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Effects of the Suspension System and Housing on the Noise Output of Rotating Vane Compressor

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EFFECTS OF THE SUSPENSION SYSTEM AND HOUSING ON THE NOISE OUT PUT OF ROTATING VANE COMPRESSOR.

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INTRODUCTION

Increased consumer awareness of household appliance noise coupled with pressures to build more efficient, quieter compressors is causing manufacturers to take a closer look at the noise produced by compressors. Suzuki (1) has shown that the compressor is one of the major noise sources in refrigerators, freezers and air conditioners. Johnson and Hamilton (2,3) and Shryock (4) have made extensive noise measurements of fractional horsepower rotary vane compressors. Tree and Hamilton (5,6) have summarized the findings of the previous work. One of the findings of this work was the fact that the acceleration spectrum of various parts of the compressor shell have a high correlation with the sound spectrum. That is, the compressor shell is rating most of the acoustical energy. The shell itself cannot be the noise producer. It is being driven or forced to vibrate by other parts of the compressor. There are two major paths along which the acoustical energy flows from source to shell. They are (1) the air borne path between pump and case and (2) the suspension system or other solid path (tubing, etc.) between pump and shell.

Since earlier work did not indicate the importance of these two paths, the present work was undertaken.

APPROACH AND APPARATUS

The approach used in this study was to isolate the effect that the shell has on the sound spectrum of a stock compressor. To accomplish this isolation two different compressors were used. Both were similar to those tested by Johnson and Hamilton. One was a stock compressor which had a hermetically sealed shell and the other was an open case compressor, which was a stock compressor with the shell removed.

The stock compressor was a fractional horsepower unit using refrigerant 12 as its working fluid. Figure 1 is a cutaway drawing of the stock compressor. This compressor operates in much the same way that a typical rotary vane refrigerant compressor operates. As the rotor rotates and a vane passes minimum clearance a small leaf valve opens the suction port and gas enters the cylinder. As the rotor turns, the vane sweeps out a crescent shaped volume and the gas is compressed. When the discharge pressure is reached, the leaf valve on the discharge port opens and the high pressure gas leaves the cylinder. This "first discharge" is sent through a pre-cooler and then back into the hermetically sealed shell for cooling purposes. The discharge from the shell, the "second discharge", is then sent to the rest of the refrigeration system. The compression process in this compressor occurs twice per revolution. Notice also that the shell on this compressor is solidly mounted to the pump unit, i.e. no internal suspension is used. Acoustically, this solid mounting provides a low resistance mechanical transmission path between the pump unit and shell.

The open case compressor was essentially the same pump unit as the stock compressor only with the shell removed. The only other major difference between the two pump units was that the open case pump had a strengthened cylinder head assembly to withstand the high pressure differential it would experience in as atmospheric environment. Additional subsystems were also added to preform the duties of the non-existant shell. For example, an oil separator was used to separate the oil out of the working fluid and a cooling device was added to cool the pump. These subsystems were acoustically isolated from the compressor so that only the open case compressor was under investigation. Figure 2 is a cutaway drawing of the open case compressor.

It is assumed in this study that the shell has negligible energy feedback to the pump structure and that the strengthened head assembly of the open case compressor has a negligible effect on the mechanisms which are actually producing the sound within the pump. In other words, it is assumed the character (frequency content and amplitude) of the sound being produced by the pump unit is not effected by the shell. Using this assumption any differences in the sound data for the stock compressor can be attributed to the shell and/or mounting configuration.

PRESENTATION OF DATA

Figures 3 and 4 are typical narrow band spectra for the stock and open case compressors respectively.

In a compressor of this type, there are two characteristic frequencies which need to be defined. The fundamental shaft frequency is 57.5 Hz., due to the fact that the rotational speed of the compressor is 3450 RPM. The pumping frequency is 2 times the 57.5 Hz. fundamental. This comes about because the compression process occurs twice per revolution. It follows then, that the even harmonics of the 57.5 Hz. fundamental are also the integral harmonics of the pumping frequency.

From Figure 3, it can be seen that almost all the harmonics of the fundamental shaft frequency, even and odd, from the 18th to the 119th appear. In comparing Figures 3 and 4 it is seen that in Figure 4, the open case compressor spectrum, only the even harmonics of the 57.5 Hz. fundamental are significant. If a more detailed comparison is made, one can see what is actually happening.

