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High Frequency Acoustic Excitation in Centrifugal Compressor and Adjacent Piping Vibration

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ABSTRACT

In a centrifugal compressor, interactions between impeller blades and stationary parts such as diffuser vanes will generate high frequency pressure pulsations. The frequencies of these pulsations, so called “blade passing frequency” (hereafter BPF), depend on the rotating speed and number of impeller blades. The BPF of a centrifugal compressor is typically beyond 1 kHz, and the resulting pressure pulsations can excite high frequency vibrations in the compressor itself and in the adjacent process piping. In the piping system, generally, there are many natural frequencies around the BPF, and the accompanying vibration modes can be complex shell wall type. In some cases, excessive piping vibration and noise radiation can be observed due to excessive BPF pulsation amplitudes. In the worst-case scenario, cracks in the pipe and eventual failure can occur. Any pipe failure is a serious safety concern and can further cause downtime for the plant operation. It is, therefore, very important to control the piping vibration in the engineering phase of the design.

The OEM experienced a pipe failure within its compressor package due to excessive piping vibration caused by pressure pulsation originating from a centrifugal compressor in a chemical plant. The root cause investigation and countermeasures were collaboratively discussed among the end-user, EPC contractor, and OEM with involvement of a 3rd party consulting company. The parties solved the high vibration problem in the piping by means of the following countermeasures.

- Modified the compressor’s diffuser construction in compressor from vaned type to vaneless type in order to reduce the vibration source
- Increased the pipe thickness in order to reduce the vibration response
- Install a rubber expansion joint in order to isolate the piping system from vibration source (compressor)

In addition to the above three major countermeasures, the fabrication methods such as welding, post-weld-treatment and inspection procedures were improved as well. After implementation of the countermeasures, their effectiveness was confirmed by on-site measurements.

This paper discusses acoustic excitation in a centrifugal compressor, resultant high frequency vibration in the accompanying piping system, along with details surrounding the field case of pipe fracture and its countermeasures as well as root cause analysis.

INTRODUCTION

For many years centrifugal compressors have been widely used in the industry. In such a compressor, pressure rise is achieved by adding kinetic energy to a continuous fluid through a high-speed rotor or impeller. This energy is then converted into potential energy, in other words static pressure rise, in the diffuser and volute casing. As indicated, in contrast to positive displacement equipment, centrifugal compressors are considered to act as continuous flow machines, and therefore, expected to generate only limited pressure pulsations. As a result, in the design phase of this type of compressor pressure pulsations and accompanying vibrations and fatigue stresses are often not a design criterion, and therefore, not taken into account neither for the machine itself nor for the equally important connected installation, such as the piping, supporting structure, compensators, valves, instruments, etc. However, various historical cases have proven that several mechanisms inside the compressor can cause large pressure pulsations and finally damages. Examples are surge and stall in the low frequency range, but also high frequency, tonal components due to the interaction between the high-speed flow and static objects in the flow path.

Since the beginning of the development of centrifugal compressors, the aim is to achieve maximum energy efficiency. A vaned diffuser is an often-used method to improve the efficiency of the kinetic to potential energy conversion. This efficiency is enhanced by minimizing the space between the moving blades of the impeller and vaned stator (minimal tip clearance). However, this optimized efficiency is at a cost such that flow interaction between the moving and static parts can cause large amplitude and high frequency pressure pulsations at very distinct tonal frequencies, well known as Blade Passing Frequencies (BPF). Next to that, when these pulsations coincide with an acoustic and/or mechanical resonance frequency in the volute or the connected piping, further amplification and finally fatigue failure can occur. Due to the high frequencies of these pulsations, and thus the high number of cycles, this failure can happen within a very short period of time.

Due to this high frequency nature of the BPF, it is not so straightforward to consider the acoustics and mechanical behavior of the installation already in the design phase. Depending on the pipe diameter and wall thickness, many acoustic and mechanical resonance frequencies may be present in the frequency range of interest. Moreover, these 3D acoustic modes couple well with the circumferential pipe wall modes. As a result, the acoustic energy is efficiently transferred into vibrational energy of the pipe. Accurate and deterministic prediction of all these (coupled & uncoupled) modes is a tedious job and the predicted resonance frequencies are very sensitive to uncertainties in dimensions, material properties, damping, etc. Next to that, finally as-built details and quality play a key role in the fatigue resistance of the installation. The welded connections, in particular, are very vulnerable to large amplitude and high frequency loading. So, a robust design of large diameter piping is critical for the lifetime of the centrifugal compressor installation and should be taken into account in an early design stage, especially for high speed and vaned diffuser applications.

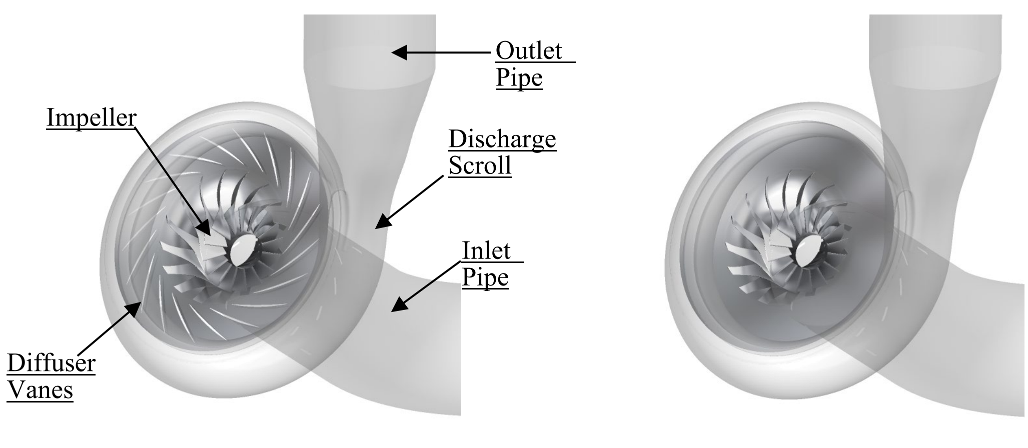
HIGH FREQUENCY ACOUSTIC EXCITATION IN CENTRIFUGAL COMPRESSOR AND ASSOCIATED PIPE VIBRATION

There are methods to visualize the high frequency acoustic excitation in a centrifugal compressor generated by the interaction between rotating blades and stationary parts in pressure field. Unsteady CFD analysis is one of the common practices and a reasonable method to visualize the pressure field in the machine. CFD analysis can provide qualitative evaluation of the magnitude of interaction. Another approach is measurement, either with tests rigs in a laboratory or a commercial machine operating in a field. The quantitative evaluation can be achieved by testing, however the information obtained from the measurements is limited by applicability/availability of sensors such that testing would be a time/cost consuming way.

