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Case Study of Solving Screw Compressor Package Vibrations

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Abstract

Screw compressors are used in industrial applications requiring large volumes of gas at high compression ratios. This rotary-type positive-displacement machine results in a more steady flow than in reciprocating compressors, however, risks of vibration and fatigue failure due to the compression process remain.

This paper presents a case study involving three nominally identical oil-flood screw compressor packages in a cooling system for a de-ethanizer facility. High vibration on one of the packages led to a failure on a small-bore connection attached to the oil separator vessel. The case study describes:

- The initial assessment of the vibration after the failure
- Shell-mode vibration measurements of the vessel
- Computer simulations of the pulsation and mechanical systems
- Optimized solutions to reduce the pulsation-induced forces and vibration response
- Implementation of recommendations
- Resolution of the problem

Purchasers and operators of screw compressors can apply the recommendations presented in this case study to minimize the risk of vibration-related failures on new or existing packages.

Introduction

Vibration testing was conducted on three screw compressors, units 301, 311, 321, at the Willams Oak Grove facility near Moundsville, West Virginia. These units are used for refrigeration in the De-Ethanizer section of the plant and are crucial to the process. Although the process can be performed with only one unit operating at a time, operations ran two of the three units, each unit partially loaded. A photo of unit 301 is shown in Figure 1.

High vibration and failure of the equalizer line between the discharge line and the oil separator vessel initiated the investigation. A photo of this pipe is shown in Figure 2. A crack was reported on top of the equalizer line nozzle on the discharge line of unit 301. After the crack occurred on unit 301, daily non-destructive tests (NDT) were instituted by Williams on each of the units. NDT showed a potential crack on top of the equalizer nozzle on the vessel for unit 321.

Figure 1. Unit 301 overview

Figure 2. Equalizer line between the discharge line and oil separator vessel

Background

Initial vibration testing was done to evaluate vibration on the equalizer line. These units are wet screw compressors which are driven by a fixed-speed motor operating at 3579 rpm. The screw compressors have a four-lobe male rotor. Vibration measurements showed high vibrations on the equalizer line nozzle of the vessel at pocket pass frequency which is four times compressor speed (3579 x 4=14,316 cpm=239 Hz). The dominant frequency of vibration and pressure pulsation is referred to as ‘pocket pass frequency’ or ‘lobe pass frequency’. This is because the number of gas pockets between the main and gate rotor or the number of lobes on the main rotor that are exposed to the piping system during one shaft revolution define the frequency of these pulsations.

Along with traditional piping vibration measurements, the operating deflection shapes (ODS) of the equalizer line were measured and created. The ODSs suggested more detailed testing on the shell of the oil separator vessel should be performed. Vibration of the vessel shell, although below guideline levels, was observed and measured at pocket pass frequency.

Field testing and analysis

In order to take ODS measurements of the oil separator, the vessel was divided into a measurement grid or mesh, as shown in Figure 3. Vibrations were measured on all locations shown in Figure 3. The acquired vibration data was visualized with the computer program ME Scope. One of the resultant ODSs for unit 311 is shown in Figure 4 and Figure 5. Internal and external attachments to the vessel made the measurements slightly more complicated than would be expected for a simple cylindrical shell.

A simplified model of the vessel was evaluated using finite element analysis (FEA) in order to assess the mechanical mode shapes of the vessel. The model did not include internal and external attachments, the liquid level or gas pressure inside the vessel. Despite these differences, the FEA showed similar mechanical natural frequencies and mode shapes near the frequency of the measured high vibration on the vessel and equalizer line, 239 Hz. One of the FEA mode shapes calculated at 242 Hz is shown in Figure 6.

In addition, empirical calculations were done to determine the acoustic transverse waves inside the oil separator vessel. The calculated acoustic transverse wave was found to be near the frequency of high vibration and pocket pass frequency. Assuming the speed of sound of 840 ft/sec for the discharge gas (propane), the acoustic transverse mode for the vessel could be between 230 Hz for four radial acoustic nodes and 270 Hz for five radial acoustical nodes (with one axial node). This calculated acoustical natural frequency matched well with both FEA mechanical natural frequencies and field measurements of high vibration. Based on the measurements and calculations, it was concluded that high shell-wall vibrations were due in part to excitation of the shell at the shell natural frequency by resonant acoustic transverse waves.

