

# Vibration-Induced Fatigue Failure of Discharge Bottle Fitting Connections – A Case Study

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## Introduction

Vibration measurements are the most common method to accurately and efficiently quantify fatigue risks on piping and vessels subject to vibration. Typically, recorded vibration levels are compared against industry-standard acceptance criteria.

For complex geometries of piping and vessels, or when shell mode vibrations are involved, the use of vibration guidelines as an acceptance tool for fatigue risk can be limited. In some cases, where stress raisers and structural discontinuities are present, vibration guidelines can be non-conservative.

A series of fatigue failures was found on welded fitting of reciprocating compressor discharge bottle nozzles. Since units of identical design operate throughout the area any additional fatigue risk to these units had to be assessed. Due to the complex nature of the geometry, a finite element model was built, as the use of vibration measurements alone was not deemed accurate enough to quantify the fatigue risk.

The case study in this paper describes the methodology used to quantify the fatigue risk on units with similar pulsation bottle designs, through both field testing and analysis. Vibration measurements and dynamic strain measurements were carried out, and results were compared with strain and vibration levels calculated by a finite element analysis. An acceptance guideline for the specific geometry was determined, which could then be compared with other units in operation. The paper also includes considerations taken into account for minor geometric differences between the remaining units in operation.

Lessons learned regarding the design of fittings and welded discontinuities on suction bottles are presented. The paper further discusses recommendations for performing vibration measurements on complex geometry piping and illustrates a methodology to be applied when vibration measurements alone cannot be used to determine fatigue risk.

## Background

A series of fatigue failures was found on reciprocating compressor discharge bottle nozzles. The failures occurred at coupling connection points of asymmetrically located nozzles. These couplings are used for thermo-wells, which are commonly installed to measure discharge temperatures on compressor pulsation bottles.

As shown in

Figure 1, a crack appeared on one of the nozzles in the operating area. This was surprising because no visual signs of high vibration on the bottle or cylinder heads were reported. The distance from the coupling to vessel reinforcement pad (repad) weld varied from sample to sample. Failures were noted on samples at a range of coupling to repad distances. The welding procedure, especially the performance of post-weld heat treatment on the coupling is unknown.

Figure 1 shows a photo from a metallurgical report with a similar coupling that is spaced closely to the repad. Figure 2 shows a cross section of a similar coupling. Sectioning of the crack indicated a high potential for stress raisers due to close proximity of the repad and coupling plus overlapping of the heat-affected zones. Units of identical design operate throughout the area and therefore any further fatigue risk to these units had to be assessed. Due to the complex nature of the geometry, the use of vibration measurements alone was not deemed accurate enough to quantify the fatigue risk. A test program was developed to quantify the fatigue risk using bonded strain gauges and vibration transducers. The important aspects of the test program were to:

- Ensure a representative range of compressor operating conditions is covered during the test
- Obtain measurements considering the range of discharge bottle wedge support adjustment
- Ensure that sufficient vibration measurements were taken to identify the operating deflection shape (ODS) of the bottle
- Quantify the level of strain versus vibration so that stand-alone vibration measurements could be used to quantify risks on similar pulsation bottles
- Validate the FEA model
- Use the FEA model to assess the risk on similar bottles

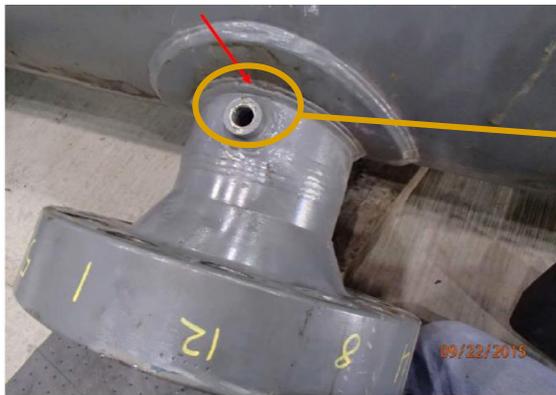


Figure 1: Fatigue crack located between discharge pulsation bottle nozzle reinforcement pad and coupling

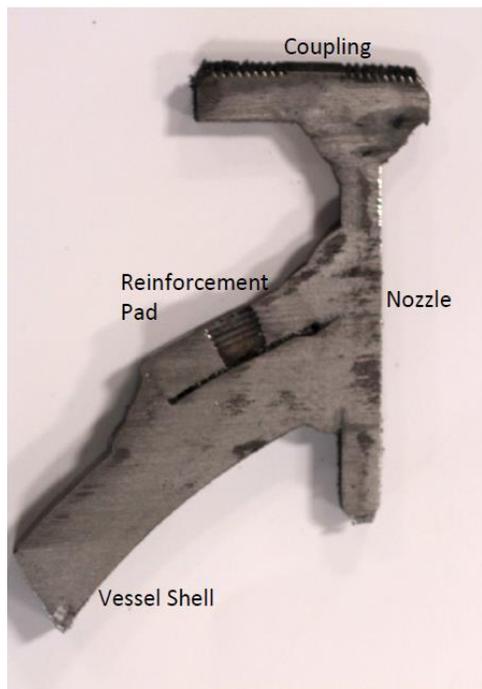


Figure 2: Cross section of a similar bottle, repad, nozzle, and coupling connection

## Measurement setup

A compressor station with similar discharge pulsation bottle coupling and reinforcement pads was located. Vibration levels on the two existing units at this compressor station were compared and the unit with higher vibration levels was selected for more detailed measurements.

For the test, a multi-channel data acquisition system was used to record vibration, dynamic strains and compressor speed. Vibration levels and dynamic strains were measured using ICP velometers and bonded strain gauges. Figure 1 shows a schematic of the vibration transducers. The velometers were mounted on the surface of the cylinder head and vessel body using magnets. The strain gauges were bonded to the nozzle in the longitudinal and circumferential directions adjacent to the fitting. An additional strain gauge was bonded to the nozzle away from the coupling. Figure 2 shows the location of the strain gauges. An overall view of one of the test setups are shown in Figure 3.

The signals from these sensors were then sent through a signal conditioner (and strain gauge amplifier for strain channels) to a multi-channel data acquisition system which was linked to a computer for further processing. The vibration and dynamic strain levels were recorded across a range of operating conditions both with and without wedge supports. Velometer positions were chosen with the goal to enable the measurement vibration modes of the bottle and cylinders.

