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Analysis and Control of Hermetic Compressor Sound Power Levels

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ANALYSIS AND CONTROL OF HERMETIC COMPRESSOR SOUND POWER LEVELS

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ABSTRACT

The consumer market is demanding that the noise level of appliances be reduced to lower and lower levels to satisfy new expectations for environmental quality. This has placed a burden on the manufacturers of compressors for air conditioners and heat pumps to reduce noise radiated by their units.

In order to accomplish this task, the approach has been to apply advanced signature analysis and structural dynamic analysis techniques to the compressors to learn the manner in which the excitation is produced and transmitted, and how the noise is radiated by the structure. Lastly, as a result of information gained through analysis of the system and its response, design changes are implemented in the compressor which significantly reduce the noise radiated by it.

An example is given where hardware changes were made to an existing hermetic compressor in order to reduce its noise level. These changes consisted of reducing the transmissibility between the internal compressor components and the shell, and reducing the response of the shell through damping. Results are given which demonstrate the amount of noise reduction attainable with this approach.

INTRODUCTION

Today consumers are requiring a better environment. Noise pollution is one of the main factors by which appliances affect the environment. Air conditioning and heat pump units are especially important in this respect because of their prominent position within a room or outside a house. Increased pressure from original equipment manufacturers, and a desire to compete in the international market, primarily Europe and Japan, where low radiated sound power levels are a necessity, has resulted in a need for reduced sound power levels in the typical hermetic air conditioning and/or heat pump compressor. As a result, an intensive sound reduction program was undertaken to reduce compressor sound power levels.

In the past, design for reduced radiated sound power has been based primarily on experience with prior units and experimentally oriented programs. These programs were oriented to reduction, or modification of, suction and discharge gas pulsation, under the premise that these were major sources of vibration excitation, which resulted in excessive radiated sound. However, work carried out previously on refrigeration compressors (1, 2, and 3) indicated that other mechanical excitation sources may play a significant role.

New micro-computer based signature analysis and dynamic structural analysis capabilities presented an opportunity for a more thorough evaluation and understanding of these sources, the resultant transmission paths, and the structural participation of the shell. Additionally, since structural resonances of the shell were suspected of being important in the radiation of sound, constrained layer damping, and other techniques appeared to be attractive methods of sound reduction. It was anticipated that these new instrumentation techniques and sound control methods would lead to more efficient design changes for sound control.

THE SOUND POWER LEVEL REDUCTION PROGRAM

Work began on a typical air conditioning/heat pump compressor that had been recently placed into production. This particular compressor had a somewhat larger hermetic shell than the rest of a similar family, and radiated more sound than was considered desirable. Thus, it was considered an ideal candidate for a sound reduction program, utilizing the latest technological approaches. A cross section is shown in Figure 1. The compressor is fairly typical, being mounted on side spring supports for low frequency vibration isolation, with a top spring for stabilization. The discharge tube design, which was critical from a static and dynamic standpoint, required a fourth torque control spring to limit stresses during starting and stopping of the compressor.

Sound power level measurements (Figure 2) indicated that the "A" weighted overall sound power levels of the compressor were about 81.1 dBA with a considerable amount of energy in the 800 to 2500 Hz. third octave bands. Narrowband analysis (Figure 3) indicated a particular frequency around 900 Hz, as a major contributor, many peaks in the 1000 to 1400 Hz. area, and others around 1875 and 2200 Hz.

In order to more fully understand the noise sources, and their transmission paths, a program was initiated to investigate sources, transmission paths, and methods of sound reduction.

MEASUREMENT OF OPERATING PARAMETERS

A standard compressor was placed inside a semi-anechoic room. A refrigeration system used to provide accurate operating conditions was also mounted inside the room and was treated with an absorbent/barrier material to prevent its noise from contaminating measurements of compressor sound.

Measurements of refrigerant pressures, vibration, and sound were made during operation of the compressor at the standard ARI air conditioning operating point. Locations for these measurements are given in Figure 4. Narrowband analysis of these measurements was performed with a digital signal analyzer. A sampling of the spectra from these analyses is given in Figures 5 through 7.

Several important facts about sound generation, transmission, and radiation by the compressor were learned from these data. It was found that the predominate frequencies in the sound spectra were the same as those found in vibration on the shell of the compressor. These frequencies showed no consistent relationship with the fundamental operating frequency of the compressor or its orders. The spectra of the pressure measurements on the other hand consisted almost solely of response at the fundamental frequency and its orders.

Transmissibility functions were computed relating vibration of the shell to vibration of the internal components, and sound outside the shell to sound inside the shell. These indicated that there was poor isolation between internal components and the shell and thus there was significant transmission of vibration energy to the shell by structural paths. The sound measurements on the other hand indicated that there was no significant transmission of acoustic energy from the internal cavity to the outside.

Frequency response measurements were then performed on the compressor at the same locations where the vibration measurements were made. Two of these are shown in Figures 8 and 9. From these results several important resonances were identified which correlated with prominent noise frequencies. Modal analysis was then used (as described in Ref. 4) to identify the character of the structural response at these frequencies. Following is a brief description of the major resonance modes.