This chapter provides typical outcomes from both unsteady state CFD analysis and testing in laboratory with regards to acoustic excitation in a centrifugal compressor. The following sections will focus on the interaction between impeller blades and diffuser vanes by referring to CFD analysis and testing on both vaned diffuser and vaneless diffuser.

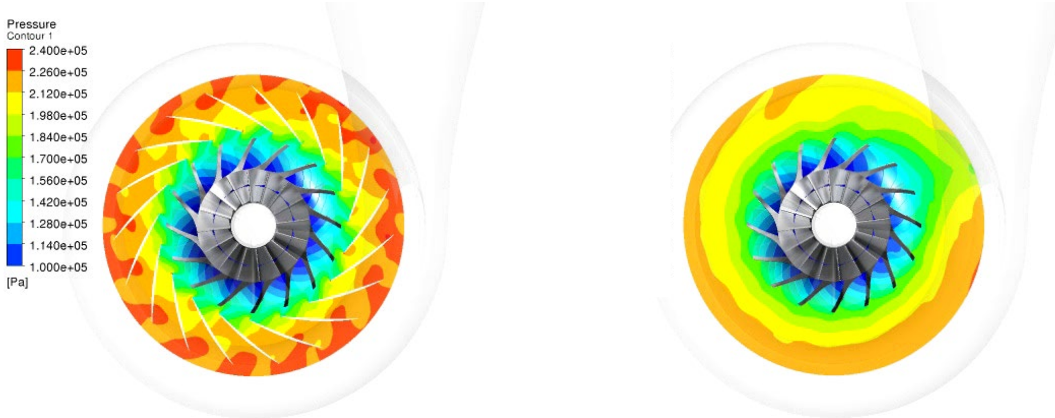
Unsteady State CFD Analysis

A single compressor stage, which consists of an impeller (16 blades), diffuser (vaned or vanless) and discharge scroll casing, was modeled together with connected pipe on inlet and outlet via unsteady CFD analysis as shown in Figure 1. The turbulence model and y+ for the CFD were respectively Stress-Blended Eddy Simulation (SBES) and about 30. The computed static pressure field in the impeller/diffuser and the wall surface of discharge scroll/pipe is illustrated respectively in Figure 2 and Figure 3. In the case of a vaned diffuser, there is a periodic pressure distribution in the circumferential direction with 16 harmonics due to the blade and vane interactions; and the pressure pulsation propagates to the discharge scroll and downstream pipe as well. In the case of a vaneless diffuser, on the other hand, the smooth pressure distribution is shown in the diffuser, discharge scroll and pipe. Figure 4 shows the time vs static pressure at the tie-in point between the discharge scroll and pipe along with its spectrum for both the vaned diffuser case and vaneless diffuser case. The large pulsation with 16 harmonics is clearly generated in vaned diffuser case, and it is also noted that the small pulsation with 16 harmonics is visible in the spectrum even in case of a vaneless diffuser, but the magnitude is significantly different from that of a vaned diffuser.



