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APPLICATIONS OF MODAL TECHNIQUES TO NOISE CONTROL OF HERMETIC REFRIGERATION COMPRESSORS

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ABSTRACT

That there exists interaction between vibration and structure-borne or structurally radiated sound is generally recognized. However, it is often difficult to predict the complex interactions between excitation forces, structural responses and the generation of sound. This paper will discuss such interaction especially in light of modal characterizations of the structural vibrations. In addition, a case history of application of these techniques to control of noise emissions from a hermetic reciprocating compressor is discussed.

INTRODUCTION

A large percentage of machinery noise is structurally radiated. A vibrating structure generates sound by moving the particles of air near the surface. This creates an audible pressure pulsation which moves away from the structure at the speed of sound. The phenomenon of structurally radiated sound is dependent on the complex interaction of structural vibration and sound generation and propagation. This paper will consider techniques for the control of unwanted sound generated by vibrating machines.

One of the primary noise control philosophies is to reduce the harmonic motion of the structure. This is most efficiently understood using modal characterizations. The structural vibration of individual modes can be reduced either by keeping the natural frequencies well removed from the excitation frequencies, applying damping treatments, and/or by reducing the coupling between the excitation mechanisms and each mode.

The acoustical considerations are usually characterized by the radiation efficiency parameter which is defined as

$$\sigma = \frac{\Pi}{V_t^2 \rho_0 c A} \qquad (1)$$

where σ is the radiation efficiency parameter, Π is the power radiated from an object with surface area A, and V_t is the RMS velocity of the radiating surface. Radiation efficiency is a gross measure of the sound generating effectiveness of the structure in a certain mode of vibration at a given frequency. At frequencies where the wave length in the structure is shorter than the wavelength in air (low frequency), sound generated by one part of the surface will propagate out of phase with sound generated by adjacent parts of the surface. Thus, the radiated sound amplitude will be small despite the fact the surface velocities may be large. On the other hand, at frequencies where the wavelength of sound in air is shorter than the wavelength of the waves in the structure, most of the sound energy will propagate in phase in some direction as shown in Figure 1 and the radiation efficiency will be approximately one. A typical radiation efficiency curve is shown in reference 2.

Often, the modes of highest amplitude will not be the most important noise producing modes. Thus, acoustic diagnostic procedures such as acoustic intensity or virtual coherence [1] are very important for defining the noise problem before solutions are sought. The frequency of demarcation between the low and high radiation efficiency regions is referred to as the coincident frequency and is the frequency where the wavelengths in air and in the structure are the same. Other effects such as edge effects, sound transmission through barriers and the effects of inhomogeneties are also primary considerations in some problems and are discussed in depth in various texts [2-4].

The dilemma of structurally radiated sound control is that many times action to reduce motion will actually increase sound production. For example, stiffening a cylindrical shell to reduce the coupling of the shell to certain excitations many actually increase sound radiation because the radiation efficiency has increased for those modes. The following case study for a hermetic reciprocating compressor will illustrate some of these phenomena.

An Overview of Compressor Noise

A typical hermetic compressor has a multiplicity of noise generation sources and noise transmission paths. The block diagram shown in Figure 3 illustrates how compressor noise can be generated by mechanisms in the compression process and how the noise is transmitted by intermediate paths to the shell [5].

The pulsing nature of the fluid entering and exiting the pumping process can manifest itself as very substantial harmonic noise. Turbulent flow through the valve ports results in broad band noise. Valve impacts against seats and stops, and impacts of the piston and crank generate vibrational energy which may travel structural paths to the shell. Reciprocating compressors which have significant reciprocating masses, exhibit substantial compressor body motion, which exerts forces on its supports, resulting in noise from the compressor shell or within the compressor mounting system. In general, the character of the sound is predominately harmonic. Although the magnitudes of these harmonics have an inverse relationship with harmonic number [6], resonances in components which make up the intermediate paths tend to reinforce some harmonics.

Four possible paths of energy transmission are identifiable in the compressor, the springs, the lubricating oil, the refrigerant discharge tube, and the refrigerant gas. The compressor

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suspension springs are a possible mechanical path by which vibrational energy may be transmitted to the shell. The suspension system is designed so that the suspension frequency is well below the rotational frequency of the compressor. However, at frequencies much higher than the suspension frequency, the springs exhibit a number of axial and transverse spring modes. The effect of such modes is to short-circuit the expected isolation effect and to transmit significant forces to the shell at high frequency [7]. The spring resonances are particularly harmful if their natural frequencies correspond with the natural frequencies of the shell modes.