- 700 Hz. - deformation of the bottom half of the case.
- 900 Hz. - breathing mode of the bottom half of the case with all panels moving in-phase.
- 1025 Hz. - deformation of top, sides, and end moving in-phase.
- 1530 Hz. - deformation of the lower half of the case in the area opposite the lift hook.
- 1850 Hz. - deformation of upper half of case.

As a result of the measurements, it was concluded that noise from the compressor was in large part due to response of the compressor shell at its resonances to broad-band excitation from the internal pump/motor assembly. Noise due to a mechanism such as this can be controlled in several ways:

1. Reduction of excitation of the case by attenuating the source.
2. Reduction of excitation of the case by improved isolation between the internal components and the case.
3. Reducing response of the case through an increase in its stiffness.
4. Reducing response of the case by increasing the damping of its resonances.

Since inherent damping in the case was fairly low, it was decided to pursue the path of increasing damping to reduce the noise level. It was felt that a significant reduction in sound level could be achieved through damping of the shell at minimum cost and alteration of production procedures. It was also decided to determine if improving the isolation between the compressor body and shell would be a cost effective method of achieving lower noise levels.

COMPRESSOR MODIFICATIONS AND RESULTS

The compressor shown in Figure 1 was then modified in the following manner:

Isolation Modifications

- o The standard coil springs that are used to mount the compressor body to the shell were replaced with rubber mounts with the same stiffness characteristics, but higher damping in an attempt to reduce mid to high frequency transmission.
- o The standard all-metal discharge tube was replaced with a tube that consisted of a six inch portion of automotive air conditioning rubber hose. This modification increased the flexibility of the discharge tube system and again reduced the magnitude of forces transmitted between compressor body and shell.

Shell Damping Modifications

- o A constrained layer damping system was designed and applied to the outer area of the compressor shell. The system covered all sides of the bottom and top shell, as well as the top of the top shell, and was designed specifically for the temperature ranges encountered during normal operation. A typical configuration of these damping treatments is given in Figure 10.

The design of a constrained layer damping treatment depends on the temperature of the surface to be treated and the frequency range over which it must be effective. This is due to two factors:

- o The effectiveness is dependent on the wavelength of the resonant modes contributing to vibration of the surface.
- o The properties of the material comprising the damping layer are temperature and frequency dependent as shown in Figure 11.

Figures 12 and 13 show driving point frequency responses on the shell initially and with the shell damping treatment installed. It is obvious that the treatment produced a significant reduction in response at the resonant peaks.

- o Oil foaming has been found to act as an extremely good vibration damping mechanism for hermetic compressor shells.

  In this particular case oil foaming was achieved by forming a coil of copper tubing with .010" diameter holes drilled along its length and placing it in the compressor lubricating oil. One end of the tube was closed off and the other end was connected to the compressor discharge plenum. This system provided sufficient refrigerant gas flow to add small bubbles to the lubricant and cause it to foam vigorously.

- o It was thought that the motor cover shown in Figure 1 may also have been a contributor to measured sound levels. Consequently, it was replaced by a motor cover that had been stamped from a constrained layer steel sheet.

Initial tests were carried out without oil foaming, since it would be desirable to achieve sound reduction without the performance penalty required with the foaming technique. The third octave band results are shown in Figure 14. It can be seen that A-weighted sound power level has been reduced from 81.1 dBA on the original compressor to 75.8 dBA on the modified compressor. Comparison of Figure 14 and Figure 2 shows that much of the sound energy radiating from the shell above 800 Hz. has been reduced. Narrowband analysis indicates that the particularly strong peak at around 900 Hz. has been dramatically reduced, while all frequencies above 1250 have been reduced to some extent. Those frequencies in the 1500 to 2500 Hz. range were reduced by about 10 dBA.

Finally, to determine what minimum sound levels might be achieved, the gas bubbling system was installed. These test results are shown in Figure 15 and indicate a further reduction in sound power levels to 73.9 dBA. Comparison of Figures 14 and 15 indicate that sound has been reduced in all frequency bands from 1000 Hz. up. However a strong 800 Hz. frequency still remains. Later investigation indicated that this particular frequency was a result of a shell resonance at 810 Hz. Modal analysis indicated that the mode shape was such that neither the constrained layer or oil foaming were sufficient to damp that frequency. Later attempts at removing that frequency by oil foaming have been more successful.

CONCLUSIONS

Advanced signature analysis and structural dynamic analysis techniques can be valuable in sound reduction efforts in complex mechanical equipment.

Shell damping can be an effective method of sound reduction in hermetic air conditioning and refrigeration compressors. Constrained layer damping techniques appear to be particularly useful for this purpose.

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1. E. B. Brookbank, "Forced Vibration of Thin Elastic Shells with Application to Fractional Horsepower Hermetic Refrigeration Compressor Shells", Dissertation, Ohio State University, 1978.

2. F. Saito, S. Maeoa, N. Okubo, T. Vetsuji "Noise Reduction of Hermetic Compressor by Improvement on its Shell Shape", Proceedings of 1980 Purdue Compressor Technology Conference, 1980.

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4. M. Richardson, "Modal Analysis Using Digital Test Systems", Proceedings of Seminar on Understanding Digital Control and Analysis in Vibration Test Systems sponsored by Goddard Space Flight Center and Jet Propulsion Laboratory.

