An Improved Shape for Hermetic Compressor Housings

D. C. Lowery

An Improved Shape for Hermetic Compressor Housings

David C. Lowery, Research Engineer
Tecumseh Products Company, Research Laboratory
Ann Arbor, MI 48104

ABSTRACT

An improved acoustic design for a hermetic compressor shell has been developed. The entire shell surface is described by a single mathematical expression which insures that there are no discontinuities in either the surface or its curvature. The curvature of the improved housing is maximized within the geometric constraints of the size of the interior pumping mechanism and the specified maximum outside dimensions. The resulting housing, in the case of a small refrigeration compressor, has natural frequenices about one octave higher than exist in conventional housings. These higher resonant frequencies coupled with proper acoustic design of the rest of the compressor can result in significantly lower sound power levels.

INTRODUCTION

Compressors transmit sound in many ways: through connecting tubes, through the mounting feet and through the air; but the major sound radiator is the compressor housing. Two methods are available to lower such housing radiation. The first involves reducing the amount of energy which reaches the shell by making improvements to the compressor inside the housing. However, as Figure 1 shows, there are a number of interconnected paths through which sound energy can reach the housing, and determining which component or path is significant to the overall sound level can be a difficult task.

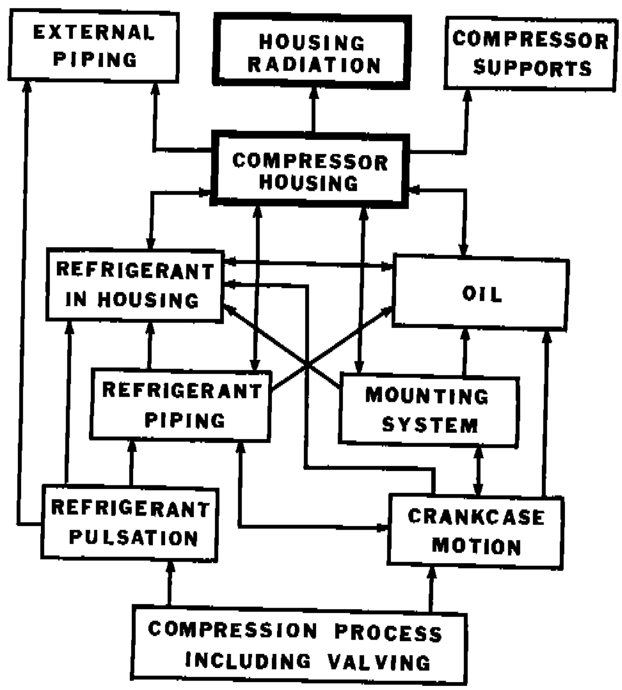

FIGURE 1. SOUND TRANSMISSION FLOWCHART

The second approach is to improve the housing by reducing its response to the sound energy reaching it. The housing has long been recognized as a significant influence on compressor sound radiation [1-2] and attempts have been made to find quieter housings [3-5]. In general, the housing responds in two distinct modes. At frequencies below the onset of bending modes, the housing vibrates as a rigid body with no deformation from its "static" shape. In this mode, relatively large amounts of energy are required to sustain the housing vibration. Sound problems in this frequency range are usually associated with large forces, for example, those produced by a resonance of the refrigerant within the housing cavity. Because the housing is softly supported in order to reduce vibration transmission, its inertia controls the amount of rigid body motion. Once the onset of bending modes is reached, the second form of housing radiation begins. At these frequencies the housing deforms from its "static" shape into other shapes described by "resonance modes". These modes require much less energy to sustain and, therefore, account for most of the compressor sound problems. This type of sound problem has also proven to be the most difficult to solve. This paper presents a housing design philosophy that shifts the resonance modes up in frequency, thereby eliminating an entire range of sound problems.

REDUCED HOUSING RESPONSE

Figure 1 represents a simplified description of the noise transmission paths for a hermetic compressor. Each path can be described as a filter with a frequency response function relating the output sound or vibration to the input. In the case of a housing, the inputs are the forces and pressures exerted on the housing by the internal mechanism, and the output is the sound radiated by the shell. Because this is a complicated system with both localized and distributed inputs, the mechanical transfer function relating shell motion to individual localized forces is usually used to represent the frequency response of the housing. It has been well established that the high frequency noise problems occur at the resonance frequencies of the mechanical transfer function. Two methods are generally available to reduce the mechanical response of the housing: (1) additional damping and (2) additional stiffness. Added damping works to reduce the response at the resonance frequencies where the noise problems occur, but while it reduces the amplitude of the resonance peak, it also broadens the frequency range where the resonance mode is important.