Lubrication oil can transmit vibrational energy to the shell by providing a direct path from the pumping mechanism. The discharge tube or "shock loop" acts as a mechanical transmission path to the shell for energy generated by compressor body vibrations, and by pulsations within the shock loop.

The refrigerant gas acts as an airborne energy transmission path. Because of the large impedance mismatch of the gas and the shell material, this path is usually not important except at gas cavity resonance conditions. The hermetic shell of a compressor is essentially a small acoustical enclosure surrounding the noise producing sources and intermediate paths. The shell keeps a significant percentage of the noise incident on it from the inside from radiating. However, the cavity produced is an ideal situation for generation of gas resonances. Cavity resonances, when excited, can generate enough force to drive the shell, despite the impedance mismatch between the shell and the gas.

All of the intermediate paths transmit energy to the shell. Thus, the characteristics of the shell influence almost every aspect of the radiated noise. The coupling of the intermediate paths to the shell excite the shell. The shell modal parameters characterize the vibration response of the shell. The radiated sound is dependent on the structural response and the radiation efficiency of the shell.

NOISE CONTROL OF A HERMETIC RECIPROCATING COMPRESSOR

The compressor under investigation was a 1.5 HP hermetic reciprocating refrigeration compressor. The compressor noise levels are shown in the one-third octave band spectra (Figure 4). A narrow band spectrum of the total radiated sound power is shown in Figure 5. The mechanisms causing noise in the frequency band from 1000 Hz to 1600 Hz is of particular interest in this study.

Modal Analysis of the Shell

Since the shell is the final radiator of sound, modal analyses of the shell were conducted to determine the characteristics of the shell. The shell modes were then compared to the radiated sound spectra to determine if the high noise levels were influenced by shell characteristics, or by the excitation mechanisms and intermediate paths. All analyses were made using a PCB instrumented hammer, and an HP-5451C FFT analyzer with University of Cincinnati Modal Analysis Software.

Several circumferential modes were found within the frequency band of interest. As an example, Figure 6 illustrates the 3,0 mode. The fundamental mode of the shell top was also found. While some coupling exists between the top and circumferential modes, the top mode is practically independent of the circumferential modes. Table 1 is a list of the frequencies of the circumferential modes that were found for various shells.
Note that there may be several frequencies found for a given mode. Repeated mode shapes (i.e. modes of the same shape, but slightly different frequency and orientation) occur because of the elliptical nature of the shell.

TABLE 1. Coincidence Frequency and Natural Frequencies for the Circumferential Modes

Perimeter = 0.6858 meters C { air } = 343 m / sec		
MODE	COINCIDENCE FREQUENCY	RANGE WHERE MODE OCCURS
1	500	
2	1000	1400 → 1235
3	1500	855 → 1120
4	2000	1310 → 1451
5	2500	1913 → 1960

At this point, it appeared that the circumferential shell modes could be a significant factor in the radiation of sound. The internal components of the compressor were mounted such that significant energy could be transferred to the sides of the shell, providing energy to excite these modes. The top mode could also contribute to the sound radiation, but would have to be driven primarily by gas cavity pulsations, since the coupling with the circumferential modes was small.

Radiation Efficiency Considerations

To try and link the shell modes found with the radiated sound spectra, directivity patterns were plotted from the radiated sound spectra. The directivity information didn't correlate well with the mode shapes found, except for the case of the 2.0 mode. This phenomenon is an indication that the shell modes other than the 2,0 mode were below the coincidence frequency

Table 1 also lists the coincidence frequency, and the range of frequencies were experimental modal analysis indicated the mode occurred. Only the 2,0 mode occurs at frequencies above coincidence for the shell. From Figure 2, the radiation efficiency of each mode may be estimated. The radiation efficiency of the 3,0 mode is approximately 5%, indicating very poor radiation to the air. The 4,0 and 5,0 modes have radiation efficiencies of 10% and 30% respectively. The shell top mode radiates above coincidence, which makes the top an efficient radiator of sound. These estimates correlate well with the experimental results of the radiation studies.