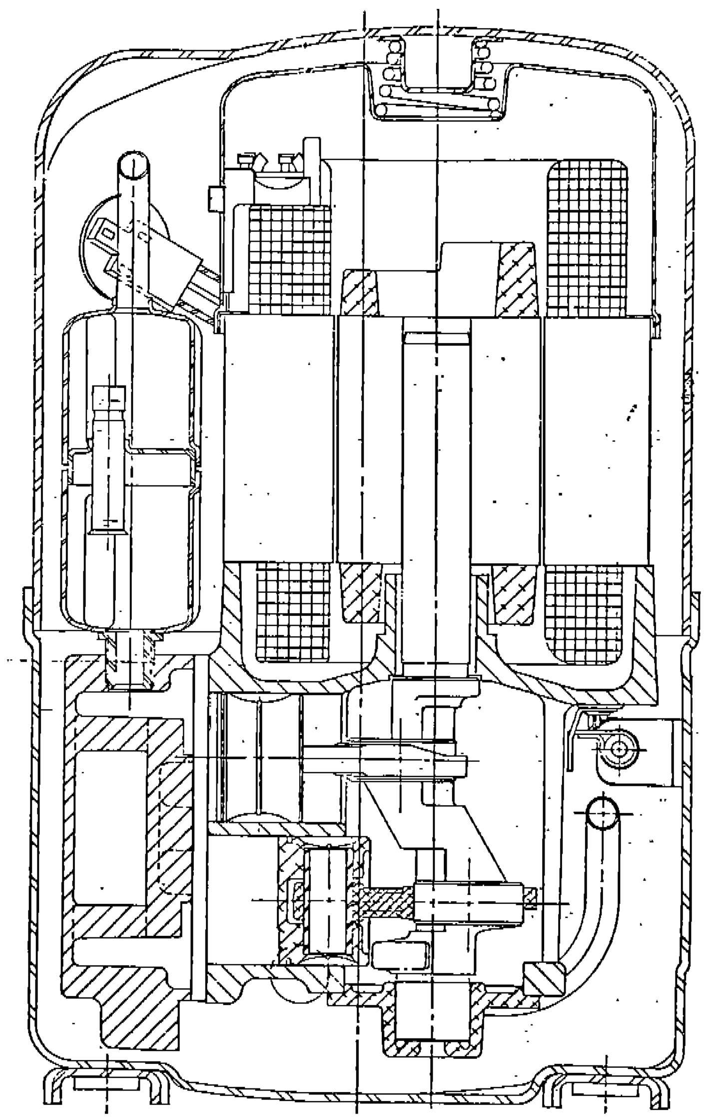

FIGURE 1

HERMETIC AIR CONDITIONING COMPRESSOR

LW	LWA	FREQ.
55.3	36.2	100
72.7	56.6	125
46	32.6	160
45.9	35	200
68	59.4	250
61.7	55.1	315
57.3	52.5	400
59.4	56.2	500
68.7	66.8	630
75.1	74.3	800
69.6	69.6	1000
72	72.6	1250
70.4	71.4	1600
72.1	73.3	2000
72.1	73.4	2500
65.8	67	3150
63	64	4000
62.5	63	5000
59.8	59.7	6300
60.8	59.7	8000
59.5	57	10000
81.7	81.1	TOTAL

