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NOISE AND VIBRATION CHARACTERIZATION
AND STATISTICAL ENERGY ANALYSIS OF A SCROLL COMPRESSOR

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Statistical Energy Analysis (SEA) is used to diagnose the operational dynamic characteristics of a scroll compressor. The SEA model is benchmarked against test data for external force as well as internal pressure pulsation excitation (with air). Sound power contributions from various regions of the operating compressor are determined by extrapolation of measured surface vibration levels. These results are compared with SEA predictions for internal force and pressure pulsation excitation sources. The distinct characteristics of the two sources, revealed by the SEA model, help separate them in the operating machine -- which is otherwise difficult to do because they are highly correlated. This information is useful for noise control, since the effectiveness of potential solutions is different for each source. Design modifications are evaluated using the SEA model.

INTRODUCTION

Compressor noise control can be particularly challenging because of the presence of multiple coherent excitation sources. In scroll compressors, discharge pulsations, scroll flank forces, and coupling forces excite to some degree all harmonics of rotation. The fact that these sources are correlated makes it very difficult to separate their effects by test. The high frequency range of concern, to 4 KHz and beyond, limits the practicality of traditional dynamics models such as Finite Element Analysis (FEA) or Boundary Element Analysis (BEA). Given that different methods are required to control each source and the structural response to them, achieving low noise in a cost-effective manner can be difficult. Statistical Energy Analysis (SEA) offers a practical means of gaining insight into the process of scroll compressor noise generation which can be applied to refining the design.

STATISTICAL ENERGY ANALYSIS MODEL

SEA is a relatively new method for calculating the spatial- and frequency-averaged flow of dynamic power and the resulting distribution of response levels throughout a system (1). Both vibratory and acoustic power can be included in the same model. Compared to 'discrete' methods like FEA or BEA, SEA has the advantage that it can be applied to (and in fact becomes increasingly accurate at) high frequencies. More importantly, SEA models of even very complex systems comprise relatively few elements and model computation time is almost negligible (2,3). The disadvantages of SEA are its limited applicability at low frequencies and the subjective nature of the modeling process which derives from the simplicity of the model. Further, the SEA result being in terms of average responses means that individual resonances and mode shapes are not calculated. As a result, SEA is often most effectively applied to trend analysis, which can be particularly useful early in the design process, as opposed to detailed prediction of response peaks.

A 10 ton scroll compressor and the SEA model are shown schematically in Figures 1a and 1b, respectively. The model is comprised of 38 'subsystems', each representing a particular class of mode (resonance) group. These include plate-like bending, pipe-like bending, in-plane compressional and shear, and both internal and external acoustic mode types. SEA calculates the average power flow and distribution throughout this system, equating input power (excitation) with power dissipated (structurally and acoustically). The result is average vibration or pressure levels in general regions of the system.
Two types of excitation were applied to the model for the present study. One was internal scroll forces, which represent a combination of scroll flank contact and coupling sources. These forces were applied to the in-plane modes of both the fixed and orbiting scrolls. The other excitation mechanism was pressure pulsation applied to the acoustic mode group in the discharge cavity. The analysis was done using the SEAMI® program (4).

MODEL VALIDATION: NON-OPERATIONAL TESTS

Damping loss factors for the model were assigned based on decay-rate measurements on the test compressor, as well as by experience. The damping levels at the top of the unit and at the lower shell (see Figure 1b) are presented in Figure 2. It was found that the former possesses relatively low damping as compared to the latter.

Where practical, drive-point impedance measurements were used to validate key subsystems in the model. Figures 3a and 3b compare various impedance measurements on the top of the unit and on the lower shell, respectively, to those calculated for the corresponding SEA mode groups. At the top, the model represents the test average reasonably well above 1500 Hz, although sharp peaks, due to the low damping level, are seen. Below 1500 Hz, a lack of modes causes the model to increasingly under-predict the top impedance. This is typical of SEA, as the theory is based on transmission of power between groups of resonances. The lower shell impedance prediction is quite close to test, owing to that area's higher modal density and damping. Also shown is the impedance measured in close proximity to where the shell contacts the fixed scroll casting, indicating the (also typical) finding that the SEA in-plane impedance is slightly high. This is due to the difficulty of measuring in-plane impedances in practice, as the much softer bending modes tend to mask the in-plane component.

To validate the model's internal acoustic coupling to vibration, an air test was run with a loudspeaker driving the discharge cavity. Figure 4 shows measured and predicted vibration levels per unit of internal pressure for the top and the lower shell. Overall, the prediction of response at the top is reasonable to a surprisingly low frequency, given its relatively low modal density. The model also provides a good prediction of the 20-25 dB difference between top and lower shell vibration at and above the 2000 Hz band. Individual resonances in the test compressor cause the model (limited to averages across modes) to underpredict in the 1600 Hz band. Below 1300 Hz, due either to low modal density or subtleties in structural coupling across the high-impedance fixed scroll casting, the model tends to over-predict lower shell vibration for this excitation source.