The studies indicated that only the 2,0 mode, and the shell top mode were significant radiators of sound. Combining this data with other studies of the internal components of the compressor, it was hypothesized that high order suspension spring modes were coupling with the 2,0 circumferential mode of the shell. In addition, it was hypothesized that the shell top mode was being excited by gas pulsations, and by some coupling with the circumferential shell modes. Two modifications were made to the compressor to test the hypotheses.

Suspension System Modification

The suspension system for this compressor consists of 3 equally spaced tension springs, attached just below the center of gravity of the compressor at the midpoint of the shell. Since such a design could provide significant energy input to the 2,0 shell mode, the system was changed.

Modal analysis of the shell showed that the lower corners were rigid relative to other parts of the shell. Thus, the tension springs were replaced by compression springs, which transferred energy directly to the base of the shell. Such an arrangement provides little excitation energy for the 2,0 mode.

Total radiated sound power measurements were made on the modified unit. The narrow band spectrum is shown in Figure 7. In the frequency band of interest, both the harmonics peaks and the noise floor were lowered. A general flattening of the spectrum was also obtained which is psychologically advantageous. Since the spectrum of Figure 7 is the total radiated sound power, it does not show the most important aspect of the modification. Sound pressure levels in the 1000 to 1600 Hz frequency band were significantly lower when measured at the sides of the shell. At locations near the top, the levels remained the same due to the influence of the shell top radiation. Subjectively, the noise level of the modified compressor was somewhat lower than the original.

Shell Top Modification

Drivepoint accelerance measurements taken on the shell top indicated that the shell top vibration mode of concern was like the fundamental bending mode of a fixed elliptical plate. A twin cantilever beam, tuned absorber was designed and installed on the shell top. The response of the new system vs the original is shown in Figure 8. The peak at 1450 Hz has been moved down in both frequency and amplitude.

Having shown that the tuned absorber effectively controls the shell top mode, radiated sound power measurements were repeated. (Remember that the suspension system modifications had already been made, so any effects of that modification will affect further results). The new sound power level spectrum is shown in Figure 9. The harmonic peak at 1450 Hz was reduced to the level of the surrounding harmonic peaks.

A one-third octave band comparison of both modifications with the original is shown in Figure 10. In the frequency band of interest, the levels decreased as much as 10 dB. The low frequency bands increased, due to poor design of the alternate spring mounting which should be easily corrected.

CONCLUSIONS

Modal analysis techniques are an excellent tool for noise control diagnostics, provided some guidelines are followed. Modifications to the structure that are intended to reduce noise levels should be based not only on the modal response, but on the radiation effects, as well. Care must also be taken so that only the desired characteristics are modified. Computer aided design procedures to evaluate changes made to the structure are approaching maturity and will be an invaluable aid in predicting and evaluating modifications.

It should also be noted that the solutions to noise problems using modal procedures considers only the transmission paths, and not the sources themselves. Source control solutions are often much more effective than path control solutions. Coherence analysis or other noise diagnostic procedures should be used if the noise sources are to be treated.

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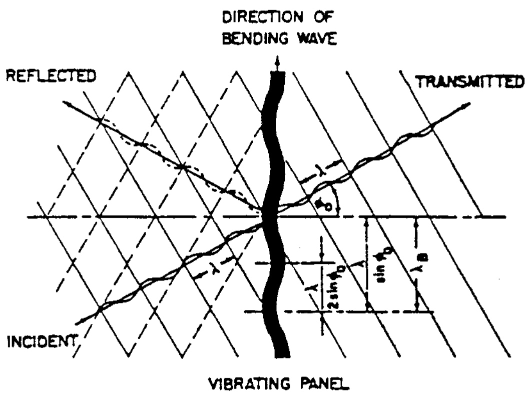