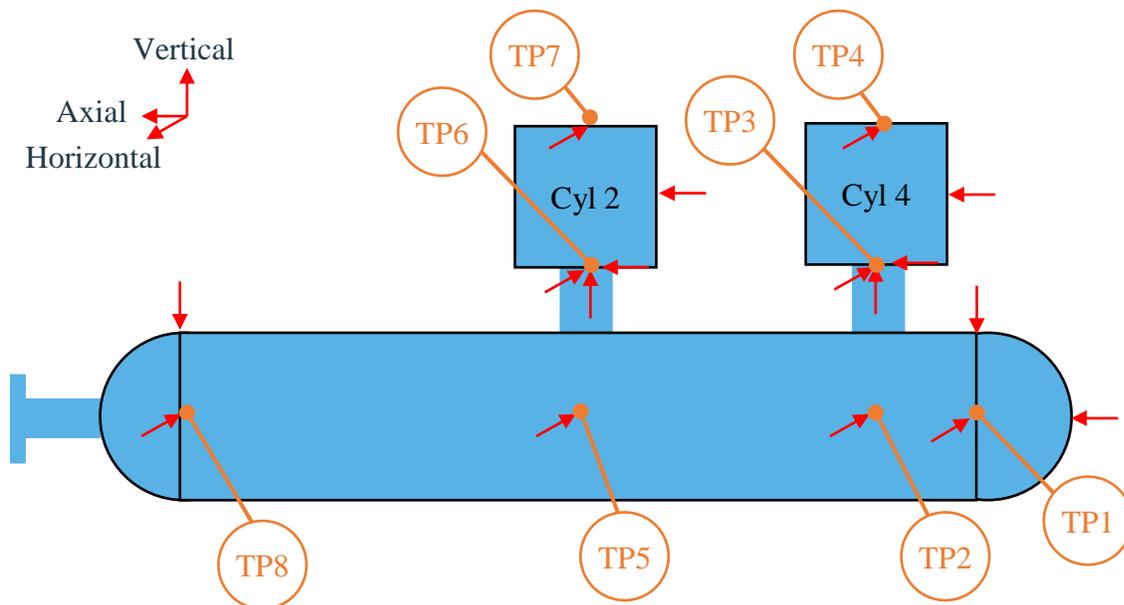


Figure 1: Schematics of velometer locations

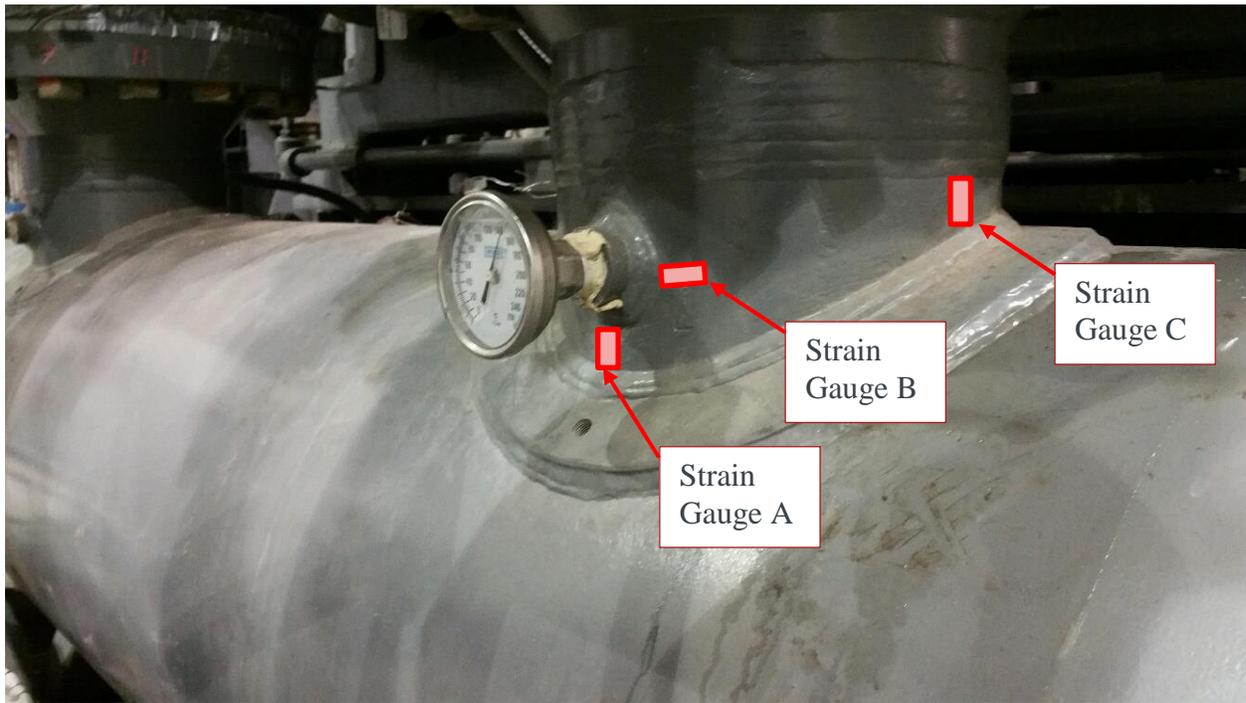


Figure 2: Strain gauge attachment positions

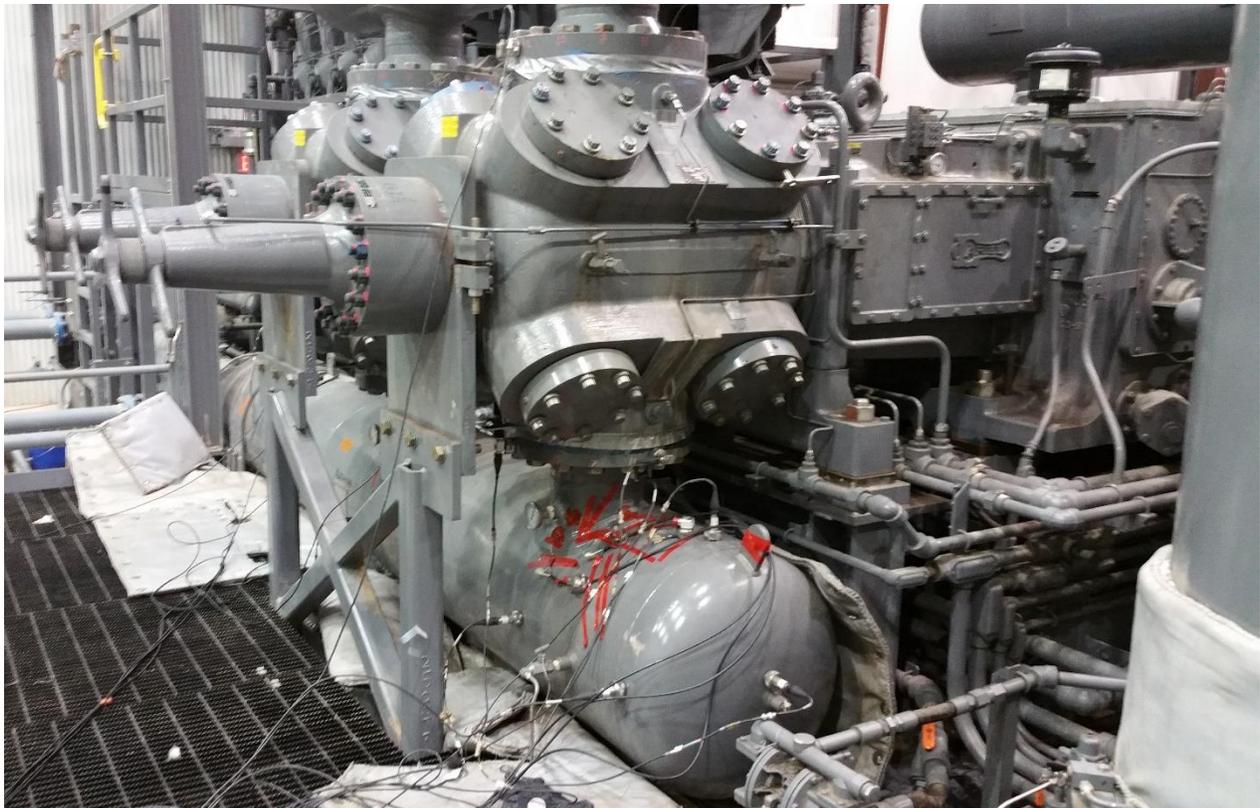


Figure 3: Overall view of one of test setups

## Test results – vibration and dynamic stress

To determine the operating condition related to the highest vibration, vibration and dynamic strain levels were initially evaluated by a speed sweep from 900 rpm to 1000 rpm. The vibration and strains were measured with the discharge bottle wedge supports installed and removed (no restraint on the bottle). Figure 1 shows a waterfall plot of Strain Gauge A at Cyl 4 when the wedge supports were removed. The highest strain happened at about 960 RPM, with an amplitude of  $30\mu\epsilon$  O-P and at 192 Hz which represents 12x compressor speed. The distribution of the peaks over the frequency range shows there was not a single frequency leading to potential failures. High-amplitude strains occur at several orders of run speed. Similar results were observed when the unit was operated with the wedge supports installed.

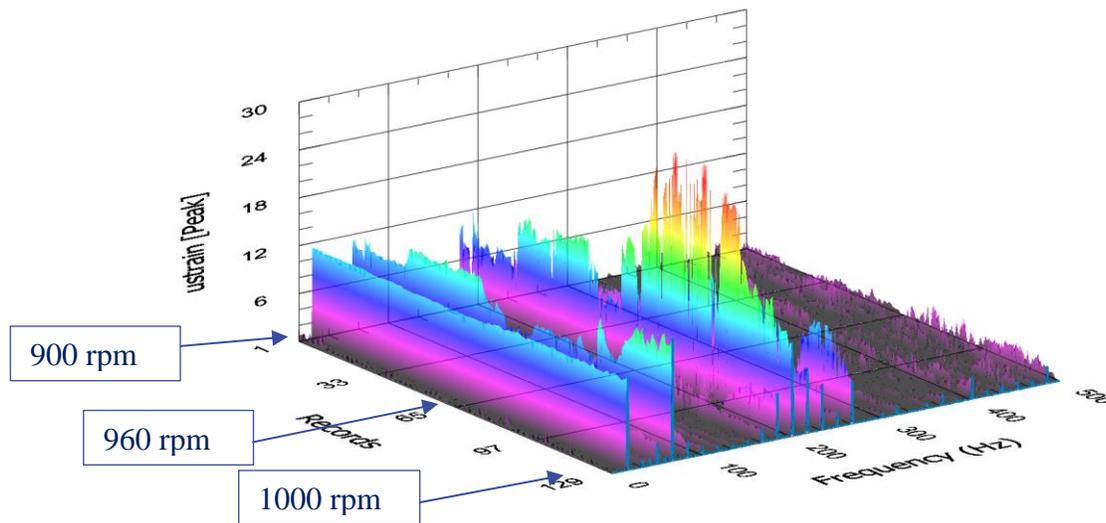
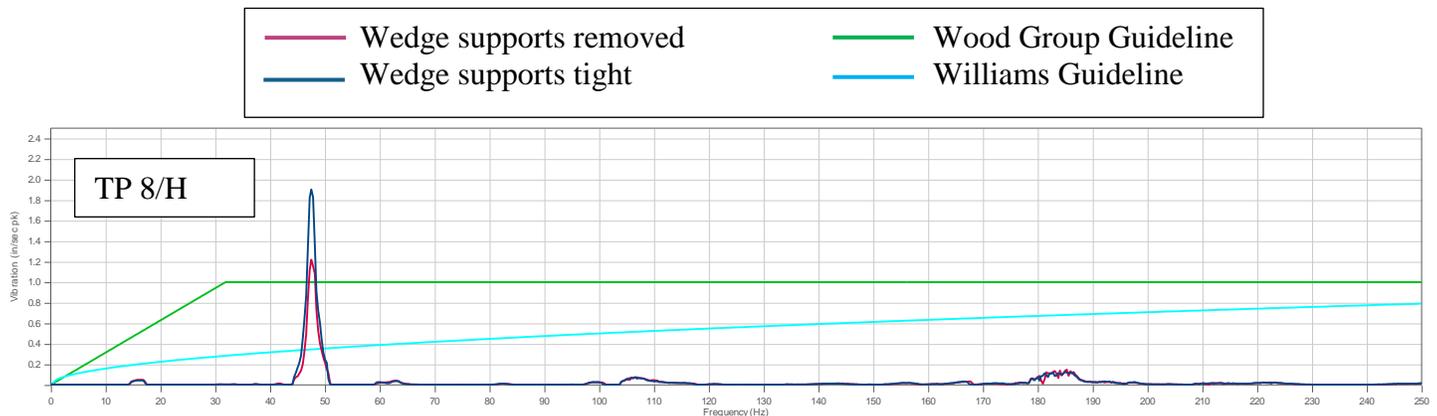


