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ABSTRACT

Sources of noise at discrete frequencies in compressors may be generated by acoustic energy in the discharge pulse exciting mechanical or acoustic resonances within the compressor assembly. In this case history, a known noise source was used in the vicinity of the discharge port to excite those resonances in a bench-type test where they could be easily identified. This test could be run in air at a room ambient rather than at the temperature and refrigerant environment of a full prototype. Modelling clay was used to experimentally modify acoustic cavities since actual compressor operation was not required. This allowed rapid evaluation of several different design proposals. A final recommendation was made and verified through testing of a sample of full prototypes.

INTRODUCTION

Rotary compressors are widely used in the world today in room air conditioning applications. Quiet operation of these units, especially in densely populated areas, has long been a concern and is steadily increasing in importance in the marketplace. As a result, compressor manufacturers are constantly seeking to reduce radiated noise, including development of methods to quickly evaluate proposed improvements. The method discussed below allows rapid screening of design proposals aimed at modification of acoustic space resonances and reduces the quantity of actual compressor prototypes which might otherwise be built for evaluation.

Objective of Study

This study was directed towards the identification and elimination of a noise source in a rolling piston type compressor used in room air conditioning applications. Figure 1 is a cross-sectional illustration. A significant portion of radiated noise was concentrated in the 1250 Hz 1/3 octave band, some also in the 1600 Hz band. The radiated spectrum is shown in Figure 2. The preliminary investigation began with the use of accelerometer measurements on the outer shell to determine the specific source of the radiated sound.

Preliminary Investigation

It was found that the top surface of the shell, compared to other regions, was most active in the 1250 Hz 1/3 octave band. Possible excitation sources could have been either internal gas pulsations or vertical vibration of the shell near the top. Accelerometer measurements taken on the sides of the shell, in a vertical direction, and near the top ruled out the vertical shell wall as a source at 1250 Hz.

In addition, it was found that the top cap of the shell responded strongly at between 1250 and 1600 Hz when excited by an impulse. Figure 3 shows the measured response. This response existed even when the cap was excited in a free state, unattached to the shell. The top of the shell would be relatively "transparent" to internal pressure pulsations between 1250 and 1600 Hz.

Time domain vibration measurements made with an internal timing mark revealed that the top of the shell was most active during the time the discharge valve was open. This is a common characteristic of rolling piston type compressors and has been discussed in some detail by Sano et al [1].

Finally, narrow band discharge pressure pulsation data taken on a running compressor without an internal discharge muffler revealed strong pulsations between 1250 and 1600 Hz as shown in Figure 4. This was an indication of a possible internal gas cavity resonance.

The working hypothesis at this point was that the upper surface of the shell was responding to pressure pulsations within the shell. The energy was generated by broadband discharge flow through the valve. Between the muffler and shell spaces, there was no effective attenuation of the 1250 Hz component of the pulse, perhaps there was resonant amplification, which excited the top shell at a characteristic resonance frequency.

Without going into a detailed study of possible internal resonances, the investigation shifted to the discharge muffler on the premise that even if a resonance did not exist there, overall performance could be improved by increasing the muffler transmission loss within the 1250 Hz band. Also, since the compressor was already fully tooled and in production, changes to the muffler, a simple steel stamping, would likely be the most economical and quickly implemented.

Possible changes to the top cap did not seem to be the best approach. It would be difficult to eliminate mechanical resonance. Any changes might only shift the resonance frequency and perhaps place the same problem in a different band.

An investigation reported by Asami et al [2] revealed that relatively minor changes to the design of the discharge muffler could result in significant changes in radiated sound in the range of 600 to 1800 Hz. The relative complexity of possible resonance modes in a part which seems simple in shape often defies modal analysis of the gas spaces. There are generally no clear, independent paths for radial or circumferential modes to develop.

TEST PROGRAM

Approach

Simply stated, a random noise source was used to excite the gas space within the discharge muffler. Measurements for any given muffler configuration were compared against the source with the muffler removed. In this manner, the transmission loss for any configuration at a frequency band of interest could be evaluated. Also, if any given modification were to induce a new resonance within the muffler, examination of the broadband response would reveal it.