Figure 1. Transmission of Sound through a Panel above the Coincidence Frequency

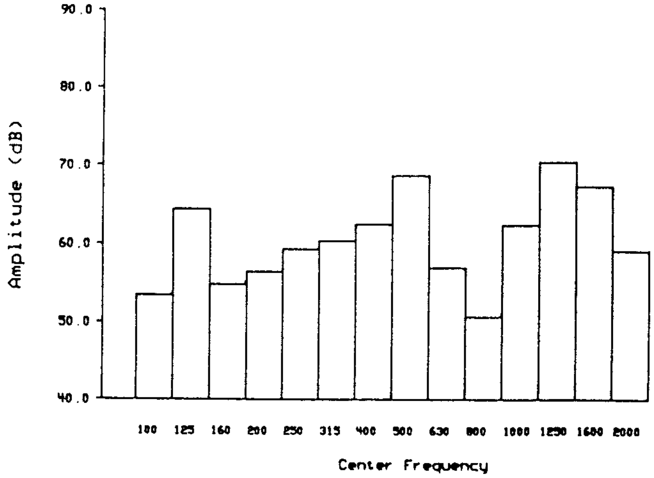

Figure 4. Total Radiated Sound Power, One-Third Octave Band Spectrum

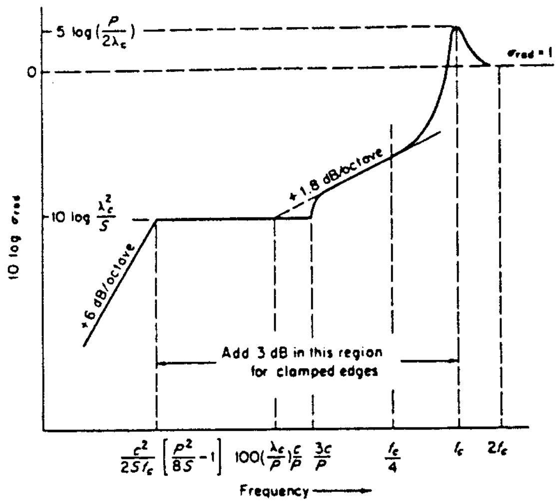

Figure 2. Radiation Efficiency vs Frequency

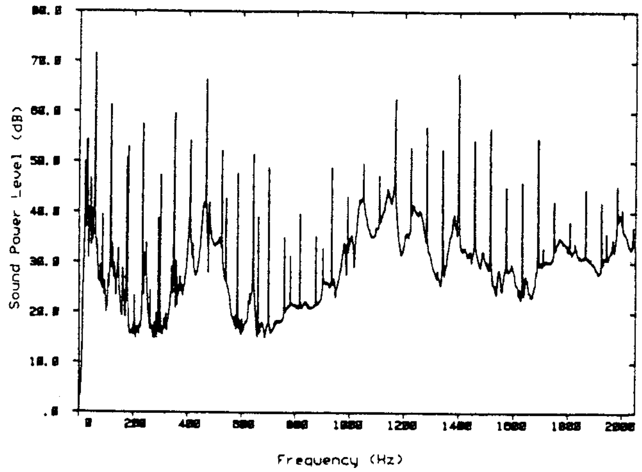

Figure 5. Total Radiated Sound Power, Narrow Band Spectrum

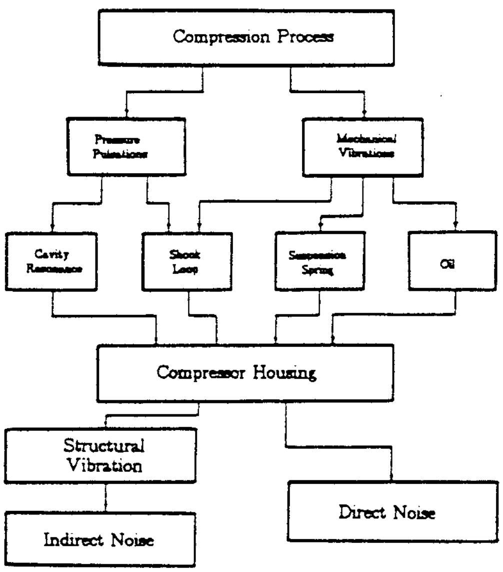

Figure 3. Noise Paths in a Hermetic Compressor

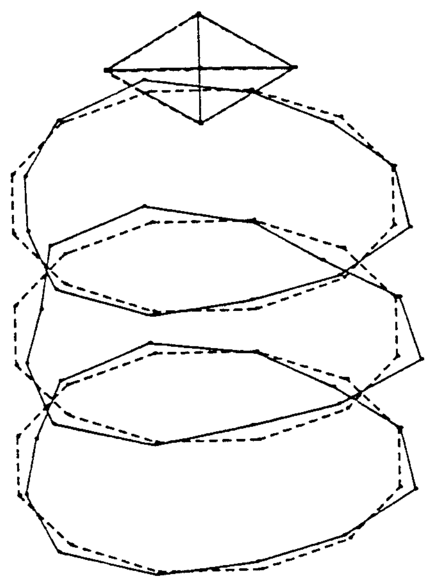