Vaned Diffuser Case   Vaneless Diffuser Case

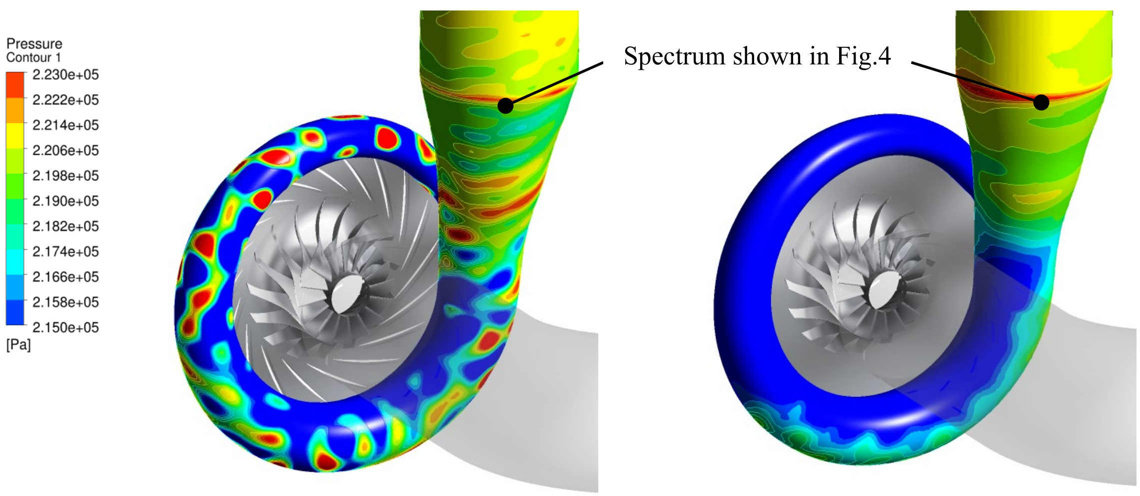
Figure 1. Model for CFD Analysis



Vaned Diffuser Case

Vaneless Diffuser Case

Figure 2. Static Pressure Field in Impeller and Diffuser



Vaned Diffuser Case

Vaneless Diffuser Case

Figure 3. Static Pressure Field on Wall Surface of Discharge Scroll and Pipe

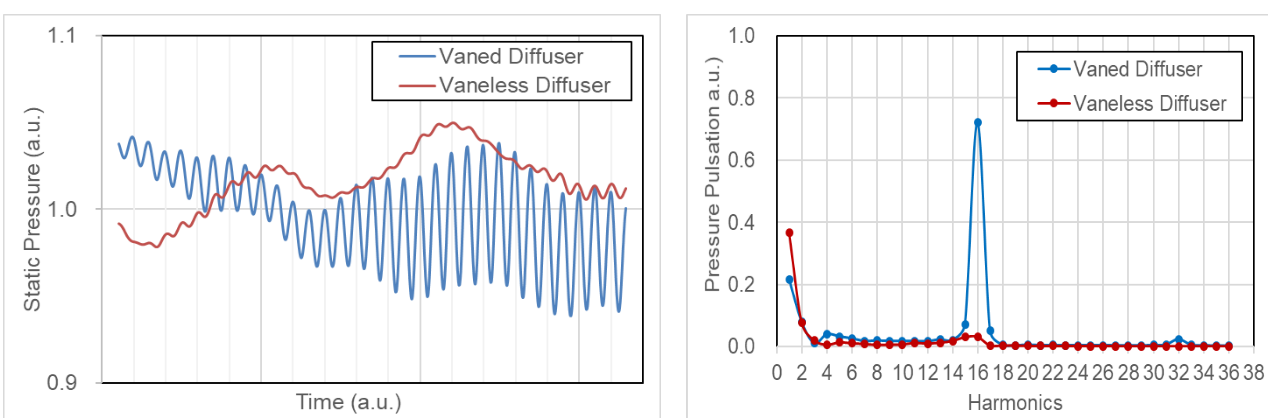


Figure 4. Static Pressure vs Time and Spectrum (at the Discharge Flange)

Testing in Laboratory

The testing campaign took place in the OEM's laboratory with a single stage compressor shown as Figure 5 and Table 1. The test compressor was driven by a turbine open to atmospheric air at inlet, and the outlet was connected to piping system of laboratory equipped with valves, a silencer and so on, then finally air was discharged to atmosphere. The overview of the test setup with instruments is shown in Figure 6. Pressure transducers for measurements of pressure pulsation were installed not only in the pipe but also in the diffuser of the compressor.

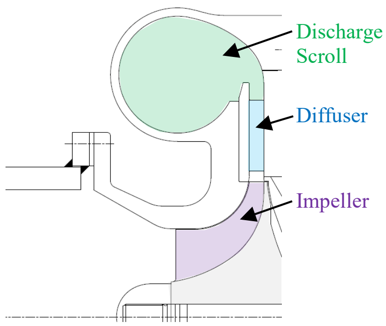


Figure 5. Compressor Stage (Meridional View)

Table 1. Basic Specification of Test Compressor

Impeller Diameter	195 mm
Number of Blades	16
Diffuser Type	Vaned and Vaneless
Test Speed	40,920 rpm
BPF	10,912 Hz
Inlet Pressure/Temperature	ATM
Pressure Ratio	2.5

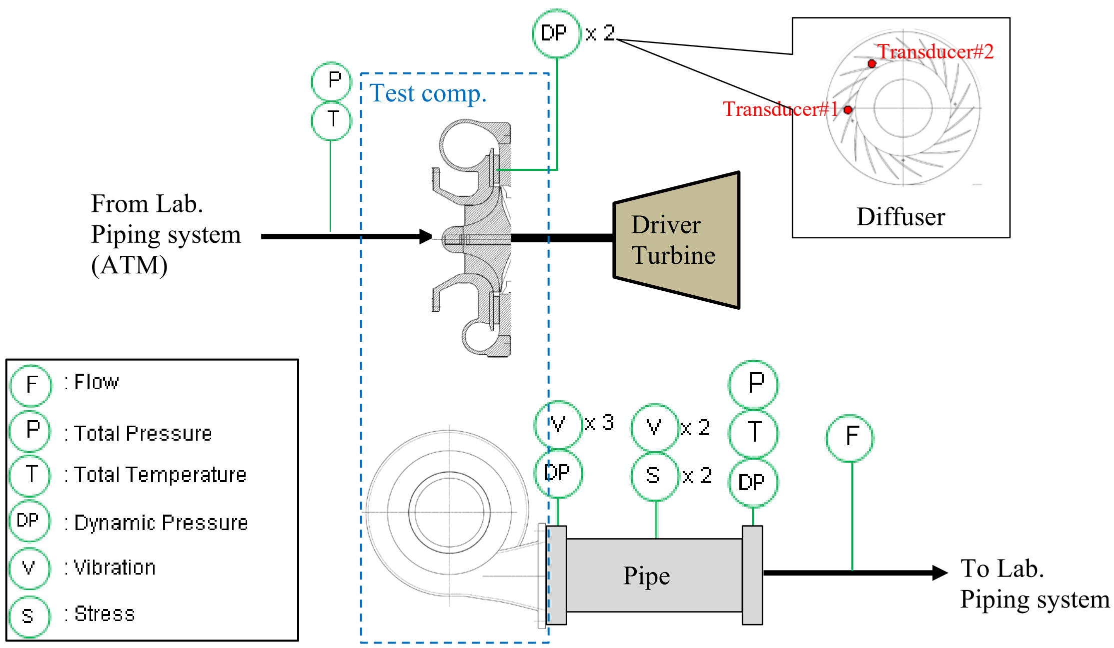
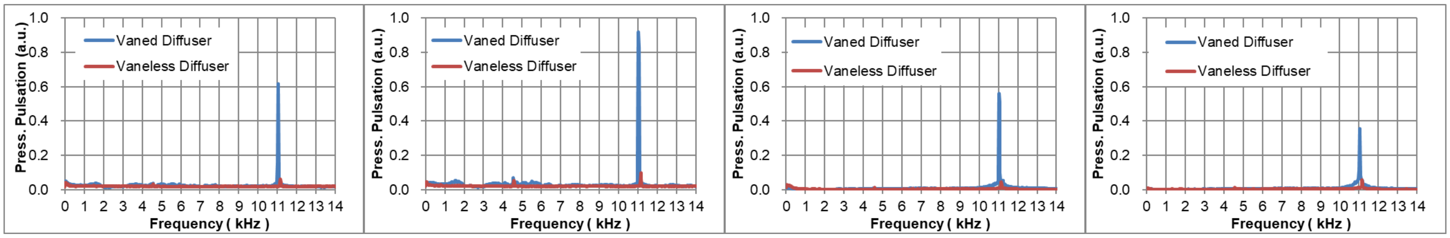