Uetsuji et al [3] used the same technique to excite gas resonances within the entire compressor. A loudspeaker was enclosed in a sound-proof box. The sound was piped into the cylinder and to the discharge port, acoustically exciting the entire compressor. The sound pressure distribution in the cavity was measured and used to diagram internal mode shapes.

A somewhat similar approach is described by Hamilton [4]. There, a running compressor was used to provide flow and pressure pulsations to excite resonances in gas cavities of a non-running compressor. Only sound radiated by the non-running compressor was measured. Changes in resonance frequency with respect to temperature (and thus acoustic velocity) were studied.

The original muffler design was a simple dome with two recesses for mounting bolt holes and a central hole to allow passage of the upper bearing sleeve. A single hole near one of the mounting bolts served as the discharge into the shell cavity. See Figure 5. The most obvious modifications would involve shaping or contouring the muffler in such a way that some volume would be reduced, since the muffler already occupied nearly the maximum volume available to it.

With this approach in mind, the muffler modifications were made through the use of modelling clay to fill various parts of the muffler chamber and change its internal shape and volume. This method of modifying acoustic cavities has proven to be very effective and repeatable in previous acoustic investigations performed by the author [5].

Since testing was done in air, the results had to be modified for the acoustic velocity in Refrigerant 22. For this study, the acoustic velocity in the refrigerant vapor was taken to be 550 feet per second (168 meters per second), which is approximately half the acoustic velocity of air.

Test Setup

Test were conducted alternately in either an anechoic room (free field) or semi-anechoic room (semi-free field). A random noise generator and amplifier was used to power a horn driver. The driver was attached to a 5/8 inch (16 mm) copper tube through threaded fittings and located outside the test room. The copper tube was run into the room and reduced down to a 1/4 inch (13 mm) tube. Steel wool packed loosely into the ends of the tube served to damp out tube resonances.

The tube was then run into a partial assembly of the compressor cylinder, shaft, piston, bearing, valve, and muffler. The tube was positioned in the discharge valve port with the valve reed removed. The stop was left in place with a spacer underneath in place of the reed to maintain the normal stop position.

For the free-field measurements, a 1/2 inch (13 mm) normal incidence microphone was placed approximately six inches (152 mm) from and in line with the muffler discharge. For the semi-free field measurements, a 1/2 inch (13 mm) random incidence microphone was attached to a long hand-held rod and tied to the compressor subassembly with a 12 inch (305 mm) string. The microphone was then swept over the subassembly in a 180 degree arc. While this latter method was found to be much less sensitive to setup variations, both methods yielded similar results.

Microphone signals were analyzed using a narrow band frequency analyzer and the results plotted graphically. Visual comparison of frequency domain plots obtained using modified and unmodified mufflers was used to determine whether any change in transmission loss resulted from a particular change. As a later refinement in technique, raw data was downloaded from the analyzer to a PC-based spreadsheet and transmission loss was analyzed digitally.

For each modification, a test was first run without the muffler in place to provide a baseline or "zero attenuation" measurement. Next, the then-current production muffler was tested to provide an ongoing check of repeatability. Finally, the modified muffler was tested.

Sequence of Testing

The muffler cavity was roughly divided into six tangential zones as illustrated in Figure 6. Modelling clay was used to fill each of these zones and the response to the random noise input was measured. In addition to this series, additional modifications were tested. Included were:

(1) Selected zones filled to half radial width.
(2) Selected zones filled to half circumferential width.
(3) Reduced height, or "headroom".
(4) Reduced overall diameter of cavity.
(5) Relocated discharge hole.
(6) Selected combinations of above modifications.

The muffler design in use featured clearances at both the outside and inside diameters, that is, the seams between the muffler and the compressor frame were not sealed. These gaps were sealed with lead tape in some other tests.

As hoped, some modifications resulted in large changes in the insertion loss at around 1250 Hz for Refrigerant 22. The change in insertion loss for one of these sets of modifications (tested in air) is shown in Figure 7. Note that the peak at around 2500 Hz for the test in air corresponds to around 1250 Hz in refrigerant vapor.