OPERATIONAL SOUND POWER, VIBRATION, AND DISCHARGE PULSATION

In addition to standard ARI-530 reverberant room sound power tests, external vibration measurements were used to characterize the operational behavior of the unit. Although radiation efficiencies deviate increasingly from unity at low frequencies, it is useful to compute the power which would be radiated from the main surfaces assuming perfect efficiency, based on their average vibration level:

$$\Pi_{\text{rad}} = \sigma \rho c A \langle v \rangle^2$$

where:
- $\sigma$ is the radiation efficiency (assumed to be 1 for the present study)
- $\rho$ is the mass density of the fluid
- $c$ is the wavespeed of the fluid
- $A$ is the surface area
- $\langle v \rangle$ is the rms velocity level over surface $A$

One such result is shown in Figure 5, which compares vibration-extrapolated radiation from the top with that from the lower shell. The total (rss) of these is also compared with the measured sound power level, showing very good ± 2 dB agreement from 500 to 8000 Hz. The vibration extrapolation indicates that: 1) lower shell radiation dominates the 630 Hz band and below, 2) the top dominates the 1.25K and 1.6K bands, and 3) the top and lower shell are of comparable strength at and above 2 KHz. This information can be useful in noise control.
To gain further insight, we turn to the SEA model. In evaluating this model relative to test, the central question is how to separate the force and pressure excitation mechanisms analytically. One revealing approach is to back-calculate the excitation which would be required to match the measured lower shell vibration, assuming that each source were acting alone. The vibration results for the scroll force case are shown along with the test data in Figure 6a. With this excitation source, the SEA model significantly under-predicts the top vibration level across most of the 1 - 4 KHz range.

Figure 6b shows how the model responds to internal pressure excitation, again in the amount sufficient to match lower shell vibration. For this case, the model significantly over-predicts top vibration.

The results for the two excitation cases show that at least two significant sources are present, each of which affects the compressor structure differently. To validate this finding, operational discharge pulsation levels were measured directly. To obtain a stable result, the rms of three transducers was obtained for comparison with the SEA model. Figure 7 plots the measured data, along with the above-mentioned SEA pressure back-calculated from measured lower shell vibration, and the reverberant SEA level calculated by the model given scroll force-only excitation. The fact that the test data lie between the two SEA results is consistent with two sources being present: the operational pressures are not sufficient to drive the structure to the measured lower shell vibration levels, but are significantly above those which would result from force-only excitation.

Since discharge pressure has a relatively small effect on lower shell vibration (as compared to its effects on the top), it is acceptable to combine the two sources by applying to the SEA model the measured discharge pressures and the lower shell back-calculated scroll forces simultaneously. The result is shown in Figure 8. This excitation case matches the actual relationship between top and lower shell vibration very well from 1 to 8 KHz.

From the model one may conclude that, in this compressor, top vibration is strongly influenced by discharge pulsation, while lower shell vibration is controlled by force-type sources.

**Predicted Effects of Design Modifications**

Noise reduction alternatives evaluated by the SEA model include various source control and structural modification concepts. The following results were obtained for two structural modifications considered:

<table>
<thead>
<tr>
<th>Modification</th>
<th>Excitation Source</th>
<th>L_w change, dBA</th>
</tr>
</thead>
<tbody>
<tr>
<td>2x top thickness</td>
<td>discharge pressure only</td>
<td>-5.5</td>
</tr>
<tr>
<td></td>
<td>scroll forces only</td>
<td>-0.5</td>
</tr>
<tr>
<td></td>
<td>combined pressure &amp; scroll forces</td>
<td>-2</td>
</tr>
<tr>
<td>add 13 lb. to fixed scroll assembly</td>
<td>discharge pressure only</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>scroll forces only</td>
<td>-1.5</td>
</tr>
<tr>
<td></td>
<td>combined pressure &amp; scroll forces</td>
<td>-0.5</td>
</tr>
</tbody>
</table>

Given the appreciable differences between the predicted modification effects for each source, effective noise optimization of the compressor, in the face of strict cost constraints, requires that both the dynamic system and all significant excitation mechanisms be reasonably well modeled.

**Conclusions**

SEA was used to separate the effects of coherent internal pressure and scroll force-type excitation in an operating scroll compressor. Using the SEA system model which takes account of both, potential noise reduction alternatives were evaluated analytically as part of the design process. A particularly attractive aspect of SEA is the minimal design definition and computational resources required, so general design direction, even in the concept stage, can be given.
REFERENCES


Figure 1a: Scroll Compressor

Figure 1b: SEA Model of Scroll Compressor

Figure 2: Damping Loss Factors from Decay Rate Testing vs. SEA Model for Top and Lower Shell Bending Modes
Figure 6a: Exterior Vibration, Test vs. SEA with Internal Scroll Force - Excitation to Match Vibration of Lower Shell

Figure 6b: Exterior Vibration, Test vs. SEA with Internal Pressure - Excitation to Match Vibration of Lower Shell

Figure 7: SEA vs. Test Discharge Pressure, Pressure and Scroll Force Excitations to Match Lower Shell Vibration

Figure 8: Exterior Vibration, Test vs. SEA with Two Sources - Pressure (Measured) and Scroll Forces (to Match Lower Shell Vibration)