ENGINEERING & DESIGN SERVICES
FIELD SERVICES
OPTIMIZATION SERVICES

Case Study
Keeping it running smoothly

Centrifugal Compressor Pulsation/Vibration Problems

Summary

In 2008, three centrifugal compressors at this compressor 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 excessive shell mode piping vibration 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 also highlights why a centrifugal vibration study may be good practice during the initial design (or retrofit). This study predicts interferences between the compressor and shell mode piping natural frequencies and potential excitation sources such as flow induced or, and, blade passing pulsations. 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.

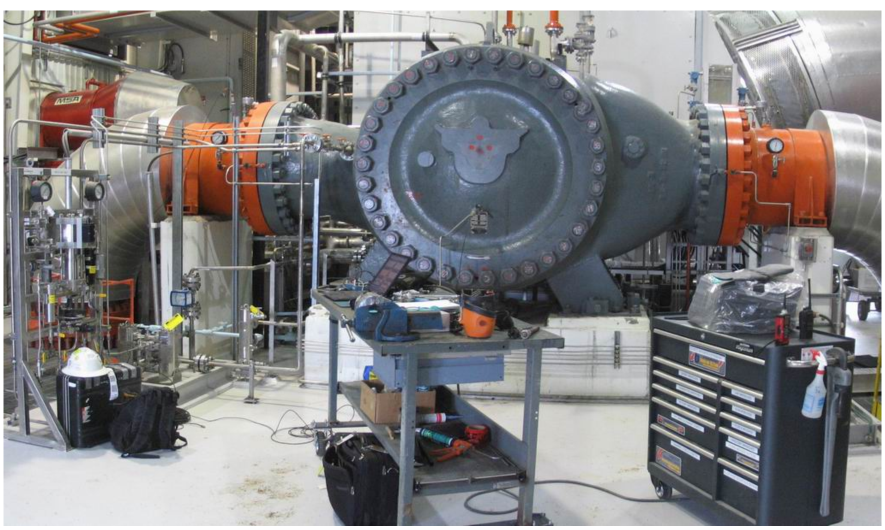

Figure 1: One of three centrifugal compressors that experienced vibration problems due to mechanical and acoustical resonance

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MACHINERY ANALYSIS

Vibration and Pulsation Problems

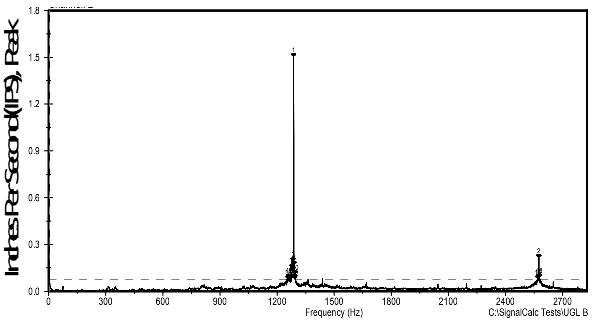

Figure 2: Vibration on Compressor Discharge Piping

The vibration and pulsation on the unit increased significantly when the compressor was run at or near full speed of 4500 rpm. The operating condition was just off the best efficiency point of the compressor.

As shown in Figure 2, the vibration levels were as high as 1.5 ips pk on the discharge piping and varied around its circumference. This is equivalent to an estimated 30g pk at blade passing frequency. This vibration amplitude is above recommended guidelines for the piping system.

Sources of Vibration Problems

The primary vibration sources on centrifugal compressor systems include:

1. Pulsations due to Blade Passing Frequencies
2. Vortex Shedding (Flow Induced Vibration)
3. Surge

1. Pulsations due to Blade Passing Frequency

The main excitation source in this case was due to the blade passing frequency. The primary frequency of vibration was at blade pass frequency, 1260 – 1280 Hz (17X run speed) and the secondary frequency of vibration was at twice blade pass frequency (34X run speed).