Figure 6. 3 Mode of Compressor Shell

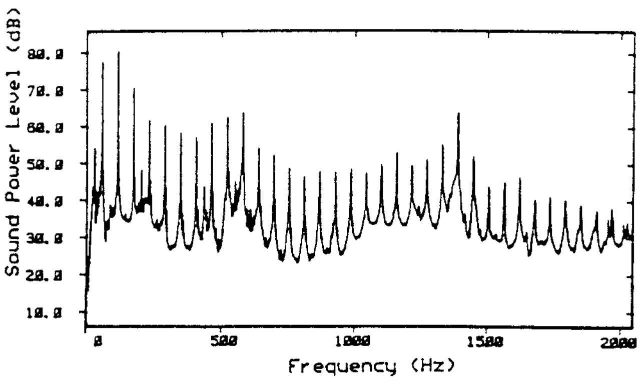

Figure 7. Radiated Sound Spectrum with Suspension Modification

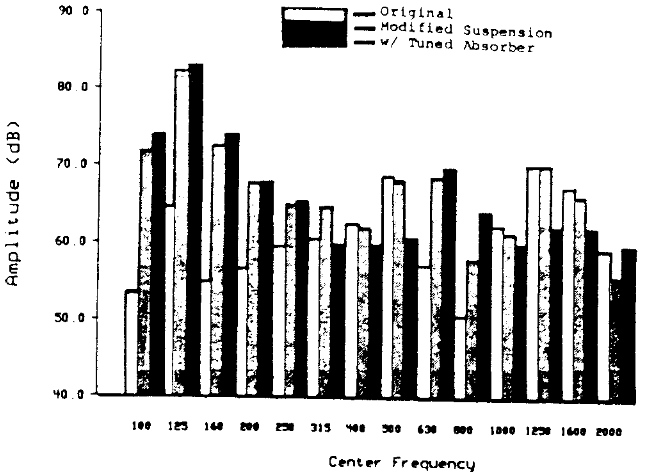

Figure 10. One-Third Octave Band Spectrum of Compressor with Modifications

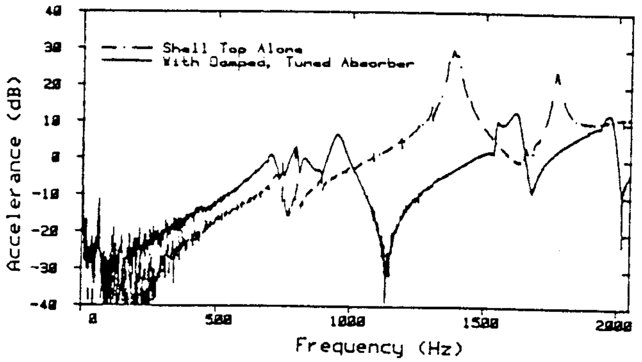

Figure 8. Drivepoint Accelerance With and Without Tuned Absorber

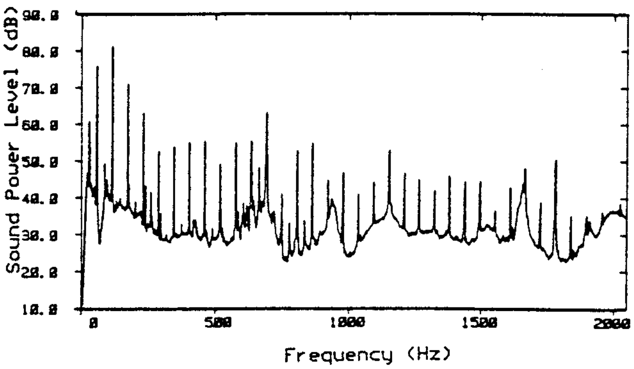

Figure 9. Radiated Sound Spectrum with Suspension Modification and Tuned Absorber