Centrifugal Compressor Case Study

by:

Bill Eckert, P.Eng., Ph.D., Principal Engineer
Beta Machinery Analysis
Calgary, AB Canada
beckert@betamachinery.com

Gas Machinery Conference October 4-6, 2010 Phoenix, AZ

Key words: vibration, pulsation, flow induced vibration, centrifugal compressors, vortex shedding, blade pass frequency

Abstract

In 2008, three centrifugal compressors at a pipeline station were retrofitted with higher head impellers. For the next two years the owner experienced continual vibration problems that caused failures with RTDs, transmitter and position switches. Most of the failures were on the discharge side, but failures did occur on the suction side as well. The cause of the failures was assumed to be flow induced pulsations (also referred to as vortex shedding). Many attempts were made to modify the thermowells and RTDs, but they were not successful in reducing the failures.

In early 2010, Beta Machinery Analysis traveled to the site and conducted a vibration and pulsation analysis. After assessing the situation, it was determined that shell mode piping vibration excited by blade pass pulsation was responsible for the problems and not flow induced pulsations as original assumed.

This case study outlines the factors that contributed to the vibration problem and recommended solutions; it predicts interferences between the compressor and shell mode piping natural frequencies and potential excitation sources such as flow induced or, and, blade passing pulsations; and it also highlights why a centrifugal vibration study may be good practice during the initial design (or retrofit) as it is much easier and less costly to make adjustments at the design stage compared to searching for, and solving, the problem in the field.

Introduction

Centrifugal compressor installations are subject to various forces. Because of the geometry and speeds, the excitation frequencies are often very high. Some of the forces that can act on the compressor and associated piping are as follows:

1) Mechanical forces due to imbalance of the rotor, or rotor whip or whirl. These are normally well controlled and if excessive are dealt with quickly due to the probability of damaging the compressor.

2) Flow-induced turbulence can occur at flow past objects intruding into the flow path or discontinuities in the flow path. This may result in high levels of broad band energy that excite acoustic and, or mechanical natural frequencies in the system.

3) Rotating stall forces can be generated by centrifugal compressors. This typically happens because of flow instabilities in the compressor at low flows resulting in excitation at frequencies sub-synchronous to the compressor run speed.

4) Momentum changes due to rapid opening and closing of valves, for instance, create transient pressure waves that are often called “water hammer”. The pressure rise can be large resulting in significant forces on the piping.

5) Pulsations from the compressor impeller itself can excite acoustic and mechanical natural frequencies of the system. The frequency of excitation is determined by number of blades in the impeller and, or the diffuser, and shaft speed. The strength of the pulsation is determined by the internal geometry and operating point. Generally operating the compressor off its best efficiency point will increase the source strength of pulsations.

In the past most of these factors were not examined at the design stage. More recently industry has recognized the issues that can occur and has begun to specify design studies to minimize the probability of issues. However, some of these phenomena are difficult to predict at the design stage due.

This paper will present a case study of vibration issues at a centrifugal compressor station which examines some of these forces.

1.0 Case Study History

There are three single stage centrifugal compressors at a pipeline station. The units are driven by gas generators at speeds of between 3600 and 5000 rpm. Suction pressure is approximately 615 psig with discharge approximately 830 psig. All three compressors were revamped in 2008. The update consisted of installing a new compressor wheel with higher head (lower flow) in each unit and replacing the drivers with higher power gas generators. Significant changes were made to the yard piping. However, the piping immediately adjacent to the compressors in the compressor buildings was not modified. After the compressor revamp there have been numerous failures of temperature instrumentation and valve position switches. Most of the failures have been on the discharge side of Units A1 and A2. But failures have occurred on the B unit (with a larger compressor case) and on the suction side RTD's as well. There have also been failures of position switches on valves downstream of the compressors (within the compressor buildings). There are no known issues on the yard piping outside the compressor buildings.

Several modifications have been made in an attempt to reduce the number of failures:

- shorter thermowells (75mm insertion length vs 150 mm).
- oil filled thermowells
- RTDs with higher acceleration tolerance
- relocating the thermowells from the compressor casing to the piping

Noise was also a concern in the compressor buildings after the revamp. The primary frequency was found to be at diffuser pass frequency (19X run speed). The suction and discharge piping inside the buildings was insulated to reduce noise levels.

Beta Machinery was tasked to look at the vibrations and make recommendations to reduce failures.

2.0 Investigation

One of the likely causes for failure was the thermowell configuration resulting in flow induced turbulence. The flow past the thermowell causes vortices to be shed at a characteristic Strouhal frequency. If that vortex shedding frequency matches the mechanical natural frequency of the thermowell, then a resonance occurs. Resonance results in high vibration that can lead to fatigue failures. However, two different thermowell geometries were installed and it was reported to make no difference to the vibration. The Strouhal (vortex shedding) frequencies were calculated for both thermowell geometries at suction and discharge conditions. Those were compared to the estimated mechanical natural frequencies (1st cantilever mode) of the thermowells. Table 1 shows the comparison. The mechanical natural frequencies were well separated from the vortex shedding frequencies meeting ASME PTC 19.3. So, there was no apparent link between the flow-induced turbulence and the vibration problem.