After completion of the test series, those modifications which resulted in significant attenuations of transmitted noise around 1250 Hz were prototyped using then-standard mufflers and epoxy filler to duplicate the forms generated with the modelling clay. These were tested in running compressors to verify reduction of radiated noise at 1250 Hz.

RESULTS

The most significant results were obtained with modifications which affected zones 1, 2, 5, and 6. Changes to the overall diameter of the muffler cavity also had noticeable effect. Changes to the vertical height, muffler discharge location, and edge sealing had no appreciable effect.

A final design which incorporated reductions in volume primarily in zones 2 and 5 was placed in production. Changes in the radiated sound spectrum are shown in Figure 8. On another larger, but similar, model, a small change in the overall cavity diameter was implemented which provided a similar benefit.

CONCLUSIONS

A good correlation was observed between results obtained using the random-noise excitation in air and prototypes tested in a running compressor. While the series of clayed-in modifications was reminiscent of a cut-and-try approach, the ease and speed of test evaluation provided by this method greatly reduced the disadvantage normally posed by such an approach. If a particular modification failed to yield a useful result or insight, the investment in the test was extremely small compared to a test in a running compressor. Also, a methodical testing of various zones within the acoustic cavity helped determine quickly which areas influenced the transmission of acoustic energy in the frequency band of interest.

ACKNOWLEDGEMENTS

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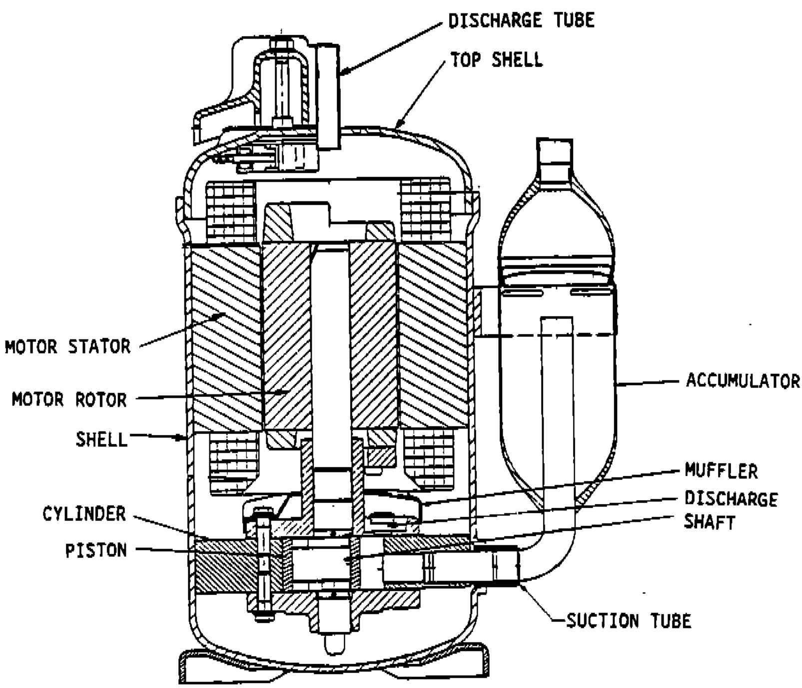