Figure 1: Maximum peak dynamic strain recorded during speed sweep

Figure 2 shows a comparison of vibration levels when the wedges are tight versus when they are removed at two different test points. Note that both vibration measurements at both locations show higher than guideline vibration levels. Vibration at TP8, which is at the overhung end of the bottle (See Figure 1) occurs primarily at about 50 Hz. However, for TP1, which is closer to Cyl 4 Nozzle (see Figure 1), the vibrations primarily occurred at about 190 Hz. Although some differences can be observed at lower frequencies between two different wedge adjustments, there are no significant differences at the higher frequencies. As shown in Figure 1, the highest strain peaks were observed at higher frequencies. This indicates that the mode shapes resulting in the peak dynamic strains were not affected by the global restraint of the wedges and supports.



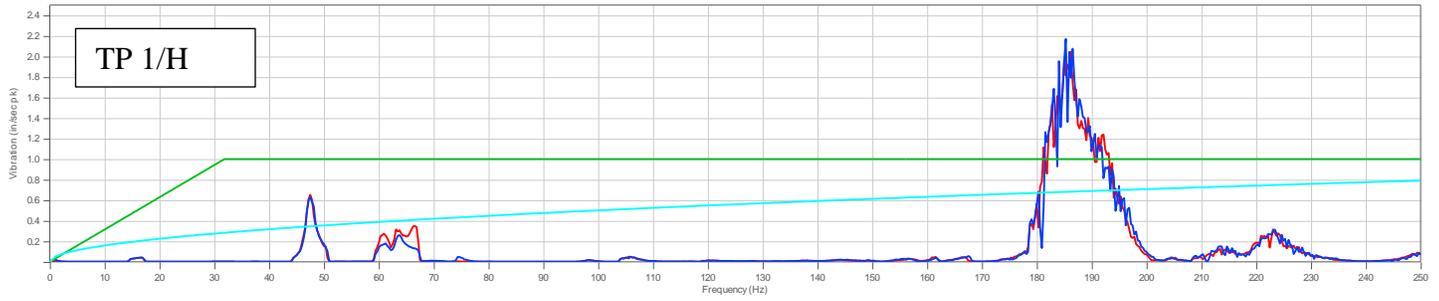


Figure 2: Peak hold vibration plots at TP8/H (top) and TP1/H (bottom), with wedge supports tight and removed

Figure 3 shows the measured strain at Strain Gauge A, Strain Gauge B, Strain Gauge C. Figure 4 shows the vibration at different test points for the same test. The peak strain values correlated well with higher levels of overall vibration (approximately 2.7 ips pk) on channel 4 which was measured horizontally on TP 1 (see blue curve on top plot of Figure 3). This peak vibration coincided with the peak strain at 192 Hz. Since vibrations were measured at multiple frequencies, more accurate modeling of the pulsation bottle was necessary in order to capture a response at such a high frequency.

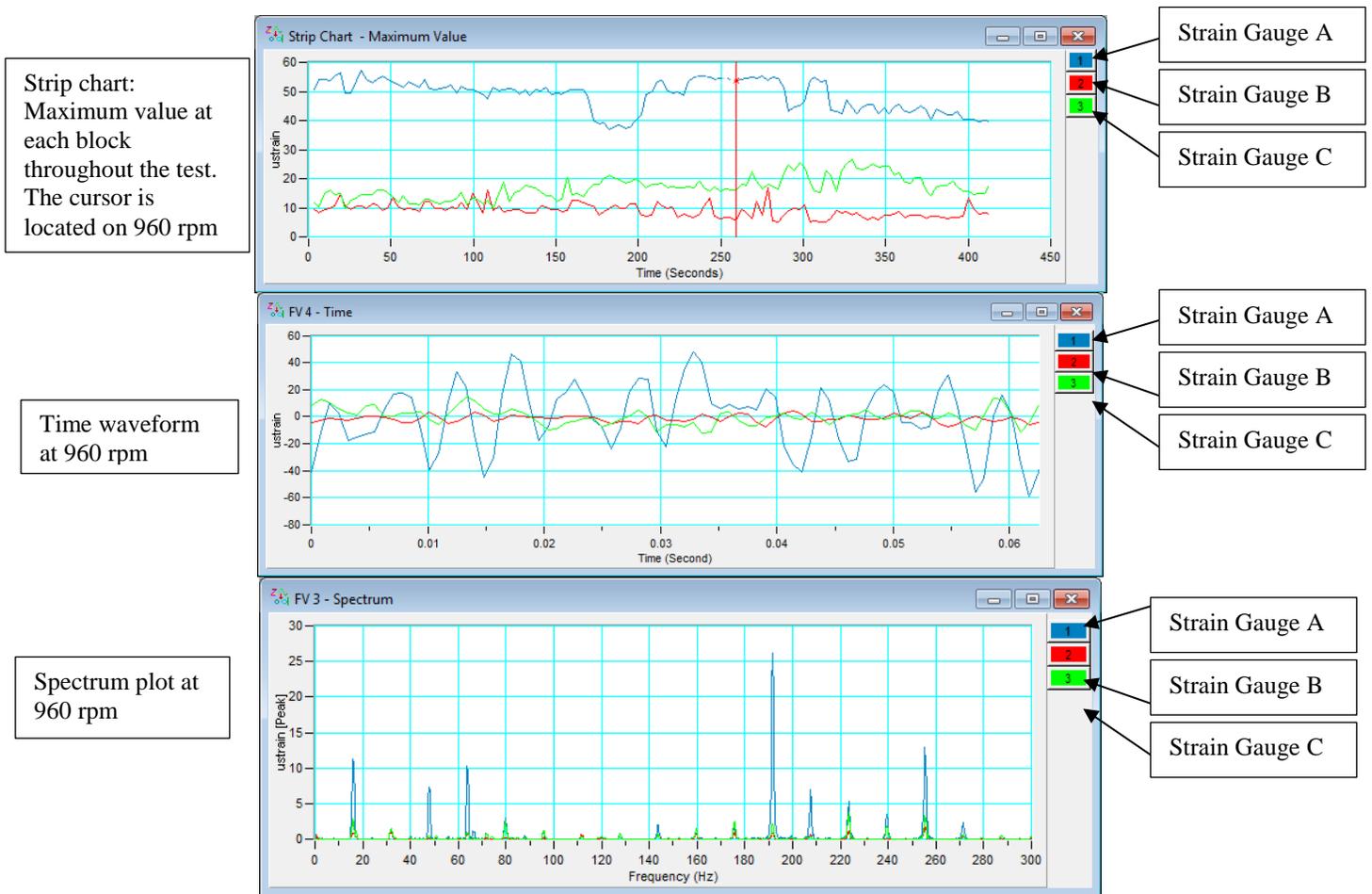


Figure 3: Top: strip chart throughout the test, middle: time waveform at 960 rpm, bottom: spectrum plot at 960 rpm