Table 1. Mechanical Natural Frequencies and Vortex Shedding Frequencies of Thermowells

	Original 150 mm Thermowell	New 75 mm Thermowell
Mechanical Natural Freq. (1st Bending Mode of a Cantilevered Beam)	715 Hz	2840 Hz
Vortex Shedding Freq. at Suction Conditions (54°F, 615 psig, and 2500 mmSCFD)	370 Hz	
Vortex Shedding Freq at Discharge Conditions (100°F, 826 psig, and 2500 mmSCFD)	310 Hz	

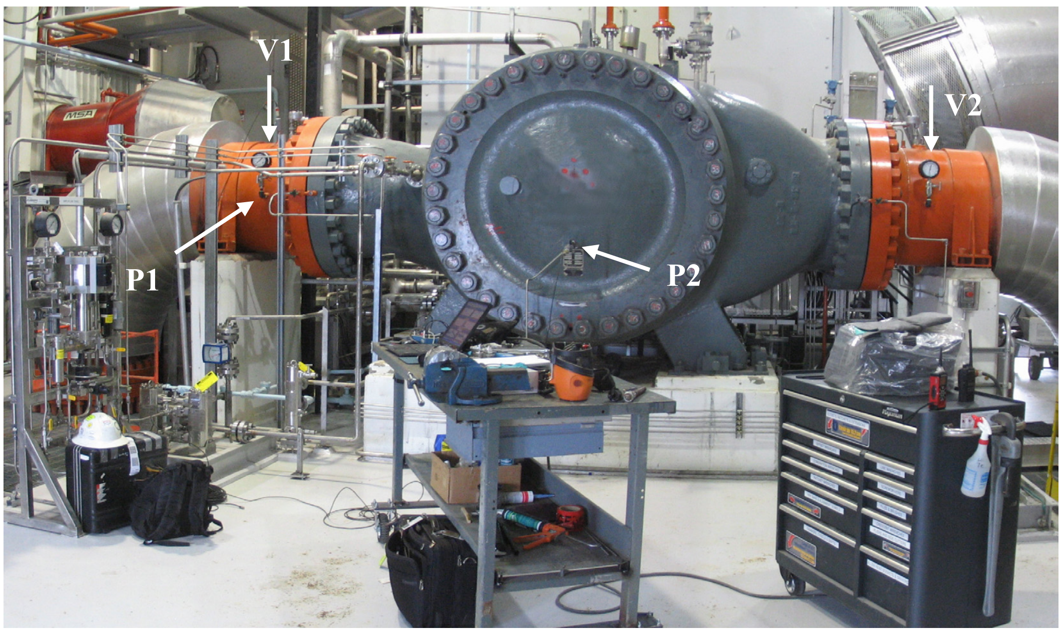


Figure 1. Unit A2 with initial test points marked.

Subsequently, testing of one centrifugal compressor (A2) was done on January 12, 2010 with the primary objective to look for pulsation induced issues. Onsite discussions with plant personnel revealed that a loud noise and vibration of the piping developed at higher operating speeds – again supporting the likelihood of pulsation. Figure 1 shows the unit tested with suction on the right hand side of the photo and discharge on the left. Initially, peak hold pulsations were measured at the compressor discharge and compressor suction eye (marked as points P1 and P2 respectively on the photo) over a speed range of 3600 – 4500 rpm. As the speed increased, the pulsation in the discharge line increased and spiked at the upper end of the speed range (see Figure 2). The peak pulsation was 24 psi pk-pk (compared to a discharge pressure of approximately 825 psig). There is no guideline, but that pulsation is approximately 3% of the mean pressure. In comparison, the pulsation in the suction eye was considerably lower throughout the tested speed range (less than 0.2% of mean pressure) and did not develop a significant single peak (see Figure 3). The frequency of the peak pulsation in the discharge was 1274 Hz which was 17X the run speed and equaled the number of vanes in the impeller (blade pass frequency or BPF). The noise in the building changed when the spike in the pulsation occurred to what sounded like a single tone, but no sound pressure measurements were conducted at the time.

