CENTRIFUGAL COMPRESSOR NOISE REDUCTION BY USING HELMHOLTZ RESONATOR ARRAYS

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ABSTRACT

Centrifugal compressors used in the pipeline market generate very strong noise, which is typically dominated by the blade passing frequency and its higher harmonics. The high level of noise is not only very disturbing to the people living close to the installation site but can also potentially cause structural failures in the piping. A novel design of an acoustic Helmholtz array (AHA) has been developed to address this type of noise problem. Computational studies show that the installation of the AHA liner on the compressor diffuser walls is very effective in reducing the noise level of the compressor, especially the dominant blade passing frequency noise. The acoustic liner design has actually been built and tested. The data clearly show that the use of acoustic liners is indeed very effective in the reduction of compressor tonal noise. This effort compares the noise levels of a pipeline compressor with and without the acoustic liner of Helmholtz resonator arrays and quantifies the effectiveness of the liner in terms of decibel reduction. The effects of the liner on compressor aerodynamic performance are investigated and discussed.

INTRODUCTION

The market demand for pipeline transmission and gas production turbocompressors has historically required new compressors with increased aerodynamic and mechanical performance. Design improvements have included the use of low solidity diffuser (LSD) vanes located directly downstream of the impeller to boost peak performance and to increase the stability characteristics of the compressor. These requirements have led to configurations that operate at high peak efficiencies but possess noise levels that can exceed levels set by Federal guidelines (FERC, 1978). In addition, as gas facilities are located closer to denser populations, local government agencies are placing additional requirements on the manufacturer to manage the compressor noise. These stringent requirements along with the escalating number of lawsuits have forced the designer to recognize that noise has become a primary design consideration.

The most common solution for reducing the noise levels of a turbocompressor is to use external noise control measures such as sound enclosures and wrappings to isolate the machine (Ver, 1973). This approach works on the basic principle of attenuating the noise transmission by use of a sound barrier. Sound enclosures are very costly and impose accessibility issues that often result in only the mechanical driver being enclosed. Even if the machine is totally enclosed, the acoustic energy will still propagate downstream in the form of gas pulsations, which serve as the principle forcing function to the downstream pipe. If the source is of enough amplitude to excite one of the many natural frequencies of the system, the piping will vibrate and can potentially lead to a mechanical failure. The implementation of silencers and/or beefing up the pipe thickness has been used to prevent this type of problem (Beranek and Ver, 1992). These devices can work very effectively to reduce piping noise but will add a performance penalty, and will eventually clog over time due to impurities in the gas. These impurities are often a mixture of solid particulate and liquid.

Another way to reduce noise is to eliminate or minimize the noise sources occurring inside the machine itself. Figure 1 shows sound pressure data for a pipeline booster compressor with LSD vanes for different flow rates. The interaction between the impeller and the stationary vanes form the so-called rotor/stator interaction problem. The resulting noise signature is a superposition of discrete frequency tonal noise peaks on a broadband noise floor. As indicated in Figure 1, the tonal noise peaks occur at the impeller blade passing frequency and its higher harmonics. It is these frequencies that will dominate the overall noise level of a pipeline compressor, as well as potentially excite downstream piping natural frequencies.

Acoustic liners in the form of Helmholtz resonator arrays can be a very effective means to reduce turbomachinery noise, especially the dominant noise component occurring at the blade passing

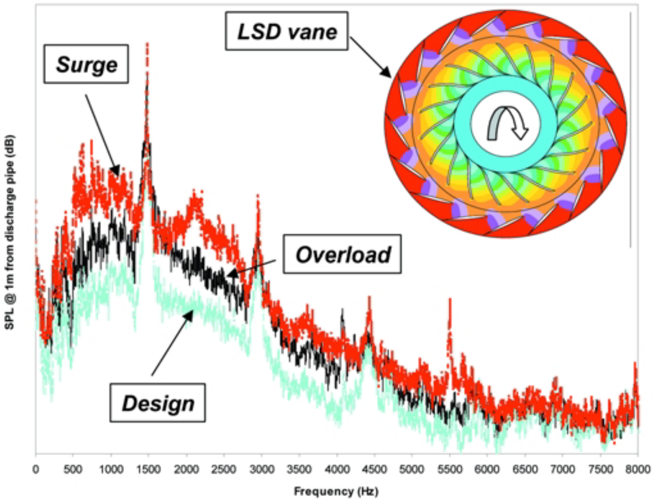

Figure 1. Centrifugal Compressor Noise Signatures under the Same Speed but Different Load Condition.

frequency. Helmholtz resonator arrays can provide remarkably high noise attenuation in a targeted frequency range with a limited requirement of space. This great advantage almost perfectly meets the requirement of the selective sound reduction at the blade passing frequency for centrifugal compressors. This can be achieved by tuning the Helmholtz arrays so that their maximum sound attenuation occurs around the blade passing frequency and at higher harmonics.