Figures 5 and 6 are harmonic comparison plots for the even and odd harmonics respectively. From these two figures it is evident that, for the most part, the shell and/or mounting system is passing the even harmonics of the shaft frequency (the pumping harmonics) and amplifing the odd harmonics. The mechanism by which the shell or mounting system does this amplification of the odd harmonics is unknown at the time of this writing.

Figure 7 is a cross plot of one-third octave band sound spectra for stock and open case compressors. Figure 7 gives a good overall view of how the shell and mounting system effect the sound spectrum of the stock compressor. It can be seen also that the shell system adds approximately 5 dBA to the overall noise level.

CONCLUSIONS

From the data presented in this paper one must conclude that the shell and/or mounting configuration used in this rotary vane compressor is adding 5 dBA to the overall level of the compressor. Also from the narrow band data it seems the shell or mounting system is in some way amplifing the odd harmonics of the fundamental shaft frequency.

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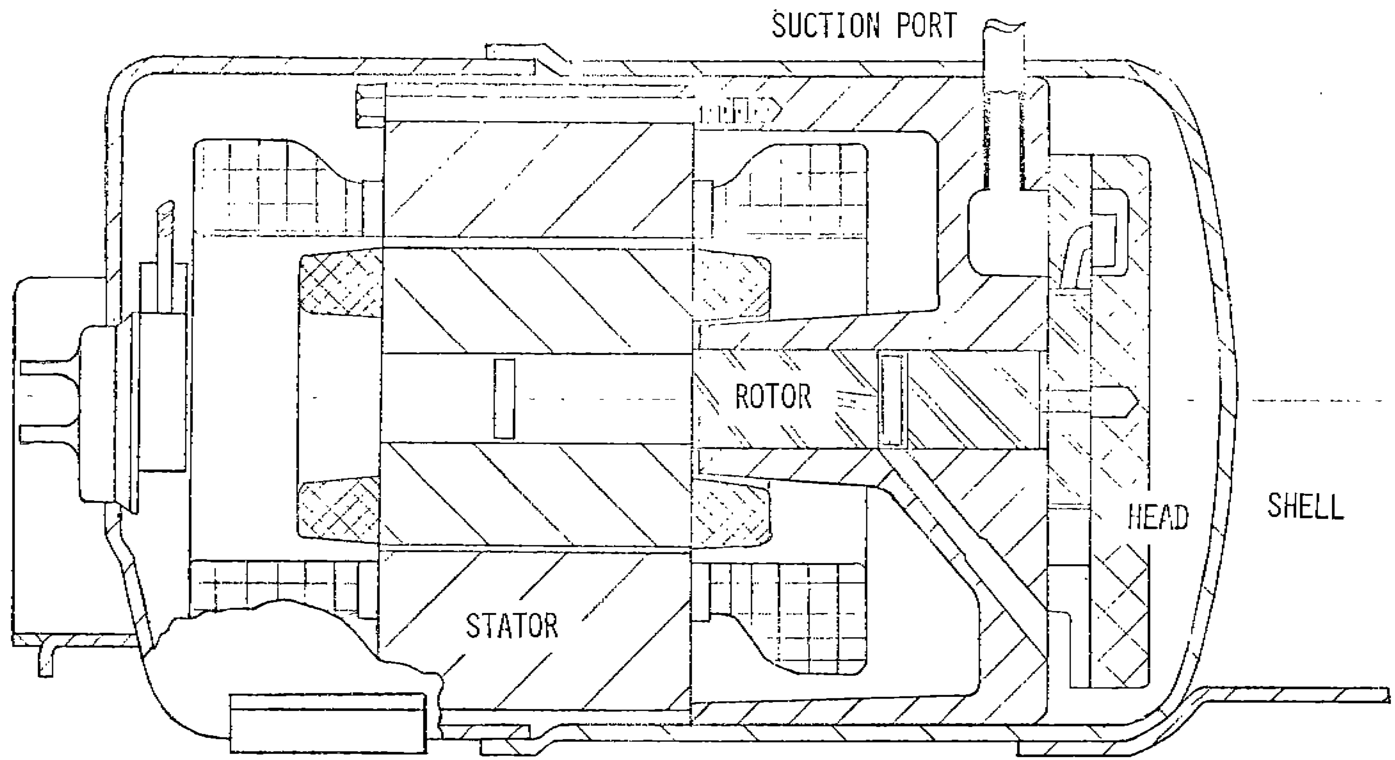

FIGURE 1. CUTAWAY DRAWING OF STOCK COMPRESSOR.