Increased stiffness serves to increase the frequency of the resonance modes while reducing the response below the first resonance frequency. This increases the frequency of the sound problems associated with bending modes to an area where there is less energy available to excite them. The result is a drop in the sound level across the frequency band extending from the lowest natural frequency of the "soft" housing to the lowest natural frequency of the "stiff" housing. At these higher frequencies the resonance modes should also be spaced farther apart which can cause additional reductions in the sound level. Acoustic treatments are also more effective at high frequencies. Because increased stiffness raises the frequency for the onset of sound problems, and thereby reduces their frequency range, it should be considered before undertaking more costly measures such as increased damping.

INCREASED HOUSING STIFFNESS

Ideal shapes such as the sphere and the ellipsoid have long been recognized as good housing shapes [5] but are usually too large with excessive surface area at sizes which will enclose the pumping mechanism. On the other hand, conventional housings have conformed closely to the shape of the inner mechanism and have usually been straight-sided cylinders, often with elliptical horizontal sections. In contrast to the ideal shapes, conventional housings have had little extra internal volume or external surface area.

The stiffness of compressor housings can be improved by increasing their curvature (reducing their radii of curvature). For straight-sided cylindrical housings the most effective method of adding curvature is to radius the flat sides resulting in the "barrel" structure shown in Figure 2. The stiffness of the barrel shape can be further increased by eliminating the abrupt changes in radius of curvature that occur at the "blend-points" shown in Figure 3. Such "blending" is common in conventional housing design, but is not present in spheres and ellipsoids. When considering their strength, compressor housings are usually modeled by thin shell theory which ignores bending stresses. Flügge [6] points out that thin shell theory predicts two different stress states at the blend point depending upon which radius of curvature is used. In real housings bending stresses resolve this

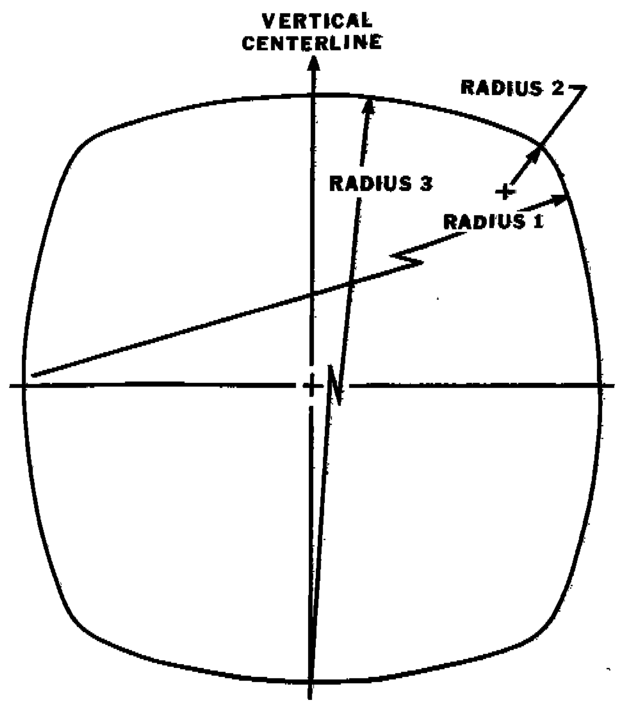

FIGURE 2. SECTION OF AXISYMMETRIC "BARREL"

