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M. A. Craun
Virginia Polytechnic Institute and State University

L. D. Mitchell
Virginia Polytechnic Institute and State University

IDENTIFICATION OF SOUND PROPAGATION PATHS WITHIN A RECIPROCATING COMPRESSOR VIA MULTIPLE-INPUT/SINGLE-OUTPUT MODELING

M. A. Craun
Graduate Research Assistant
L. D. Mitchell
Randolph Professor of Mechanical Engineering
Virginia Polytechnic Institute and State University

ABSTRACT

Through the use of multiple-input/single-output (MISO) modeling, the propagation paths of sound within a reciprocating refrigeration compressor have been investigated and ranked. By investigating the nature of sound propagation within reciprocating compressors, it is hoped that strategies for compressor sound reduction can effectively be formulated .

From experimental data of compressor far-field sound output, suspension spring forces, and internal pressure fluctuations, a MISO model has been developed. From this model, the importance of the suspension system to the compressor far-field sound spectrum has been identified. In the frequency range above 800 Hz, forces passing through the suspension system appear to be the dominant contributor to shell excitation and sound radiation.

Based upon this finding, it is recommended that modified suspension systems be considered as an avenue for compressor sound reduction efforts in the future.

INTRODUCTION

Consumer demand has dictated the development of reduced-noise products. The desire of the compressor industry to provide a such a product has made necessary the study of sound propagation paths within compressors, as part of a total program to understand and minimize compressor sound output. The following study will hopefully provide useful information about sound propagation within reciprocating compressors, and aid in the effort to develop effective sound reduction strategies.

Since compressor sound radiation is the direct result of high-frequency vibration of the compressor shell, a key question in the compressor sound reduction problem concerns the forcing function of the shell vibratory modes. Namely, do the pressure fluctuations within the shell contribute the most to shell radiation, or do the structure-borne forces passing through the compressor suspension system have more influence? The following study has employed multiple-input/single-output (MISO) processing of experimental test data to provide some answers to this question.

ANALYSIS METHOD

Figure 1 depicts a typical multiple-input/single-output (MISO) system. This model consists of k inputs ($x_1(t),x_2(t),...,x_k(t)$) passing through k linear frequency response functions (FRFs) denoted by $H_1,H_2,...,H_k$. The outputs of these FRF's ($y_1(t)$, $y_2(t)$, ...,$y_k(t)$) sum with each other to produce the single-output of the model, $y(t)$. An uncorrelated contribution of unknown origin ($n(t)$) can be included in the summation and is called the noise input. From this model, frequency-domain descriptions of the input/output spectral relationships can be developed. For example, the spectrums of the model inputs can be described by (ignoring $n(t)$)

$$G_{iy} = \sum_{j=1}^{k} H_j G_{ij} \qquad (i=1,2,...,k) \qquad (1)$$

where G_{iy} represents the cross-spectrum between the ith model input and the output, and G_{ij} represents the cross-spectrum between the ith and jth inputs. By experimentally collecting spectral data, these quantities are easily formed, thus allowing for the solution of $H_1, H_2, \ldots, H_k$ from Eq. (1). These are then used in the relation describing the output auto-spectrum

$$G_{yy} = \sum_{i=1}^{k} \sum_{j=1}^{k} H_i^* G_{ij} H_j \tag{2}$$

which is the MISO relation used for the ranking of inputs in their order of importance to the output. For a more complete discussion of MISO relations, refer to Bendat and Piersol[1].

By allowing the ith FRF in Eq. (2) to equal zero, a prediction is made of the output auto-spectrum G_{yy} with the effects of the ith input removed from the system. In a similar fashion, any combination of inputs can be removed, and the effect on the output auto-spectrum noted. Those inputs which result in the greater reduction in G_{yy} when removed are then considered dominant inputs to the output of the model.

EXPERIMENTAL TESTING

In order to utilize the MISO modeling technique described above, it was necessary to collect spectral data from a compressor operating under normal conditions. This made necessary the instrumentation of a production compressor. In order to accurately measure the dynamic forces passing through the compressor suspension system, piezoelectric triaxial force gages were inserted in each suspension spring path. To measure the pressure fluctuations which might drive the shell vibration, three piezoelectric pressure transducers were placed inside the compressor: one within the compressor shell cavity, one near the compressor suction line, and one within the compressor exhaust line. Other transducers were also present in the compressor, but are not pertinent to the results presented here[2].