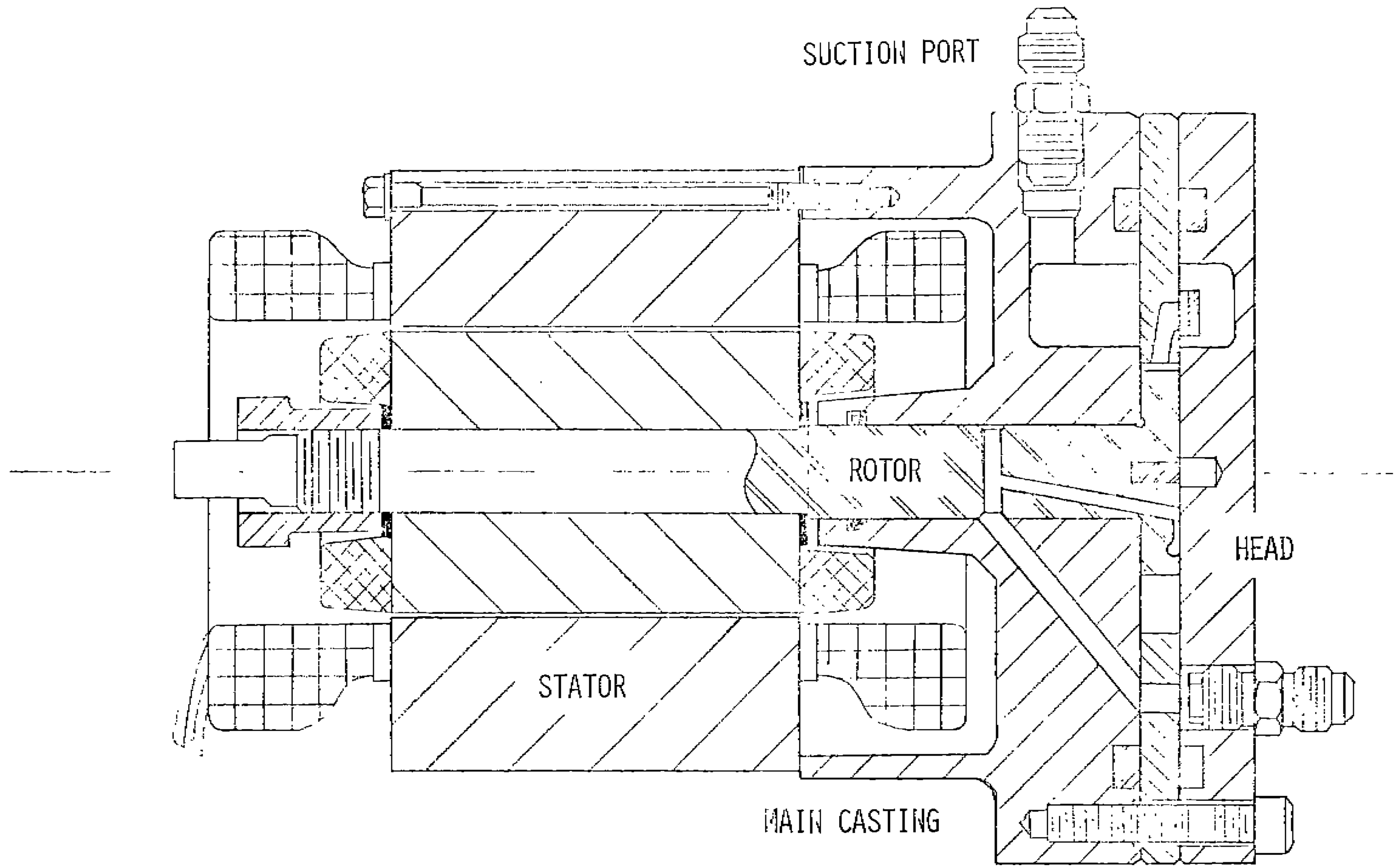

FIGURE 2. CUTAWAY DRAWING OF OPEN CASE COMPRESSOR.

