Proceedings of
ASME TURBO EXPO 2002
June 3-6, 2002, Amsterdam, The Netherlands

GT-2002-30278

On Reducing Piping Vibration Levels – Attacking the Source

Zheji Liu and D. Lee Hill
Dresser-Rand
Olean, NY 14760

Roman Motriuk
TransCanada Pipelines Ltd.
Ontario, Canada

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 noise is not only very disturbing to the people living nearby the installation site but also causes expensive structural failures in the downstream piping. A novel design of Helmholtz array has been developed to address this type of noise problem. Computational studies show that the installation of the Helmholtz array acoustic liner on the compressor diffuser walls is very effective in reducing noise level of the compressor, especially the dominant blade passing frequency noise. The acoustic liner design has been built and tested at an installation site by the customer. The data clearly shows that the use of acoustic liners is indeed very effective in the reduction of both the noise and the vibration levels of the machine.

NOMENCLATURE

A	Dimensionless proportionality constant
C	Speed of Sound
C_d	Orifice discharge coefficient of the perforated holes
D	Diameter of perforated holes
dB(A)	Decibel, A weighted
h	Cavity depth
K_i	Dimensionless entrance loss coefficient
K_{ex}	Dimensionless exit loss coefficient
M	Mach Number
mm	millimeter
Ns	Specific Speed
P	Excitation acoustical pressure
R	Impedance resistance
RPM	Revolution Per Minute
sec	second
T	Height of perforated holes
X	Impedance reactance
Z	Impedance
ε	Porosity or percentage open area of the perforated facing
μ	Fluid viscosity
ρ	Fluid Density
θ	Normalized resistance, $R/\rho c$
σ	End correction for the perforated facing
χ	normalized reactance, $X/\rho c$

INTRODUCTION

The operation of a centrifugal compressor inherently delivers an unsteady flow that can manifest itself into various forms of strong pressure pulsations. These pulsations unavoidably travel to the downstream piping structure and will excite the piping to vibrate. High levels of vibration often result in premature failures or even catastrophic failures if a natural frequency of the piping system is excited. Cracking will occur in the pipe itself, but can also occur in pipe fittings, valves, and pipe nipples. The cost associated with these types of repairs can quickly add up if the source or system is not permanently modified. In addition, delays in operation will cause additional losses in revenue. Field studies reported by Motriuk (1994), Patela and Motriuk (1996), Zhou and Motriuk (1996), and Marsher (1996) have shown conclusively that pipe vibration is predominately driven by the upstream compressor. Several of these reports also cite that better communication between the compressor manufacturer and the piping designer could have minimized a lot of the problems.

Manufacturers of turbo-compressors have also established that acoustical energy is the primary source for high-frequency vibration problems. This source is a function of blade passing frequency and its higher harmonics. This source is in the form of pressure pulsations and has peaks at blade passing frequency and its higher harmonics. The magnitude of the pressure pulsations is typically a function of machine rpm and geometric parameters. Figure 1 and 2 show generalized vibration level trends for a machine with a vaneless diffuser as a function of machine rpm and flow rate. High frequency piping disturbances are directly linked to the rpm and the specific speed of the machine(load) where broadband spectra are linked to the flow rate. The recent work by Field et. al. (2001) shows that inlet flow disturbances have virtually no impact on the sound pressure spectrum at the compressor discharge.

Efforts to eliminate or control high-frequency acoustical energy have resulted in the use of spoilers. These devices are inserted directly downstream of the compressor and are designed to disrupt the high-frequency pressure patterns entering into the piping. The work of Motriuk (2000) does report that these devices can reduce vibration levels but do have the undesirable tendency to fail in high-cycle fatigue. Another popular solution is the use of an absorptive silencer. They are installed in same manner as the spoilers but work as a dissipative component. Because of the inherit impurities in the gas, these devices have a very high maintenance requirement. If designed improperly, they can actually add pressure noise called singing. The work of Mangiarotty (1968) provides a good discussion on this subject. The last type of solution is to modify the piping system. Additional structural support can be achieved with thicker components or with use of shell clamps to add local rigidity. While many vibration problems have been solved with these approaches, they do not offer a preemptive solution.