The flat surfaces on both walls of the diffuser are ideal locations to mount Helmholtz array acoustic liners because it places the component nearest to the noise source. The closer a noise controlling device is to the noise source, the more effective the device. This approach reduces the possibility that noise will bypass the silencing device allowing it to transmit through an alternative path. Different from the external noise control measures such as a sound enclosure, this is an internal noise control approach that addresses noise at the root level and therefore can be considered in the design phase of a compressor.

Tonal noise problems are very common in the aircraft industry for axial turbomachinery (Hanson, 1999; Schulten, 1982). The typical configuration of the Helmholtz Array acoustic liners, which have been successfully applied, is a three-piece sandwich structure as depicted in Figure 2. The honeycomb cells in the middle are bonded with a perforated facing sheet on the top and sealed off with a backplate. Even though this type of constructed liner has been proven to withstand high temperature environments, there are several concerns that prevent this type of liner from being directly applied to the compressor. The first concern is that it is still not rugged enough. The structural integrity of the liner has to survive extreme events, such as depressurization caused by emergency shutdowns. Another concern is the rigidity of the composite structure. The diffuser passage of a centrifugal compressor is relatively narrow. Good aerodynamic performance requires that the diffuser walls be rigid, uniform, and smooth. Any diffuser wall warpage or waviness caused by the deformation of a nonrigid structure will deteriorate the aerodynamic performance of the compressor. Therefore, it is anticipated that the high-pressure environment existing in centrifugal compressors requires a liner that is structurally stronger and more rigid than those used for aircraft applications.

Based upon the above considerations, a new type of Helmholtz array acoustic liner, in the form of a rugged unitary configuration, has been developed by Dresser-Rand. The installation methodology is similar to LSD vanes. First a recess is milled on each side of the diffuser wall to accommodate the liner. The structure is then inserted and bolted to the underlying diffuser wall.

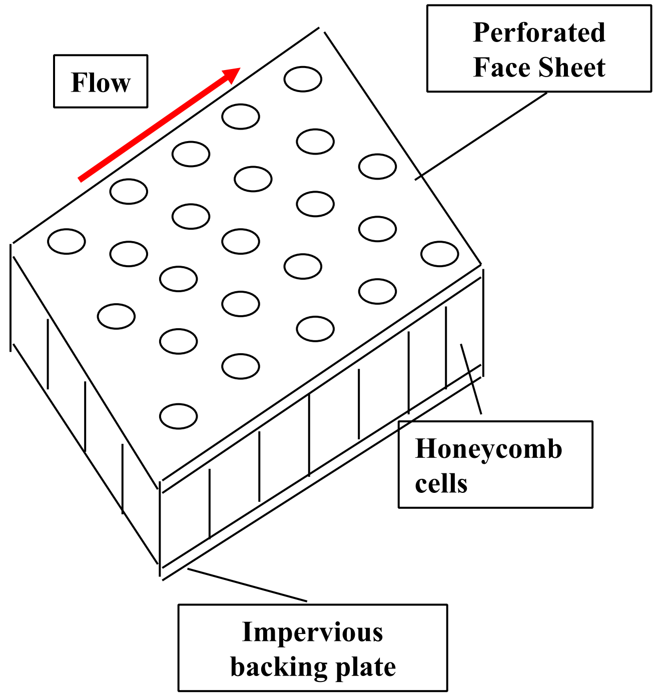

Figure 2. Conventional Design of Acoustic Liners.

The large cells formed in the plate essentially behave like dead volumes to the mean flow but are partially transparent to noise. As sound waves oscillate through the smaller perforated holes, a certain amount of acoustic energy is transformed into vorticity and is dissipated. In addition, the impedance mismatch realized by the acoustic liner on the diffuser walls causes the acoustic energy to be reflected to the source, resulting in less acoustic energy being propagated to downstream flow paths.