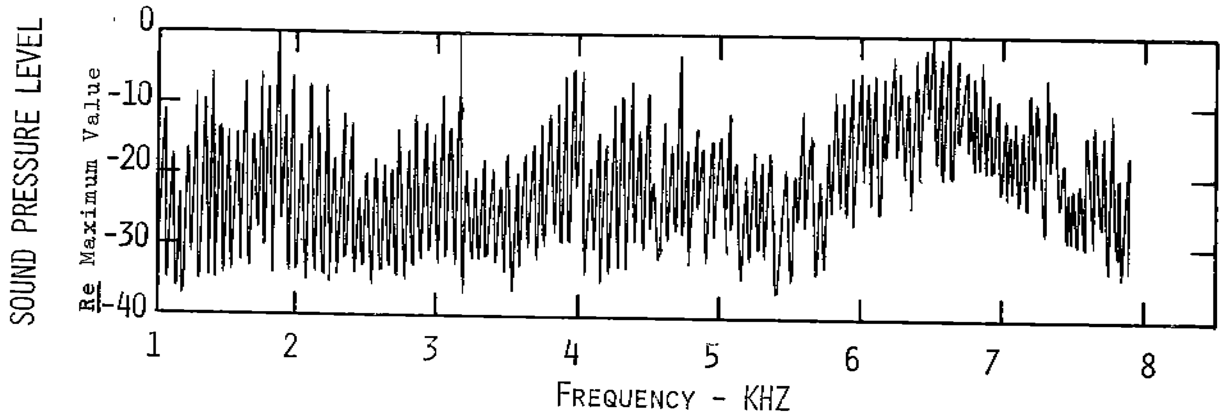

FIG. 3 TYPICAL NARROW BAND SPECTRUM OF STOCK COMPRESSOR

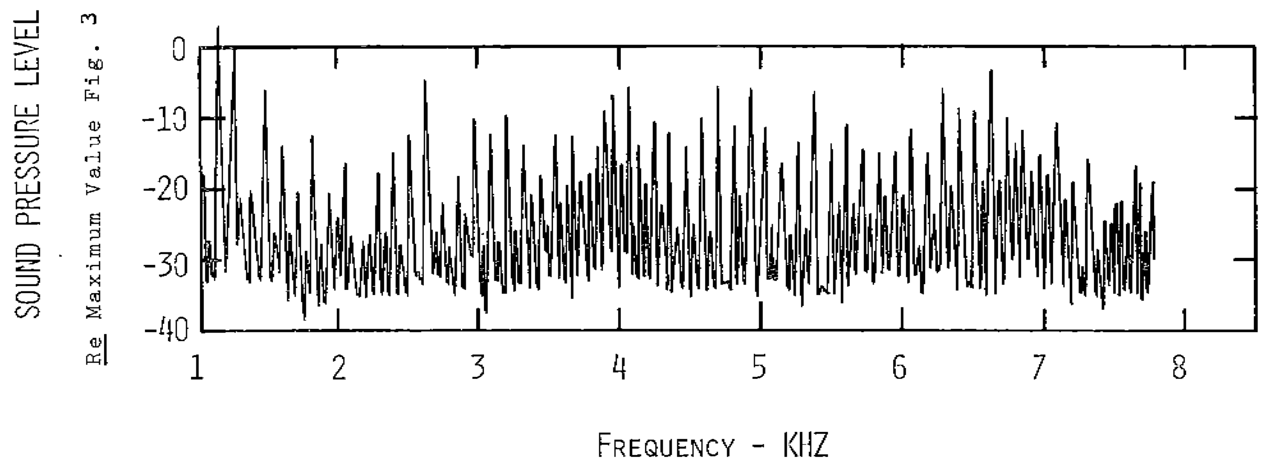

FIG. 4 TYPICAL NARROW BAND SPECTRUM OF OPEN CASE COMPRESSOR