The work of Biba et. al. (2000) presents an exhaustive design effort for a pipeline compressor. This effort attempts to jointly consider both performance and noise as design parameters. Given the fact that the compressor noise source is of the highest magnitude at the blade passing frequency or higher harmonics, acoustic liners in the form of resonator arrays were proposed to reduce the dominant noise component at the blade passing frequency. The proposed device works on the principle of a Helmholtz resonator. The work of Liu and Hill (2001) presented a complete in-house experimental study that proved that the new invention was very effective in reducing tonal noise. The study showed the overall noise of a pipeline compressor was reduced by about 10 dBA. Note that this is the same acoustical energy that drives pipe vibration problems. The concept of using a resonator with perforated openings can be traced back to the work of Ingard (1954).

The present study uses the Acoustic Helmholtz Arrays (AHA) to solve a re-occurring piping failure due to excessive vibration. In the following section the details of the problem are presented first. All available initial vibration data is shown and discussed along with the details of the design of the AHA. Next, the experimental data taken from the field is presented. The data shows that the AHA is effective in eliminating the forcing function for the problem. Consequently, when AHA liners are installed in a machine, piping designers no longer have to worry about the harmonics driven by the blade passing frequency.

ANALYSIS

Vibration Problem

The centrifugal pipeline compressor investigated in this work is a standard single-stage unit designed for high flow pipeline applications. The machine shown in Figure 3 has Low Solidity Diffuser (LSD) vanes located just downstream of the impeller to enhance performance. Before this compressor was revamped, the facility had not experienced any high frequency vibration problems. When the operators went to bring this machine online in Pembroke, Canada, high levels of piping noise were detected and were subsequently treated with acoustic lagging. Figure 4 schematically locates the region where the high levels of vibration occurred. Due to the unexpected strength of high-frequency excitations, a dry gas seal attachment on the pipe failed. The failure initiated at the root of the weldolet causing a pipe shell crack. This failure was temporarily addressed by reducing the rotating speed of the machine and by repairing the weldolet attachment through increasing its cross sectional area (Figure 5).

An initial testing effort was performed to characterize the source of this problem. Accelerometers were located in the failed region to establish the level of vibration. Table 1 shows the results from this effort and clearly show that the vibration amplitude increases with the rotational speed. The maximum level measured is approximately 100 mm/s, which is too high to allow the compressor to run at full load. Figure 6 shows the acclerometer spectrum plot for the 6088 rpm test run. From the trace, the maximum vibration levels do occur at the blade passing frequency.

Design Modification with Acoustic Liners

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 a finite value. After normalized with respect to the characteristic impedance ρc of the acoustic media can be written as follows:

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

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:

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

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

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

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 2-5.

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 2 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 4 can be found in the published literature such as the work of Hubbard (1995). With the values in Table 2 and 4 set, the values in Table 3 are iteratively determined using the impedance model described in equations (1) – (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 AHA liner, will also affect the overall attenuation of the acoustic liner.

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. Because of its complexity, the wave has to be solved numerically using realistic boundary conditions. The work of Liu and Hill (2001) provides a complete discussion on this phase of the design calculation. This study used the same finite element model to iteratively optimize the parameters in Table 3 until an acceptable design was achieved.

Figure 7 shows the final AHA design. The side shown faces the gas flow path. Note that there is a large amount of smooth surface left. All data taken to date has consistently shown that the liner does not degrade aerodynamic performance. The installation of the liner follows the same process as an LSD vane assembly. A typical rail-fit is used for split-case designs. Figure 8 shows the compressor layout with the AHA liner. The actual effectiveness realized is a direct function on how well the AHA design is tuned to match the noise characteristics of the machine.