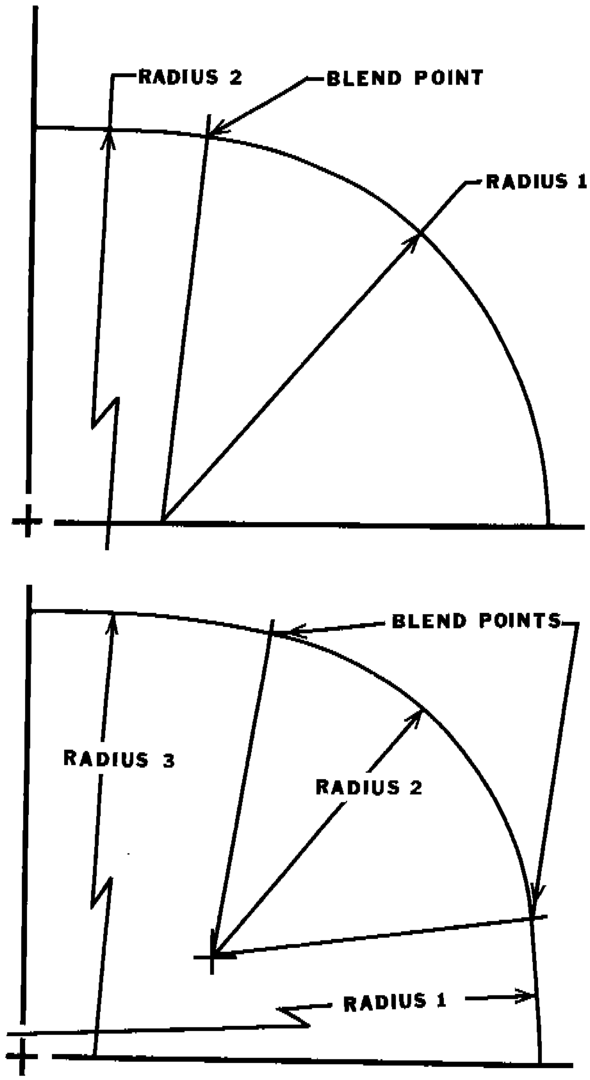

FIGURE 3. TYPICAL SECTIONS OF CONVENTIONAL HERMETIC COMPRESSOR HOUSINGS

conflict but result in a higher state of stress than in a thin shell. The higher stresses lead to higher deflections within the housing which finally result in a housing with lower stiffness. The stiffness, therefore, can be increased by replacing the abrupt changes in curvature with smooth, continuous changes which result in lower bending stresses. The housing can then be accurately modeled by thin shell theory because it more closely follows "stress streamlines". The stiffest shell is achieved by maintaining smooth variation in the curvature while also maximizing the curvature over the entire surface of the housing.

NEW HOUSING DESIGN

In order to provide maximum stiffness, a viable compressor housing design must (1) be of reasonable size while still providing sufficient clearance for the internal mechanism, (2) maintain the maximum possible curvature over the entire surface and (3) follow a shape that insures that the curvature of the surface varies smoothly with no discontinuities. These goals can be realized by modifying the equation of an ellipsoid to include higher-order terms. The result is an equation that describes the entire surface with mathematically continuous derivatives at all levels, thus insuring that the curvature function is also continuous and that discontinuity stresses do not become a problem. The higher-order terms essentially square up the housing shape making it practical in terms of maximum size, volume and surface area.

There are a number of additional advantages to a mathematical description of the housing. First, the housing is precisely defined over the entire surface. In designs employing compound curvature, care must be taken to specify one radius to define a generating curve and then to specify a centerline around which this curve is swept. Specifying only two radii on the major sections can define two ambiguous surfaces depending upon which radius is used first. With a mathematical description of the surface any point on the surface can be calculated exactly. A second advantage is that other housing parameters such as curvature, surface area and volume can be calculated. The properties of various designs can then be compared, for example, to maximize curvature or minimize surface area. In this way maximum curvature can be obtained while providing sufficient clearance to the internal mechanism. It is also possible to perform "sensitivity" studies to determine which constraints have the greatest influence on various housing properties. Modification of important constraints might then be considered in order to design a stiffer housing.

EXAMPLE HOUSINGS

Figure 4 shows a conventional housing for a compressor family that covers a capacity range of 200 to 800 BTU/hr (59-230 watts). These housings are approximately 7.7 inches (20 cm) long by 6.1 inches (15 cm) wide and range in height from 6.5 inches (17 cm) for the lower capacities to 8.0 inches (20 cm) for the higher capacities. Figure 5 shows an improved design housing for a new family of compressors designed to cover the capacity range of 200 to 400 BTU/hr (59-120 watts). It is smaller than the conventional housing having dimensions of only 6.4 x 5.9 x 6.0 inches (16 x 15 x 16 cm). Because of its smaller size alone, it should have natural frequencies somewhat higher than the conventional housing.