Figure 6. Schematic of Test Rig Setup Overview with Measurement Points

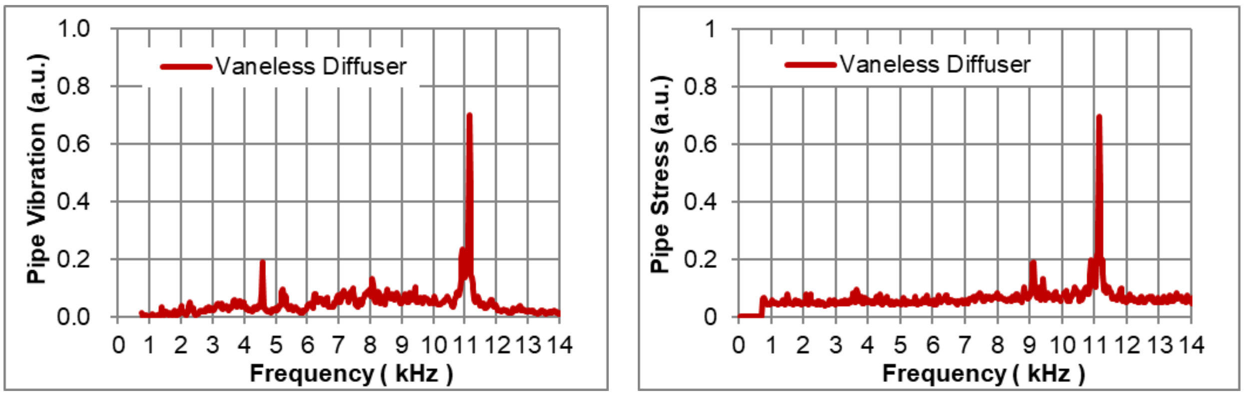
Figure 7 shows spectrums of pressure pulsation in the diffuser and inlet/outlet flange of downstream pipe for both vaned diffuser case and vaneless diffuser case when the compressor was running near the design point. As seen in results of CFD analysis, the peak at 11 kHz corresponding to BPF is remarkable in case of vaned diffuser and a small peak is still visible in case of vaneless diffuser. The magnitude of pressure pulsation for vaned diffuser is more or less 10 times larger than that for vaneless diffuser. It is also noted in vaned diffuser case that the pressure pulsation is gradually decreasing as distance from the acoustic source (diffuser) increases.

Figure 8 representatively shows one of the spectrums of vibration and stress for the vaneless diffuser case. Comparison in vibration and stress between vaned diffuser and vaneless diffuser was not made since the stiffer pipe was used for tests with vaned diffuser than tests with vaneless diffuser thus vibration response could not be well measured in testing with vaned diffuser. The spectrum of vibration and stress shows the peak at 11 kHz, just like the pressure pulsation, however the peak is not steep like pressure pulsation but with certain width of band. It is considered that numerous modes of vibration should be excited by pressure pulsation since there must be many natural frequencies near BPF and therefore a band of vibration response should be created.



Diffuser (Transducer#1) | Diffuser (Transducer#2) | Pipe Inlet Flange | Pipe Discharge Flange

Figure 7. Spectrum of Pressure Pulsation for both Vaned and Vaneless Diffuser (Measured by Test)



Pipe Vibration | Pipe Stress

Figure 8. Spectrum of Pipe Vibration and Stress for Vaneless Diffuser (Measured by Test)

The gain of achieving lower pressure pulsations and lower accompanying mechanical loading with the vaneless diffuser is at a cost of adverse effect on the aerodynamic performance as shown in Figure 9. Both the pressure coefficient and efficiency with the vaneless diffuser are lower than with the vaned diffuser. Here the difference in efficiency was in the order of 2~3%. On the other hand, the operating range was extended by applying the vaneless diffuser, because the choke flow region moved to right side in the graph. In the testing, surge flow could not be verified due to the limitation of instruments, therefore the minimum flow shown in the graph was not the surge limit point.

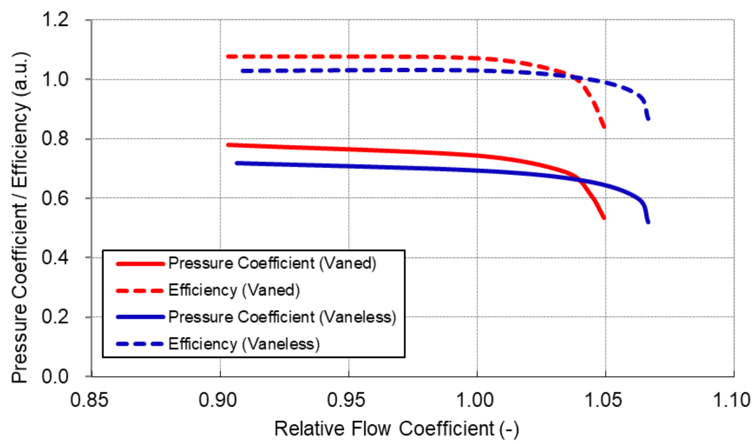


Figure 9. Compressor Aerodynamic Performance (Measured by Test)

PIPE FRACTURE FAILURE BY HIGH FREQUENCY VIBRATION

In a chemical plant, some of the operating parameters on the process air compressor indicated abnormalities in pressure and temperature of process air after a few months had passed since the machine started while the compressor was running near the center of OEM's performance map (well within the operating map), and the machine was immediately stopped and inspected. The damage, holes as well as cracks, in the process pipe, which was supplied by OEM to directly connect the compressor stages, were found, then the root cause investigation, verification of countermeasures and validation of countermeasures by on-site measurements were collaboratively carried out by end-user, EPC contractor and OEM with involvement of a 3rd party consulting company.