On the discharge side the pulsation was as high as 20 psi pk-pk at blade pass with the overall pulsation being 30 – 40 psi pk-pk. On the suction side the frequency content was the same as the discharge, but with lower amplitudes.

2. Vortex Shedding (Flow Induced Vibration)

Two kinds of vortex shedding problems can develop in centrifugal compressor piping systems.

i. Dynamic forces from vortices generated behind thermowells can resonate with the mechanical natural frequency (MNF) of the thermowell; thermowell failures causing gas leaks can result.
ii. Vortices across the open end of side branches without mean flow can excite acoustical resonances in the piping, which can, in turn, excite MNFs of the piping. Unacceptably high piping vibrations result when resonances develop [ref 1]. Flow-induced vortex shedding can also excite the torsional natural frequency [ref 2].

Both can be avoided using proper techniques in the design stage. In this case study, it was determined that the failures and vibration problems were not due to vortex shedding.

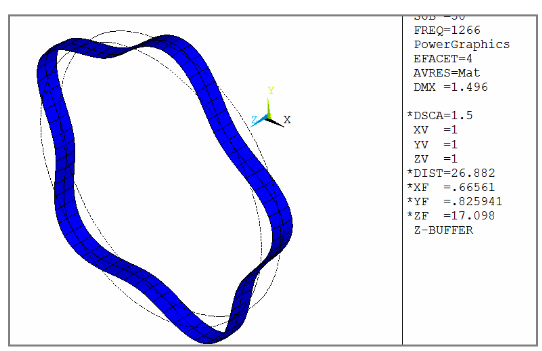

Figure 3: Selected ring to show the 5 lobes of the mode shape

3. Surge

Surge in a compressor creates large, low frequency pulsations and vibrations. However, avoidance of surge is much more important than mitigating short term vibrations that result.

Pipe Shell Mode Natural Frequency

The calculated natural frequency for a 5 lobed circumferential mode shape was approximately 1266 Hz as shown in Figure 3 and in the interference plot, Figure 5.

Stress Levels

A finite element model was used to determine stresses in the pipe shell when the mode was excited using actual measured displacements as shown in Figure 4. That calculated stress was well below the endurance limit for carbon steel pipe. Although this indicates there is a very low probability of failure of the pipe itself, the excessive pipe vibration causes serious problems with branch attachments, leading to the failures.

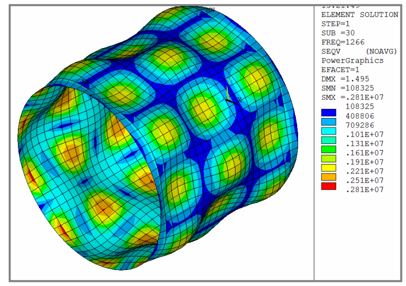

Figure 4: Piping mode shape

Acoustic Natural Frequency

At the discharge conditions, the speed of sound is 1430 ft/s. With that speed of sound, the discharge pipe has a transverse acoustic natural frequency of <u>1260 Hz</u>. The mode shape <u>matches</u> that of the pipe shell mode natural frequency.

Interference Plot

Figure 5 is an interference plot showing the coincidence of the discharge pipe shell mode, acoustic natural frequency and blade passing frequency at a compressor speed of approximately 4400 – 4500 rpm.