The body of this work presents a development effort that has focused on reducing both compressor noise and piping vibration using an acoustic Helmholtz array (AHA). An extensive investigation is presented to demonstrate the effectiveness of the invention. The bulk of this study was performed inhouse. The intent of the test program was to document the effectiveness of the device to reduce noise for both vaned and vaneless diffuser configurations. Prior to the experimental investigation, computational studies were first performed to help design the acoustic liner and to determine the effectiveness of the liner on compressor noise reduction. The details of the analysis are presented next and then are followed by the results from the experimental effort.

ANALYSIS

The insertion of an acoustic liner on a diffuser wall changes the diffuser wall from being acoustically rigid to acoustically transmissive. Consequently, the placement of the liner changes the diffuser wall impedance from infinite to the following, after normalized with respect to the characteristic impedance c of the acoustic media (Kinsler, et al., 1982).

$$\frac{Z}{\rho c} = \frac{R}{\rho c} + i\frac{X}{\rho c} = \theta + i\chi \qquad (1)$$

The impedance of the acoustic liner has a real part called resistance and an imaginary part called reactance. The normalized reactance of the acoustic liner is defined by the following expression (Hubbard, 1995).

$$\chi = \frac{X}{\rho c} = \frac{2\pi f\left(T + \varepsilon d\right)}{c} - \cot\left(\frac{2\pi f}{c}h\right) \qquad (2)$$

The normalized resistance θ is obtained by finding the roots of the following equation:

$$\left(\theta - \frac{a\mu T}{2\rho c\sigma C_d d^2}\right)\sqrt{\theta^2 + \chi^2} - \frac{K_i + K_e}{2c\left(\sigma C_d\right)^2}\frac{p}{\rho c} = 0 \qquad (3)$$

The parameters appearing in the above equations, which affect the impedance of the acoustic liner and, consequently, the attenuation effect of the liner, are summarized in Tables 1, 2, and 3.

The acoustical property of the Helmholtz array acoustic liner is represented by its impedance, which is controlled by four geometrical parameters, five acoustic and flow environment parameters, and four empirical parameters. The parameters in Table 1 are known prior to the design. They are either specified directly or can be calculated from other known quantities. The values used need to represent the fluid, and acoustic parameters are taken from the region where the device will be located. Necessary empirical parameters listed in Table 2 can be found in the published literature (Hubbard, 1995). With the values in Table 1 and 2 set, the values in Table 3 are iteratively determined using the impedance model described in Equations (1), (2), and (3). In addition to the above parameters from the impedance expressions, the diffuser width and the radial length of the diffuser wall segment lined with the AHA liner will also affect the overall attenuation of the acoustic liner.

Table 1. Acoustic and Flow Environment.

Variable	Description
ρ	Fluid density
c	Speed of sound
μ	Fluid dynamic viscosity
p	Excitation acoustic pressure
M	Mach number

Table 2. Empirical Parameters.

Variable	Description
ε	End correction for the perforated facing
C_d	Orifice discharge coefficient of the perforated holes
a	Dimensionless proportionality constant
$K_i + K_e$	Dimensionless entrance loss and exit loss

Table 3. Geometrical Parameters.

Variable	Description
d	Diameter of perforated holes
T	Thickness of face sheet or height of perforated holes
σ	Porosity or percentage open area of the perforated facing
h	Cavity depth

With the initial design parameters determined, the effectiveness of the design has to be evaluated. This is achieved by modeling the propagation of the acoustic pressure in the diffuser region using the wave equation. This equation is given below in cylindrical polar coordinate from Morse and Ingard (1986).

$$\frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} - \frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial p}{\partial r}\right) - \frac{1}{r^2}\frac{\partial^2 p}{\partial \phi^2} - \frac{\partial^2 p}{\partial z^2} = 0 \qquad (4)$$

where r, Φ, and z represent, respectively, the radial, circumferential, and axial directions of the diffuser, t is time, p is the acoustic pressure, and c is the speed of sound.