Machine Construction and Specification

The inter-connecting pipe between 1st stage and 2nd stage is directly mounted on the scroll casings of the integrally geared type process air compressor as shown in Figure 10. Table 2 shows the basic specification of the compressor stages. The 1st stage compressor is constructed by an open impeller (number of blades :16) and a vaned diffuser, as the combination between rotating speed (7,528 rpm) and number of impeller blade (16) the BPF is approximately 2,000 Hz. The vibration frequency which will be discussed in this paper is BPF of 1st stage compressor.

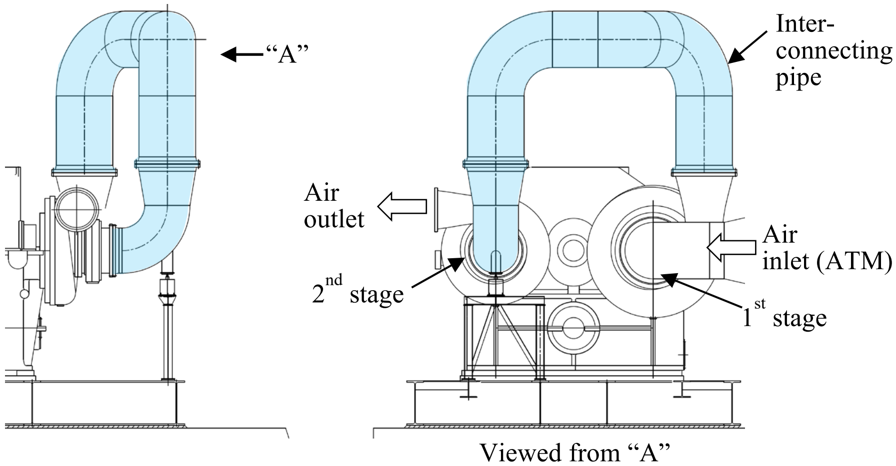


Figure 10. Machine Construction

Table 2. Basic Specification of Compressor Stage

Item	1st stage	2nd stage
Speed (rpm)	7,528	9,345
Impeller type (-)	Open	Open
Flow Coefficient (-)	approx. 0.1	approx. 0.08
Impeller blade number (-)	16	16
BPF (Hz)	2,007	2,492
Diffuser type (-)	Vaned	Vaned

High Frequency Vibration Mode

Figure 11 presents the calculated displacement and stress distributions of the vibration at natural frequencies around 2,000 Hz which are analyzed by FEA. As there are generally numerous natural frequencies existing in such high orders, about 100 natural frequencies are calculated within +/- 1% of 2,000 Hz in this case. The high frequency vibration modes in the pipe show complex shell wall modes and generates local high stress points.

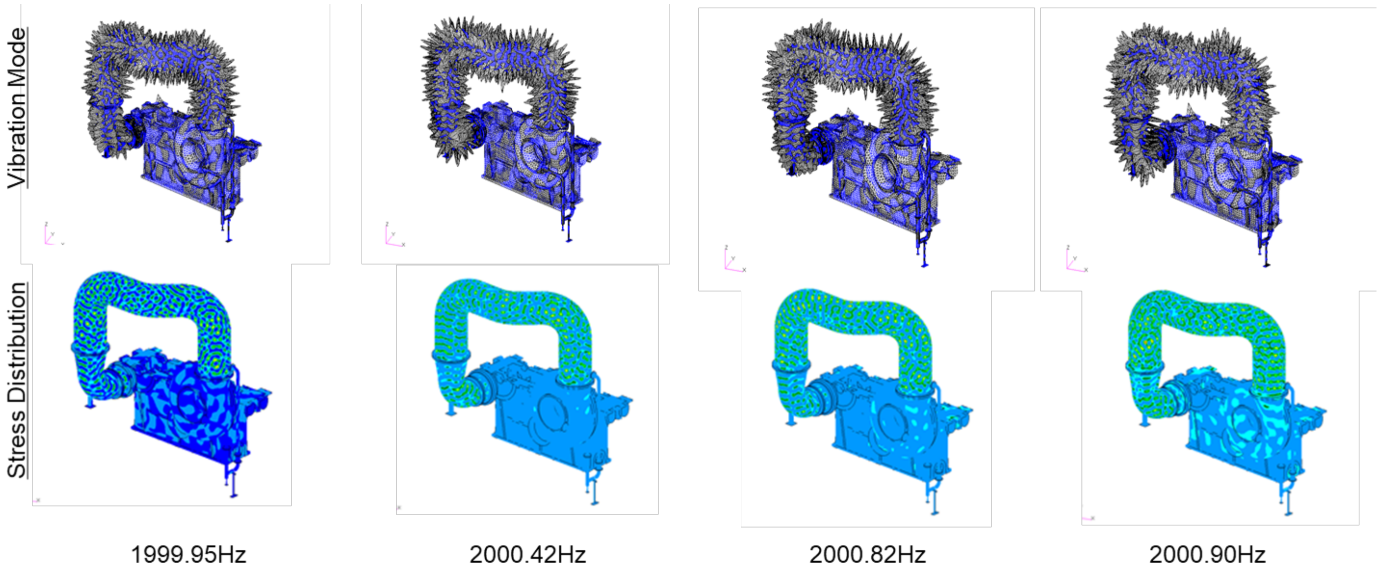


Figure 11. Calculated Vibratory Displacement and Stress at Natural Frequencies around 2,000 Hz

Pipe Fracture Failure Mode

After dismantling the inter-connecting pipe from compressor package, the pipe was visually inspected. It was confirmed that in addition to the cracks, small pieces of the pipe were dropped off from the pipe as shown in Figure 12. The detailed inspections were carried out on the piece of fractured area and following points were identified (Figure 13).

- The cracks originated at the boundary of circumferential welding on the pipe, which was revealed by radiation patterns, which are a typical feature of a crack origin.
- The crack propagated along the welding seam at the beginning of failure with beach marks, which are a typical feature of fatigue fracture.
- The crack propagation tracked in small circles along with the vibration mode at BPF.