RESULTS

The AHA designed in the previous section was built and installed in the pipeline compressor at the Pembroke facility. A third-party consulting group was contracted to obtain the field data. Both vibration and noise was taken during all testing. Accelerometers were mounted in four locations of the discharge piping. Figure 5 indicates the instrumentation located around the repaired weldolet. Noise data was also taken in both the near and far field. The microphone setup for the near field sound pressure data is shown in Figure 9. From the data taken, it is possible to determine the effectiveness of the AHA. Sound intensity data obtained from scanning 1.2 square meter control surfaces is necessary to determine the acoustical energy reduction and as a function of rpm. The before and after vibration data will be able to conclusively show the impact of the acoustic liner on the amplitude levels specifically in the failed region.

The sound pressure, gas pressure, and vibration signals were processed using a Hewlett Packard 35670A analyzer. The dynamic signal analyzer was configured to receive narrowband frequency spectra, baseband to 6400 Hz. with 800 lines of resolution. Each measurement was averaged over a 3-minute period. The spectra appeared, however, to stabilize within 15 seconds. The measurement data was simultaneously recorded on a Sony digital audio tape recorder for post processing. This data was used to review the time history and to compute the 1/3 octave band spectra.

The sound power emission was calculated using a Hewlett Packard 3569A Real Time Analyzer and was configured to acquire four 1/3 octave band sound intensity spectra per component. (200 Hz. to 5000 Hz.). The spectra were corrected for the respective surface area and the sound power determined. The intensity measurements were made using the procedures described in ISO 9614-2.

These experimental tests were designed to establish the performance of the AHA as a function of rotational speed. Due to the ambient temperature conditions, the highest speed obtainable was approximately 7000 rpm. Table 5 provides the necessary aerodynamic conditions for all test points. As the rotational speed increases, the load of the machines also rises. In addition, the data from these tests will establish the range of effectiveness of the liner. One of the unique features of this design is that multiple frequencies can be attenuated with a single piece of hardware.

The sound pressure spectrum is shown in Figure 10 for all data points. All maximum peaks occur at each respective blade passing frequency. Also indicated in Figure 10 is the tonal noise being excited at the higher harmonics of the blade pass frequency. A better comparison of the sound power peak levels is shown in Figure 11. As the rpm is increased, the sound power is shown to increase. At the highest rpm, however, the sound power level is shown to drop. This is a direct impact of the AHA. Higher rotational speed points were not taken due to environmental limits on the driver.

The vibration data will also peak at the blade frequency of the test point. The peak data for four accelerometers is shown in Table 6 for all test points. From the data it is clear that the levels of vibration are significantly lower than Table 1. Also shown in the table is the trend that vibration levels drop as the rotational speed gets closer to one of the design attenuating frequencies. This correlates with the sound power data. Although no sound power data was taken for the pre-AHA testing, it is expected that the sound power peak tonal noise was significantly higher than the level shown in Figure 11.

The AHA liner presented in this study was installed into the compressor on 05/1/00. Low levels of vibration have been maintained without any maintenance required. This implies that the AHA is rugged enough to withstand typical gas impurities of a pipeline. This machine has also been able to be run at the compressor's full-load design condition thus allowing the full effectiveness of the AHA to be realized. This fact alone proves that the AHA is very effective in reducing acoustical high-frequency energy.

CONCLUSIONS

A piping failure occurred at a pipeline compressor facility in Pembroke, Canada. Field tests identified the vibration levels in the discharge piping and that they occurred at the blade passing frequency of the compressor. A novel approach was developed to attenuate the noise energy at the source. The field data taken with the AHA installed showed that the peak sound power level dropped at the maximum rpm tested. The vibration data also supported this trend. The overall vibration levels were significantly lower indicating a good range in effectiveness of the liner. Finally, two years running at full load without maintenance, or piping failure, show that the device is very rugged and that it works.

ACKNOWLEDGMENTS

The authors would like to thank Dresser-Rand for the permission to publish this paper. We would also like to express our appreciation to Navcon for providing excellent support during the field testing of the liner. Finally, we also like to acknowledge all of the support from our colleagues who made this effort a success.