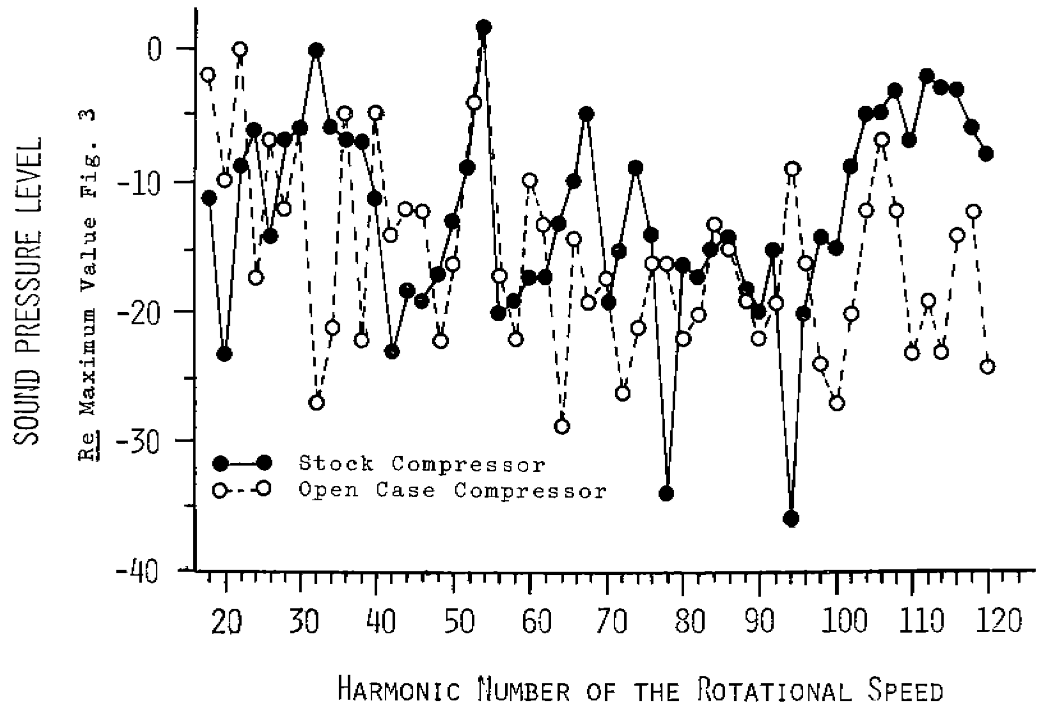

FIG. 5 CROSS PLOT OF THE EVEN HARMONIC OF THE ROTATIONAL SPEED FOR STOCK AND OPEN CASE COMPRESSORS

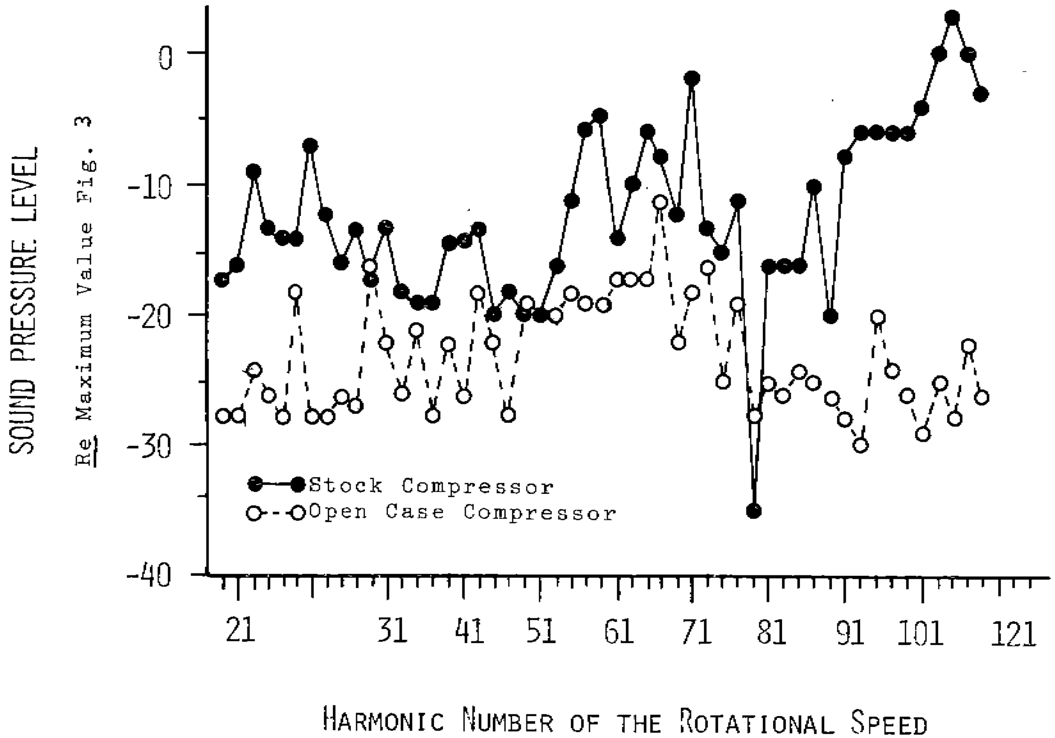

