Improving Screw Compressor Housing Design Using Simulation

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The sound radiation from a rotary-screw compressor was simulated using structural finite-element and acoustic boundary-element analysis. The modes of the housing were measured with and without the screws. Finite-element modal analysis results were compared to the experimental modes with good agreement. Results demonstrated that the internal components affected the bending modes but not the cylinder modes of the compressor housing. The sound radiation from the compressor housing was then predicted using acoustic boundary-element analysis. The results indicated that the second and fourth pumping frequencies were close to structural modes. Subsequent simulations investigated the effect of design changes on the housing via a thicker cylinder and the addition of ribs.

Over the past two decades, the role of numerical simulation in noise control engineering has expanded. Though numerical simulation is no substitute for measurement, simulation can reduce the number of expensive prototypes and associated costs of bringing products to market. Simulation is best used in conjunction with measurement, since it rarely predicts the absolute noise level for a product. After a satisfactory numerical model has been obtained and correlated with measured results, simulation is most properly used as a guide to suggest design changes and their impact on noise relative to other design options.

In the work reported here, a Johnson Controls Inc. screw compressor was investigated. The simulation model of the compressor was first validated experimentally. Simulation results were compared to measurements to adjust the model. The model was then used to simulate the acoustic performance for four redesigns. Since the compressor is modally sparse, the finite- and boundary-element methods (FEM and BEM) were used for the simulation. FEM was used to model the compressor structure.

Developing a structural FEM model of a compressor is challenging, since components inside the compressor housing like the screws, motor, and stator are difficult to model. Accordingly, we chose to model the compressor housing alone, neglecting the screws, motor, and stator. An experimental modal analysis was used to determine the effect of removing these components. The experimental modal also provided necessary information regarding the damping properties of the compressor.

A subsequent BEM analysis was used to predict the sound radiation from the compressor. The BEM is a numerical approximation used to solve the Helmholtz equation (the governing differential equation for linear acoustics in the frequency domain) and is well documented in the literature.[1,2] Though FEM can be used to model the acoustic domain, BEM models are easier to construct, since only the boundary of the acoustic domain needs to be discretized instead of the acoustic domain itself. Additionally, the Sommerfeld radiation condition is automatically satisfied so that the exterior domain need not be bounded.[1,3]

This article documents the validation and the successful application of the model to drive design changes for the rotary-screw compressor. The model was used to investigate the effectiveness of thickening and adding ribs to the compressor housing to reduce radiated noise.

Modal Analysis

The screw compressor investigated is shown in Figure 1. Experimental modes were obtained from a "static" (non-running) compressor using impact testing. As the figure indicates, the screw compressor was isolated using air mounts. Two separate modal analyses were conducted. For the first, the modes for the *in situ* compressor (screws, motor, and stator intact) were identified. The modal analysis was then repeated for the compressor housing alone (screws, motor, and stator removed). The screw compressor was excited at locations normal to the top surface and to the side (showing the label) surface. Data were collected on the top and side surfaces using an accelerometer at 41 and 36 positions respectively.

A FEM mesh (Figure 2) of the YCAV compressor housing was created for dynamic analysis from a defeatured solid model developed using Pro/ENGINEER. The mesh was created using linear tetrahedral elements and then transferred to ANSYS. Inside ANSYS, mid-side nodes were added converting the elements to parabolic. The two separate parts of the compressor housing were assumed to be connected together rigidly. Other options were attempted, such as using constraint equations at each bolt location. However, the predicted modes did not compare as closely to the measured results. This is not surprising, since the two sections of the compressor housing are connected tightly with 19 bolts. Material properties were adjusted slightly so that the FEM model correlated with the measured results. The elastic modulus, mass density, and Poisson's ratio were 135 GPa, 7190 kg/m^3, and 0.25 respectively.

Comparisons of the experimental mode shapes with the results obtained using FEM simulation are summarized in Table 1. There was generally good agreement between FEM simulation and the measured cylinder modes (Modes 4, 5, 6, and 12), suggesting that the screws, motor, and stator have negligible impact on these modes. However, the measured modal frequencies were lower for the *in situ* compressor bending modes. The results demonstrate that the internal components add mass to the bending modes (Modes 1, 2, and 7). It can also be observed that the internal components add damping.