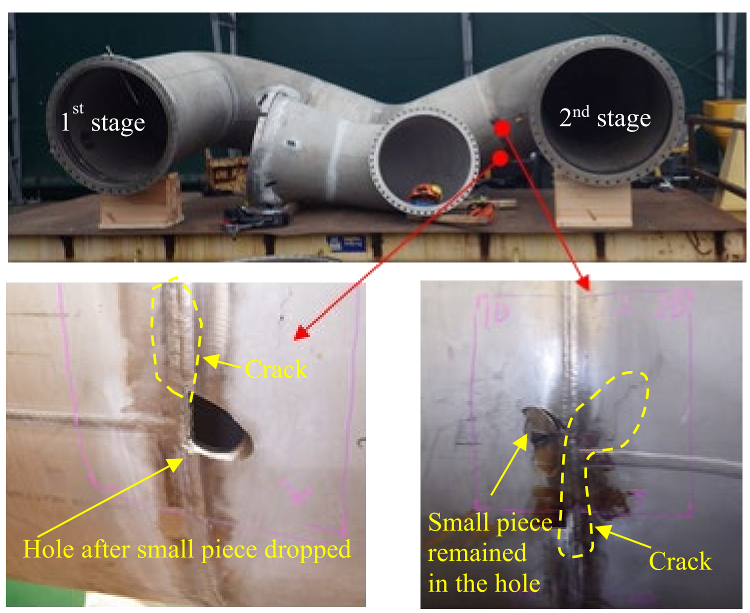


Figure 12. Overview of Fracture Failure of Inter-connecting Pipe

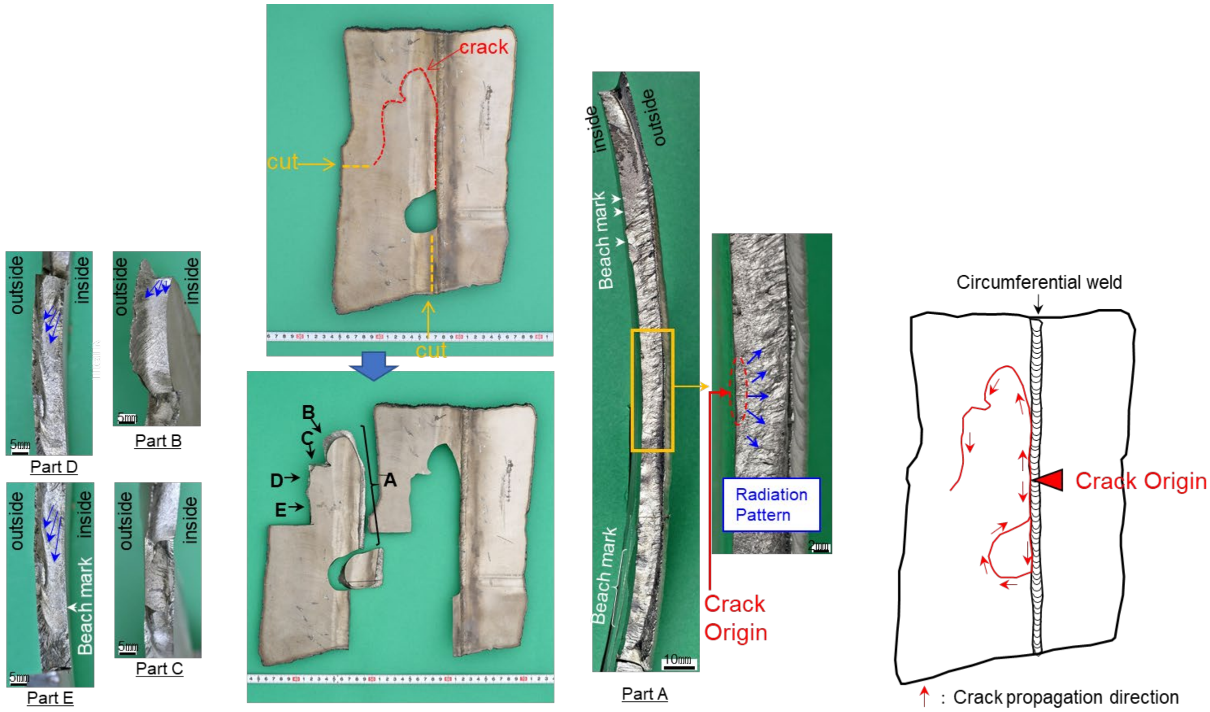


Figure 13. Detailed Inspection on Fracture Surface

Stress Measurement on Failed Pipe

Before the fracture failure happened, the stress in the inter-connecting pipe had been measured by using strain gauges. Figure 14 shows the spectrum of stress at one of the measured locations. The dominant frequency of the stress is 2,000 Hz which corresponds to BPF of the 1st stage impeller. It is also noted that there is a certain width of band around the peak, which could be caused by the excited vibration response for various natural frequencies near BPF. It could also be caused by some small amount of damping.

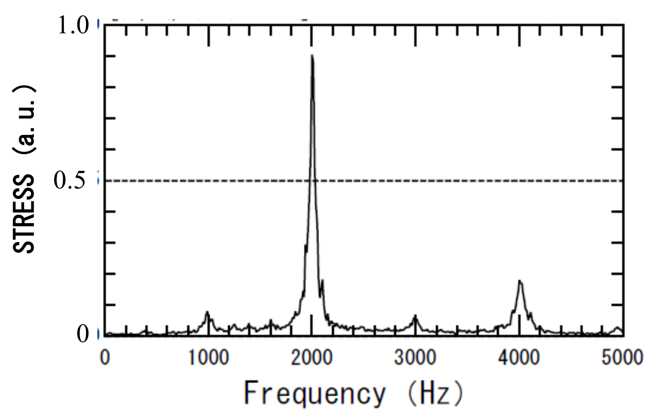


Figure 14. Spectrum of Measured Stress on Failed inter-connecting Pipe

Root Cause of Pipe Fracture Failure

The above described facts led to the following conclusions regarding the root cause of the failure.

- The failed pipe must be vibrated and stressed by pressure pulsations, which were generated by interaction in pressure field between 1st stage impeller blade and diffuser vanes, since the dominant frequency (2,000 Hz) of the measured stress was corresponding to BPF of 1st stage impeller.
- Due to the complex shell wall vibration mode for high order natural frequencies, the stress could be locally high.