Because of its complexity, Equation (4) has to be solved numerically with realistic boundary conditions. This study used the finite element module from an acoustic software program. The required three-dimensional models were made using hexahedral elements to ensure a second-ordered accurate solution. The approach used was to first analyze the diffuser region without the liner. Then the same passage with the AHA design was modeled. Figures 3 and 4 compare the sound pressure level distribution inside a vaned diffuser region with and without the AHA. The view shown presents the sound pressure levels on the hub wall. Comparison of the sound pressure level at the exit of the diffusers shows a significant noise reduction by the Helmholtz array acoustic liner. The same type of trend is also seen in the vaneless diffuser case, as shown in Figures 5 and 6. For a given operating condition, the AHA design used in the software models is iteratively tweaked until an acceptable design is achieved. The actual effectiveness realized would be a direct function on how well the empirical constants used in the model characterize the design conditions.

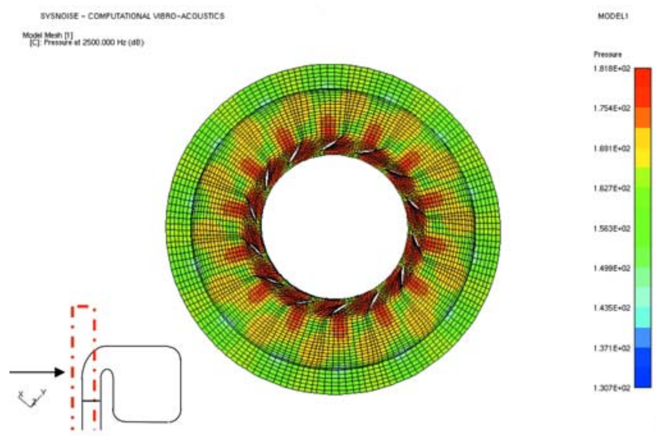

Figure 3. Computed Sound Pressure Level (dB) in a Vaned Diffuser (Without Acoustic Liner).

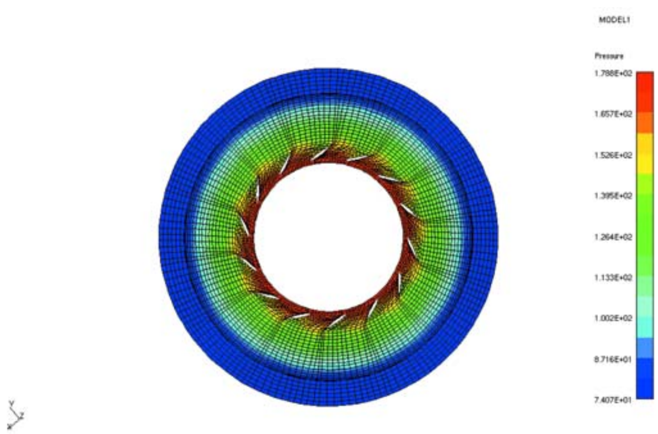

Figure 4. Computed Sound Pressure Level (dB) in a Vaned Diffuser (With Acoustic Liners on Diffuser Walls).

EXPERIMENT

After the effectiveness of the Helmholtz array acoustic liner was computationally proven, prototype hardware was designed and fabricated for experimental testing. A pipeline direct inlet “PDI” compressor was chosen as the test vehicle for the noise testing, as shown in Figure 7. Two types of commonly used diffuser arrangements, vaneless diffuser and low solidity diffuser (LSD), were fabricated for the compressor. Baseline noise data were first acquired on compressor setups without Helmholtz arrays installed. Helmholtz arrays were then installed on the diffuser walls and tests were repeated. The AHA liners for this particular case are shown mounted on the shroud side in Figure 8.

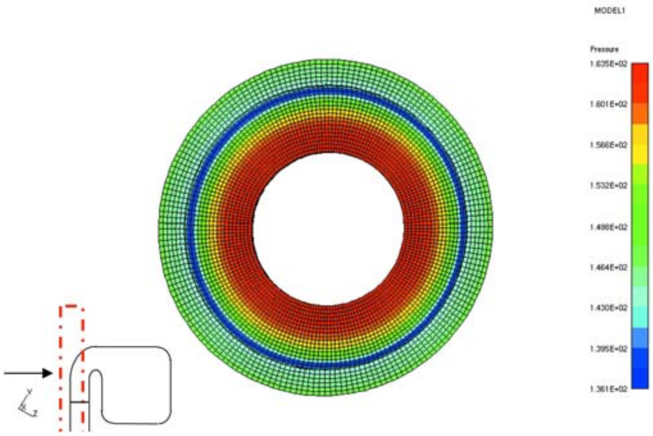

Figure 5. Computed Sound Pressure Level (dB) in a Vaneless Diffuser (Without Acoustic Liners).