Figure 1 Cross Sectional View

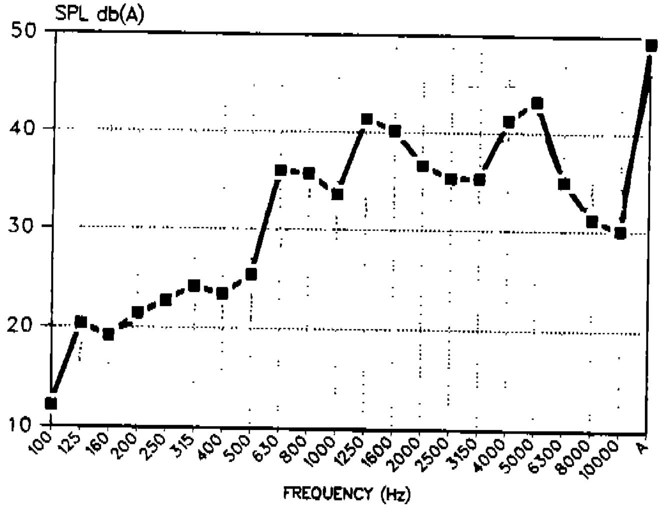

Figure 2 Original sound spectrum

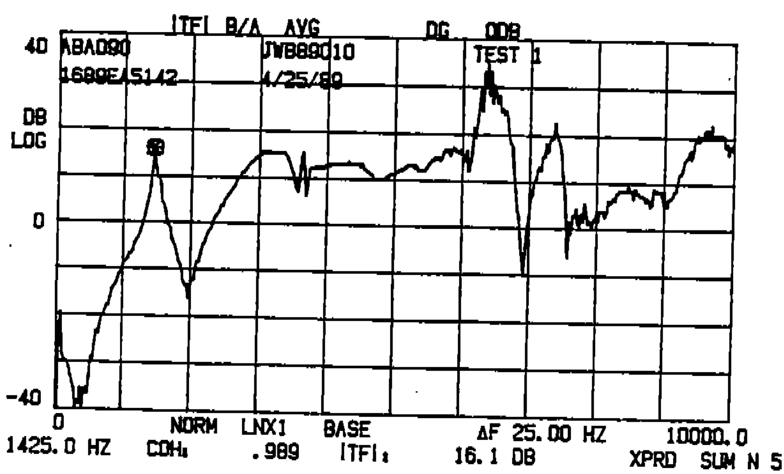

Figure 3 Inertance of Top Shell

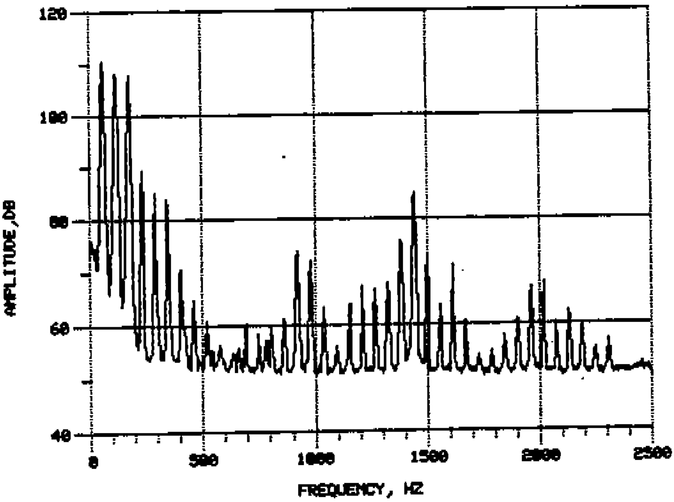

Figure 4 Discharge Pulse Spectrum

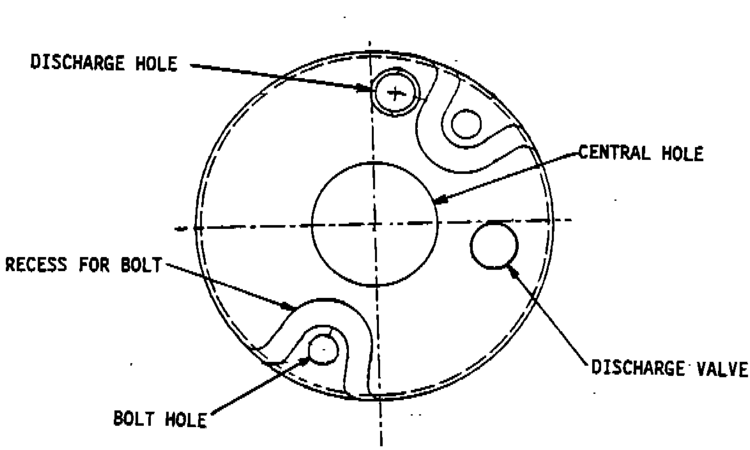