The process of instrumentation resulted in no appreciable loss of compressor capacity or efficiency, but shell dynamic response and sound output were affected. The instrumented compressor ran quieter than production models, and the response of the compressor shell changed in the higher frequencies, most dramatically above 800 Hz. Although great care was taken during the instrumentation process to avoid altering the compressor dynamics, it is speculated that the modifications to the suspension springs were the major cause of the shell dynamic changes. This is because the force gage installation required more drastic changes in the compressor's structural members than did the installation of the pressure transducers[2].

In all, four tests of the compressor were conducted: three near-field sound tests to investigate the sound radiated from the compressor sides, and one far-field test. For each test, 100 consecutive data blocks were acquired simultaneously from the microphone, force gages, pressure transducers, and other instrumentation. Data blocks were 1024 data points long, with a sampling rate of 0.0781ms/sample. This resulted in a spectral resolution of 12.5 Hz, and an analysis frequency up to 5000 Hz. In the processing of spectral records, a Hanning window was used to remedy leakage effects.

RESULTS

Some typical spectral data are presented in Figs. 2, 3, and 4. Figure 2 shows a typical suspension spring force spectrum, a horizontal force in the top suspension spring. Figure 3 presents the pressure fluctuations within the compressor housing cavity. Figure 4 shows the far-field sound spectrum acquired in the fourth test, in un-weighted dB *re* 2.9×10^{-9} psi (20×10^{-6} Pa). In Figure 2, it can be seen that the dominant harmonic in the force signal occurs at the first harmonic, the operating speed of the compressor. Also, it can be seen that there is no apparent trend in the relative magnitudes of even and odd harmonics. This is typical of all of the force signals in this test. By contrast, the pressure signal of Figure 3 exhibits even harmonics of noticeably larger magnitude than the odd harmonics, most notably below 1000 Hz. This is because the compressor investigated is a two cylinder model.

The far-field sound output spectrum of the compressor shows dominant harmonic content in the region between 700 Hz and 2000 Hz. Below around 600 Hz, the even harmonics of the operating speed are dominant, as in the housing pressure spectra. In the frequency range above 600 Hz, the even and odd harmonics appear to be more equitable in their relative magnitudes. This trend was evident in the other microphone locations as well, although each microphone location resulted in distinct spectra with different dominant harmonics. The similarity of the low-frequency sound spectrum to the low-frequency housing spectrum may indicate a possible relationship between the two spectra.

Results of the far-field MISO study are presented in Figs. 5 and 6. Figure 5 shows the far-field sound output of the compressor in dB *re* 2.9×10^{-9} psi (20×10^{-6} Pa), compared to the predicted output due only to the direct contribution of the suspension spring forces. Figure 6 shows the far-field compressor sound output compared to the predicted sound output due to the effects of the suction, exhaust, and housing pressure fluctuations. Comparison of these figures indicates that the relative contribution of the suspension spring forces to the far-field sound output is much more significant than the contribution of the pressure fluctuations. Although in both cases the broad-band sound level (the region in between the harmonics) is not affected, the harmonics in the model without springs drop by an average of 6.5 dB. The model without pressure fluctuations does not indicate any appreciable drop in sound spectrum harmonics. Low frequency data is not shown in Figs. 5 and 6 because of poor multiple coherence below 700 Hz.

CONCLUSIONS AND RECOMMENDATIONS

From the inspection of spectral data and the MISO analysis, it is concluded that the contribution of the suspension spring forces to the far-field compressor sound output outweighs the pressure contribution, especially above 800 Hz. Below 800 Hz, there are indications that compressor pressure fluctuations are the dominant propagation path of compressor sound.

