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Hybrid Experimental and Analytical Approach to Reduce Low Frequency Noise and Vibration of a Large Reciprocating Compressor

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ABSTRACT

This paper describes the experimental and analytical techniques used to identify and correct the noise and vibration concerns in a new family of reciprocating compressors. Operating sound, vibration and pressure pulsation data were acquired simultaneously on a prototype compressor. Analysis of this data led to two areas of concern. The first concern was identified using Operating Deflection Shape (ODS) analysis of the compressor in conjunction with an uncoupled vibro-acoustic model of the suction cavity. The second concern was identified using experimental impact tests on individual components in conjunction with a finite element model. In both cases, experimental data were not only used to diagnose the problem but also to validate models of the baseline configuration. The models were then used to investigate possible countermeasures.

EXPERIMENTAL INVESTIGATION

Two prototypes of a new family of large reciprocating compressors were used for the noise and vibration troubleshooting: one was intact and was used for the operating tests while one was un-welded and was used for the modal investigation of the interior sub-system.

The complete compressor was first evaluated by measuring its sound power level at different conditions of suction and discharge pressure and at 50 and 60 Hz line frequency. The measured sound power spectrum of the compressor at ARI conditions and 60 Hz line frequency is shown in Figure 1. The corresponding overall sound power level was 73.9 dB (A) and the ARI SRN was 7.6.

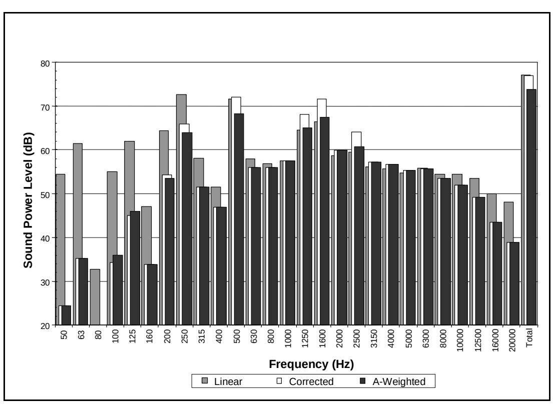

Figure 1. Sound Power level at ARI at 60Hz

The two 1/3-octave bands at 250 Hz and 500 Hz clearly suggest the presence of troublesome tonal components. These were then investigated by performing line frequency sweep tests to search for system resonant frequencies. The compressor was instrumented with four triaxial accelerometers on the housing (three around the lower part and one at the top), two high pressure microphones at the top and bottom of the housing, as well as dynamic pressure transducers in the suction and discharge lines. Four microphones around the compressor were also used to measure radiated noise, and the motor current signal was acquired as a line frequency reference. A total of twenty-one channels were acquired simultaneously using an Agilent VXI front-end driven by commercially available data acquisition software running on a NT workstation. The line frequency was swept from 65 Hz to 45 Hz in about 60 seconds, while static suction and discharge pressures were manually held constant. The time histories of all data were first stored to disk then processed into sets of frequency spectra that can be shown as 3D plots or Campbell diagrams.

The discharge pressure pulse was band passed using a digital FIR filter and used to extract the pump speed information. This pump speed was compared to the line frequency calculated from the current to produce a measure of the motor slip versus time. Figure 2 shows these two RPM versus time curves for one of the speed sweep tests. The difference between the two curves shows the motor slip, which is about 150 RPM at the 60 Hz line frequency.

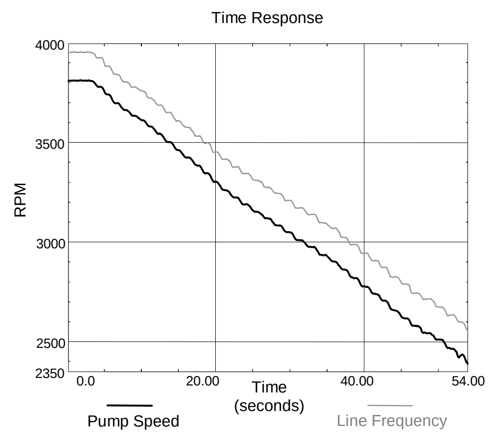

Figure 2. Nominal RPM from line frequency and effective pump RPM during sweep test.

Figure 3 shows a Campbell plot of the recorded noise for one of the microphones. The minimum (2480) and maximum RPM (3800) of the sweep are on the X-axis, the frequency from 0 to 800 Hz is on the Y-axis and the diagonal lines on the plot represent the harmonics (or orders) of the pump speed. The size of the squares on the diagonals indicates the amplitude of the orders. During the line frequency sweep, the frequency of each pump order changes with the motor RPM. If all pump orders are amplified when crossing a certain frequency, this indicates the presence of a resonance of the compressor or of one of its sub-systems. The data shown in Figure 3 clearly suggest the existence of two resonances at 380 Hz and 520 Hz, as well as a dominant 4th order, possibly being amplified by a resonance at 240 Hz. Since this is a two-cylinder compressor, the 4th order is the 2nd harmonic of the pumping frequency.