Figure 5 Discharge Muffler Diagram

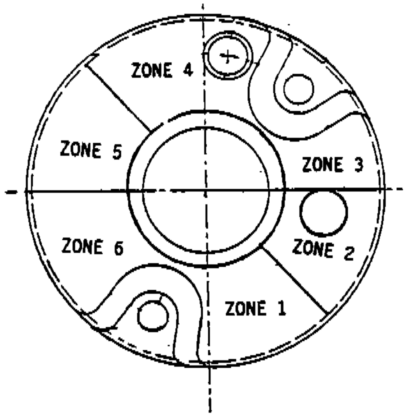

Figure 6 Muffler Zones

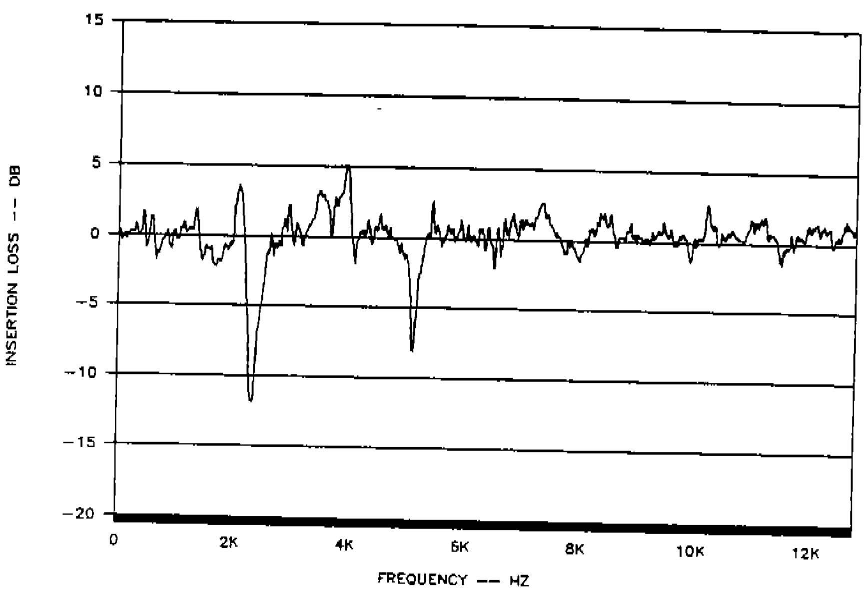

Figure 7 Modified Muffler Transmission Loss

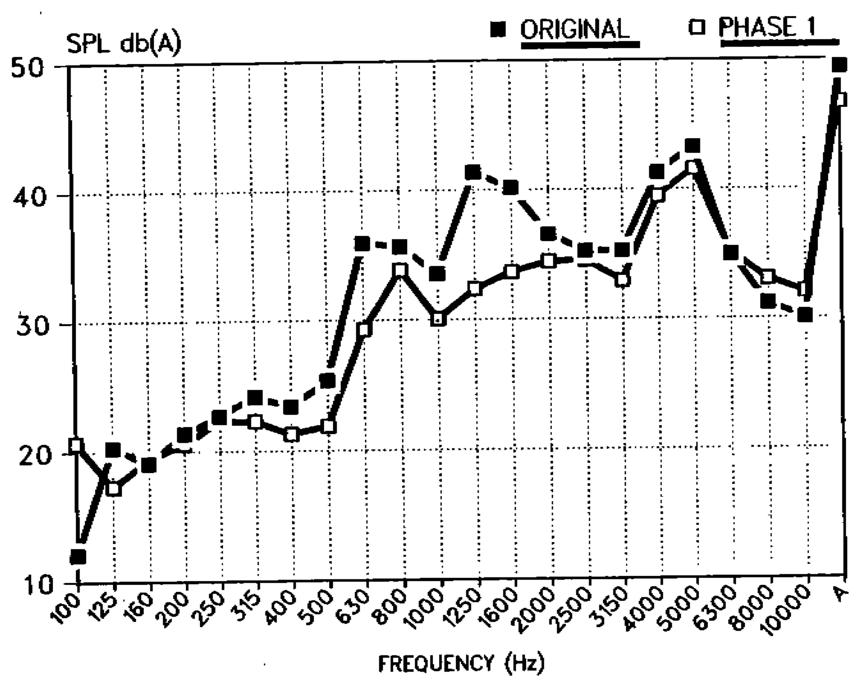

Figure 8 Modified and Original Sound Spectra