Figure 3. Location of vibration measurements on the shell of the oil separator

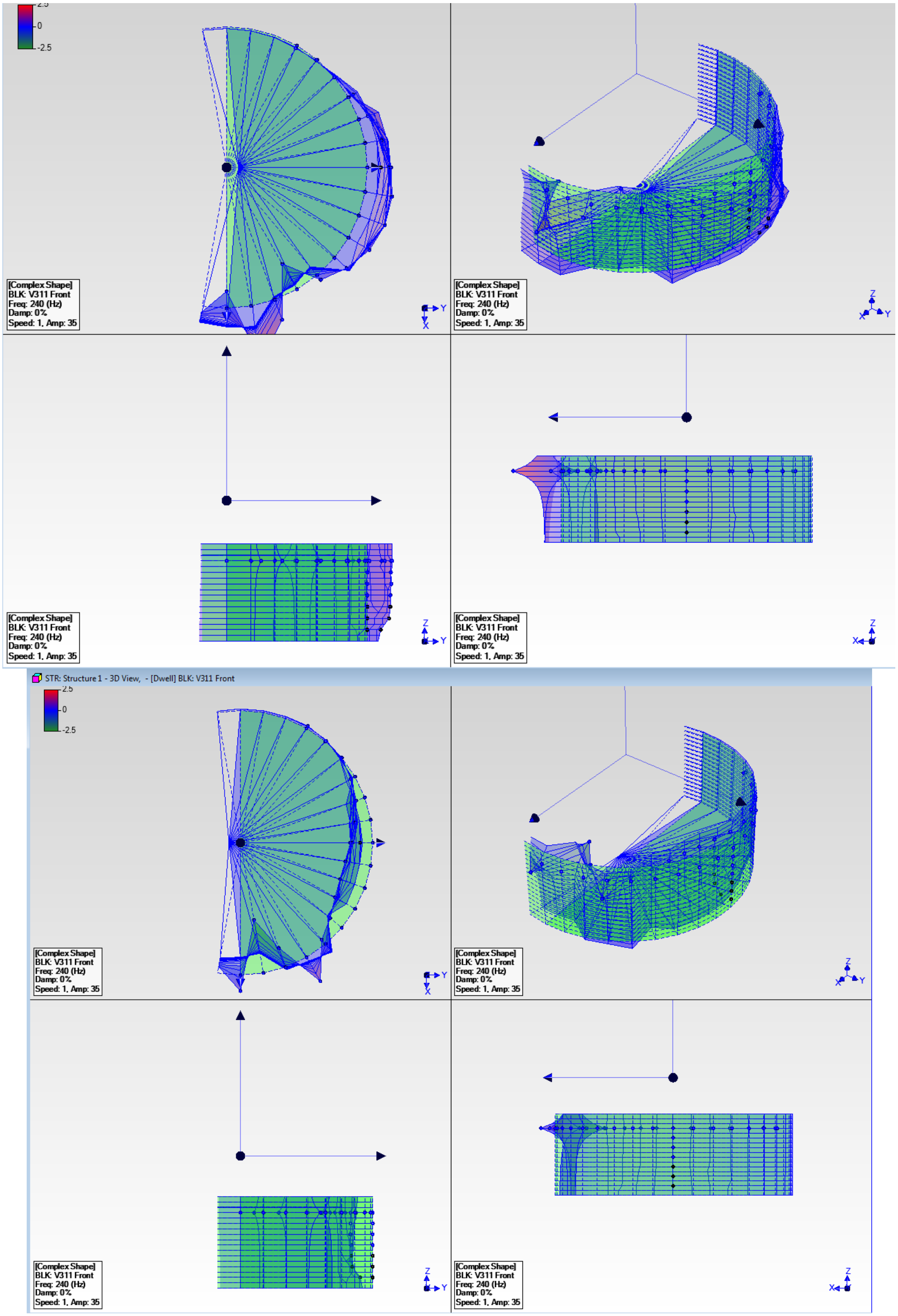


Figure 4. Measured shell ODS for unit 311 oil separator vessel, front side

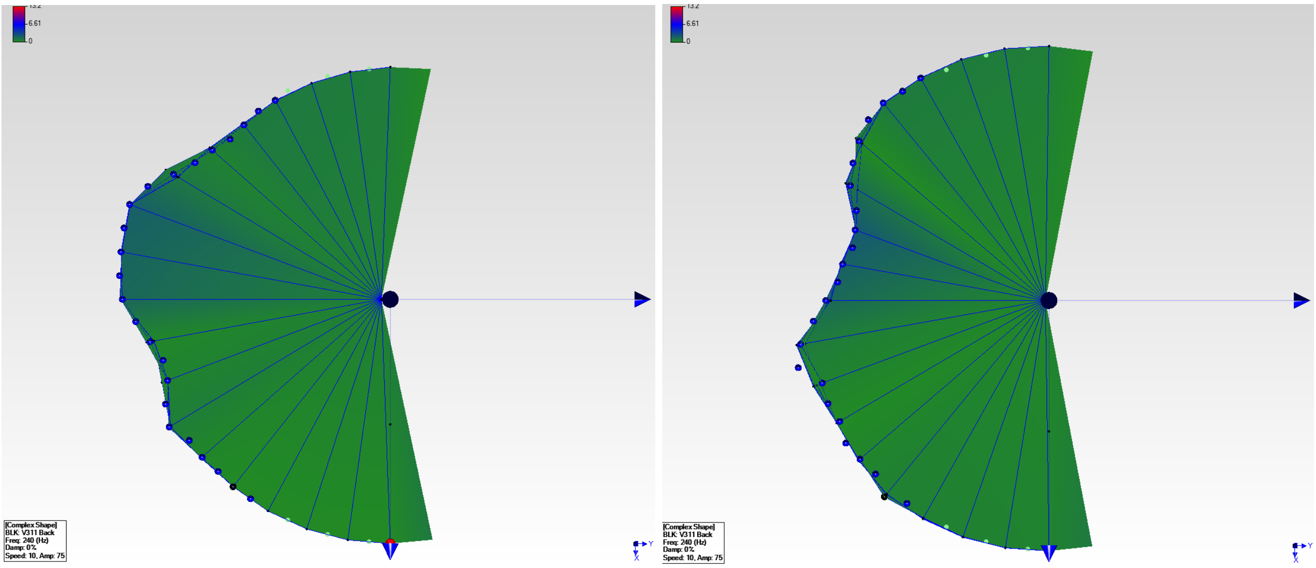


Figure 5. Measured shell ODS for unit 311 oil separator, back side

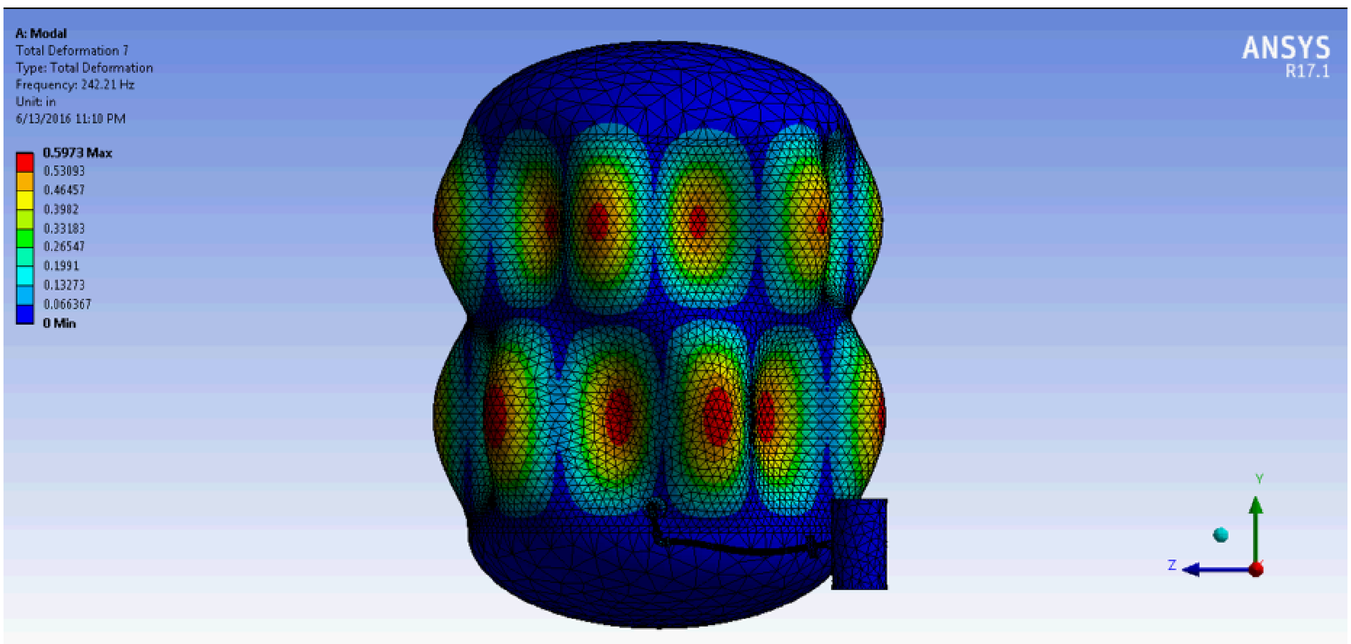


Figure 6. FEA shell mode shapes

Multiple shell modes

Based on the initial mechanical FEA showing a good correlation between the calculated mechanical natural frequencies and the measured vibration, it was decided to create a more comprehensive FE model that would include the vessel internals and nozzles. A modal