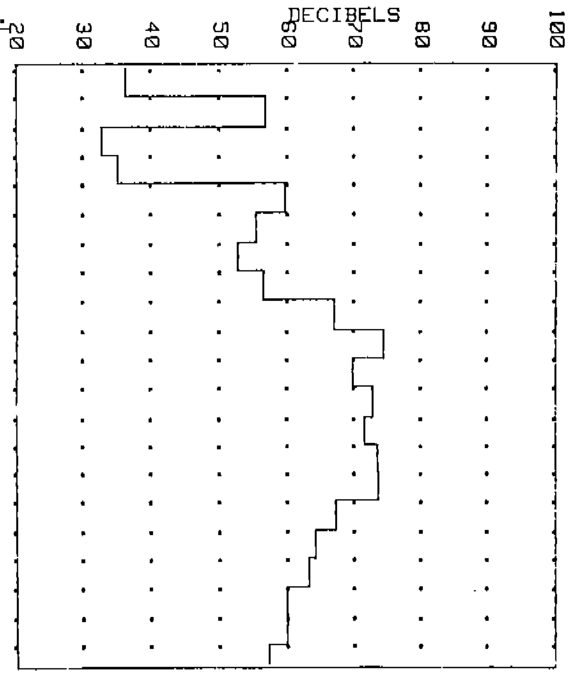

FIGURE 2
OVERALL SOUND POWER LEVEL OF ORIGINAL COMPRESSOR

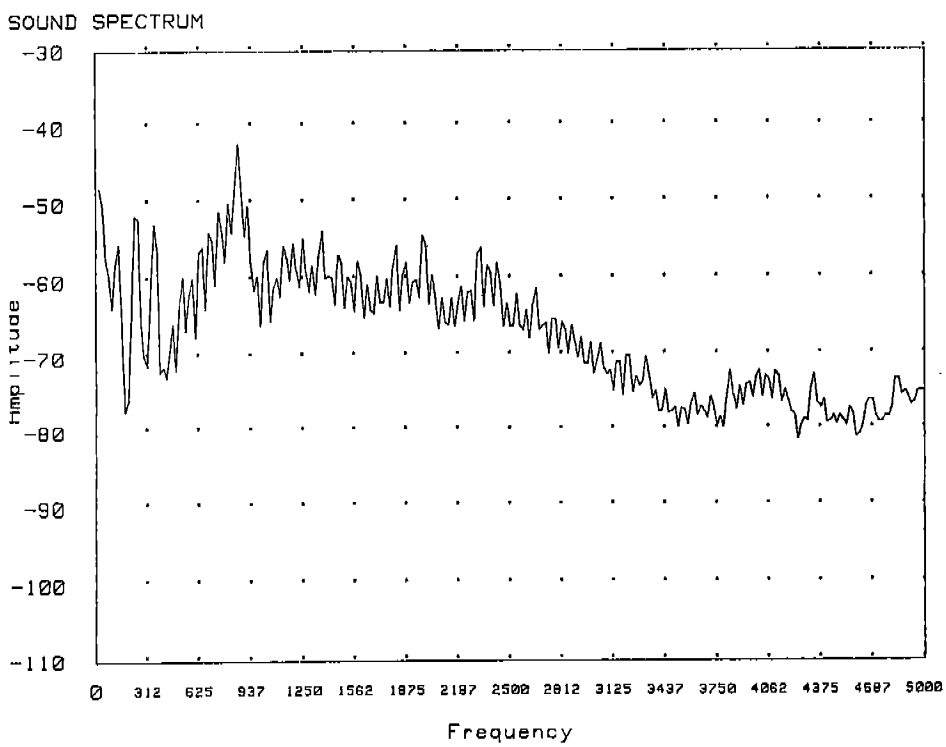

FIGURE 3
NARROWBAND SOUND SPECTRUM OF ORIGINAL COMPRESSOR

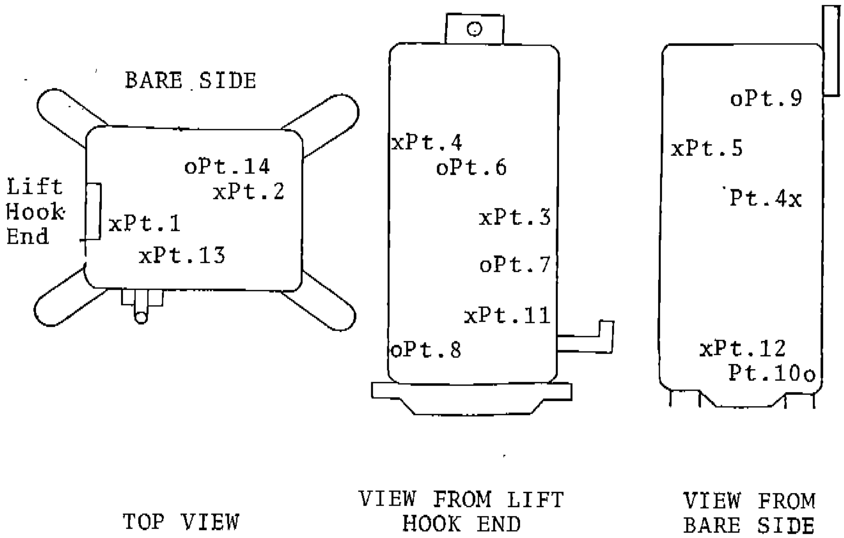

FIGURE 4: LOCATIONS OF VIBRATION MEASUREMENT POINTS

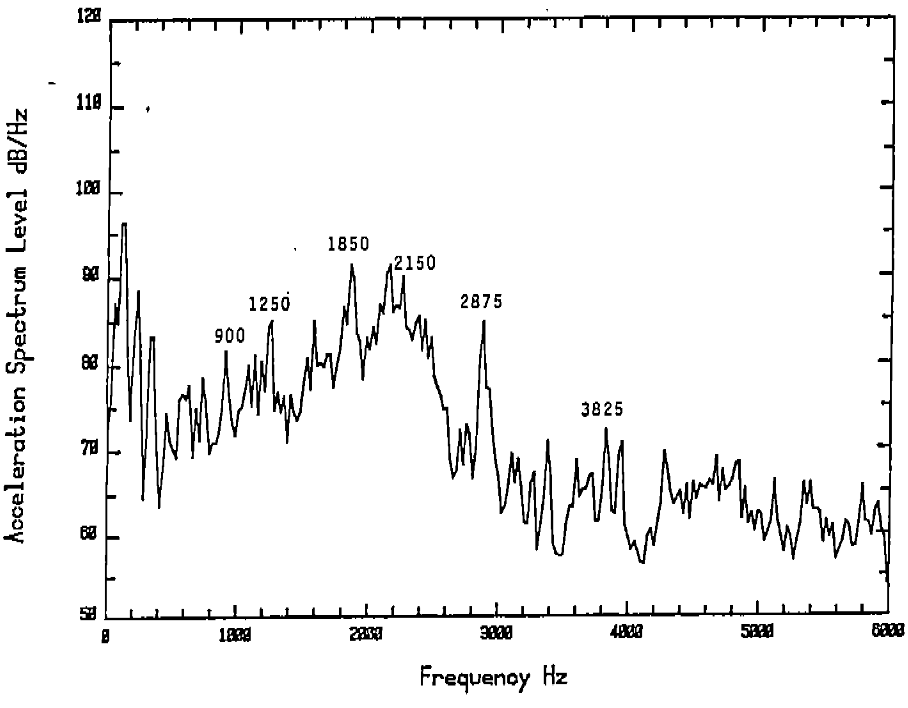

