Understanding Noise Radiation from a Hermetic Compressor

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Noise radiation from a hermetic, reciprocating compressor can be subdivided into two areas according to the type of compressor shell motion. Noise radiated by solid body motion is noise produced by solid body back and forth motion of the shell. The second area is noise produced by local deformations of the shell. In other words, the first area is concerned with noise generated by the entire shell moving as a whole; while the second area is concerned with noise generated by parts of the compressor shell moving in phase or out of phase with respect to each other.

Solid body motion of the shell predominates at frequencies below 500 Hz and is usually not important unless a resonance involving the shell, gas, and internal assembly coincides with the running frequency of the compressor or one of its multiples (i.e. twice the running frequency, etc.)

The second area of noise radiation due to shell deformation is more important and is a function of the shell design. This area of noise radiation will be the topic of this report. Typically noise radiation caused by shell deformation begins in the range of 630 to 2000 Hz. The lower value might be found in a large compressor shell designed for a ten ton air conditioning application. It might also be found in a shell design for a five ton application having a poor choice of design parameters from a noise viewpoint. It should be remembered that design parameters are frequently a compromise between performance, cost, production capability, and other factors. The higher value for the inception of deformation-caused noise usually occurs in small hermetic compressors. Examples might include compressors in refrigerators, freezers, or in somewhat larger shells having a good choice of design parameters. A typical example of the latter case would be a window unit compressor. As a general rule, increasing the shell size tends to lower the frequency at which noise radiation due to deformation occurs, while a careful selection of the design parameters tends to raise this frequency.

Shell designs currently in use exhibit a wide variety of geometries, the simplest of which is the cylinder. However, very few compressor shells are truly cylindrical. One possible exception to this is a shell which appears to be constructed from 0.25 inch pipe with welded end plates. Another widely used compressor has a circular horizontal cross section but is a two-piece shell with a centerline girth weld and an approximately hemispheric top and bottom. Other designs commonly used are oval in horizontal cross section. In addition to the wide variations in cross section, some manufacturers use different thicknesses for the top shell and the bottom shell.

Replacing the real compressor shell with an ideal cylindrical shell of equivalent radius, height and thickness, results in a model which demonstrates qualitatively the behavior of compressor shells. The closer the compressor approaches this ideal model, the better will be the quantitative agreement.
The analysis of this simple configuration is not straightforward for there are many theories to describe the dynamic behavior of cylindrical shells. However, hermetic compressor shells typically have a thickness to radius ratio of .025 to .050 and thus may be analyzed by thin shell theory without loss of accuracy in the analysis.

**Thin Shell**

Shell thickness/shell radius

.025 to .050

The thin shell theory may itself be considerably simplified by neglecting certain terms related to longitudinal inertia. The neglect of these terms is negligible for the shell height/shell radius ratio that is commonly found in compressor shells. This simplified theory is called “shallow shell theory” to differentiate it from the larger area of thin shell theory.

**Shallow Shell Theory**

Finally, it will be assumed that the top and bottom of the shell are constrained radially, that is, no radial deformation is evidenced at the upper and lower boundaries of the ideal cylindrical shell under consideration. It will also be assumed that the shell is without axial constraint.

**Freely-Supported Shell**

This particular boundary condition is called the freely-supported condition. The shell to be discussed then, is the freely-supported, thin, shallow cylinder.

To obtain satisfactory agreement between the behavior of the model shell and the actual shell, certain modifications or refinements are necessary.

**Other Factors**

Internal Pressure

Internal Liquid

Shell Curvature

Stiffeners

One modification is the inclusion of the effects of internal gas pressure on shell behavior since the pressure difference across an actual compressor shell varies depending on the suction pressure.

A second effect that needs to be considered is the effect of internal gas density and internal liquid level on the shell behavior since refrigerant density and oil level in the shell may vary as a function of operating conditions. Because most shells are not cylindrical, the effect of curvature along the vertical axis on the shell behavior also needs to be considered. Finally the effect of circumferential stiffeners on cylindrical shell behavior is useful in explaining the changes in behavior due to the shell overlap and girth weld in two-piece shells.

To readily understand shell behavior, an understanding of terminology commonly employed in shell theory is necessary. If a compressor shell is struck with a hammer, the shell vibrates at preferred, natural frequencies which are excited by the blow. These frequencies are described by two numbers which are related to observed circumferential and axial shell motion. The first number corresponds to one-half the number of circumferential nodes observed.

A node is a line along the cylinder where no motion occurs. An antinode is the line along which maximum radial shell motion occurs. The second number corresponds to one-half the number of axial nodes observed. If the pattern associated with a particular mode shows a circumferential pattern like the second figure in (a) of the example and an axial pattern like the first figure in (b), the mode identification would be the 3, 1 mode. The first number, \( n \), refers to the circumferential pattern whereas the second number, \( m \), refers to the axial pattern.

In Figure 1 the theoretical curves of natural frequency versus mode numbers for two different thicknesses of a freely-supported shallow cylinder with no pressure difference across the shell are compared to the observed results of an elliptical compressor shell with a major axis/minor axis ratio of 1.1. It should be remembered that the curves are really discontinuous and that only integer mode numbers exist.