analysis of the refined model showed multiple shell modes exist within the range of the lobe or pocket passing frequency of 239 Hz. See Figure 7 and Figure 8 for plots showing the shell mode mechanical natural frequencies (MNFs) closest to pocket passing frequency. Both example modes were seen to strongly deform around the oil equalizing nozzle, which is the nozzle that was experiencing fatigue failure. Both example modes were seen to involve (and thus can be excited to resonance by) the inlet nozzle. High shaking forces on the inlet nozzle from plane wave pressure pulsations would excite vibration on the separator shell and equalizer line nozzle.

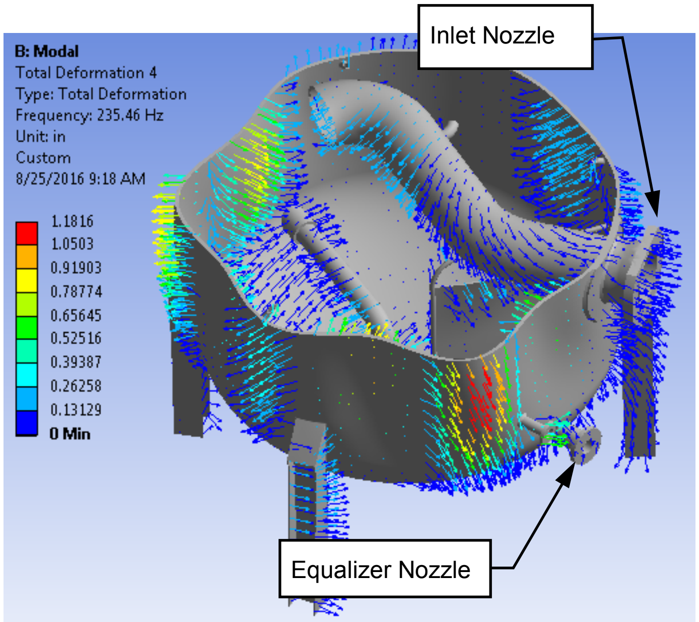


Figure 7. As found oil separator shell modes – 236 Hz

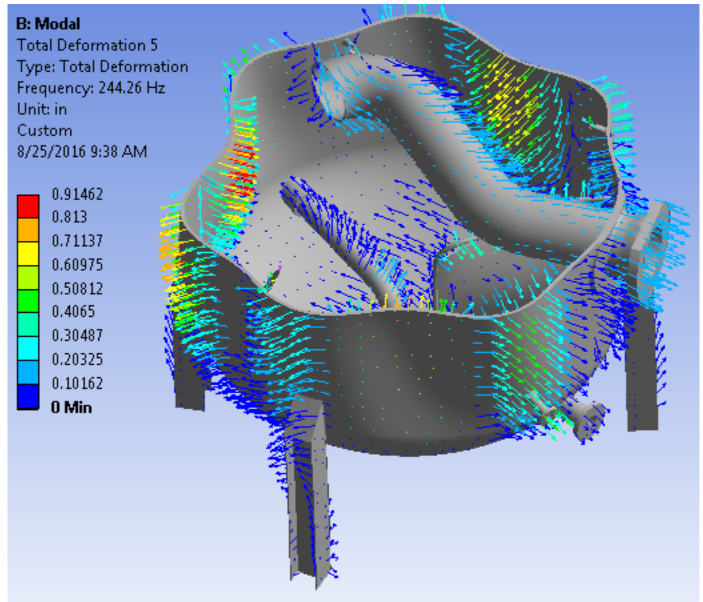


Figure 8. As found oil separator shell modes – 244 Hz

Vibration sensitivity to load

During changes in operating conditions, vibrations were measured on the shell at TP15 and TP6 in Figure 3. The vibration measurements at different slide valve positions for an operating pressure of 180 psig are shown in Figure 9. Pressure pulsations were also measured in the discharge piping. The same pattern which was observed in the pulsation measurements was observed in the vibration measurements. The highest vibrations occurred at 40%, 60%, and 80% load.