Based upon these findings, it is recommended that compressor sound output reduction efforts be directed toward modifications to the compressor suspension system. No specific suggestions regarding suspension redesign can be made as a result of the analysis presented here. There may be, however, some simple modifications to the suspension system which may produce repeatable sound reductions. Further areas of study include the investigation of other possible sound transmission paths such as the oil transmission path and the contribution of the shock-loop exhaust tube to shell radiation. Efforts to include the shock-loop tube in the MISO model discussed above resulted in numerical errors in this study. Although the analysis method is based upon sound signal processing theory, some experimental verification of the results would be ideal.

ACKNOWLEDGMENTS

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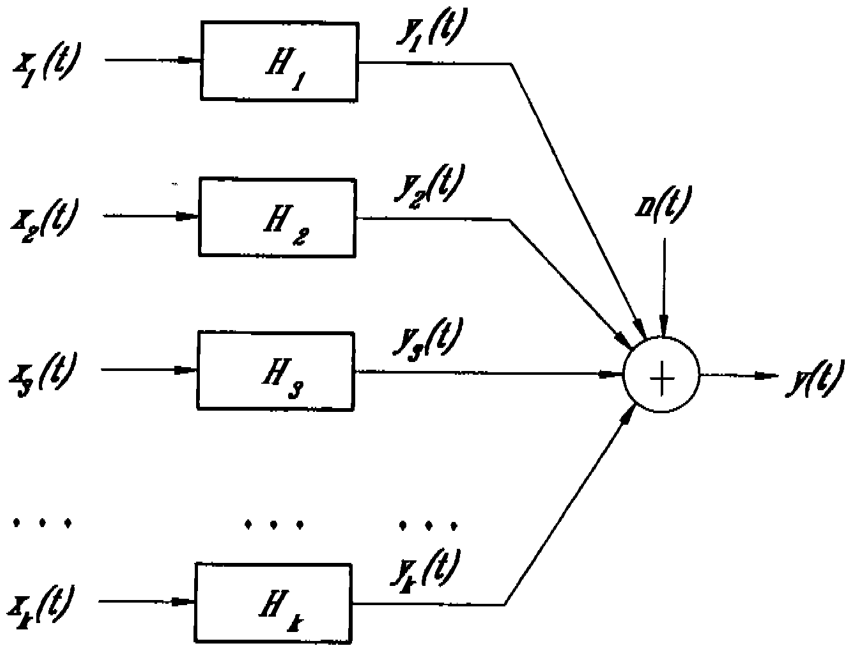