Sound Power Prediction

After the FEM model was validated, a forced-response analysis was performed using modal superposition in ANSYS. The modal damping from the *in situ* case was used in the model. Forces were applied to the suction and discharge bearing surfaces. A 1N force was applied to the suction bearing surface, and the force for the discharge bearing and surface was scaled appropriately from measured data. The forces were evenly distributed as a pressure over the bearing surfaces. Keep in mind that the developed models are linear in nature. So if the internal force is increased tenfold, the radiated sound pressure will also undergo a tenfold increase. Since the sound power is proportional to the product of sound pressure and particle velocity, the sound power results are proportional to force squared. Therefore, normalized forces were applied to the compressor housing model with the understanding that the results could be scaled accordingly later on.

A "shrink wrap" was applied to the defeatured solid model using Pro/ENGINEER. This wrapped model was meshed to create the BEM mesh shown in Figure 3. LMS SYSNOISE software[4] was used for the BEM analysis. The vibration results from the forced response were read in as a velocity boundary condition to the BEM model. The FEM and BEM meshes need not be coincident, because results from the finer FEM model can be projected onto the coarser

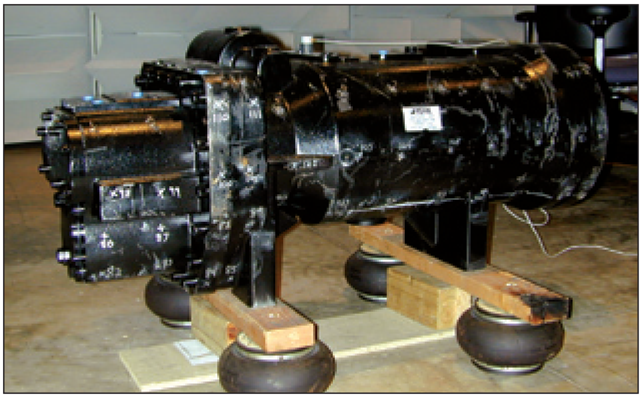

Figure 1. Screw compressor investigated for this article.

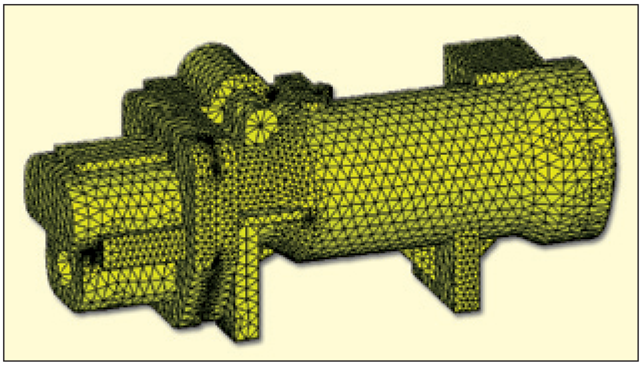

Figure 2. FEM mesh of compressor.

BEM model. Normally, a coarser BEM model is desirable due to solution time concerns. However, large degree-of-freedom radiation problems can now routinely be solved using fast multipole methods[5-8] in conjunction with iterative solvers.[9-10]

Indirect BEM[11] was used to formulate the BEM equations. Two absorbing planes were placed within the boundary to overcome the non-existence problem,[11] a well-known difficulty for radiation problems from a closed boundary if the indirect BEM is used. If absorbing planes are not added, errors will be manifested at interior acoustic resonant frequencies of the enclosed boundary. Accordingly, three absorbing planes were placed inside the BEM mesh normal to each of the coordinate directions.

The sound power results from the BEM simulation are shown in Figure 4. These results are normalized by the suction-bearing force squared. Notice that the peaks in the sound power transfer function occur at the modes of the compressor housing as expected.

Table 1. Comparison of measured and FEM modes.