REFERENCES

Biba, Y., Liu, Z., and Hill, D.L., 2000, "Pipeline Compressor Redesign with the Consideration of Both Noise and Performance," *Proceedings of the International Pipeline Conference 2000*, Vol.2, pp. 1317-1324.

Feld, H.J., Aschenbrenner, S., and Girsberger, R., 2001, "Investigation of Acoustic Phenomena at the Inlet and the Outlet of a Centrifugal Compressor for Pressure Ratio 4.5," ASME 2001-GT-0314.

Hubbard, H.H., 1995, *Aeroacoustics of Flight Vehicles*, Acoustical Society of America.

Ingard, U., 1954, "Perforated Facing and Sound Absorption," The Journal of the Acoustical Society of America, Vol. 26, Number 2, pp. 151-154.

Liu, Z., and Hill, D.L., 2001, "Centrifugal Compressor Noise Reduction by using Helmholtz Resonator Arrays," Proceedings of the 30th Texas A&M Turbo Symposium, Houston, Texas, pp. 89-94.

Mangiarotty, R.A., 1968, "Duct-Lining Materials and Concepts," NASA SP-189, pp. 29-52.

Marcher, W., and D'Orsi, N., 1996, Technical Memorandum No. 484 "Aero/Pulsation/Vibration Testing of a Compressor at Vetchland Booster Pipeline Station," CONCEPTS ETI, Vermont.

Motriuk, R.W., 1994, "Preliminary Study on Pipe Shell Vibrations: Vibration Velocity-Stress Relationship Based on Field Measurements," 5th International Conference on Recent Advances in Structural Dynamics, *Proceedings of Work in Progress Papers*, Vol. 1, University of Southampton, England, pp. 49-55.

Motriuk, R.W., 2000, "How to Design and Operate Quiet Centrifugal Compressor System," *Proceedings of the International Pipeline Conference 2000*, Vol.2, pp. 1281-1288.

Patela, G., and Motriuk, R.W., 2000, "Effect of Vaned Diffuser on Performance and Onset of Pressure Instabilities in Centrifugal Compressors," *Proceedings of the 7th FIV 2000 International Conference in Lucerne, Switzerland, Flow Induced Vibration*, pp. 733-742.

Zhou, J., and Motriuk, R.W., 1996, "Influence of Tapered Thermowell Length on Temperature Measurements," ASME PVP Conference, Montreal, Canada, Vol. 333, pp. 97-104.

Table 1 Initial Field data taken without the AHA.

	RPM	Peak Level (mm/sec)
Pt. 1	5276	31.7
Pt. 2	6088	100

Table 2. Acoustic and flow environment parameters

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

Table 3. Geometrical parameters

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

Table 4. Empirical parameters

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

Table 5. Test points for field measurements

Set Point	Shaft	BPF	HEAD	FLOW	EFF
(n/a)	RPM	HZ	kJ / kg	m^3/sec	%
1	4800	1360	8.51	6.70	44.8
2	5308	1504	12.07	7.32	50.8
3	5421	1536	9.37	7.84	39.6
4	5703	1616	14.34	7.97	51.9
5	5986	1696	13.67	8.55	45.3
6	6325	1776	13.79	9.07	41.7
7	6438	1840	25.09	8.63	66.0
8	6945	1968	45.00	7.17	84.2

Table 6. Peak Vibration Levels after the Installation of AHA

Set Point	**Narrow Band Peak Level**			
	Accel 1	Accel 2	Accel 3	Accel 4
	mm / s (rms)	mm / s	mm / s	mm / s
1	20.521	21.351	17.263	9.1353
2	14.876	8.6195	11.909	9.8586
3	15.059	19.944	22.157	12.160
4	14.085	20.177	29.912	17.677
5	28.420	27.326	23.633	9.9345
6	14.675	25.686	15.750	13.810
7	13.496	13.642	23.867	22.086
8	9.6872	6.0107	8.4255	6.3033

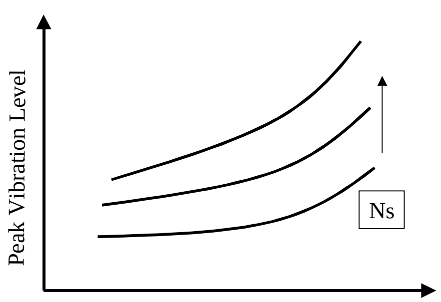

Figure 1. Generalized vibration as a function of RPM and machine load.