From this figure it should be observed that the first natural frequency of the shell along the frequency axis corresponds to the 3, 1 mode; and this frequency occurs at around 1150 Hz for the 11 gauge shell and 1240 Hz for the 9 gauge shell. The results in Figure 1 show reasonable agreement between theory and observation justifying the simplifications which were made.

In Figure 2, the behavior of several \( n, l \) modes as a function of pressure are shown for a compressor shell design with a rectangular cross section. Note that the four natural frequencies shown for this shell occur below the first natural frequency for the shells shown in Figure 1. Also, note that the 2, 1 mode is the first mode to be observed along the frequency whereas the 3, 1 mode occurs first for the oval shell design shown in Figure 1.

The Berry-Reissner theory (referenced in bibliography) predicts that:

\[ f = f_0^2 + \text{constant} \times \Delta P \]

where \( f \) = natural frequency for differential
pressure $\Delta P$

$f_0$ = natural frequency for no pressure difference

The observed results are in good agreement with this relationship. The observed increase in frequency due to increased pressure may be thought of as arising from an increased stiffness of the shell due to the internal gas pressure.

**Increased Pressure Raises Frequencies**

The importance of this result lies in the fact that it would be very difficult to design a shell such that the low order natural frequencies do not coincide with a multitude of running frequencies. In fact, due to variations in suction pressure, there will always be some condition at which the two will coincide. Sound rating a compressor at one condition gives no assurance that there may not exist another condition at which the rating will be considerably different.

Also of interest in Figure 2 is the fact that there are two 2, 1 modes and two 3, 1 modes shown. The appearance of more than the theoretical prediction of one mode for each value of $m$ and $n$ is always observed (even for cylindrical shells in which great care has been exercised in their manufacture). The appearance of these additional modes has been attributed to "imperfections" in the shell manufacture. These imperfections could be due to out of roundness or variations in thickness of the shell.

Aside from the dynamic characteristics observed between the design of the shell shown in Figure 1 and 2, there exists another important difference in the behavior of the two shells. For two shells of the same general geometric characteristics such as height, thickness, and average radius, the shell having the lowest natural frequency will have the poorest performance under static burst testing. Now this performance comes about may be seen from considering the 0, 1 mode of a compressor. Extrapolating the curves in Figure 1 to their intersection with the abscissa, one obtains the resonant frequency for the "breathing" modes in which there are no modes. In this type of motion, the shell exhibits the same type of motion as a balloon in which one alternately exhales and inhales. The shell design of Figure 1 has a much higher frequency for the breathing mode than the shell design of Figure 2. The lower the frequency corresponding to this breathing mode, the more probable is the deformation of the shell under burst testing where a high internal pressure is applied to the shell. This applies both to temporary deformation of the shell which is released when the pressure is released and permanent shell deformation which remains after pressure release. Permanent deformation was found to occur in the shell of Figure 2 at a pressure about one-fifth that of the shell shown in Figure 1. It is apparent that good shell design may have additional benefits besides control of noise radiation.

The effect of refrigerant on the observed frequencies can be seen in Figure 3 where the observed frequency of the 2, 1 modes is shown as a function of pressure for air and R-22.

The differences between the values obtained by air and refrigerant follow from the Berry-Reissner theory and again underline the impossibility of designing a tuned shell.

**Denser Gases Lower Frequencies**

In Figure 2 the increase in frequencies was attributed to an increase in stiffness due to pressure. In this case the decrease in frequency due to gas density may be attributed to an increase in the apparent inertia of the shell walls due to presence of an internal gas. Increases in oil level and liquid refrigerant-oil level should also lead to a decrease in the frequency observed for this mode.

The effects of stiffeners and shell curvature can also be understood from shell theory, and for those interested in this area, one may refer to the short appended bibliography. Circumferential stiffeners increase the lowest natural frequencies of a shell but are not economically feasible.

**Curvatures Raise Frequencies**

Increasingly complex or negative curvature in a shell design lowers the natural frequencies of a shell and should be avoided. Increasingly positive curvature raises the natural frequencies of a shell.

In summary, shallow shell theory adequately predicts the observed behavior of the natural frequencies of compressor shells as a function of design, internal pressure, etc. The results demonstrate that designing a "tuned" shell is impossible. A great deal of work remains to be done before a completely adequate theory of compressor shell behavior exists. Until that time, the final assessment and optimization of a shell design must wait. The field of noise radiation from compressor shells will continue to be an area of fertile investigation for some time.

**Selected Bibliography (in order of difficulty)**

   Simple survey of reciprocating compressor noise.

   A comparison between experiment and theory of a pressurized shell.

   One of the classic papers in the field. The freely-supported shell is investigated in detail.

   The application of shallow shell theory to a pressurized cylinder containing a fluid.


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Comparison of theoretical and observed natural frequencies for two different shell gauge thicknesses

- Theory
- Observed

![Graph showing comparison of theoretical and observed natural frequencies]
The square of the natural frequency for some lower order modes of a compressor shell as a function of pressure inside the shell (air)

NOTE: $f_o$ is the natural frequency at no pressure difference.

The effect of air and refrigerant (R-22) on the observed frequency of the lowest mode of a compressor shell as a function of pressure inside shell.