FIGURE 5: POWER SPECTRAL DENSITY OF ACCELERATION AT LOCATION #2

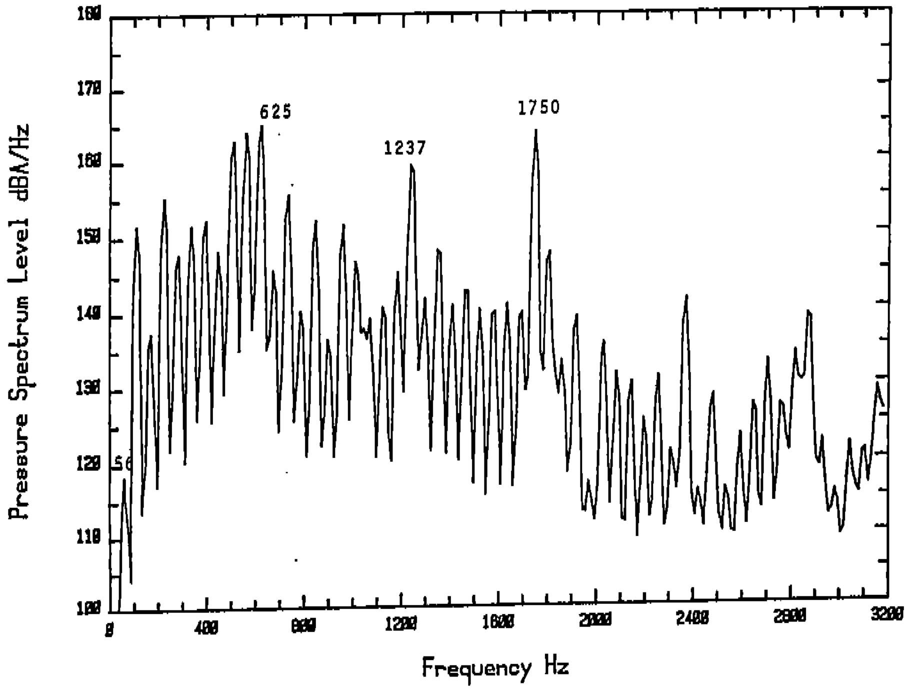

FIGURE 6: POWER SPECTRAL DENSITY OF PRESSURE MEASURED IN HEAD DURING OPERATION.

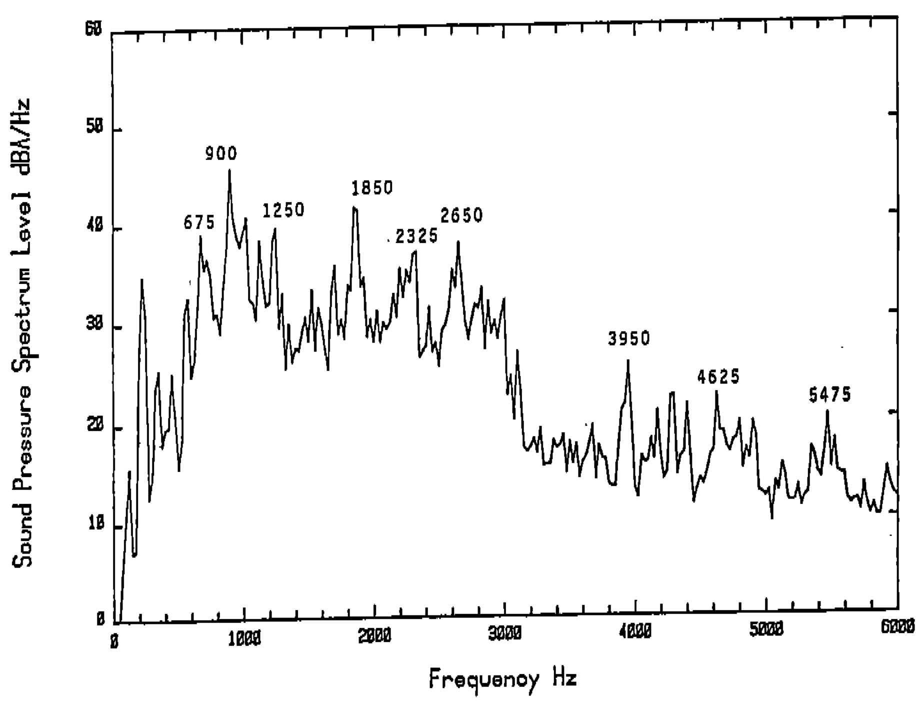

FIGURE 7: POWER SPECTRAL DENSITY OF SOUND MEASURED NEXT TO SIDE OPPOSITE LIFT HOOK.