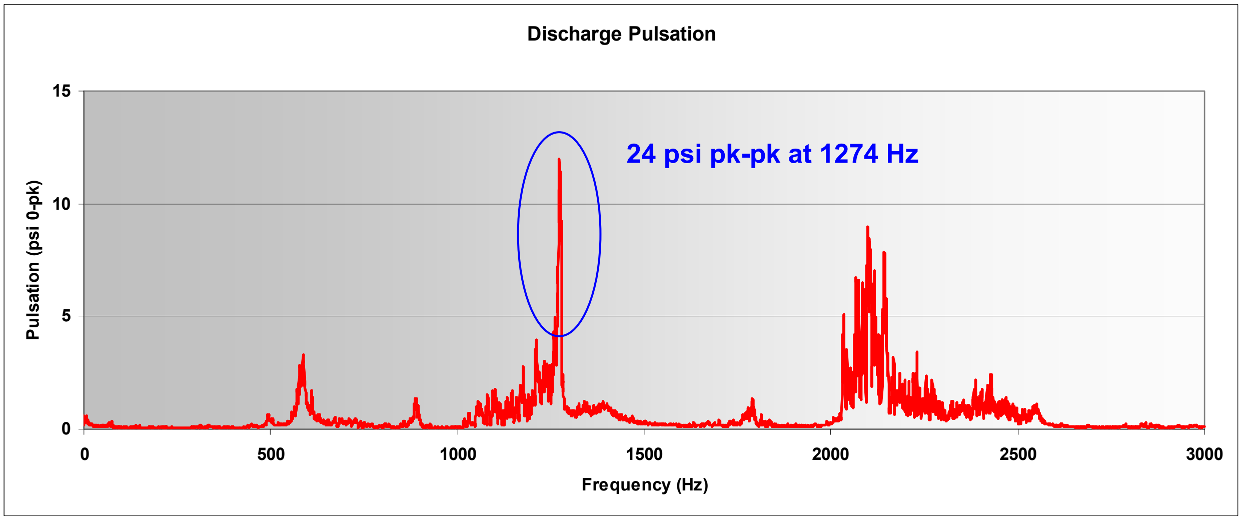


Figure 2. Pulsations in the discharge piping (P1) over 3600 – 4550 rpm.

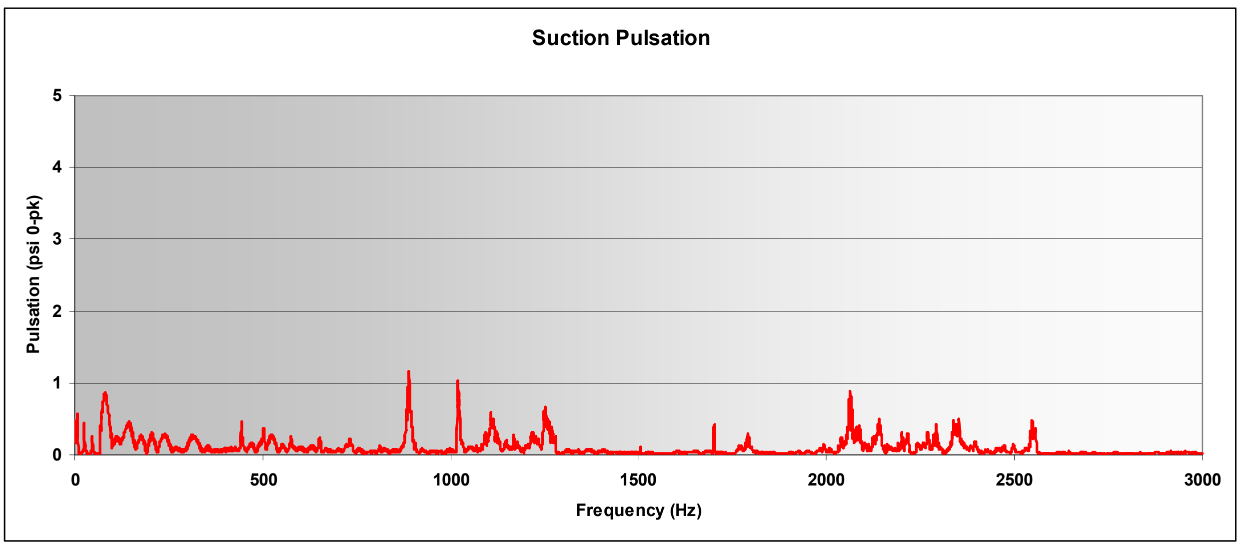


Figure 3. Pulsations in the compressor suction eye (P2) over 3600 – 4550 rpm.

Peak hold vibrations were then measured on the discharge and suction piping beside the temperature instrumentation immediately adjacent to the compressor (marked as points V1 and V2 respectively on the photo) over a speed range of 4100 – 4510 rpm (Figures 4 and 5). As with the pulsations the vibration on the discharge side was significantly higher than the suction side. The discharge piping is 36"OD with 0.75" wall thickness. The peak vibration on the discharge was 1.83 in/s pk at 1284 Hz. Again that frequency was equal to blade pass at 4510 rpm. The vibration is equivalent to 38 g's peak in acceleration terms. Instrumentation that is rated for vibratory service often has an acceleration limit of 10 g's peak. No wonder the RTD's have been failing!