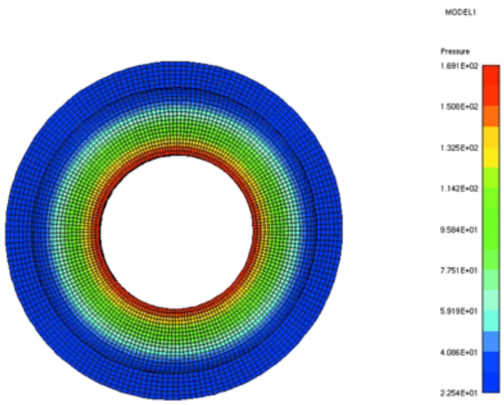

Figure 6. Computed Sound Pressure Level (dB) in a Vaneless Diffuser (With Acoustic Liners on Diffuser Walls).

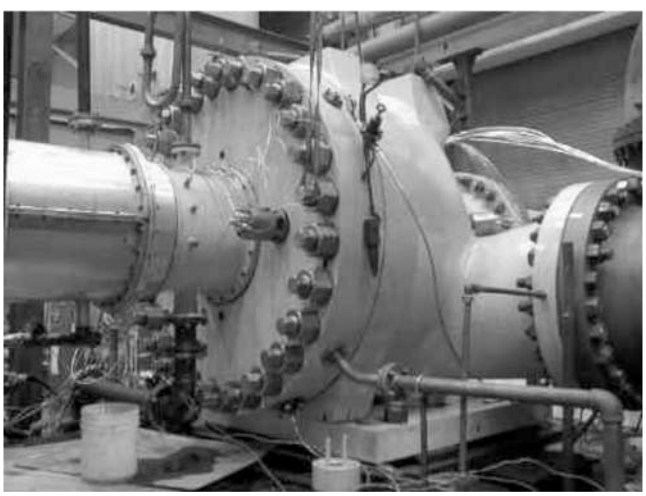

Figure 7. Compressor Test Setup.

Test Conditions and Data Acquisition

To determine how effectively the Helmholtz array attenuates noise under different operating conditions, the testing program was designed to cover multiple speed lines and multiple load conditions for each speed line. Table 4 lists all the operating points tested in each of the four tests, where Mu is the compressor head coefficient.

In total, 17 data points were taken for each hardware configuration. Care was taken to repeat the same data points in each of the

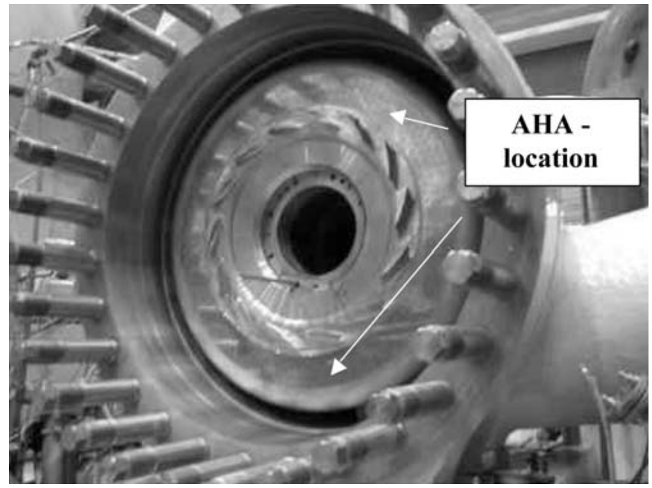

Figure 8. AHA Mounted on the Diffuser Shroud Side (Patent Pending).

Table 4. Operating Points for Each Test.

RPM	Mu=0.2	Mu=0.3	Mu=0.4	Mu=0.5	Mu=0.6
4800		Point 1	Point 2	Point 3	
5000		Point 4	Point 5	Point 6	
5217	Point 7	Point 8	Point 9	Point 10	Point 11
5400		Point 12	Point 13	Point 14	
5600		Point 15	Point 16	Point 17	

four tests so a direct comparison of noise data could be made at each point. In addition to acoustic data, unit efficiency and head coefficient were measured at every point of every test. The performance measurements were used to determine if the Helmholtz arrays had any impact on the compressor aerodynamics.