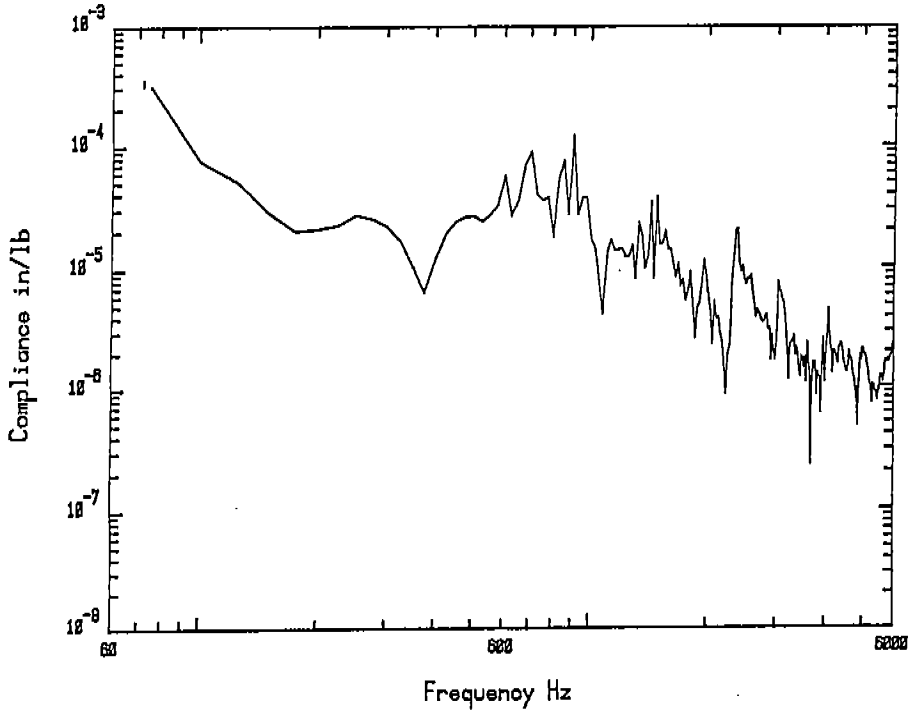

FIGURE 8: DRIVING POINT FREQUENCY RESPONSE MEASURED AT LOCATION #11.

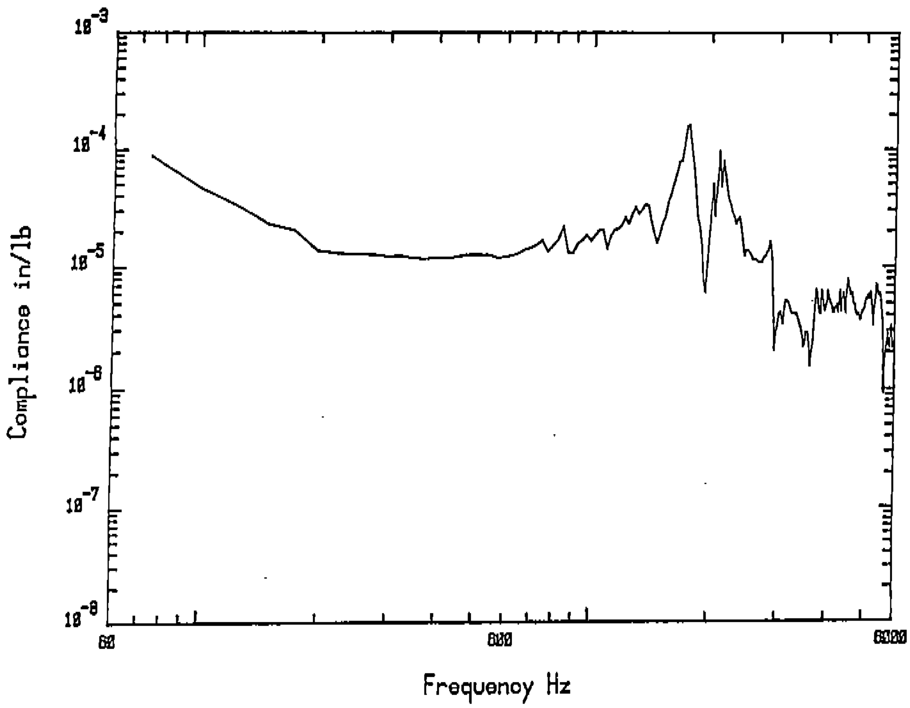

FIGURE 9: DRIVING POINT FREQUENCY RESPONSE MEASURED AT LOCATION #2.