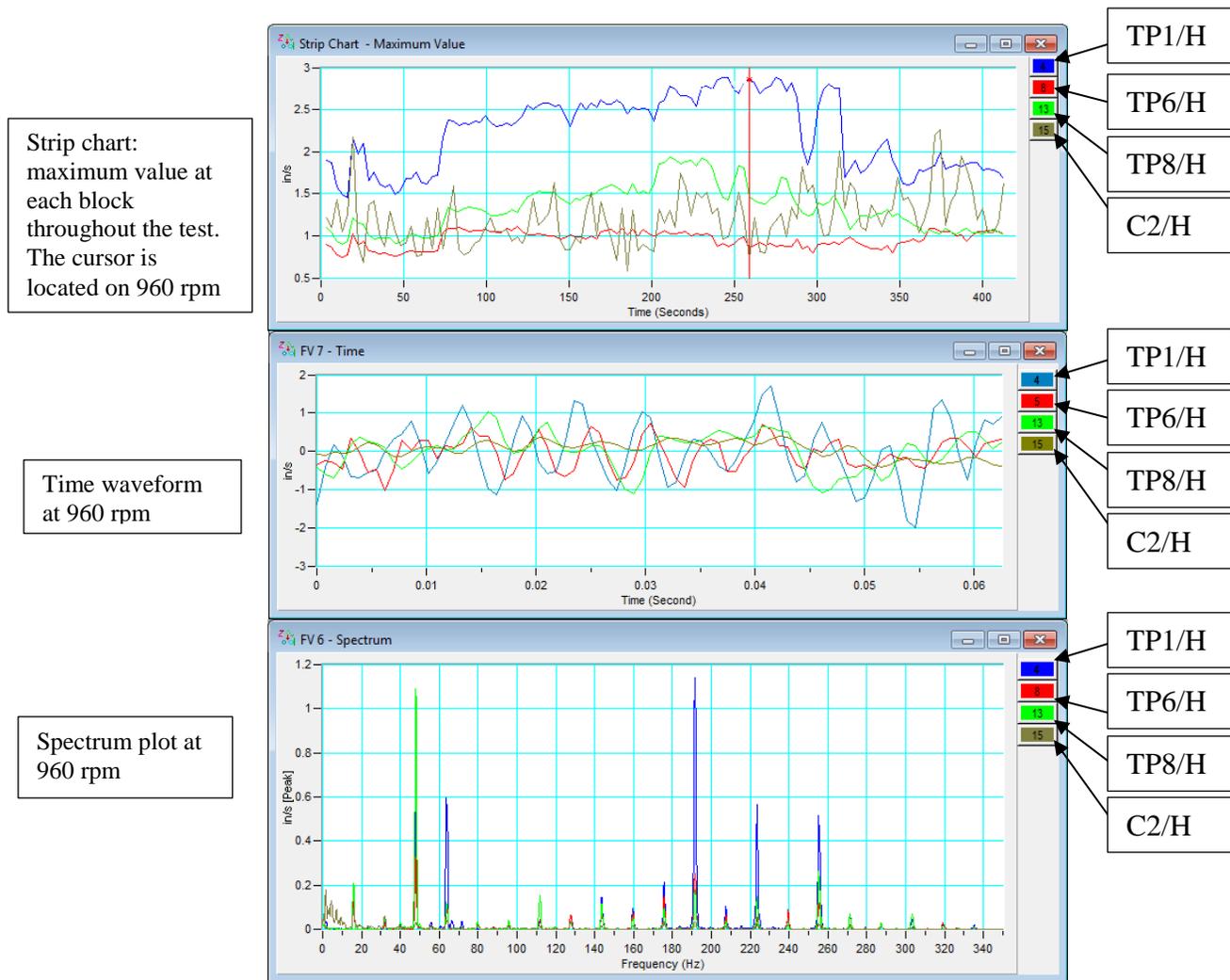


Figure 4: Top: maximum vibration throughout the test, middle: time waveform at 960 rpm, bottom: spectrum plot at 960 rpm

The factors that contribute to the magnitude of the maximum allowable dynamic stress include static and residual stresses, manufacturing defects in the welds and adjacent material, surface defects and the geometry of the connection which results in stress raisers. There are various codes which specify recommended limits for the dynamic strain based on geometry criteria and probability of failure. In comparison with Type 10.3 in Table 10 of [1], considering the connection as category F, the allowable stress range for a 2.3% probability of failure is 40 N/mm<sup>2</sup> peak to peak (200  $\mu\epsilon$  pk-pk). In accordance with the guidance in [2], for measured strains, this is recommended to be reduced to half (100  $\mu\epsilon$  peak to peak) of the above limit to account for the consequence of failure in hydrocarbon and safety critical piping. So, based on this criteria, the measured strain does not exceed guideline level. However, in the case of the bottle nozzle connections, the proximity of the fitting connection weld to the nozzle repad weld was in some instances below common industry guidelines. In this instance reference was taken from [3] where minimum of 4X wall thickness is recommended or for non-pressure parts, the smaller or 2X wall thickness or 40 mm (1.6 in). Therefore, the straight comparison of the published fatigue limit to the measured limit was not deemed to be applicable in this instance without further analysis.

Since the unit could not be shut down due to operation limitations, the impact tests were performed while the unit was running. The results of the impact tests with the unit running showed very poor coherence and was therefore inconclusive.

## FEA model

### Model description

As shown in Figure 1, a finite element model (FEM) of the cylinders and the pulsation bottles was created. For simplicity and due to their high stiffness, the cylinders were modeled as rigid rectangular boxes. They are shown in green and brown in Figure 1. The nozzles, the re-pad and the bottle were meshed with a fairly dense mesh. A closer view of more densely meshed area is shown in Figure 2. The model had been designed to accurately capture the stress concentration that will occur in that area of the nozzle. The mesh used to discretize the bottle is also fine enough to predict the global vibration modes as well as the local shell modes that might occur. This is an important consideration as it was suspected that such shell modes will occur at the frequencies of interest for this study which are around 12X or 192 Hz. Finally, the location of the baffles are indicated by the two blue rectangles shown in Figure 1. These zones will be stiffer and will have a significant effect on the shell modes of interest for this study.