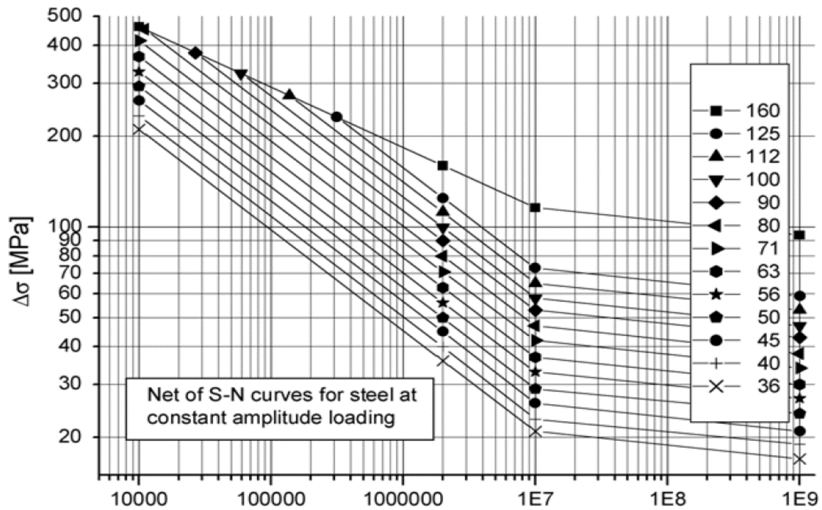
- As the result, the crack initiated from welding seams which could be one of the weakest points in the pipe and propagated along with the complex vibration mode.

Countermeasure
Since there are many natural frequencies around the BPF, it is not possible to tune the natural frequencies to eliminate the resonances. The practical approaches for the risk mitigations are reducing the excitation source, reinforcing the piping system, adding the damping in the system and so on. In this particular case, following measures were taken:

- Vaned diffuser was replaced with vaneless diffuser: reducing excitation source
- The pipe thickness was increased (1.5 times thicker than original): reinforcing the pipe
- A rubber expansion joint between compressor and pipe was installed: adding damping and mechanically decoupling the pipe from discharge scroll

The negative influence of vaneless diffuser in aerodynamic performance was compensated by the margin of the other process components.

In addition to the above, the welding procedure, treatment procedure after weld and inspection (NDT) procedure were also improved in order to achieve the higher fatigue strength at the welded locations. The fatigue strength of various weld styles can be defined by S-N curves. For example, Figure 15 provides the typical S-N curves with welding configurations with NDT requirements which are specified in IIW (International Institute of Welding). Referring to this recommendation, welding from both inner and outer surfaces were implemented, the smooth surfaces were provided by grinding and moreover dye penetrant test (PT) and radiographic test (RT) were performed on 100 percent of welds.



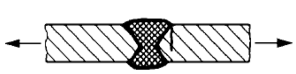
No.	Structural Detail	Description (St.= steel; Al.= aluminium)	FAT St.	FAT Al.	Requirements and Remarks
200	**Butt welds, transverse loaded**				
211		Transverse loaded butt weld (X-groove or V-groove) ground flush to plate, 100% NDT	112	45	All welds ground flush to surface, grinding parallel to direction of stress. Weld run-on and run-off pieces to be used and subsequently removed. Plate edges ground flush in direction of stress. Welded from both sides. Misalignment < 5% of plate thickness. Proved free from significant defects by appropriate NDT
212		Transverse butt weld made in shop in flat position, NDT weld reinforcement < 0.1 · thickness	90	36	Weld run-on and run-off pieces to be used and subsequently removed. Plate edges ground flush in direction of stress. Welded from both sides. Misalignment <5% of plate thickness.
213		Transverse butt weld not satisfying conditions of 212, NDT Al.: Butt weld with toe angle ≤50° Butt welds with toe angle >50°	80	32 25	Weld run-on and run-off pieces to be used and subsequently removed. Plate edges ground flush in direction of stress. Welded from both sides. Misalignment <10% of plate thickness.

Figure 15. S-N Curves for Steel and Welding Configuration IIW document IIW-1823-07)

__On-site Measurements after Countermeasure__

After implementation of countermeasures, in order to ensure the safe operation, accelerometers and strain gauges were installed on the pipe, furthermore a pressure transducer was also installed at the flange of compressor discharge scroll. In order to capture the complex high frequency vibration mode, in other words not to miss the highest stress, many strain gauges were installed as shown in Figure 16. The measurements were made from plant start up until the plant rated load was achieved. Figure 17 shows spectrum plots of pressure pulsation, pipe vibration and pipe stress under plant rated load operation. The main component of these spectra is 2,000 Hz corresponding to BPF of 1st stage as expected. It is also noted that the pressure pulsation generated by 2nd stage impeller acoustically propagated upward relative to flow direction as the small component of approximately 2,500 Hz corresponding to BPF of 2nd stage is visible in the spectra of pressure pulsation and vibration; however the spectrum of stress does not show the component with 2,500 Hz because the vibration is too low to get a certain level of stress. In comparison with original design, the measured stress was significantly reduced (95% reduction) as shown Figure 18.