Figure 1. MISO Model

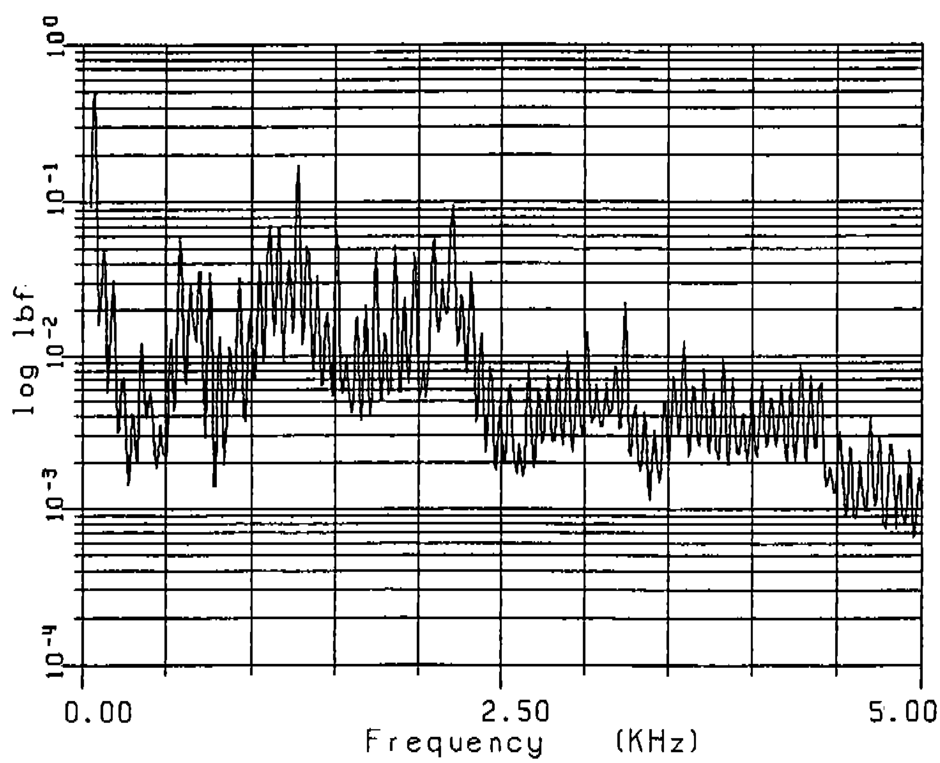

Figure 2. Typical Suspension Spring Force Spectrum (1 lbf = 4.448 N)

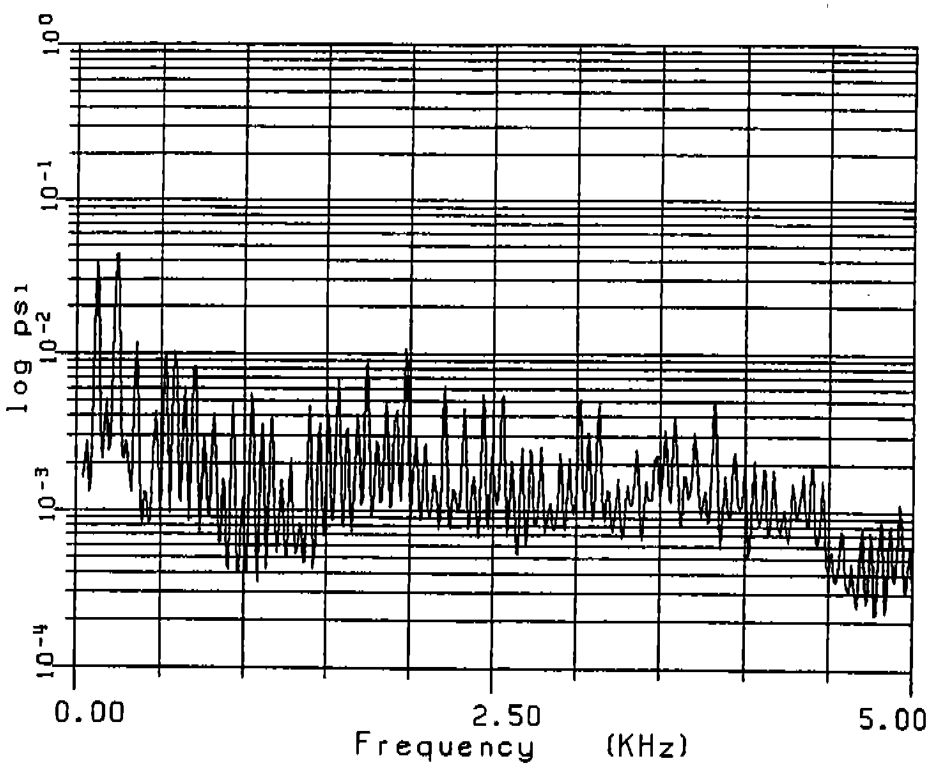

Figure 3. Housing Cavity Pressure Spectrum (1 psi = 6895 Pa)