Mode	ANSYS	*In Situ*	% Damping In Situ	Housing Only	% Damping Housing Only	Character
1	593	548	2.0	593	1.4	Bending
2	618	604	2.9	633	0.6	Bending
3	741	740	1.8	744	0.4	Torsion
4	979	980	2.0	979	0.5	Cylinder
5	1054	--	--	1051	0.4	Cylinder
6	1144	--	--	1143	0.9	Cylinder
7	1214	1098	1.8	1226	0.9	Bending
8	1348	1427	2.7	--	--	Bending
9	1735	--	--	--	--	Bending
10	1789	1758	--	--	--	Torsion
11	1885	--	1.8	1714	0.4	Cylinder
12	1958	1968	1.0	1907	0.4	Cylinder

Table 2. Differnece in dB between simulated and measured discharge pressure.

First, 490 Hz	Second, 980 Hz	Third, 1470 Hz	Fourth, 11960 Hz
2	2	6	5

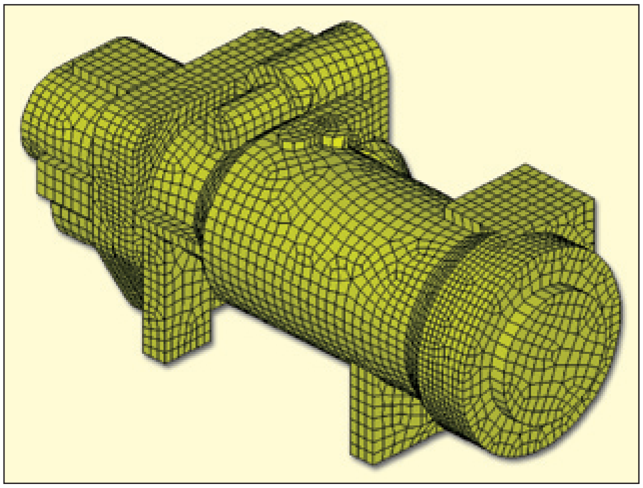

Figure 3. BEM mesh of compressor.

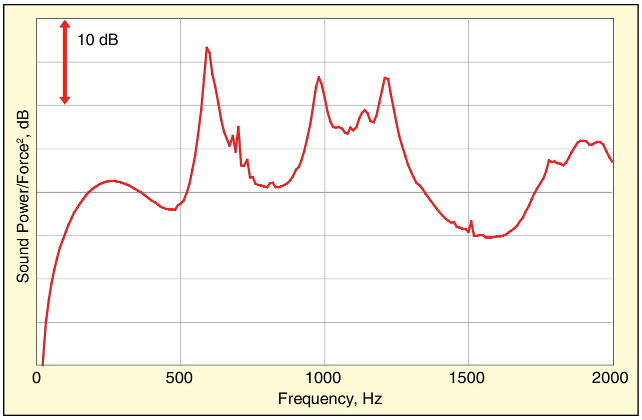

Figure 4. Sound power transfer function for original design.

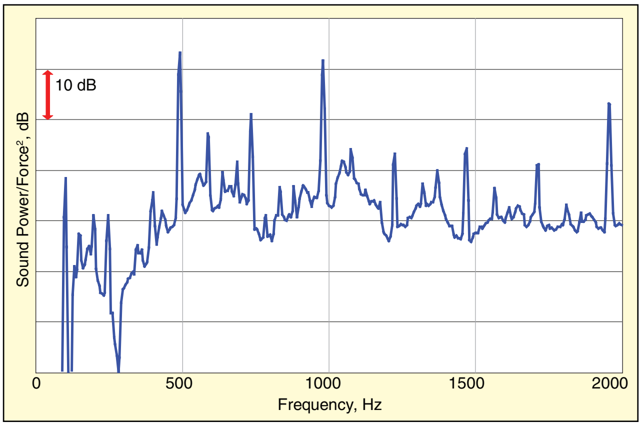

Figure 5. Measured sound power results.

The sound power from the running compressor was measured using a sound intensity scan. The pipes leading into and out of the compressor were wrapped with lagging so that the sound coming from them would not contaminate the sound intensity measurements for the compressor itself. The sound power results are shown in Figure 5. The sound power is especially high at the four pumping frequencies (490, 980, 1470, and 1960 Hz). Notice that the fourth and twelfth structural modes (Table 1) correspond to the pumping frequencies of the compressor. This suggests that the compressor design could be improved by shifting each of these modal frequencies.