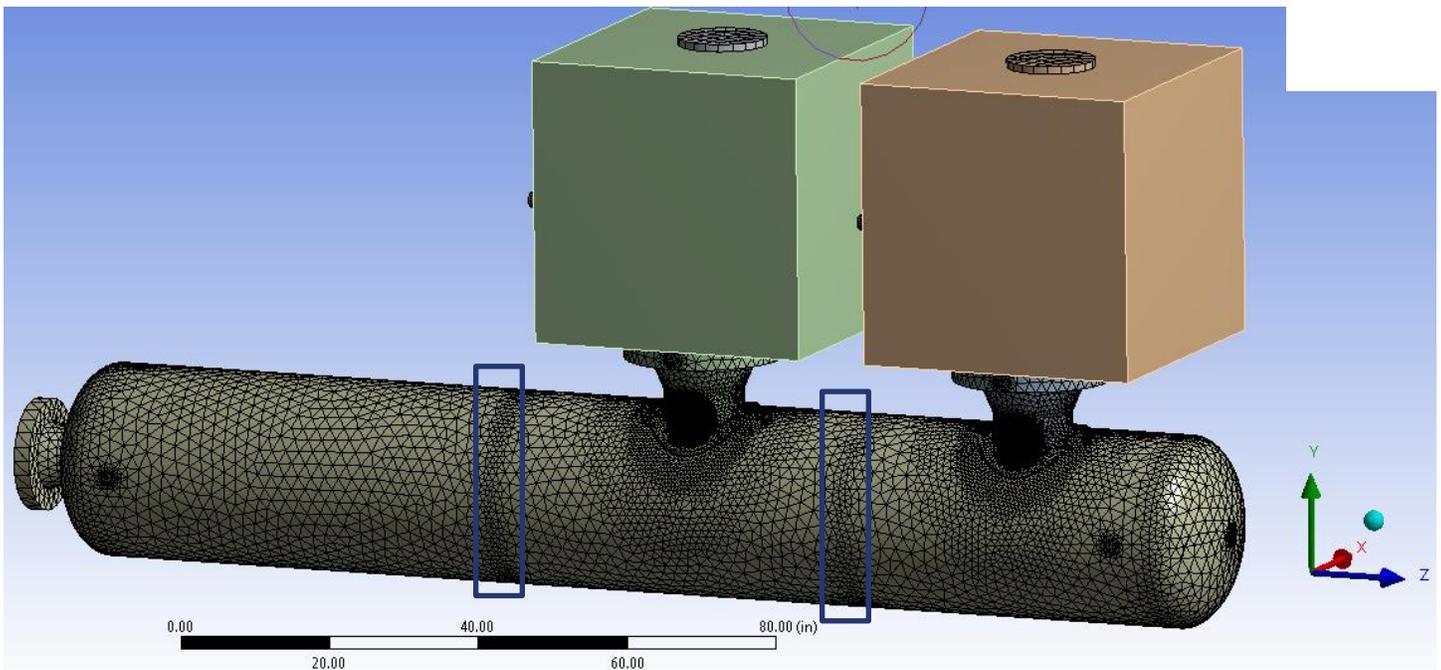


Figure 1: Discharge pulsation bottle finite element model – global view

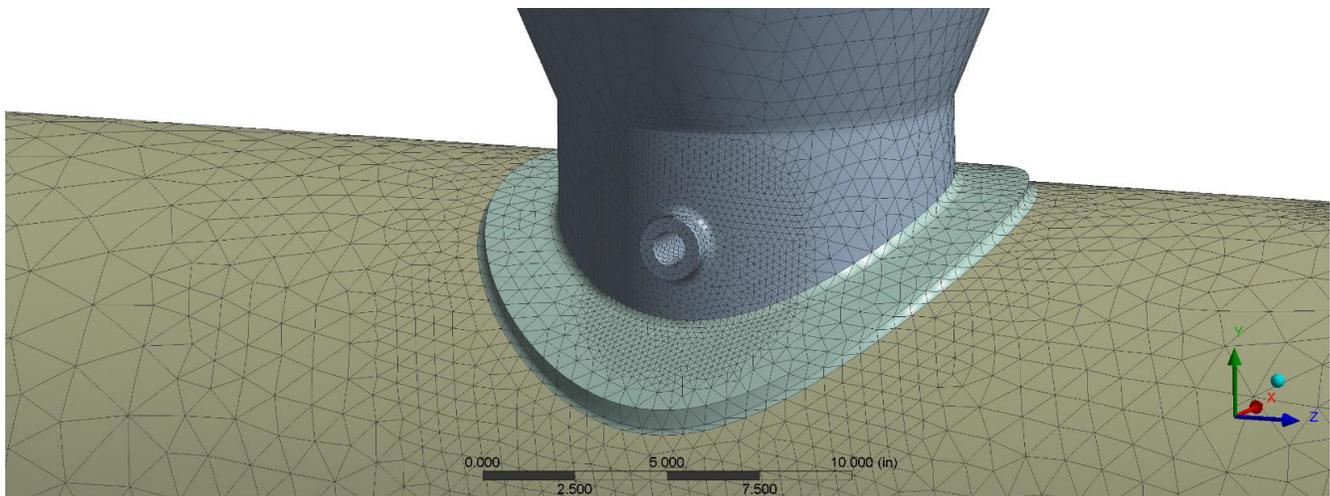


Figure 2: Closer view of the zone of interest in the mesh

The measured velocities at every test point were applied to the model as time-domain boundary conditions, as shown in Figure 3. A transient analysis of the FEM was then executed. The velocity boundary conditions which were applied to the model are similar to the velocity waveforms shown in the middle image in Figure 4. The time history analysis was selected for this project since there were multiple dominant frequencies in velocity and strain signals.

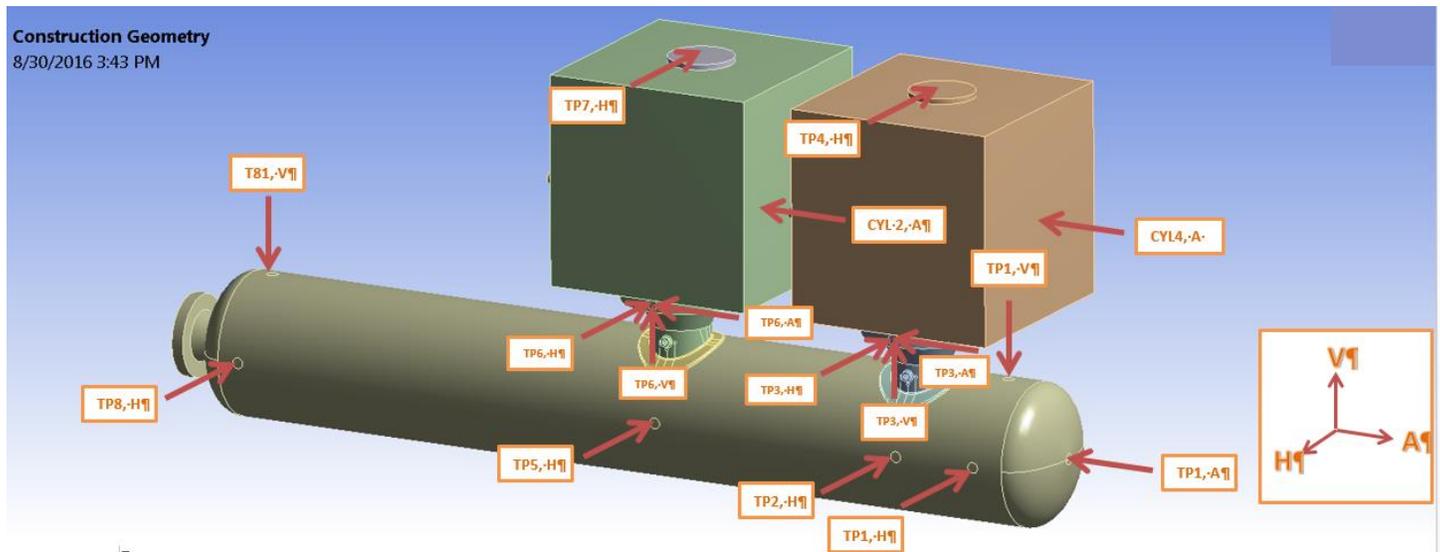


Figure 3: Measured time domain velocities applied to FEM

## Validation

The calculation of the mechanical natural frequencies was done without applying any boundary conditions. This corresponds to a so-called free-free condition that will not result in accurate global vibration modes of the bottle and cylinder assembly. However, it is thought that the shell modes of interest in this study will be predicted accurately since the field measurements confirmed the wedge adjustments will not have an effect on the higher frequency modes. The first shell mode was predicted at 204 Hz. The X component of the mode is shown in Figure 1. This corresponds to a shell mode that occurs in an area of the bottle located directly below Cyl 4 nozzle, as shown in Figure 1.

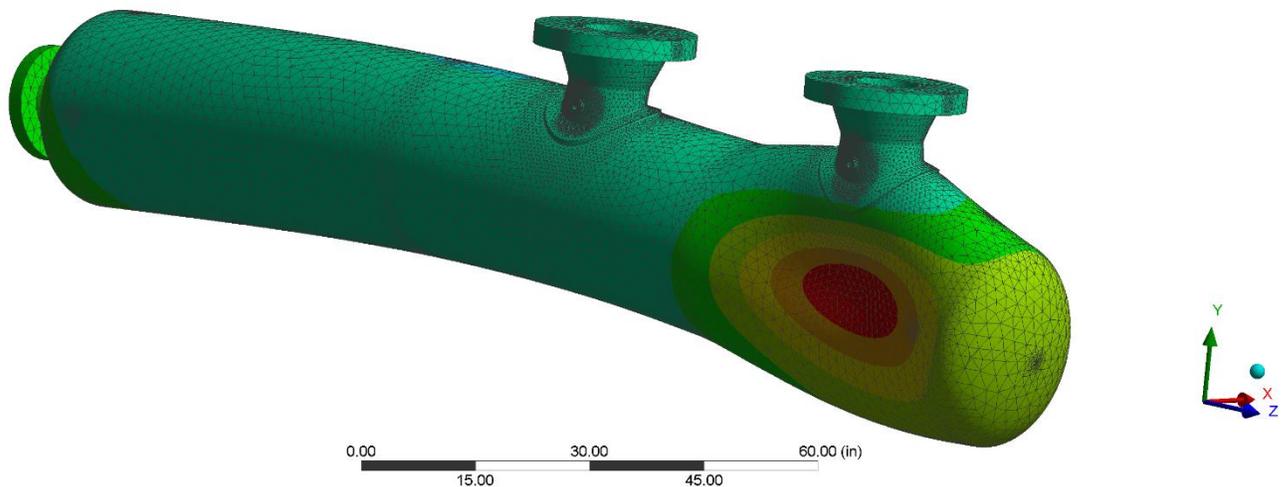


Figure 1: First predicted shell mode – 204 hz

The measured and predicted strain levels are shown in Figure 2 for Cyl 4 and Cyl 2 nozzles. The results shown have been obtained by averaging the normal vertical strain over the area from where the strain gauge was installed during the field measurements ( $\frac{1}{2}$ " diameter circle between the coupling and the repad as shown in Figure 3). The experimental values are the output of measurements at Strain Gauge A, as shown in Figure 2.