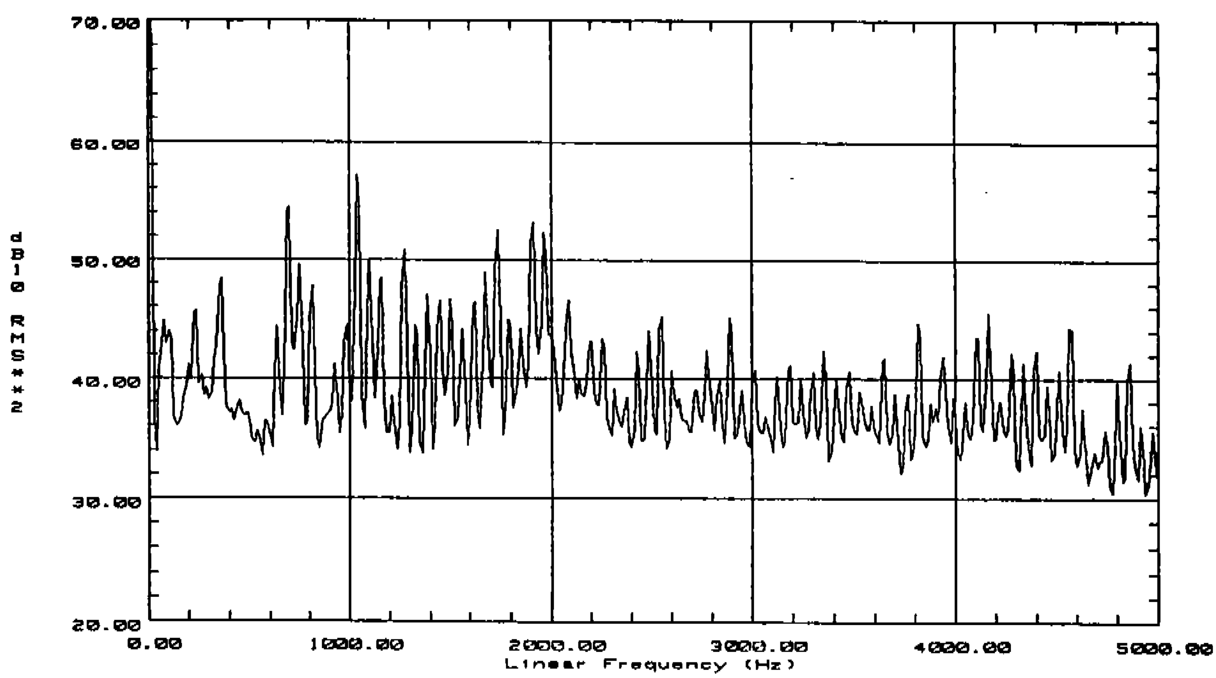

Figure 4. Far-Field Compressor Sound Output Spectrum

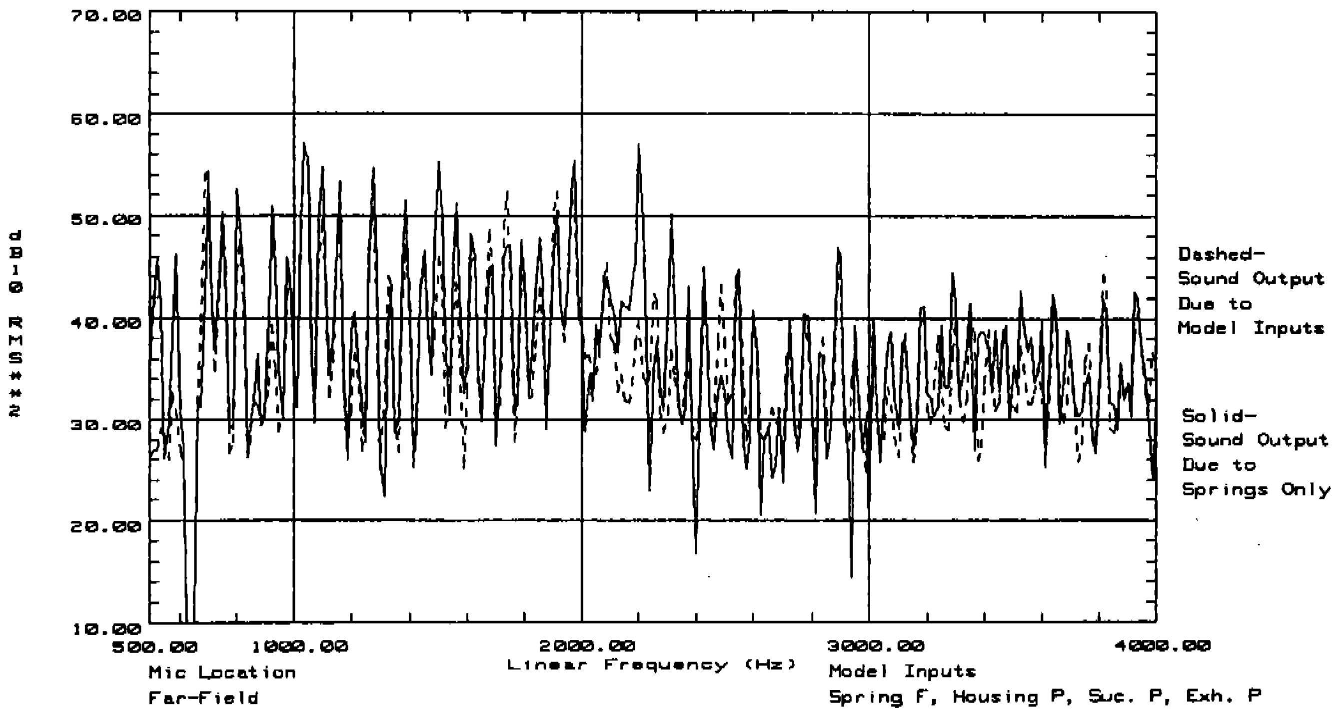

Figure 5. Prediction of Sound