For each data channel, the orders shown on the Campbell plot were then sliced and plotted versus frequency. Some of the sliced orders for one of the microphone locations are shown in Figure 4. The resonances at 385 Hz and at 520 Hz are clearly seen. The resonance at 520 Hz falls in the 500 Hz band, which is one of the two areas of concern identified at the start of the investigation. On the other hand, the 385 Hz resonance was not initially identified and may be of concern only at 50 Hz line frequency (at which it corresponds to the 8th order).

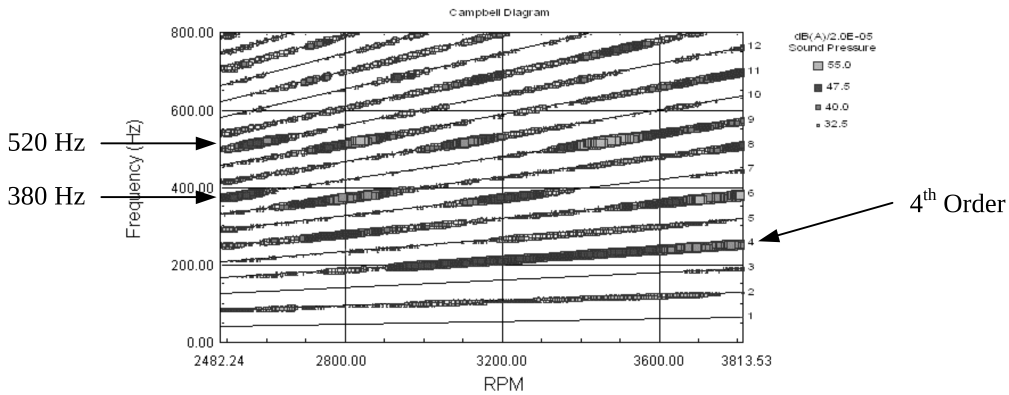

Figure 3. Campbell plot of sound pressure level at one microphone location

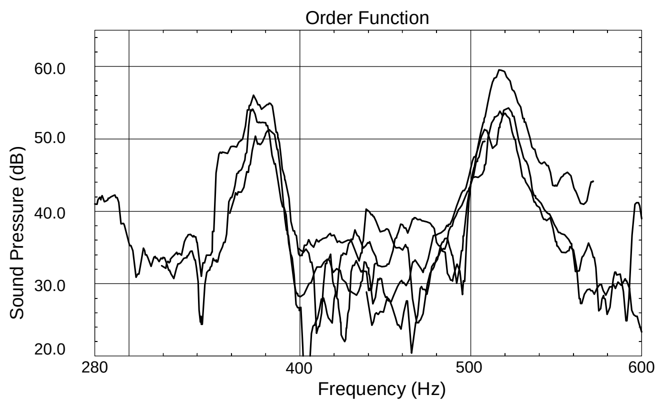

Figure 4. Orders 6, 7, 8 and 9 versus frequency - sound pressure level at a microphone

As for the 4th order, the amplitude curves for the pressure at the top and the bottom of the housing increase fairly constantly as the speed increases with a constant phase difference (Figure 5). This indicates a vertical acoustic mode of the cavity inside the housing. Furthermore, an Operating Deflection Shape (ODS) analysis of the housing confirms a strong vertical motion of the housing at this same frequency, most likely excited by the suction cavity mode. This is the likely cause of the high sound pressure level measured in the 250 Hz band.

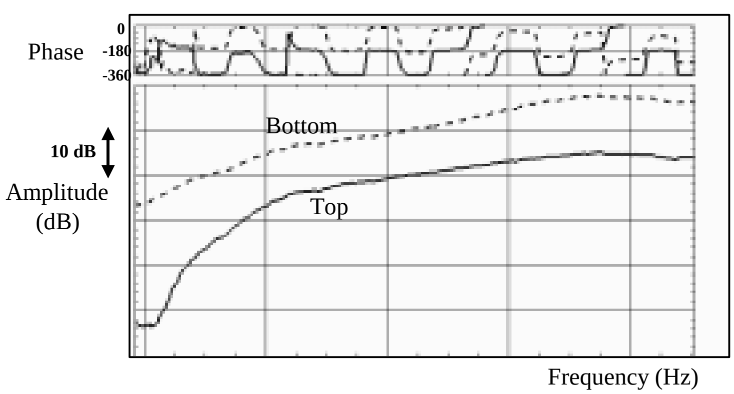

Figure 5. 4th order slice versus frequency for top and bottom housing pressure

Finally, the un-welded compressor was used to investigate the structural response of some of the internal sub-systems. Using a calibrated hammer and a laser vibrometer, the structural response of the discharge shock-loop was measured; it is displayed in Figure 6. The shock loop exhibits a mechanical resonance at 520 Hz, which is the likely cause of the high level in the 500 Hz band.