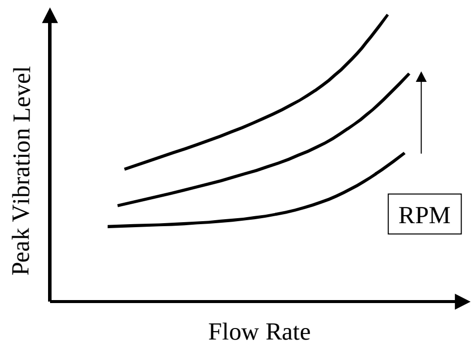

Figure 2. Generalized vibration as a function of flow rate and rpm.


Figure 3. Dresser-Rand Pipeline compressor located at Pembroke, Canada (TCPL)

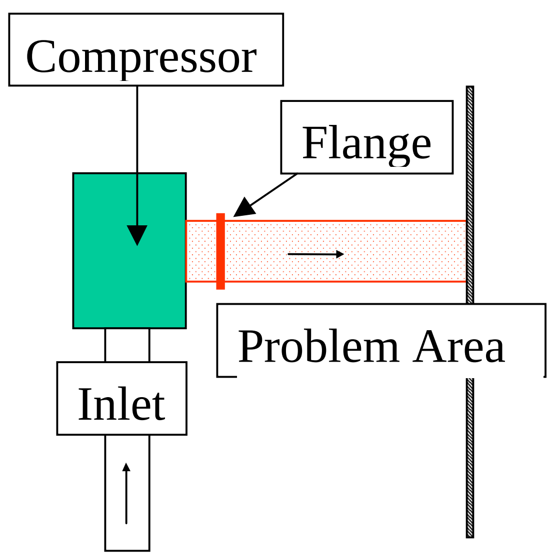

Figure 4. Failure region excited by the compressor

Figure 5. Acclerometer locations and repaired region.

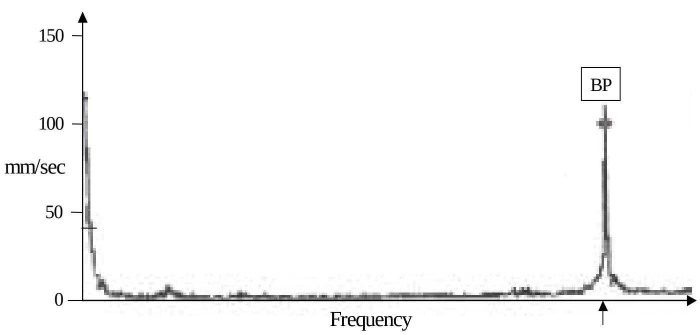

Figure 6. Initial Field Accelerometer Data for 6088 rpm.

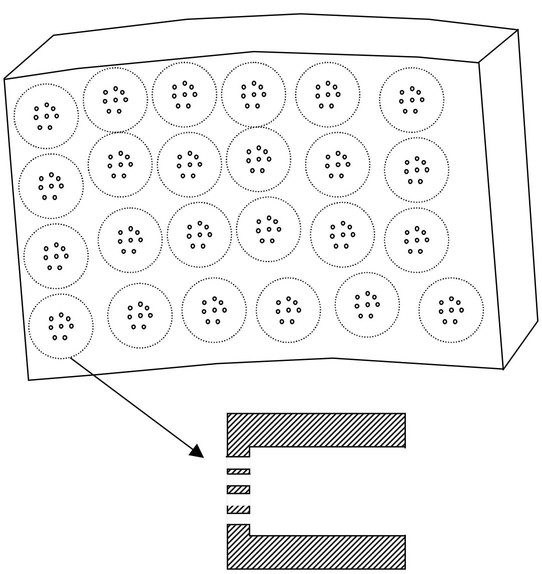

Figure 7. AHA to be installed on diffuser wall.