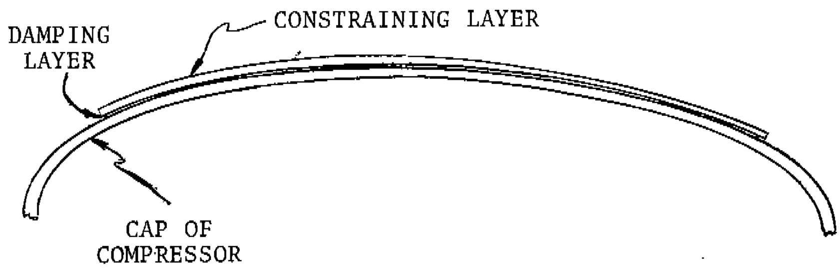

FIGURE 10: TYPICAL CONFIGURATION OF CONSTRAINED LAYER DAMPING TREATMENT

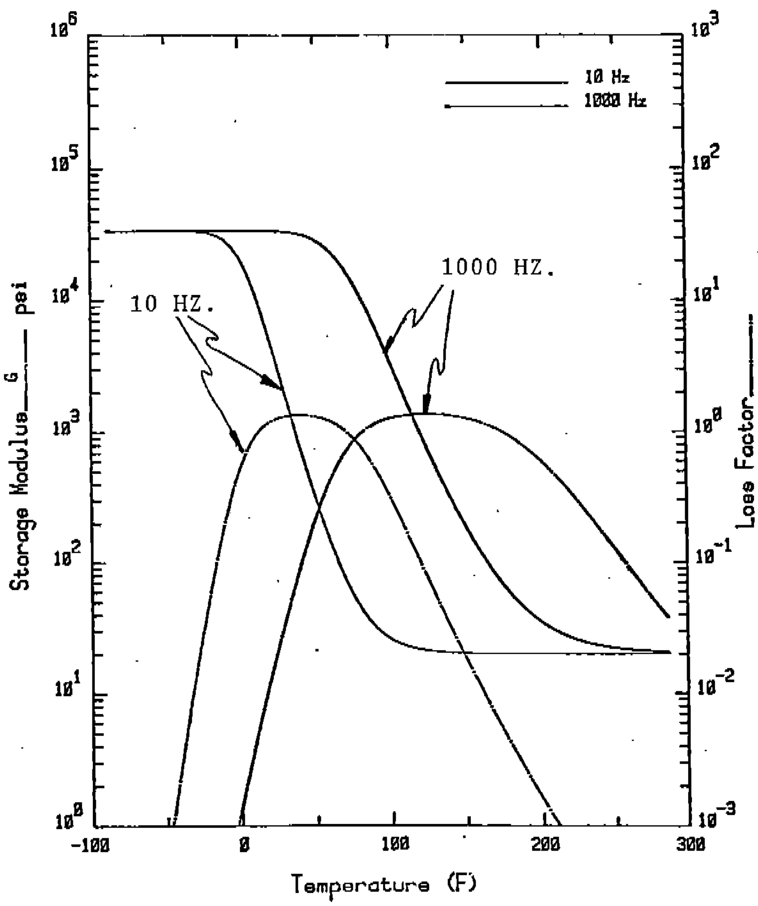

FIGURE 11: VARIATION OF THE STORAGE MODULUS AND LOSS FACTOR WITH TEMPERATURE

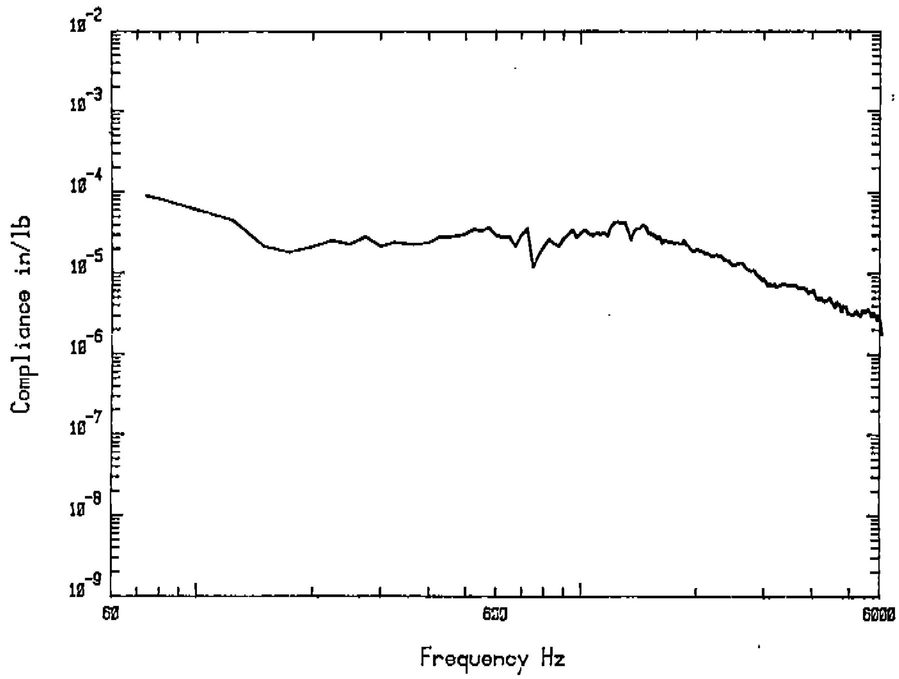

FIGURE 12: DRIVING POINT FREQUENCY RESPONSE MEASURED AT LOCATION 5 - DAMPED SHELL

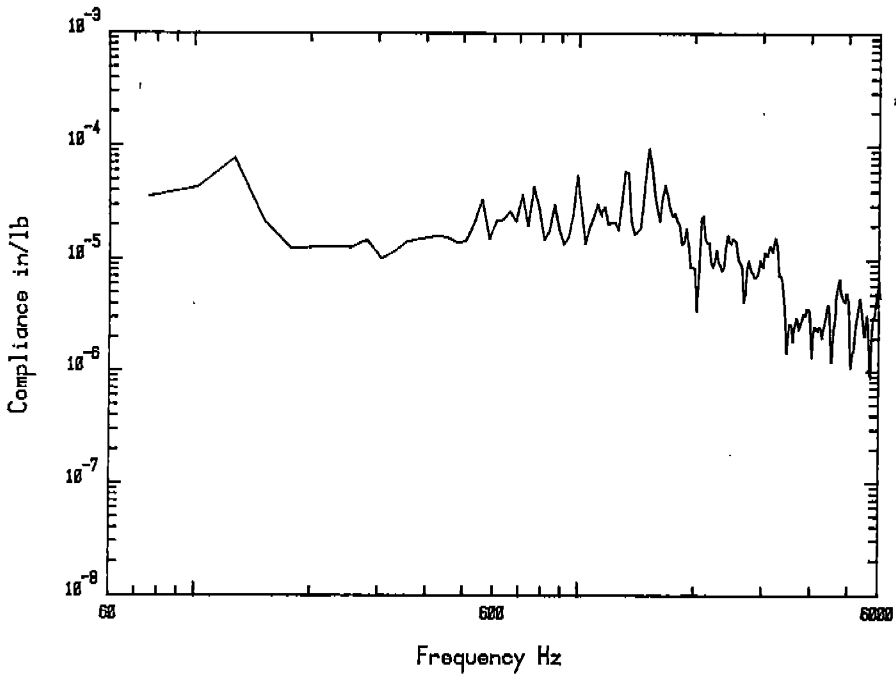

FIGURE 13: DRIVING POINT FREQUENCY RESPONSE MEASURED AT LOCATION 5 - BASELINE

LW	LWA	FREQ.
59.9	40.8	100
75.8	59.7	125
50.5	37.1	160
44.8	33.9	200
66.1	57.5	250
59	52.4	315
56.2	51.4	400
61.9	58.7	500
65.6	63.7	630
68.9	68.1	800
66.1	66.1	1000
68.7	69.3	1250
65.1	66.1	1600