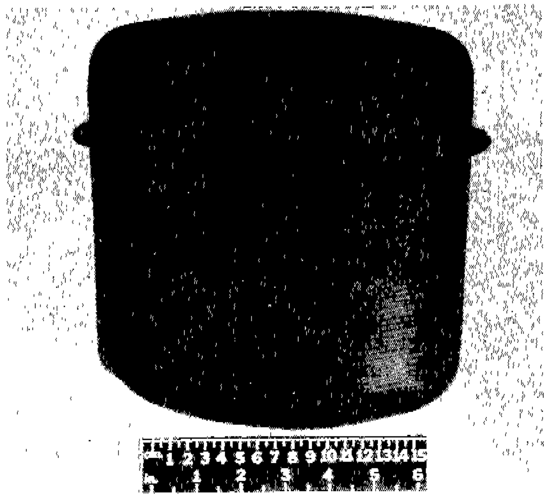

Figure 4

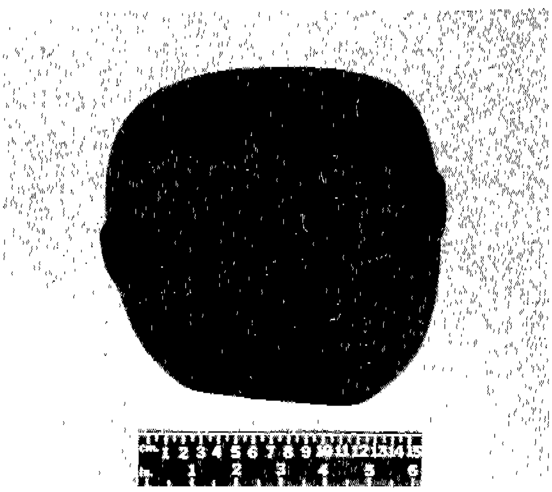

Figure 5

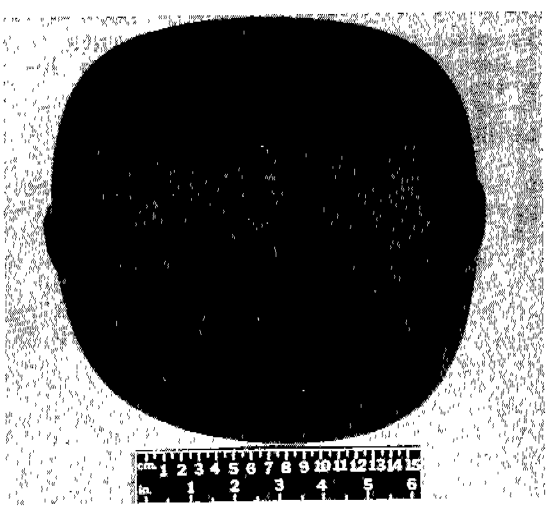

Figure 6

Figure 6 shows another improved housing designed to enclose the same compressor as the conventional housing. It is slightly larger than the conventional housing having dimensions of 8.0 x 6.8 x 8.0 inches (20 x 17 x 20 cm) and is able to handle compressors with capacities up to 1000 BTU/hr (290 watts).

Point accelerance measurements (acceleration/force) were taken at the centers of the side, end and top of each housing using an impact technique, in order to find the lowest natural frequency of each shell. Figure 7 shows the accelerance of the side of the conventional housing (Figure 4) and shows that the lowest natural frequency is near 1750 Hz. Figure 8 gives the accelerance of the end of the new small housing (Figure 5) where the lowest natural frequency occurs. The lowest natural mode for this improved housing is now over 3930 Hz, more than twice that of the conventional housing. Figure 9 gives the accelerance of the side of the new larger, improved housing shown in Figure 6. Here the lowest natural frequency is near 3650 Hz, again over twice the frequency of the conventional housing. Although this housing is slightly increased in size, the goal of significantly raising the natural frequencies of the shell through increased stiffness has been accomplished.