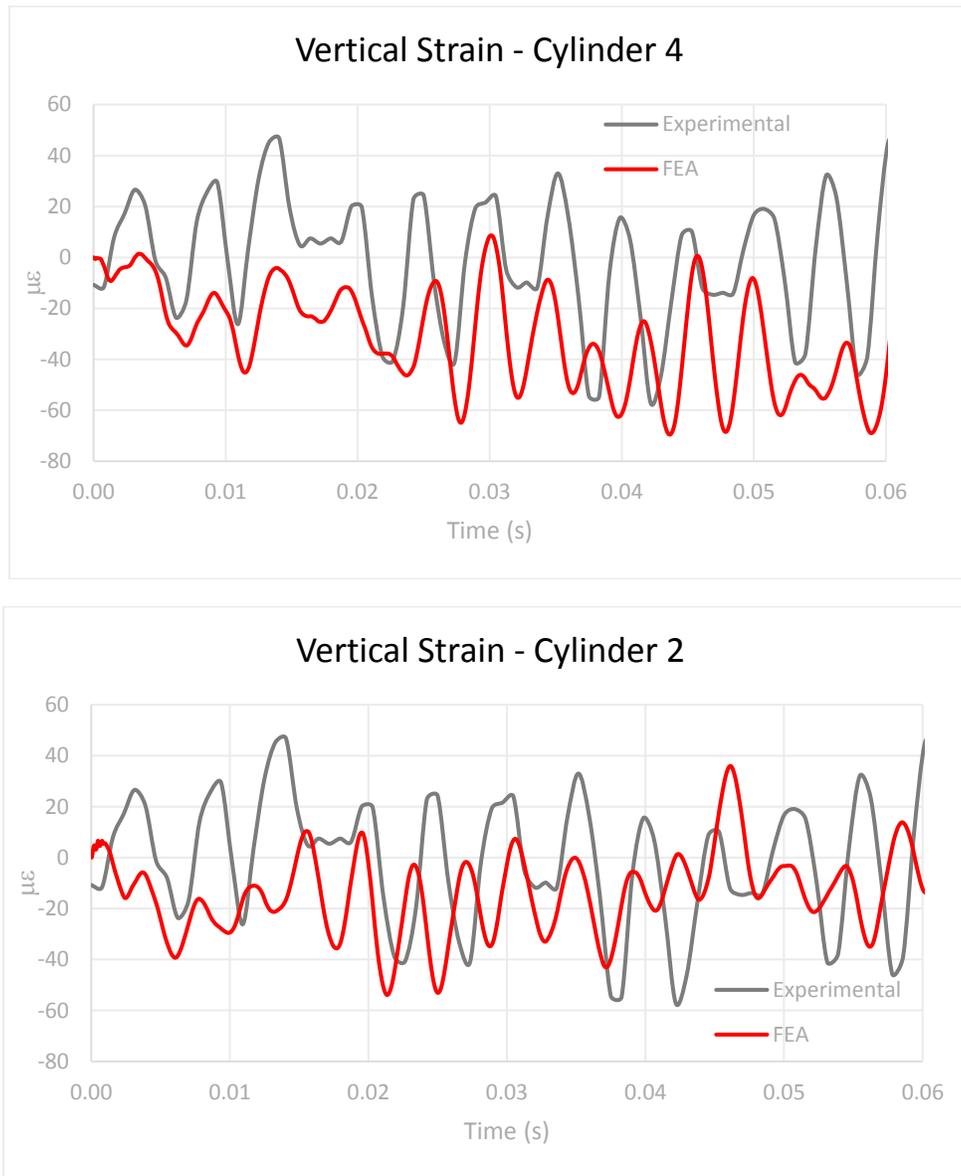


Figure 2: Transient strain (gauge a on Figure 2) - FEA vs field measurements

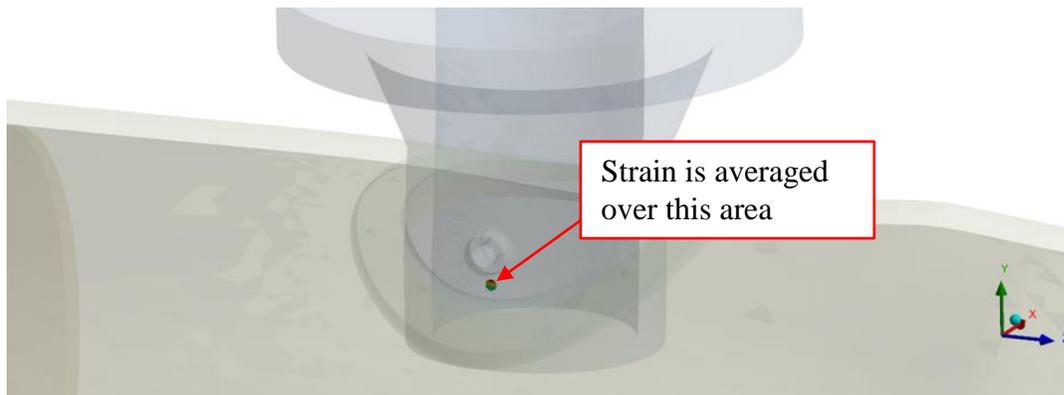


Figure 3: Zone where the strain values have been averaged

These results indicate similar patterns between the measured and predicted strain values but are not a perfect match. However, similar maximum and minimum values are obtained from both measured and predicted strain levels. In addition, based on the field measurements, an operating deflection shape (ODS) model of the unit was developed. The results of the model were compared with the results of FEA analysis and the ODS was confirmed. A snapshot of the results is shown in Figure 4.

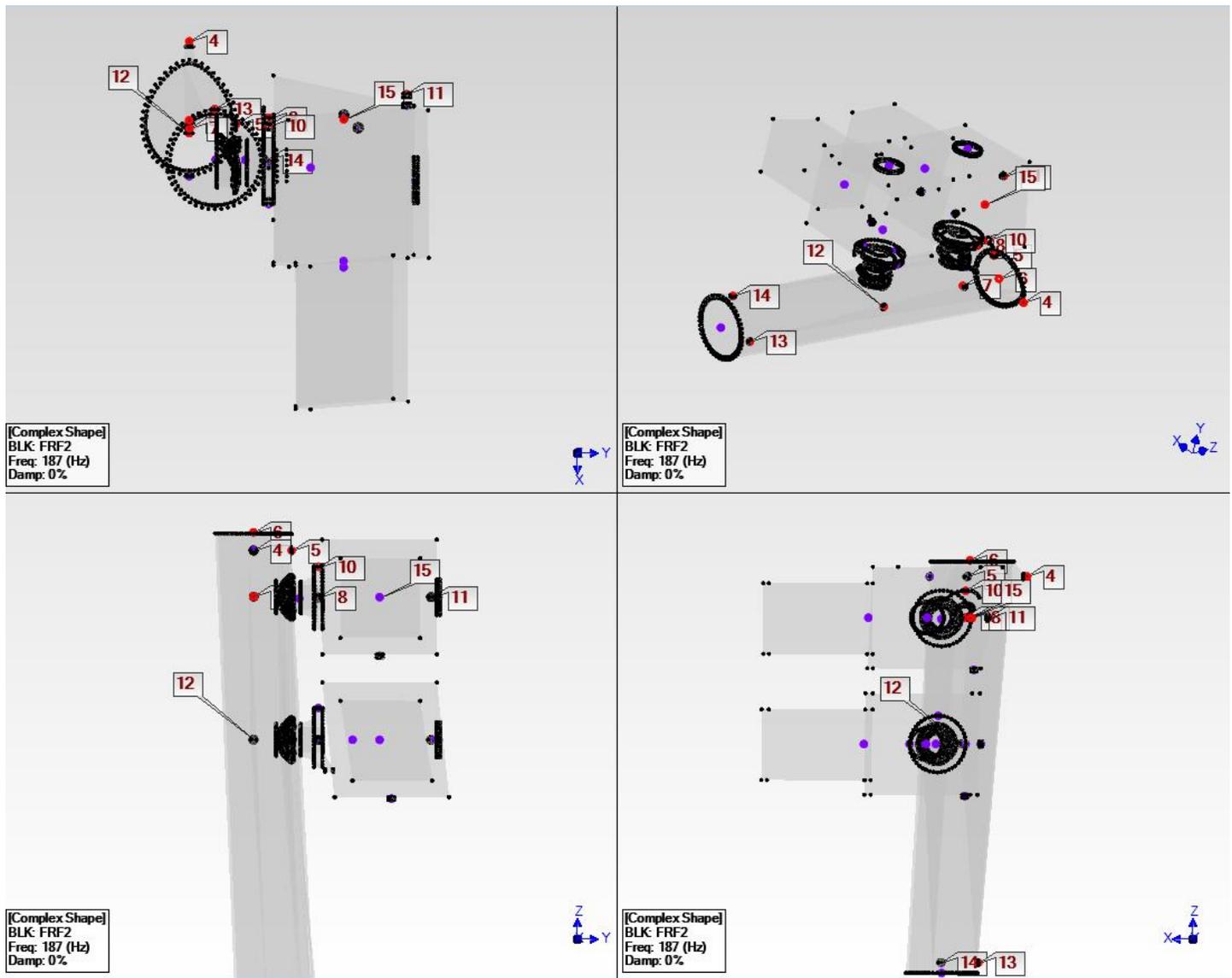


Figure 4: Measured operating deflected shape (ODS)

## Results

The equivalent maximum stress at the connection between the nozzles and the bottle is shown in Figure 1. As expected, a stress concentration occurred at the weld between the nozzle and the repad. The maximum equivalent stress value is around 6,000 psi 0-pk ( $41.3 \text{ N/mm}^2$ ) in that zone.