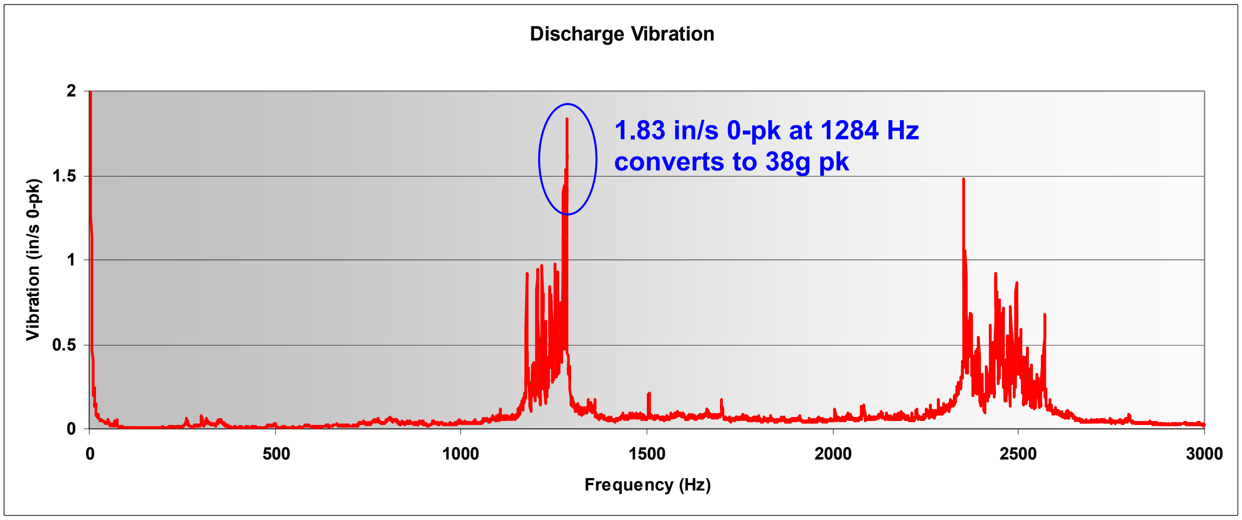


Figure 4. Vibration on the discharge piping beside the thermowell (V1) over 4150 – 4510 rpm.

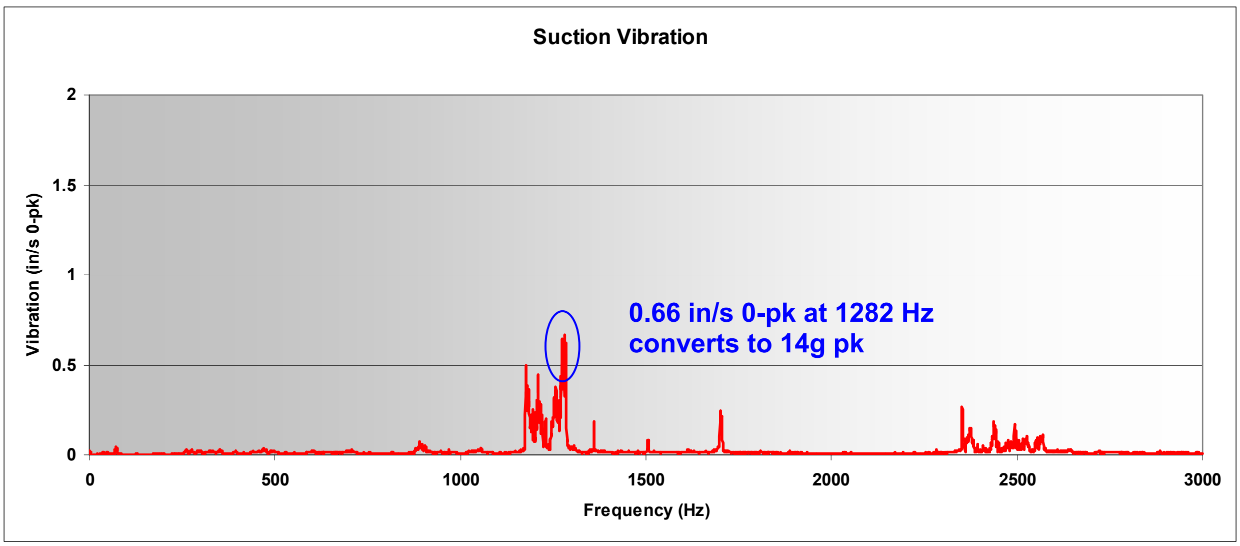


Figure 5. Vibration on the suction piping beside the thermowell (V2) over 4150 – 4510 rpm.

The next step was to conduct an operating deflection shape (ODS) analysis on the piping. The amplitude and phase of the vibration were measured circumferentially around the discharge line beside the thermowell. Vibration at the thermowell was used as a reference. The unit's speed varied during the measurements over a speed range of 4515 – 4550 rpm, but the results were good enough to show that the pipe had a five lobe vibration shape (see Figure 6).