Gathering the acoustic data involved an arrangement of four microphones around the compressor. One microphone, positioned one meter away from the suction pipe, was used to measure the sound pressure level on the suction side, as shown in Figure 9. A second microphone was positioned one meter away from the compressor casing to record the sound pressure level near the compressor housing. Two additional microphones were mounted around the discharge pipe to measure the sound pressure levels on the discharge side, with one microphone being one meter away horizontally from the discharge pipe and the other one being one meter away vertically from the discharge pipe, as shown by Figure 10. From each microphone, both narrow band and one-third octave band sound pressure levels were acquired.

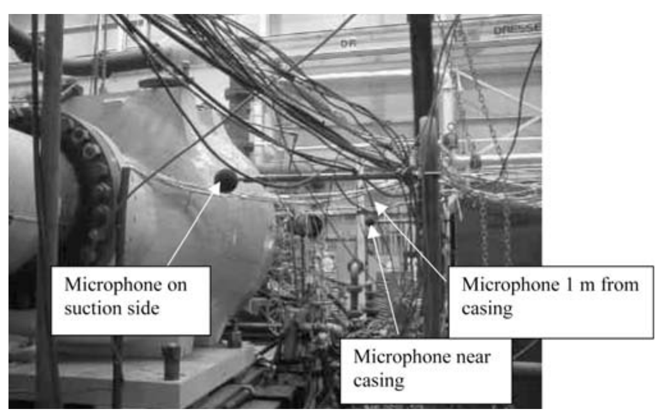

Figure 9. Microphone Positions.

In addition to the external sound pressure measurements by the four microphones, sound intensity equipment was also used to

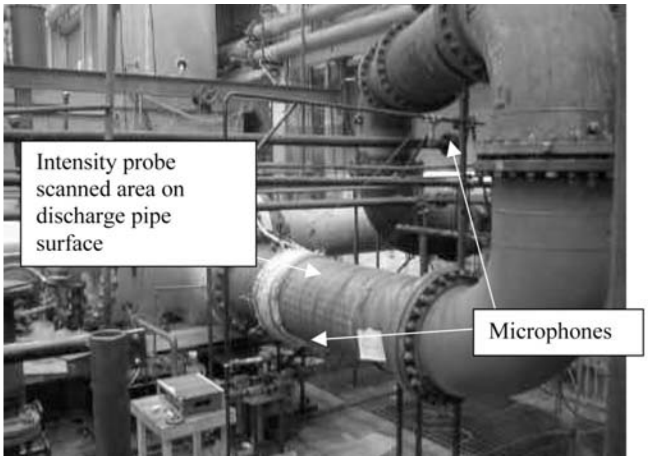

Figure 10. Noise Measurement on Discharge Pipe.

measure the sound power emission from the compressor. Three areas, one on the suction pipe shown in Figure 11, one on the compressor casing shown in Figure 12, and one the discharge pipe shown in Figure 8, were scanned with the sound intensity probe for each test.

Figure 11. Suction Pipe Surface Scanned.

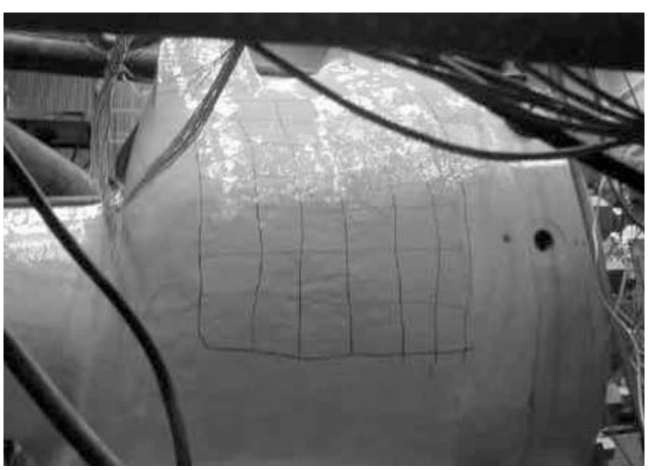

Figure 12. Compressor Casing Surface Scanned.