FIG. 6 CROSS PLOT OF THE ODD HARMONICS OF THE ROTATIONAL SPEED FOR STOCK AND OPEN CASE COMPRESSORS

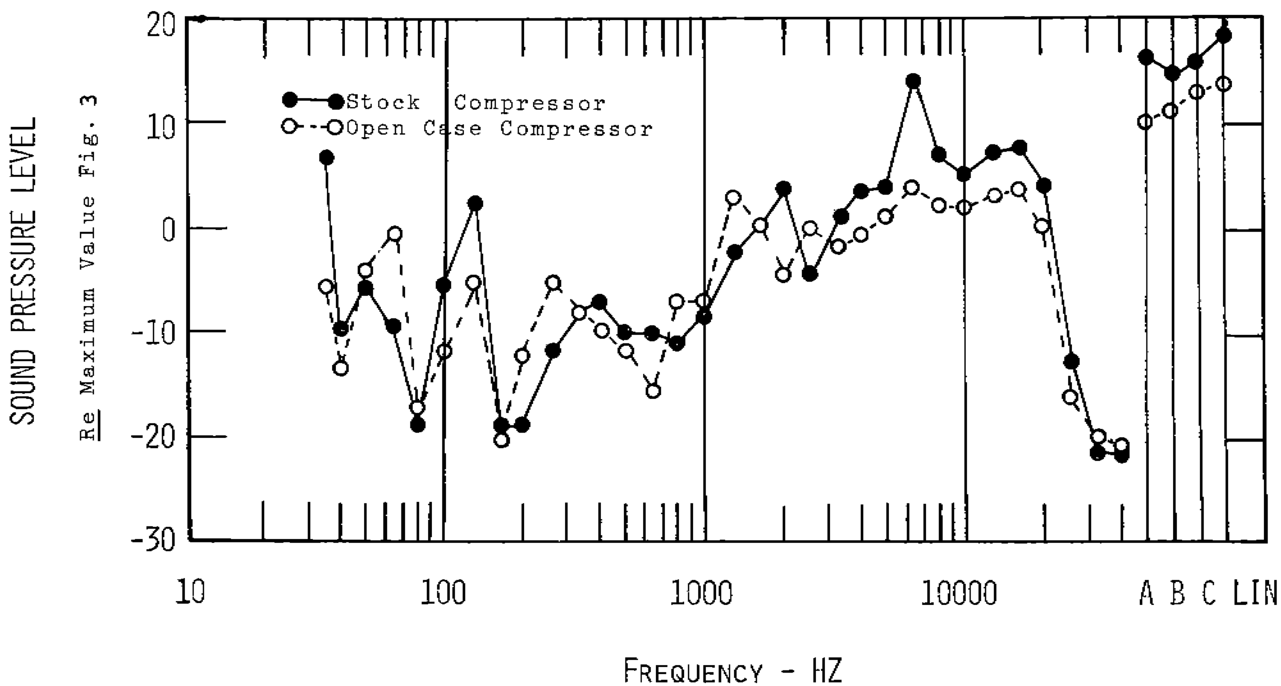

FIG. 7. CROSS PLOT OF ONE-THIRD OCTAVE BAND SOUND SPECTRA FOR STOCK AND OPEN CASE COMPRESSOR