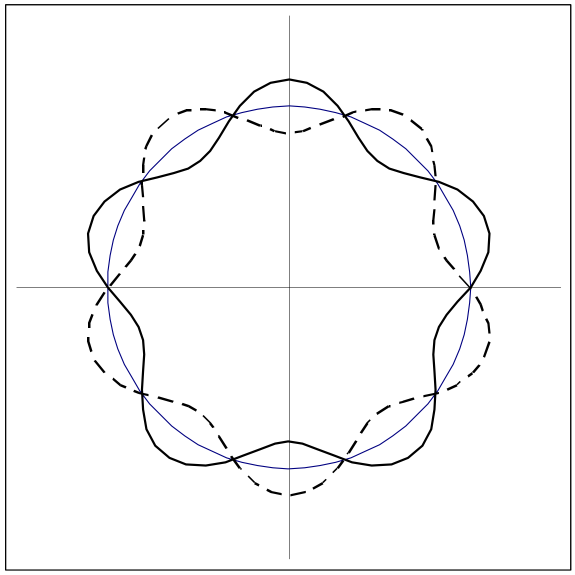
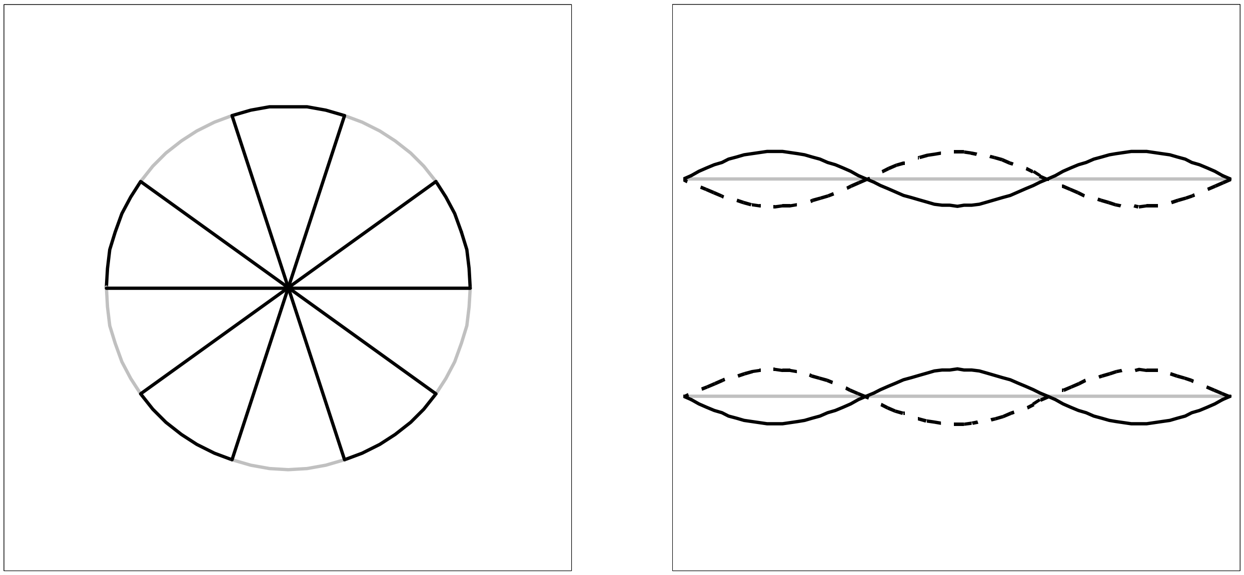


Figure 6. Circumferential mode shape of the 36" discharge pipe with 5 lobes

At this point there was enough evidence to indicate that the source of the high vibration was pulsation in the discharge piping coincident with a probable matching mechanical shell mode of the pipe. However, further measurements were conducted with vibrations recorded from the online non-contacting probes at the bearings. The vibration displacement of the compressor shaft relative to the bearing housings showed negligible amplitudes at the vane pass frequency that was observed in the discharge pulsation and vibration. The major component of the shaft relative vibration was at 1X shaft speed with acceptable amplitudes relative to the bearing clearances.

Further study of the problem was conducted to see if coincident acoustic and mechanical natural frequencies could be confirmed analytically. First calculations were performed on the acoustic natural frequencies of the discharge piping. To limit the complexity of the actual geometry, these calculations were limited to the straight run between the discharge flange the first elbow and assumed a rigid pipe wall. At the high frequencies and dimensions, we are not looking at simple plane waves in the pipe. There are many acoustical natural frequencies with transverse and axial components to the mode shape. But interestingly, one of the frequencies that came out of the calculation matched the measured frequency relatively closely (1261 Hz vs 1274 Hz). That acoustical mode had 5 high pressure lobes in one plane and 3 nodes along the axial length (see Figure 7). That obviously matched the ODS measured on the pipe at one circumference of the discharge line. How about along its length?



a) Plane acoustic mode component b) Axial acoustic mode component

Figure 7. Acoustic mode i=5 (plane), j=3 (axial), k=0 (circle)

A finite element model of the pipe shell was then run for the same length of discharge pipe and a matching mode shape was found at 1266 Hz. See Figure 8 and 9 below. The analytical results confirmed that not only were there coincident acoustic and mechanical natural frequencies, but that they had corresponding mode shapes. Therefore they would be very highly coupled. That is, the shape of the pipe shell mode in both the circumferential and axial directions matched the pressure distribution inside the pipe caused by the acoustic natural frequency.

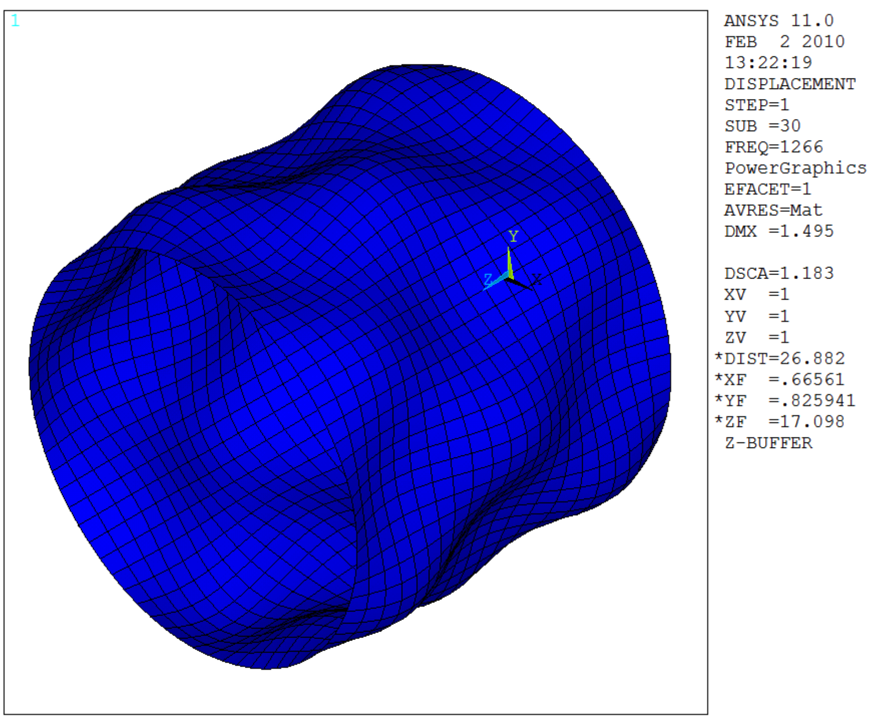


Figure 8. 1266 Hz shell mode of the pipe with 5 lobes circumferentially and 3 nodes axially