Results of Noise Reduction

A detailed analysis of the acoustic test data confirms that installation of a Helmholtz array on compressor diffuser walls reduces noise dramatically. Figure 13 shows a comparison of the overall noise levels for the four hardware configurations at 5217 rpm. Sufficient noise attenuation was provided by the Helmholtz array to reduce the noise level of the LSD arrangement below the vaneless arrangement.

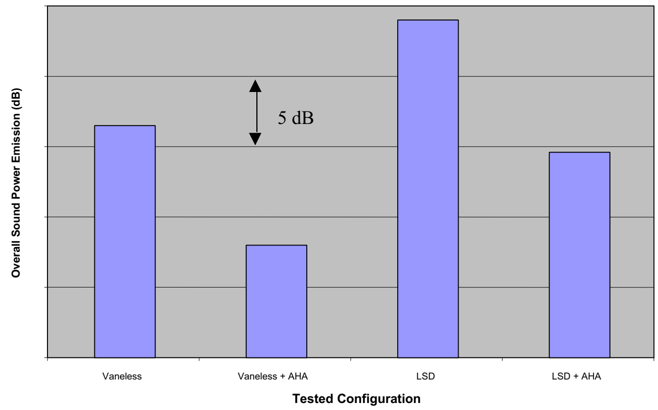

Figure 13. Overall Noise Level.

Figure 14 and Table 5 summarize the noise reduction in decibels (dB) for the different speeds by the acoustic Helmholtz arrays mounted on the LSD diffuser walls. As seen in Figure 14, the AHA acoustic liners are effective in attenuating noise across a wide range in the operating map of the compressor. The lowest overall noise reduction was 9 dB and the highest was 13.2 dB.

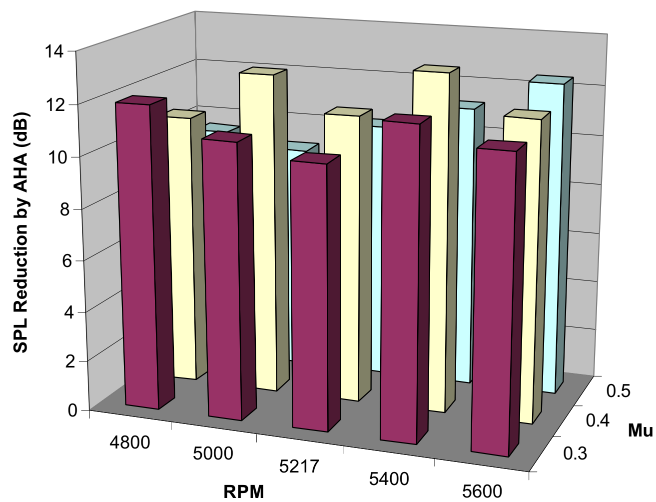

Figure 14. Overall Noise Reduction of the Compressor with LSD by Helmholtz Array.

Table 5. Overall Noise Reduction in dB by AHA.

RPM	Mu= 0.3	Mu= 0.4	Mu = 0.5
4800	11.98	10.76	9.5
5000	10.83	12.69	9
5217	10.3	11.36	10.2
5400	12	13.2	11.2
5600	11.3	11.74	12.4

Figures 15 and 16 show the discharge sound power emission from the baseline compressor design and that from the compressor with Helmholtz array acoustic liners installed on the diffuser walls. The compressor noise, particularly the noise component in the frequency band around the blade passing frequency, is significantly reduced.

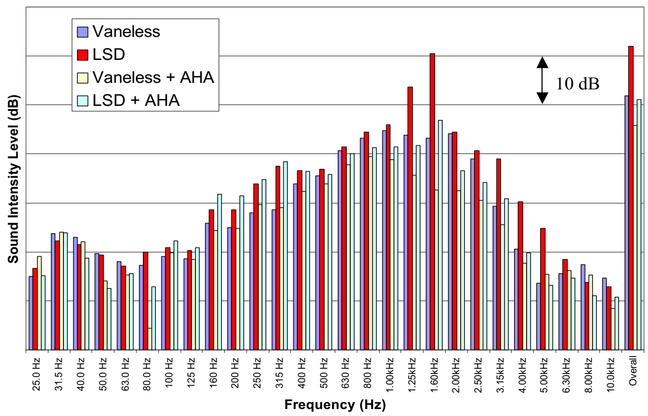

Figure 15. Discharge Pipe Sound Power Emission at Point 9 (5217 RPM, Mu = 0.4).