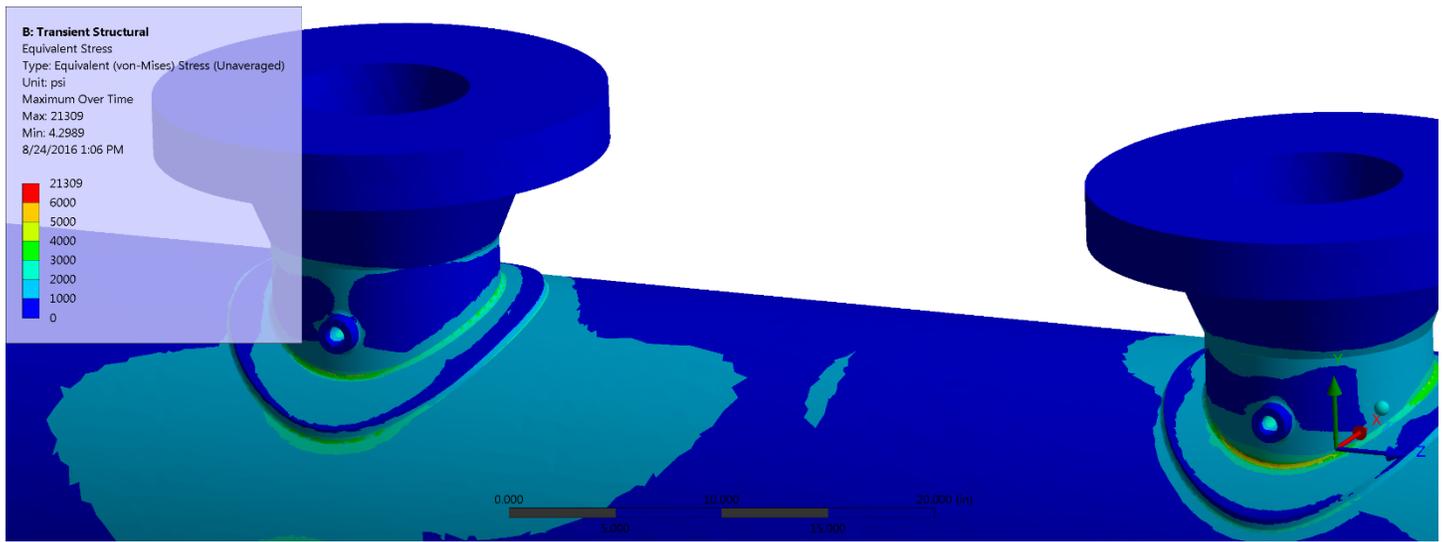


Figure 1: Near-field Von Mises equivalent stress predictions – maximum over time

A sensitivity study has been conducted to assess the variation in the stress values that result from moving the coupling up and down in the vertical direction (toward the repad and away from the repad). In the nominal design that has been studied, the coupling is located approximately 2¼ inch from the bottom of the repad weld, which is identical to the design of the as-built discharge pulsation bottle. Two additional scenarios have been studied. In one case, the coupling location moved down by 1 inch and in the second scenario it moved up by the same distance with respect to the nominal design. A detailed view of the equivalent stress distribution is shown in Figure 2. The three pictures in Figure 2 indicate that the maximum stress predicted at the weld is not affected by the positions of the coupling; however, there is a significant decrease in the average stress in the zone between the re-pad weld line and the coupling. The maximum stress values are all between 6,600 and 6,700 psi.

When comparing a predicted FEA stress against an appropriate allowable, the equivalent maximum stress can be artificially high due to a singularity such as a V-notch. It is more appropriate to use either a nominal stress or a hot spot stress when comparing against an S-N curve from a design code. The hot spot stress is the maximum stress at the weld toe considering the stress concentration effect of the discontinuity and can be obtained through extrapolation from the surface stresses [4]. As the hot spot stress is used, the weld class from table 10.3 of [1] can now be changed to a class D fatigue curve (7687 psi / 53 N/mm<sup>2</sup> peak-peak). See Table 1 below for final results comparison against this allowable. The hot spot stress value based on the Von Mises equivalent stress and a downward line on the nozzle (see Figure 3) has been calculated and is reported in Table 1. It is seen that the maximum hot spot stress value is not significantly affected by the vertical location of the coupling. The close proximity of the coupling to the repad in the lower position prevents the hot spot stress from being determined. In terms of field measurements, if the coupling was this close to the repad, this proximity would also prevent strain gauges from being affixed in gauge position A.

Table 1 - Connection position – near-field Von Mises equivalent stress predictions

| Connection position | Von Mises equivalent stress (psi peak-peak) | Hot spot stress (psi Peak-Peak) | Allowable Dynamic stress level (psi peak-peak) If no static stress exists | % of guideline |
|---------------------|---|---------------------------------|---|----------------|
| Lower               | 13,280                                      | NA                              | 7687  | NA             |
| Mid (as built)      | 13,236                                      | 4,146                           | 7687  | 53%            |
| High                | 12,934                                      | 5,494                           | 7687  | 71%            |

Although above analysis showed the dynamic stress levels are within guidelines, it is very important to note the following two points:

- These guidelines do not consider static stress in the area. If high tensile static stresses are present, according to the Goodman relation, fatigue limits can be significantly reduced. In other words, in cases where a static (mean) stress exists in the area, the allowable dynamic stress levels can be significantly lower. The static (mean) stress can be introduced due to many reasons including improper welding procedure, lack of or poor performance of post-weld heat treatment, and thermal expansion. For the situation where the coupling is located closer to the repad, the potential for high tensile static stresses increases.

- For the coupling in the lower position case, there is a potential for the presence of weld defects and notches if weld toes are in contact. This can increase local stresses significantly. This situation is not deemed acceptable regardless of the measured or predicted stresses.