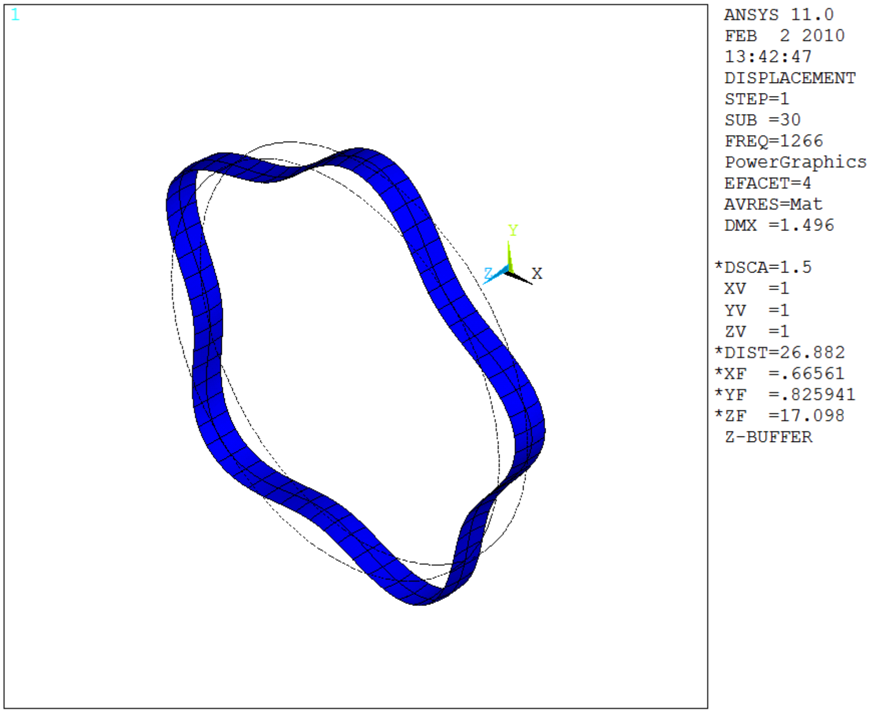


Figure 9. A selected ring of the pipe shell mode to show the 5 lobes more clearly

An interference plot was created (Figure 10) which clearly shows the “perfect storm” that occurs when the compressor is running near 4500 rpm. Pulsations at blade pass frequency (17X run speed) are the excitation source for the coincident acoustic natural and pipe shell natural frequencies of the discharge system immediately connected to the compressor discharge flange. The suction acoustic natural frequency (for similar geometry, but different pressures and temperatures) is sufficiently separated from blade the resonance until a lower speed. The shell modes of the suction pipe will be very similar to that of the discharge – so there is no speed where both the acoustic mode and shell mode of the suction pipe are excited by the blade pass frequency at the same speed. That is why failures on the suction side were far fewer. Although there was apparently enough energy transmitted from the discharge to cause a few suction RTD failures.

As stated earlier, the source strength of the pulsations will vary with the operating conditions. The operating point at the time of testing is marked on the compressor head curve (Figure 11). The compressor was operating off the best efficiency point, albeit it is not far off.