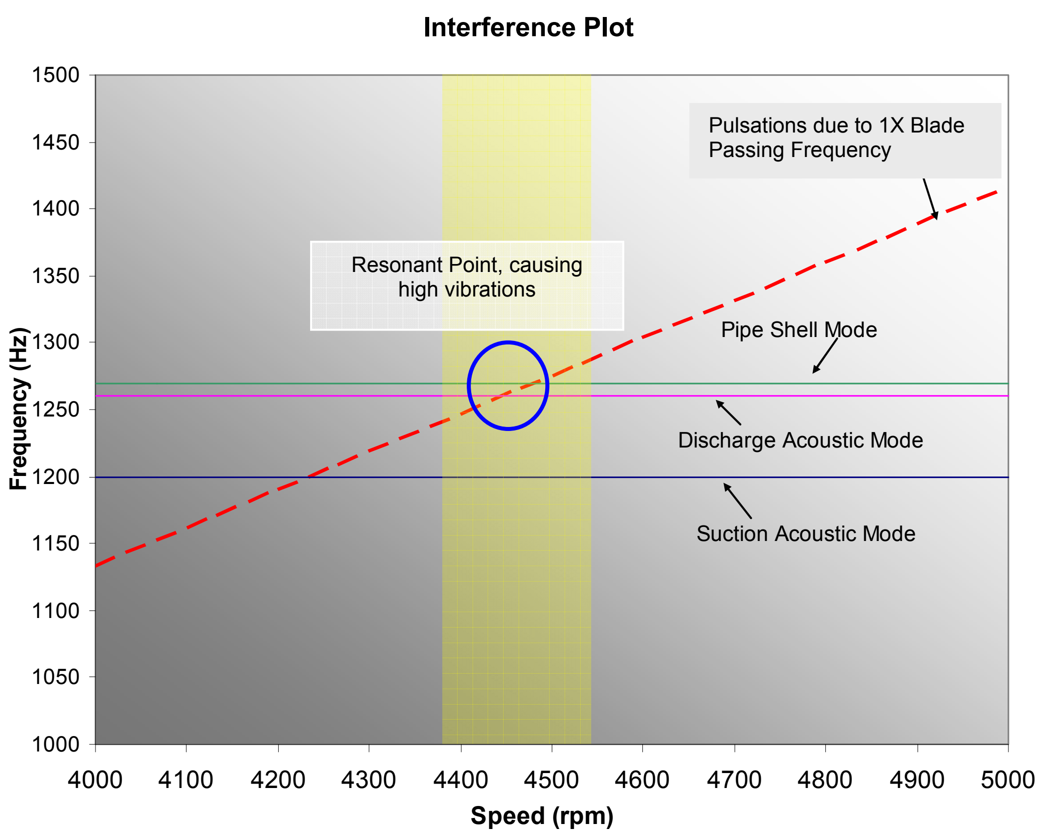

Figure 5: Interference plot

It was concluded that the piping was resonant because the shell mode (natural frequency) is coincident with 1X blade passing frequency of the impellers (17X Run Speed). The vibration was aggravated because the acoustic natural frequency also occurs at this same blade passing frequency. Additionally, the mode shapes are such that the mechanical and acoustical natural frequencies are well coupled making the piping responsive to the input energy.

If both the shell mode and acoustical natural frequencies had been well separated from each other or from the pulsation frequency, then the vibration response would have been greatly reduced.

Possible Solutions

Three solutions were identified to resolve the piping vibration:

- Changing the aerodynamics of the compressor internals to reduce or eliminate the pressure pulsations.
- Stiffening the piping to move the shell mode frequency away from resonance.
- Adding splitter plates to the inside of the pipe to change the acoustic natural frequencies of the system away from resonance.

The customer is currently evaluating these options and will begin implementing changes in the near term.

Conclusion

This example illustrates that severe vibration problems can occur on centrifugal compressors due to the interaction of the excitation force (pressure pulsations at blade passing frequency) with the acoustical and piping natural frequencies. Due to the expensive repairs and compressor downtime, vibration problems are very costly to the customer.

A good practice to avoid these problems is to perform a vibration study of the proposed layout during the design stage. A centrifugal compressor vibration study predicts interferences between natural frequencies and potential excitation sources such as, vortex shedding and blade passing frequencies.

References

A Case Study of Piping and Shaft Vibrations on a Centrifugal Compressor; Beta Machinery Analysis, William F. Eckert, P.Eng., Ph.D.; Brian C. Howes, M.Sc., P.Eng.

A Case Study of a Flow-Induced Torsional Resonance, William Eckert, Brian Howes, CMVA, 19th Proceedings on Machinery Vibration, Edmonton AB, 2001

For more performance and condition monitoring examples, contact BETA's application support team.