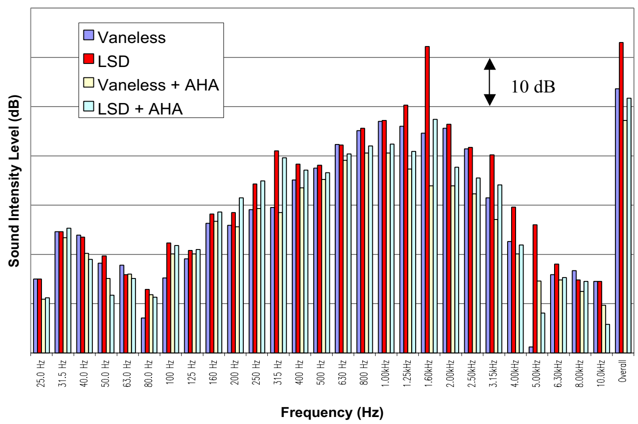

Figure 16. Discharge Pipe Sound Power Emission at Point 12 (5400 RPM, Mu = 0.3).

Results of Aerodynamic Performance

The manufacturer was also able to confirm from the performance data that the Helmholtz array had no adverse effect on the unit performance. Standard performance data were taken both at the inlet and exit flanges for all equipment configurations. Total pressure and temperature values were used to calculate Mu and polytropic efficiency. Selected nondimensional efficiency and Mu distributions are shown for the LSD vane case in Figure 17 and for the vaneless case in Figure 18. From these figures, it is clear that no appreciative differences can be seen in range or peak performance of the machine.

CONCLUSIONS

The manufacturer has successfully applied the AHA technology to reduce the noise level generated by a centrifugal compressor. The approach used to design these devices was first presented and then demonstrated. Next, the details surrounding the experimental validation effort were explained. The results from this effort clearly show that the AHA is very effective in reducing noise for compressors. The AHA acoustic performance was determined to be between 9 and 13.2 dB for the vaned diffuser, depending on the operating conditions. Similar but lower values were obtained for the vaneless configuration. Finally, the AHA was shown to have no adverse effect on aerodynamic performance for the entire flow curve. It is expected that this type of device will be used extensively for all applications where noise is a design consideration.

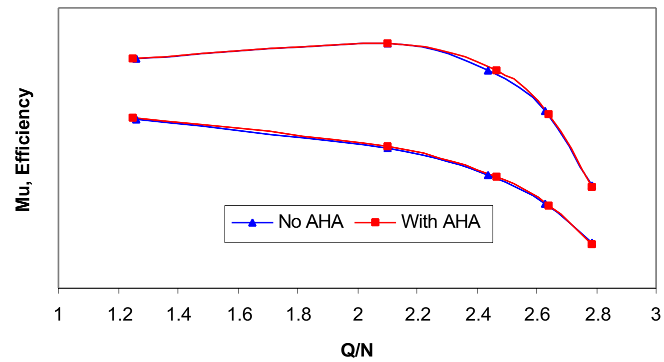

Figure 17. Effect of AHA on Aerodynamic Performance of the Vaned Diffuser at 5217 RPM.

RPM

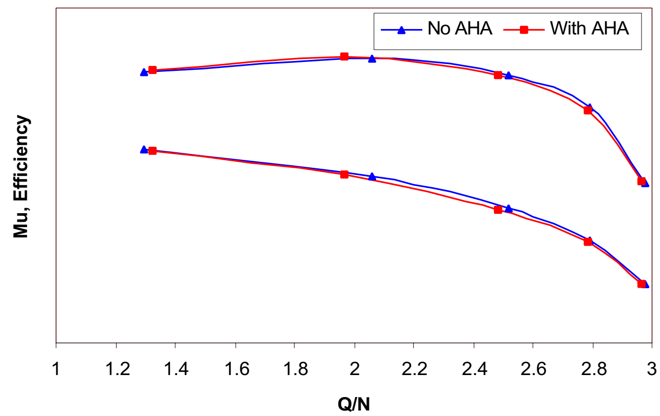

Figure 18. Effect of AHA on Aerodynamic Performance of the Vaneless Diffuser at 5217 RPM.

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