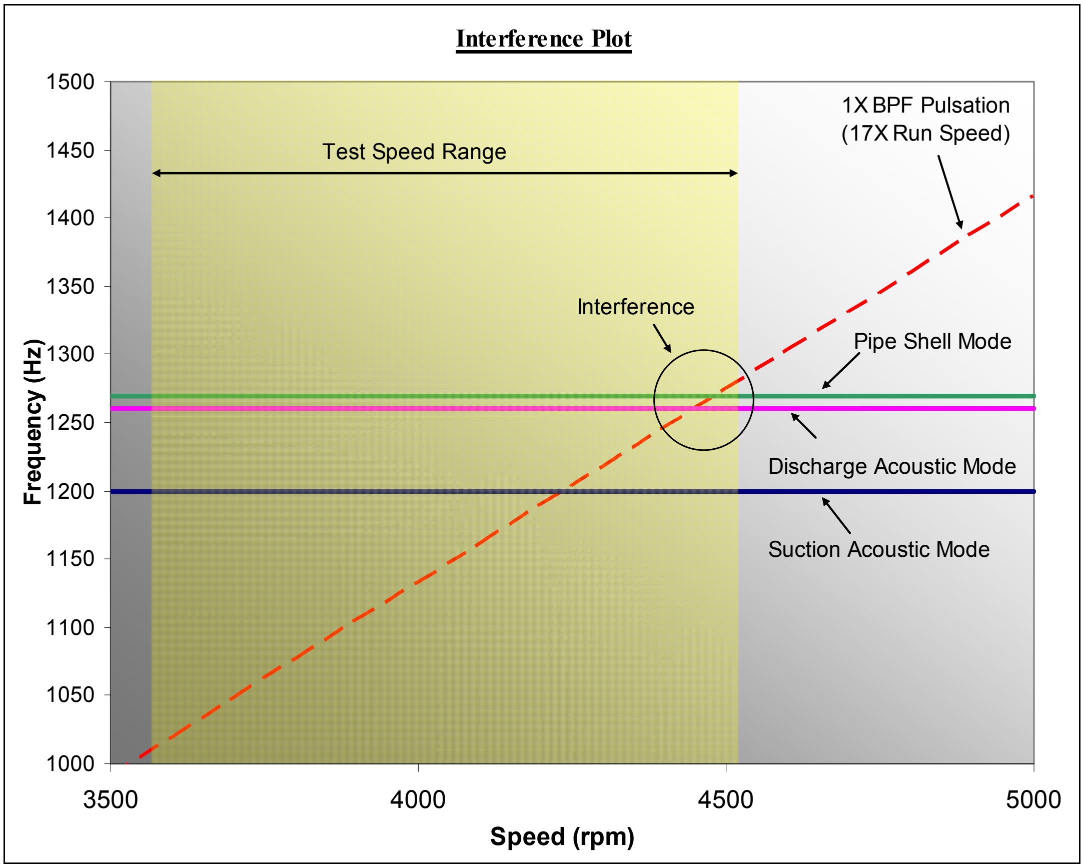


Figure 10. Interference plot showing the intersection of blade pass frequency pulsation (BPF) with the pipe shell and discharge acoustic natural frequencies between 4400 and 4550 rpm.

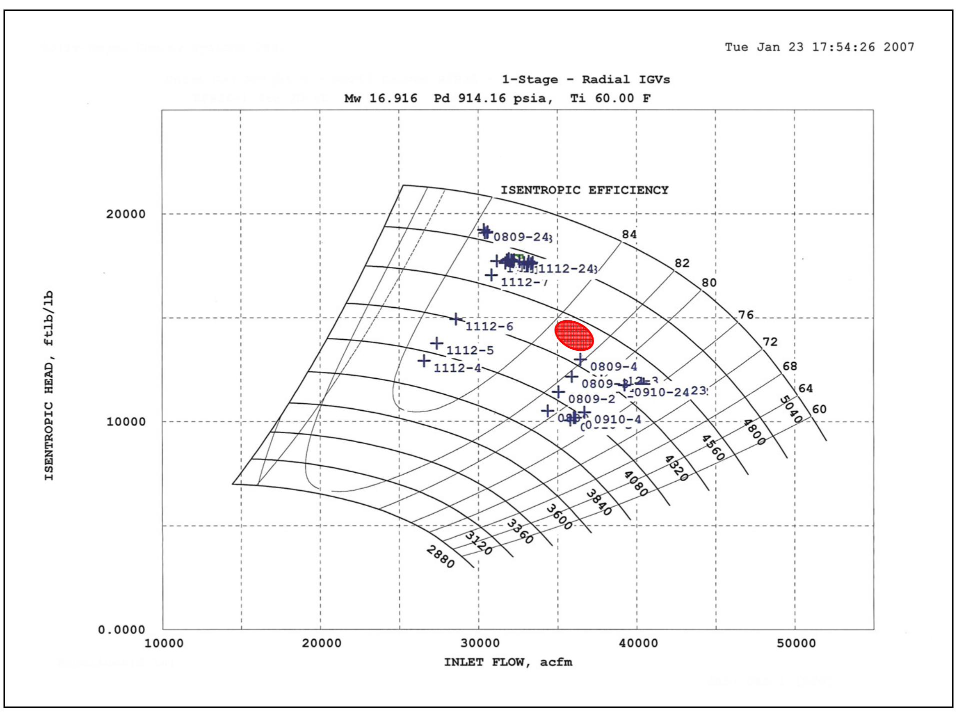


Figure 11 Head Curve for Unit A compressors. Red oval marks approximate operating conditions when high vibration and pulsation was occurring.

The client had an additional question about the piping vibration. Were the vibrations likely to cause a fatigue failure of the discharge piping itself? Since the finite element model was already available, a stress analysis was completed on the pipe shell mode that was being excited. The finite element model was used to estimate stresses in the pipe shell when the mode was excited to an arbitrary displacement as shown in Figure 12. That calculated stress was then scaled to the actual measured displacement of 1.87 x 10^{-4} in pk and found to be 710 psi pk-pk. This is well below the endurance limit for carbon steel pipe and allows for some stress intensification at discontinuities in the geometry. Although this indicates there is a very low probability of failure of the pipe itself, it does not mean that small bore connections in the vicinity will have acceptable stress range levels.