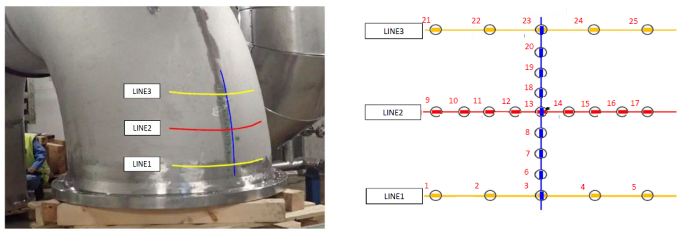


Figure 16. Strain Gauges Arrangement for On-site Measurement

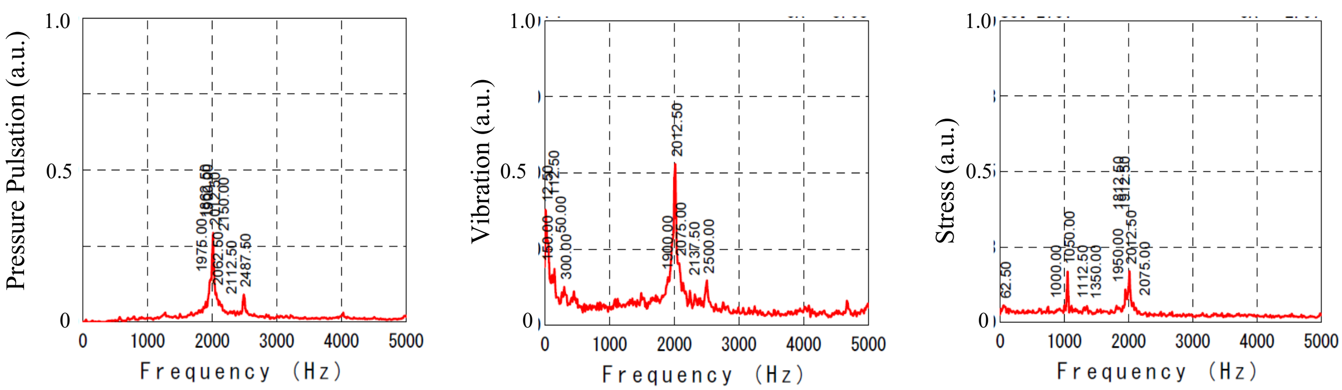


Figure 17. Spectrum of Vibration, Stress and Pressure Pulsation

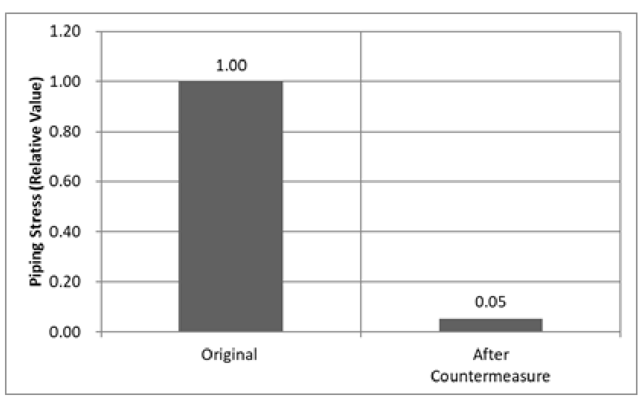


Figure 18. Comparison of Measured Stress on Pipe

Fatigue Life Evaluation on Pipe

In order to determine the fatigue lifetime of the pipe, the measured stress was analyzed by a rainflow counting method and fatigue lifetime was evaluated based on a cumulative fatigue damage theory (M. Matsuishi and T. Endo, 1968). An outcome of this analysis is shown in Figure 19 which displays the plots of counted stress with applicable S-N curve. As the measured stress was located below the S-N curve at any cycle up to one tera-cycle, the fatigue cumulative damage coefficient was calculated to be 0 (zero). It is expected that the pipe is free from fatigue failure and ensures safe operation.

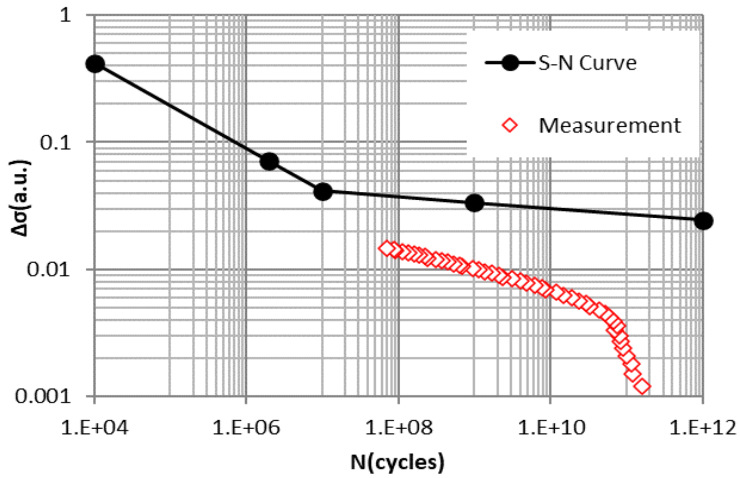


Figure 19. Fatigue Lifetime Evaluation by Rainflow Accounting Method

SUMMARY AND CONCLUSION

High frequency acoustic excitation source in a centrifugal compressor and the resulting vibration in the adjacent piping system were discussed.

- In a centrifugal compressor, a high frequency pressure pulsation is generated by the interaction in pressure fields between rotating impeller blade and stationary diffuser vanes.
- Unsteady state CFD analysis is a powerful tool to visualize the pressure pulsation in the system and provides the qualitative evaluation. Testing is an alternate method to quantitatively evaluate the pressure pulsation.
- OEM carried out both unsteady state CFD and testing campaign in laboratory in order to determine the pressure field and resultant vibration in the compression system.
- The studies were performed on both vaned diffuser case and vaneless diffuser case.
- It was confirmed that the main component in pressure pulsation coincided with BPF in both vaned diffuser case and vaneless diffuser case, however the magnitude of pressure pulsation in vaned diffuser case was significantly higher than that in vaneless diffuser case.
- OEM experienced the fracture failure on piping for a process air compressor at an end-user's site.
- The root cause investigation concluded that the failure had been caused by high frequency vibration with BPF.
- In order to eliminate the possibility of reoccurrence, several measures were implemented in regards to the acoustic system and mechanical system.
- On-site measurements were carried out for pressure pulsation, vibration and stress, and finally the stress in pipe was confirmed to be significantly lower than original design.
- It was expected that no fatigue failure would happen based on the measurement result, and the compressor has been operating successfully without vibration issue in the piping at the end-user's site for a few years since the countermeasures were implemented.
- It was suggested that controlling the acoustic behavior and the resulted vibration is important in early design stage, and additional design rules have been established in OEM's design manual based on the experience.

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M. Matsuishi and T. Endo, 1968, Fatigue of Metals Subjected to Varying Stress, Japan Society of Mechanical Engineering