63.6	64.8	2000
62.7	64	2500
61.8	63	3150
59.5	60.5	4000
58.5	59	5000
56.1	56	6300
55.3	54.2	8000
51.6	49.1	10000
79	75.8	TOTAL

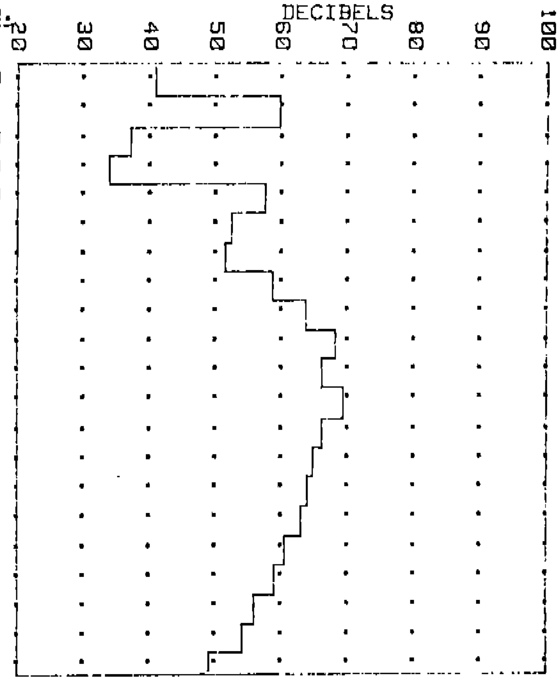

FIGURE 14: OVERALL SOUND POWER LEVEL OF MODIFIED COMPRESSOR

LW	LWA	FREQ.
61.1	42	100
76.2	60.1	125
47.1	33.7	160
47.9	37	200
69.5	60.9	250
59	52.4	315
57.9	53.1	400
60.4	57.2	500
62.7	60.8	630
71.4	70.6	800
64.1	64.1	1000
63.4	64	1250
57.4	58.4	1600
59.3	60.5	2000
57	58.3	2500
54.6	55.8	3150
53.3	54.3	4000
52.3	52.8	5000
48.7	48.6	6300
47.6	46.5	8000
51	48.5	10000
78.9	73.9	TOTAL

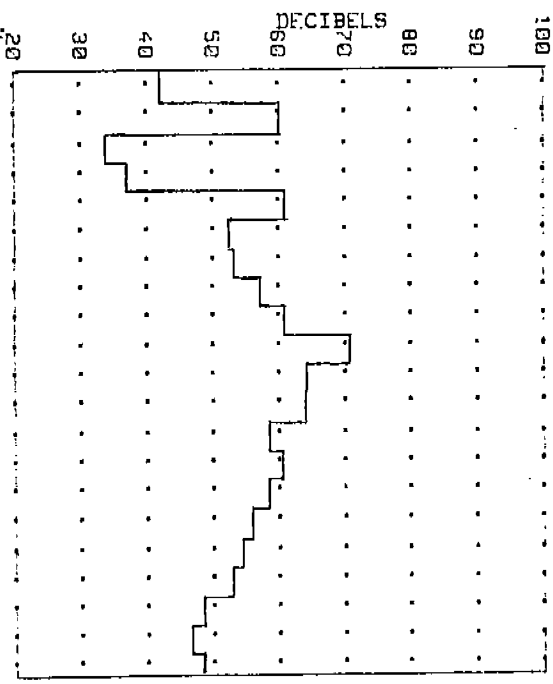

FIGURE 15: OVERALL SOUND POWER LEVEL OF MODIFIED COMPRESSOR WITH BUBBLE GENERATOR