local Masses</td>
<td>-4.0 to -8.9 %</td>
<td>• Mass loading effect most significant</td>
</tr>
<tr>
<td></td>
<td></td>
<td>if located at an antinode</td>
</tr>
<tr>
<td>Thickness variations</td>
<td>-0.3 to +1.4 %</td>
<td>• Girth with minimal thickness change</td>
</tr>
<tr>
<td></td>
<td></td>
<td>dominates, top effect not isolated</td>
</tr>
<tr>
<td>Residual Stress</td>
<td>0 %</td>
<td>• Not a factor, confirmed by stiffness</td>
</tr>
<tr>
<td></td>
<td></td>
<td>comparison and annealed housing tests</td>
</tr>
</tbody>
</table>
Theoretical Limitations of FEM

The most important theoretical limitations are due to varied formulations of the structure mass matrix, \( [M] \), and the structure stiffness matrix, \( [K] \), which define the eigenproblem, and in the algorithm chosen for eigenvalue extraction. For users of commercial software, the particular solution method and lumped- or consistent- mass matrix formulations can often be chosen. However, many theoretical factors may be buried deeply in the code with only limited documentation. In this study, eigensolutions of identical housing models by different codes, or by later versions of the same code, found natural frequency differences of more than 20%.

Other theoretical concerns involve good modeling practices. For example, numerical problems are handled by the code but the analyst can limit them by avoiding large stiffness variations in the model. The analyst is also responsible for avoiding discretization errors that result from modeling a continuous system with only a finite number of elements. For example, distorted elements are stiffer and have reduced accuracy. In addition, mesh refinements made by further element subdivision or increasing element order assure convergence of the solution. When results change little because of refinement, one has evidence (but not proof) of convergence. Additional mesh refinement becomes necessary to solve for higher modes.

The updated housing model is shown in fig. 1b although most refinements are not apparent. Eigensolution of the updated model is summarized in Table 1. The updated natural frequencies tend to agree with test data better, particularly the top mode. However, the top mode was independent in the FEM model but coupled to side motion in the experimental tests which limits correlation.

Housing Model Forced Response Analysis

In the previous sections, analytical mode shapes were compared to experimental resonance deflection shapes. The fundamental test approach, which is fully documented by Agee, et. al. [1], is to record the deflected housing while driving it at a natural frequency. Forced response analysis enables the modeler to simulate this test and insure compatibility of the data to be compared.

A mode superposition method is used to calculate forced-vibration response [2]. Mathematically, the structural equation of motion is transformed from physical coordinates to modal coordinates using the calculated eigenvectors. Orthogonal properties of the eigenvectors uncouple the structural equation into a set of independent second-order differential equations. At this point, viscous damping ratios were defined in modal coordinates. Each modal equation is then solved for the modal coordinate, or modal participation factor \( y \). The final step is to transform back to physical coordinates, which become a sum of the mode shapes weighted by the modal participation factors.

The models were excited by a unit sinusoidal excitation at a point on the girth which corresponds to the test excitation point. Forced vibration was found in the form of frequency response functions (FRFs) at a single point, which will be shown later, and response of the full structure. As expected, forcing specifically at a natural frequency usually resulted in a deflected shape similar to the corresponding mode shape. However, occasions exist when the resonance response shape differed from the pure mode shape. One notable case, shown in fig. 2, was the forced response shape at 967 Hz. This is the frequency of an independent top mode, but it shows a forced response including significant side motion from a nearby natural frequency as was seen in the experimental testing. In another case, the predicted shape for off-resonance frequencies in the 1040 to 1069 Hz range agreed with the experimental operating shape corresponding to the "shape differs" mode in Table 1. Overall, the forced response effort improved the model's agreement with test data.
Figure 2. Forced Housing Response to a Unit Sinusoidal Force at 967 Hz. Note that the top mode, M12(top), is not excited independently.

COMPRESSOR ASSEMBLY FINITE ELEMENT MODEL

The compressor assembly is modeled by combining the updated housing with a compressor mechanism, suspension springs, and discharge tube sub-models as shown in fig. 3. Helical suspension springs were represented by a series of beam elements to include surge effects, which are discussed by Kelly and Knight [3]. The discharge tube, or shockloop, was modeled by beam elements as discussed by Bernhard [4]. The internal compressor mechanism is represented by a lumped mass element since it is comparatively more rigid than the housing. Properties such as mass, mass moments of inertia, and center of gravity were calculated by a geometric solid model of the mechanism's major components. Fluid-structure interactions and internal pressurization were not included in the modeling efforts presented here.

Figure 3. Compressor Assembly Finite Element Model
Next, normal modes were calculated for the compressor assembly. The addition of internal surge effects by the springs and shockloop results in a dramatic modal density increase. Kelly and Knight [3] list about forty (flexible) assembly modes for the 0-1100 Hz frequency. For comparison, the housing model has seven flexible modes in this range. The compressor assembly modes may be grouped into four categories; (1) compressor suspension modes which describe low-frequency oscillation of the internal compressor mechanism on the mounting springs, (2) modified housing modes which correspond to empty housing modes but may show frequency reductions and interactions with the internal components, (3) internal resonances that are inconsequential, and (4) internal resonances that result in significant housing flexure. Figure 4 shows a typical assembly mode resulting from an internal spring surge. Every surge mode results in some amount of housing motion, which can be seen by the exaggerated view. The configuration of this mode is unique to the assembly and shows the importance of including spring surge effects.

![Top view](image1)

![Top view exaggerating housing motion](image2)

**Figure 4. Assembly mode with a side spring surge corresponding to 731 Hz.**

Many of the assembly modes, particularly the modified housing modes, were verified by test data. In addition, the modal density increase predicted by the assembly model agrees well with test data, when the externally visible modes are considered. Experimentally, the modal density approximately doubled upon loading. For comparison, Figure 5 shows an FRF generated by the assembly model over-laying an equivalent FRF of the housing. The internal resonances which result in significant housing flexure are apparent in the assembly FRF, while the inconsequential resonances are not visible or appear only as "glitches" on the side of major housing mode peaks using the current damping values.
CONCLUSIONS

A hermetic reciprocating compressor was analyzed by dynamic finite element modeling as part of a noise prediction project. Considerable efforts were made to improve test-analysis agreement during model development. The model adjustment approach taken was generally applicable to other compressors and a history of updates was summarized to benefit other modeling ventures. We identified some modeling parameters which may be simplified without introducing large errors, and conversely, some parameters which cannot. Specifically, the compressor housing was found to be most sensitive to geometric variations, the girth joint representation, and local masses; less sensitive to thickness variations and residual stresses. Surge by internal components was found to affect the dynamic characteristics of the compressor assembly, and are thus felt necessary to include when modeling. In addition, theoretical parameters such as finite element formulations and solution methods affect natural frequency calculations significantly and may be beyond the analyst's control. Forced response analysis (1) provided additional response parameters for direct comparison to test data, (2) provided compatible data when pure modes are influenced by adjacent modes, and (3) improved experimental agreement. Finally, test-analysis differences will always exist to some degree. As the noise prediction application proceeds, we hope to establish an agreement criteria. Present work seeks to extend the model's frequency range and improve the velocity response accuracy which is needed for sound intensity calculations.

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