Output Attributable to Suspension Spring Forces

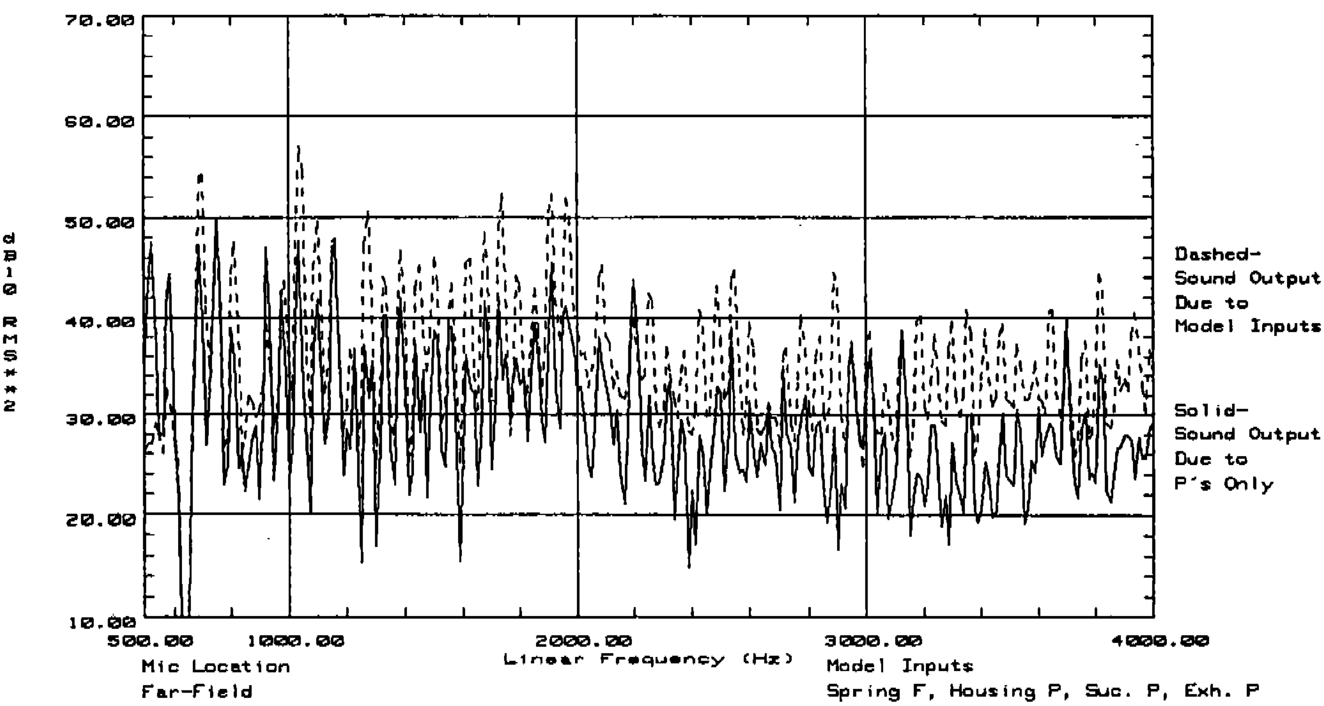

Figure 6. Prediction of Sound Output Attributable to Pressure Fluctuations