The effect of increased stiffness and natural frequencies on compressor housing sound radiation can be seen in Figures 10 and 11. Figure 10 shows the sound level comparison between conventional compressors and the new small compressor which uses the housing shown in Figure 5. The increase in natural frequency has resulted in a similar increase in the frequency of the major noise problem from the 1000 Hz 1/3-octave band up to the 2500 Hz band. The sound power levels of the 1000 through 2000 Hz bands have all dropped significantly. Above the 3150 Hz band, smaller reductions have also been achieved, but it must also be remembered that this comparison is between two different compressors and some reductions may be the result of differences other than the housing.

Figure 11 presents another comparison between four compressors first tested in the conventional housing of Figure 4 and then retested in the improved housing of Figure 6. The only modification other than the housing was to the suspension system in order to allow the compressor to fit into the new housing. Again, a large drop in the sound power level was recorded in the 1250 through the 2500 Hz 1/3-octave bands. This corresponds to the frequency range below the new resonance frequencies and is the result of increased housing stiffness. Smaller reductions at

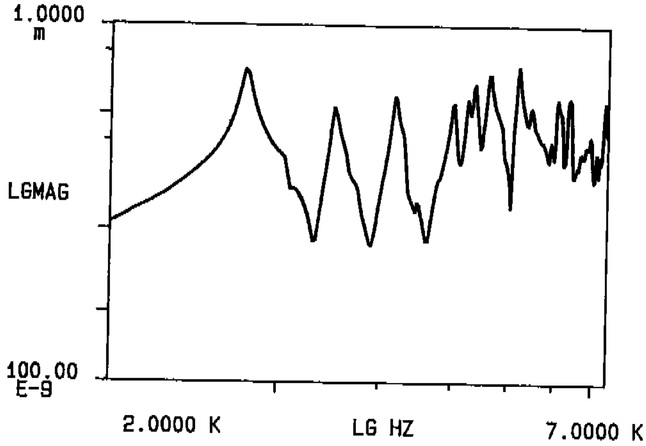

FIGURE 7. ACC./FORCE - PRESENT HOUSING

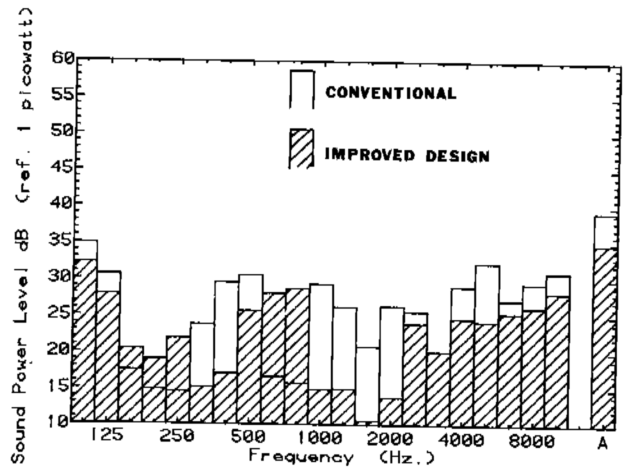

FIGURE 10. SOUND SPECTRA OF 300 BTU/HR COMPRESSOR

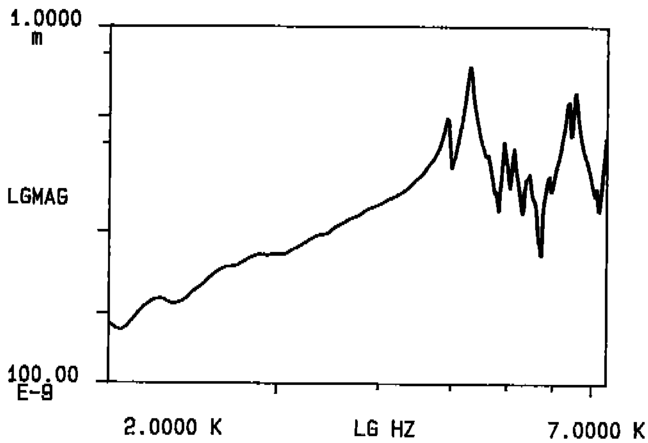

FIGURE 8. ACC./FORCE - NEW SMALL HOUSING

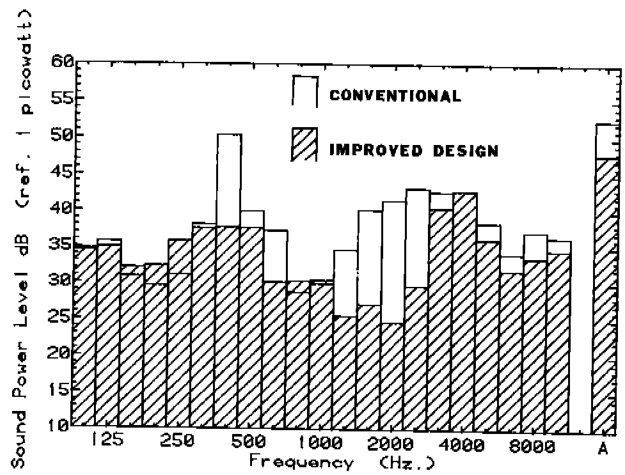

FIGURE 11. SOUND SPECTRA OF 1000 BTU/HR COMPRESSOR

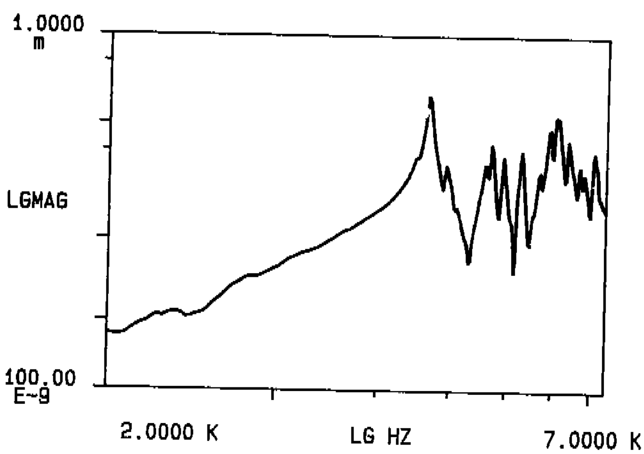

FIGURE 9. ACC./FORCE - NEW LARGE HOUSING

higher frequencies may be the result of a lower number of resonance modes in this frequency range.

CONCLUSIONS