The measured sound power results (Figure 5) were then used in conjunction with the sound power transfer function (Figure 4) to predict the dynamic pressure on the discharge surface. The differences in dB between simulation and measured pressures are shown in Table 2 at each of the pumping frequencies. The findings suggest that the simulation model is reliable.

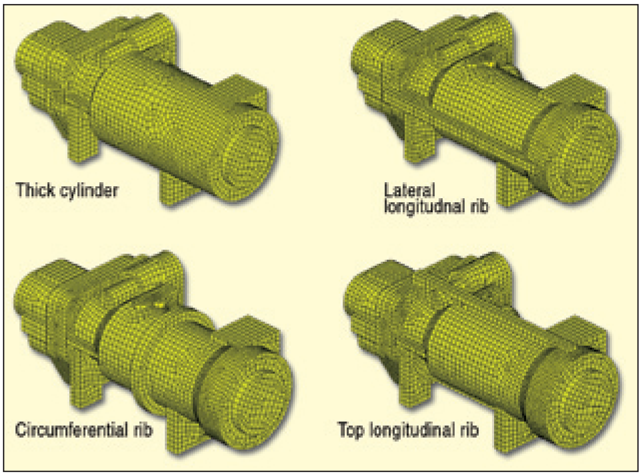

Figure 6. BEM meshes for baseline and redesigns.

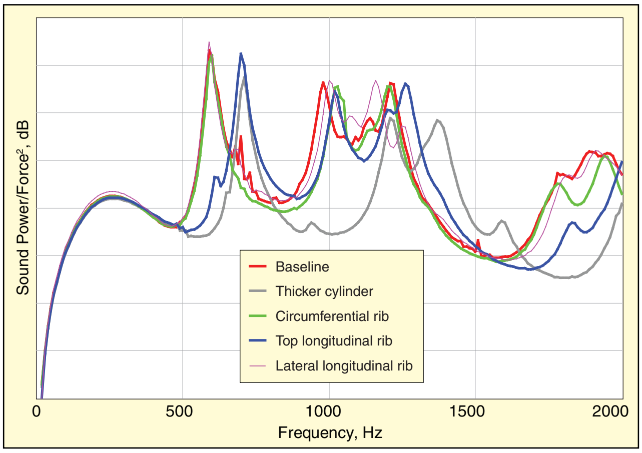

Figure 7. Comparison of sound power transfer functions for baseline and four redesigns.

Design Study

After corroborating the model with measured results, a design study was conducted. Four candidate designs were investigated (Figure 6). These included options for a thicker cylinder and for longitudinal and circumferential ribs. For each design, both FEM and BEM meshes were created and analyzed as described for the baseline case. For consistency, 2% damping was selected for all modes for the forced-response analyses instead of the measured damping.

The sound power transfer functions (sound power/force^2) for the baseline case and the four redesigns are shown in Figure 7. The results indicate that little can be done to improve the screw compressor for the first or third pumping frequencies. However, the sound power could be reduced for the second and fourth pumping frequencies by using a thicker cylinder. This would not adversely affect the sound power level at the first or third pumping frequencies.

Conclusions

A simulation model was developed for a rotary-screw compressor. A structural FEM model was used to predict the vibration. There was good agreement between the structural FEM modes and experimental measurements. The structural FEM model was then used to predict the vibration using normalized loading. The sound power was then predicted using an acoustic BEM model with the vibration as a velocity boundary condition. The measured sound power was used in conjunction with the simulation model to predict the dynamic pressure at the discharge surface. The results agreed well with the measurements.

After validating the model, a subsequent design study was performed. The results indicated that the second and fourth pumping frequencies coincided with two of the natural frequencies for the compressor housing. Simulation was used to assess four possible redesigns. The simulation indicated that a thicker cylinder would suppress the sound power at the second and fourth pumping frequencies without adversely impacting the sound at the first and third pumping frequencies.

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