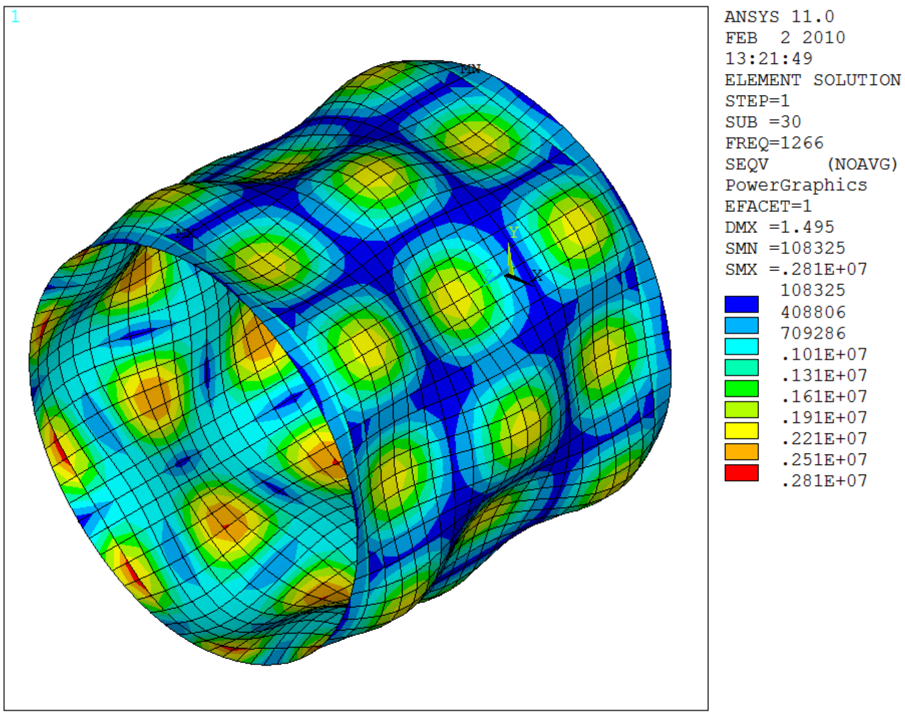


Figure 12. Stress plot for the 1266 Hz pipe shell mode.

3.0 Recommendations

The following have been suggested to the client as possible solutions. Discussions are ongoing with the client to determine further testing and modelling. These recommendations will require modeling to ensure that an adequate separation margin is achieved between any mechanical natural frequencies and their corresponding acoustic natural frequencies (similar mode shapes).

3.1 Change the wheel or aerodynamic package of the compressor to reduce the blade pass source strength. One possibility is to remove the vane diffusers in the casing. This may be tested on one machine soon.

3.2 Stiffen the piping to alter the mechanical natural frequencies of the discharge piping. Of particular concern are the shell modes near blade pass frequency. This may be accomplished by using thicker wall pipe or adding external stiffeners. A thicker spool piece (more than 0.75" wall thickness) would move the natural frequency of the problematic mode to approximately 1400 Hz. The corresponding acoustic natural frequency would be approximately 1270 Hz. This calculation is based upon changing only the straight pipe between the discharge flange and the first elbow. It would be prudent to change the wall thickness of the flange and the first elbow as well. This modification would still allow the acoustic natural frequency to be excited by the blade pass. However, the response of the pipe shell would be much less due to the separation between the mechanical natural and forcing frequencies.

3.3 Alternatively add internal stiffeners/splitters to alter both the mechanical pipe shell modes and the acoustical modes of the discharge spool cavity. One potential solution is to weld cross plates in the ID of the spool immediately downstream ofthe compressor discharge flange to the first elbow. The plates (1/2" thickness minimum) would significantly increase the acoustic natural frequencies of the spool. As well there would be some stiffening of the mechanical natural frequencies (pipe shell modes) and those would be somewhat higher than the present design. Both would then have separation from the 17X blade pass frequency. This recommendation is based upon very approximate calculations. If this is the chosen modification additional calculations should be done to check the many acoustic and mechanical natural frequencies against blade pass to ensure that another mode would not be excited.

3.3.1 Consider having future installations checked at the design stage. It is possible to predict interferences between natural frequencies and potential excitation sources such as, vortex shedding and blade pass frequencies.

3.3.2 Consider adding constrained layer damping material to the pipe spool to reduce the resonant response.

4.0 Conclusions

Standards such as the Energy Institute, "Guidelines for the Avoidance of Vibration Induced Fatigue Failures in Process Pipework", 2nd edition, give excellent tools for screening the various excitation mechanisms possible on centrifugal compressor systems such as: mechanical forces, flow-induced turbulence, rotating stall, and momentum changes. However, pulsation issues at blade pass frequencies under normal operating conditions are not included by that or other standards and are not normally studied during the station design. So although screening of the design is valuable and is recommended, it does not ensure a trouble-free installation. It can be very difficult and expensive to make modifications to solve resonance issues after a compressor system has been built. It is much easier to modify a system at the design stage to get an adequate separation margin between the source frequencies and the natural frequencies. The case study presented here would have required a detailed design study of the suction and discharge systems to reveal the coincidence of acoustic and mechanical modes with blade pass frequency pulsation. In this instance, a detailed design study will be completed after the fact to find the best solution, although the possible modifications will be limited by the constraints of the existing installation.

References

[1] Energy Institute, "Guidelines for the Avoidance of Vibration Induced Fatigure Failure in Process Pipework," 2nd Edition, 2008.

[2] ANSI / ASME PTC 19-3, "Performance Test Code - Temperature Measurement", 1974 (R1998).