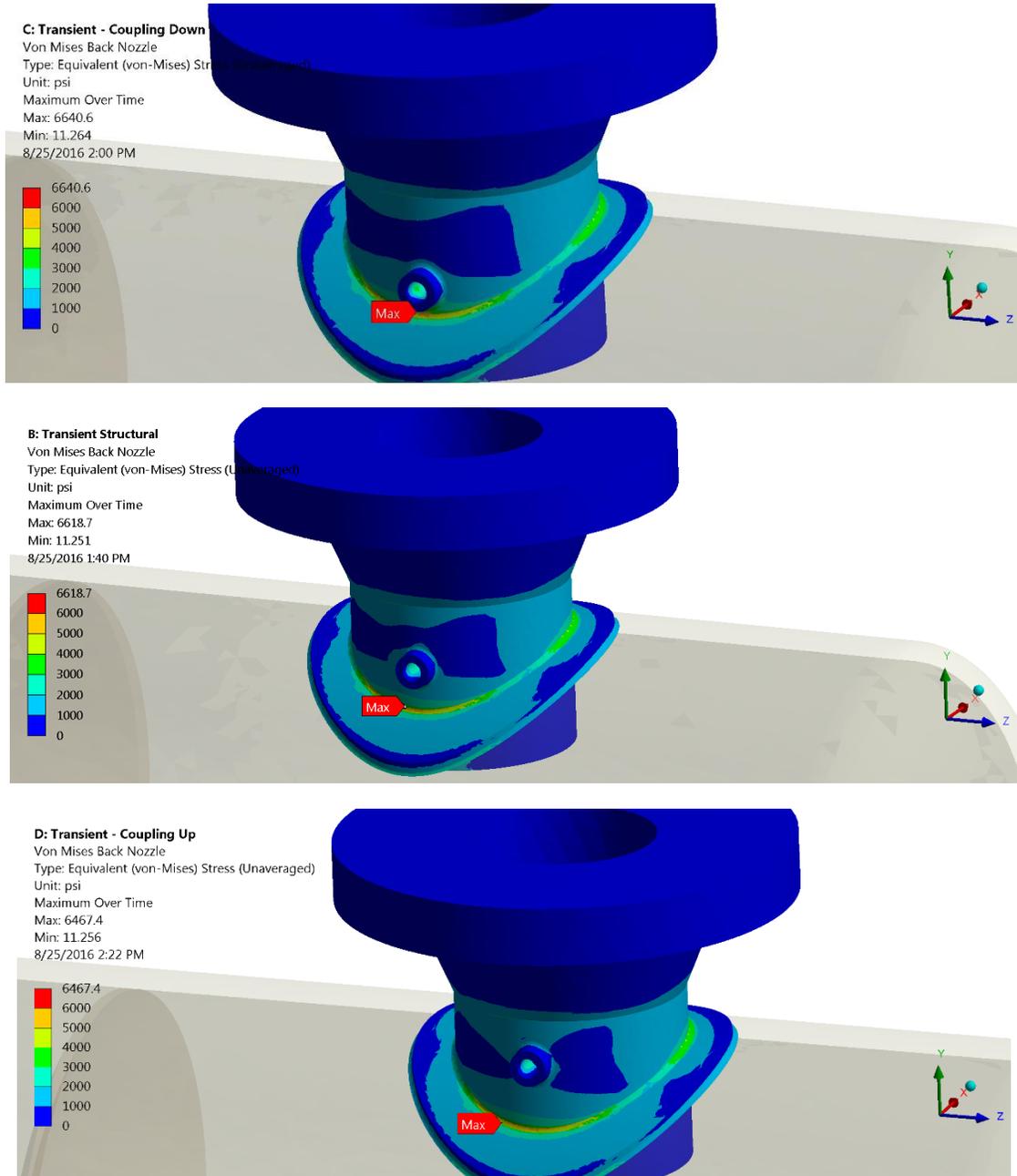


Figure 2: Stress results corresponding to different locations of coupling

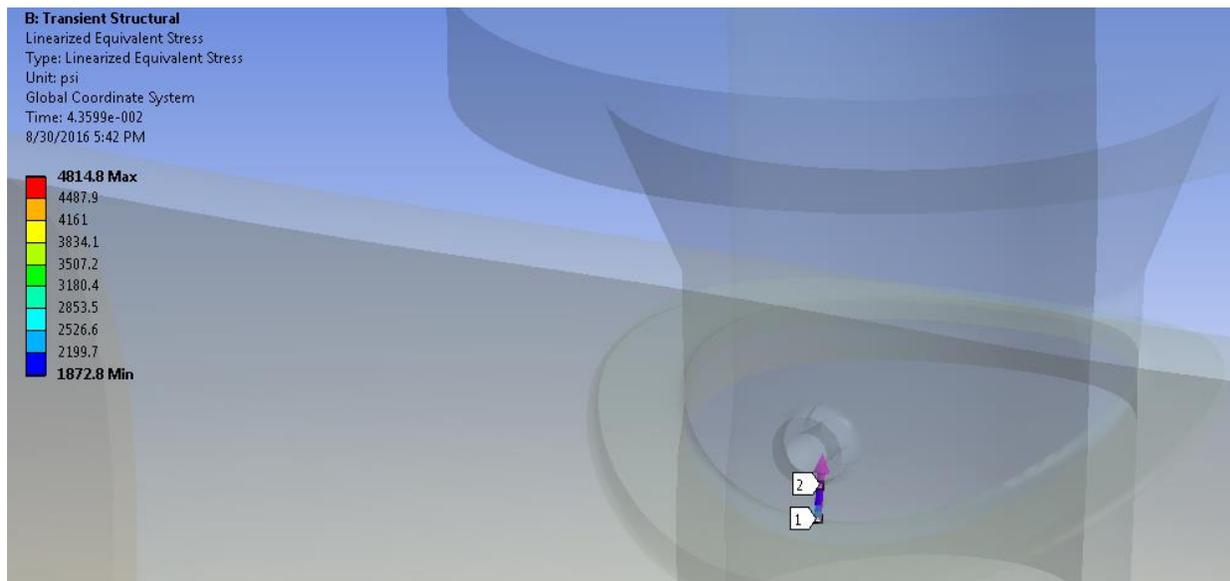


Figure 3: Line used to predict the hot spot stress value

## Conclusion

- Assuming that no static (mean) stress is induced in the nozzle intersection area, our analysis determined that the bottle in this test is not at risk of fatigue failure at the process conditions experienced. The levels of dynamic stress were calculated to be below the constant amplitude fatigue limit using the hot spot method.
- There was a minimal increase in the dynamic strains, for examples, where the coupling was located closer to the repad, however, other non-quantifiable factors which decrease fatigue life such as residual stresses, weld defects and v-notches may be present for these cases and these are therefore considered at risk of failure.
- This project has demonstrated the efficiency of FEA in predicting strain and stress levels in complex geometries. With this validated model, it is now possible to assess other conditions, or similar but slightly different configurations for fatigue failure risk.

## Project recommendations

- For the remaining similar units:
  - For bottles with the couplings welded closer than 4X wall thickness to the repad: the coupling should be removed. Meanwhile, periodic NDT of the coupling and repad weld should be performed to ensure any potential cracks are detected as soon as possible.
  - For the bottles with couplings welded farther than 4X wall thickness away from the repad: the bottles should be heat-weld treated to make sure there is no static stress in the nozzle intersection area. If no static stress exists in that area and the vibration levels are lower than what was measured at this test, the operation of the units is safe at all speeds.
  - For the bottles with couplings welded farther than 4X wall thickness away from the repad: If the vibrations are higher than what was measured during this study, or the ODS is different, the validated FEA model in combination with vibration measurements should be used to assess the stress levels using hot spot method.
- If new bottles are manufactured to replace the cracked bottles:
  - Remove the coupling on the discharge nozzle if feasible.
  - If it is not feasible to remove the coupling, make sure the coupling is welded at least 4X wall thickness away from the repad weld.
  - Review the weld procedure to ensure the possibility of induced static stress is minimized.
  - Performing a post-weld heat treatment is very important in releasing the static stress which has been induced during welding process.

- The investigation indicated that the bottle support arrangement was not a crucial factor in causing shell mode vibrations of the bottle. Therefore, improvement of support to reduce vibration levels on bottles may not be an appropriate solution in similar cases.

## General recommendations

- Although strain measurements are used as a common tool to assess the fatigue risk in the industry, they are often more time-consuming and requires in-depth planning to execute. This work confirmed that with an appropriate consideration of mesh size, boundary conditions and applied displacements, use of field-based vibration measurements in combination with finite element modelling can provide an accurate indication of dynamic stresses even when complex geometries and higher order mode shapes are present.
- Based on this issue there were modifications made to standards to improve the design of the compressor pulsation bottle. The minimum distance between welds should be no less than 4X wall thickness.
- If spacing between welds can't be achieved, a post-weld heat treatment of the location is a potential improvement. Alternatively, if a coupling exists on the compressor discharge flange it may not be necessary to design one into the pulsation bottle nozzle. Raised flange long-weld neck nozzles may be appropriate if the additional height is required to accommodate a coupling.
- Where higher frequency shell modes are believed to be excited on a vessel, a large number of vibration measurements, including phase results, are necessary to accurately determine the displacement of the vessel. Without extensive vibration measurements, use of dynamic strain measurements can be a practical way to obtain an indication of fatigue limit.
- Where welded connections are in close proximity or higher order vibration modes are present, caution needs to be taken in directly applying vibration acceptance criteria. Either strain measurements should be carried out, or sufficient vibration measurements to enable determination of operating deflective shape, combined with finite element analyses, should be carried out.
- Applicable codes exist to determine the acceptance of dynamic stress levels, however the correct stress measurement should be used. Hot spot stresses can be used to reduce some conservatism, however it is important that the measured stress is compared against the equivalent stress in the correct location and orientation on any FEA model.
- Where shell modes on vessels are excited and stress raisers exist, fatigue failures can occur on lightweight small-bore connections which would not have been identified as having poor vibration tolerant design.
- High-frequency shell modes are not visible to the human eye and may be missed in routine visual inspections. This highlights the requirement for vibration measurements not just on the machine but on the associated piping and vessels.
- Collaboration between field and design analysts is critical in completing such projects. Sufficient and correctly placed vibration measurement locations are critical in determining shell mode shapes.

## Future work

Investigation of the effect of changing bottle geometries (diameter, thickness, nozzle arrangement etc) is recommended. It would be beneficial to develop a general vibration acceptance criteria for shell mode vibration of pulsation bottles considering the higher frequency shell modes.

## References

- [1] BS7608: 2014: Guide to the fatigue design and assessment of steel products, British Standards Institution, UK.
- [2] Guidelines for the avoidance of vibration induced fatigue failure, 2<sup>nd</sup> Edition, 2008. Energy Institute, UK.
- [3] Section 11, BS2633: 1987 Specification for Class 1 Arc Welding of Ferritic Steel Pipework for Carrying Fluids, British Standards Institution, UK.
- [4] Section 18, BS EN 13445-3:2009, Unfired Pressure Vessels- Part 3: Design, British Standards Institution, UK