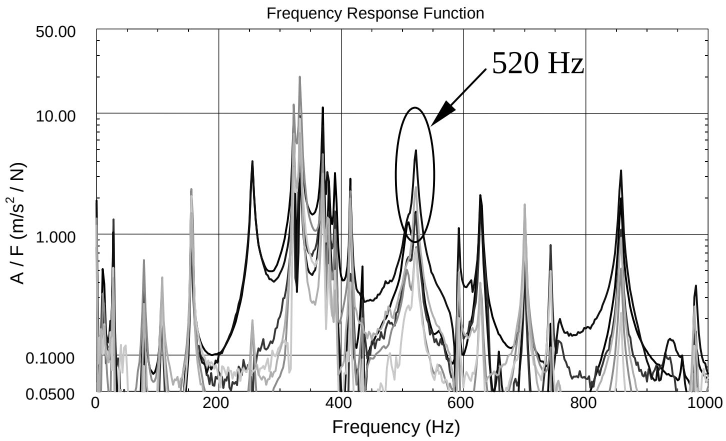

Figure 6. Structural response of shock-loop

STRUCTURAL AND VIBRO-ACOUSTIC MODELS

In order to reduce the noise radiated in the 500 Hz band, a finite element model of the shock-loop was developed and correlated to the impact data described in the previous paragraph. The model predicted a structural mode at 520 Hz. Figure 7 shows the 520 Hz mode, which is primarily lateral, anchored by the cylinder head and housing masses at the loop ends with a node along the length of the loop due to the mass of the muffler body. Several design changes were then simulated re-solving the finite element model and the best fix was then prototyped and tested.

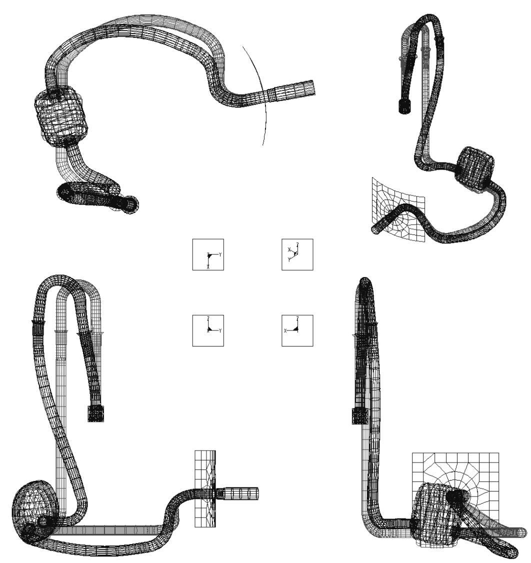

Figure 7. Shock loop finite element model
520 Hz mode

For the 250 Hz band concern, a vibro-acoustic model of the cavity between the motor-pump system and the housing was generated. Figure 8 shows the acoustic cavity mode found at 239 Hz, which exhibits a top-to-bottom pattern consistent with test data.

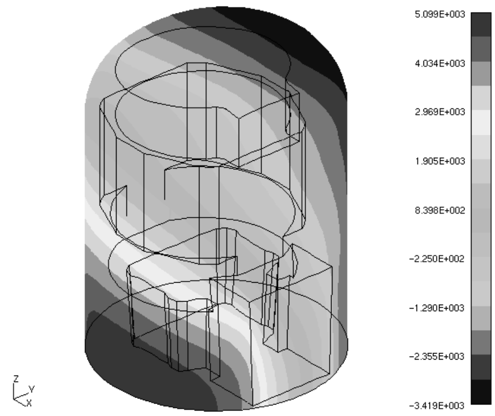

Figure 8. Housing cavity acoustic mode at 239 Hz

Several design changes were then simulated using the vibro-acoustic model with the purpose of de-coupling the acoustic cavity mode from the harmonics of the compressor running speed. Component modifications were then made based on simulation results to address the 250 Hz band issue.

CONCLUSIONS

Experimental and analytical techniques are more effective when used in conjunction with each other to identify and correct noise and vibration concerns in compressors. Multi-channel data acquisition of operating sound, vibration and pressure pulsation data is very useful for diagnosing sound problems, building confidence in analytical models, and assessing the success of component redesigns. Analytical techniques can give a more detailed understanding of the problem and streamline prototyping. When used together, experimental and analytical methods accelerate the entire solution process from problem identification to countermeasure evaluation.