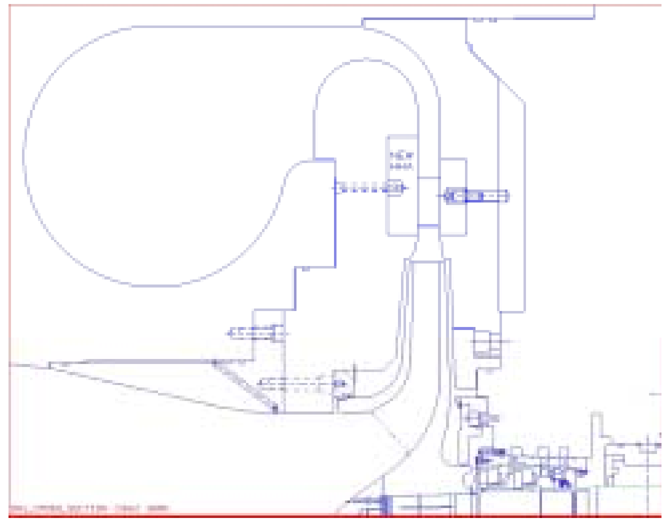

Figure 8. Compressor cross-section depicting the location of the AHA liner.

Figure 9. Near field microphone for sound pressure measurement in the discharge piping.

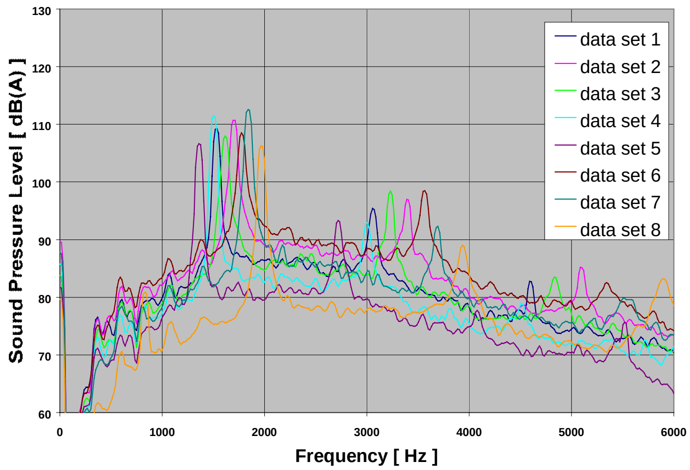

Figure 10. Spectrum comparison for all data points.

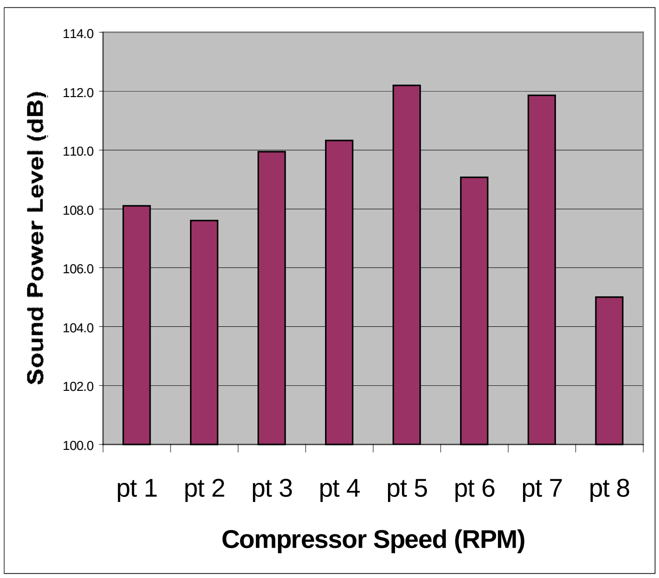

Figure 11. Impact of AHA for blade pass frequency.