A method has been demonstrated that reduces sound radiation from hermetic compressor housings by increasing housings stiffness. The stiffness is increased by maximizing the effective curvature (reducing the radii of curvature) while simultaneously maintaining a smooth, continuous variation of curvature over the entire housing surface. For small refrigeration compressors the result is an increase in the lowest natural frequency of the compressor shell of over one octave. While this improvement in housing design cannot solve all compressor sound problems, the frequency increase for the natural modes reduces the range of major sound problems. It shifts them to higher frequencies where other sound reduction efforts are more effective.

ACKNOWLEDGEMENTS

The author wishes to acknowledge the work of Dr. J.V. Gluck on real housing stresses that served as an inspiration for a discontinuity-free housing. Also, Messrs. R. Weatherwax and T. Murphy for their assistance in the preparation of this report.

REFERENCES

1. Ingals, D.J., "Understanding Noise Radiation from a Hermetic Compressor," Proceedings of 1972 Purdue Compressor Technology Conference, July 1972, pp. 69-73.

2. Johnson, C.N. and Hamilton J.F., "Fractional Horsepower, Rotary Vane, Refrigerant Compressor Noise Study," Proceedings of the 1972 Purdue Compressor Technology Conference, July 1972, pp. 74-82.

3. Saito, F., Maeda, S., Okubo, N. and Uetsuji, T., "Noise Improvement of Hermetic Compressor by Improvement on its Shell Shape," Proceedings of 1980 Purdue Compressor Technology Conference," July 1980, pp. 228-234.

4. Tōjo, K., Saegusa, S., Machida, S., Hirata, T., Sudo, M. and Tagawa, S., "Noise Reduction of Refrigerator Compressors," Proceedings of 1980 Purdue Compressor Technology Conference, July 1980, pp. 235-242.

5. Kjeldsen, K., "Capsule for Refrigerating Machines," U.S. Patent No. 3,187,995, June 1965.

6. Flügge, W., Stresses in Shells, Springer-Verlag, New York, 1967, pp. 27-28.