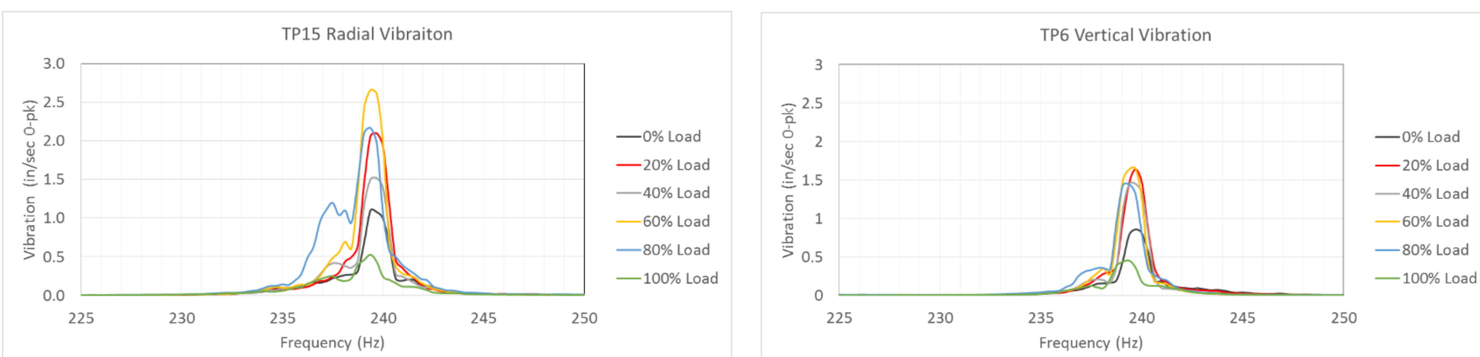


Figure 9. Vibration measurement on the vessel shell at different operating conditions, unit 311

Modal impact (bump) test

Finally, modal impact tests were done on the shell of unit 301 in order to measure the mechanical natural frequency of the shell and shell modes. Figure 10 shows the location of the impact tests.

Figure 11 shows the impact test results for TP 15. The mechanical natural frequencies from the impact tests and frequencies of the maximum measured vibrations matched, which suggests mechanical resonance of the shell. Some other shell modes were also observed at the same frequency range.

The results of the impact tests for unit 301 were fed into ME Scope to find the mode shapes. Figure 12 shows the results of impact tests on various test points around the vessel shell. A shell mode with four anti-nodal points (locations of maximum modal displacement) is very clear in this measurement. The result agrees well with the FEA results.

Figure 10. Impact test locations on unit 301

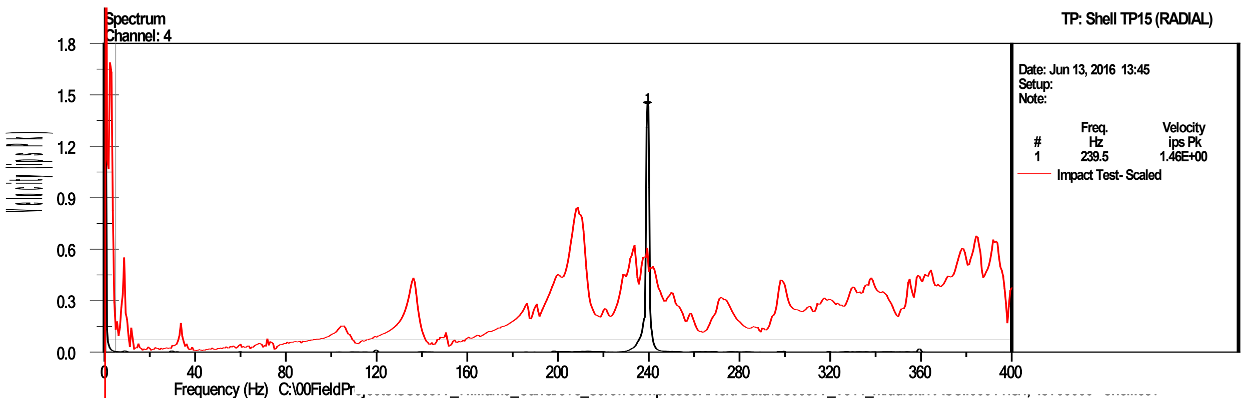


Figure 11. Overlay of shell vibrations and impact tests

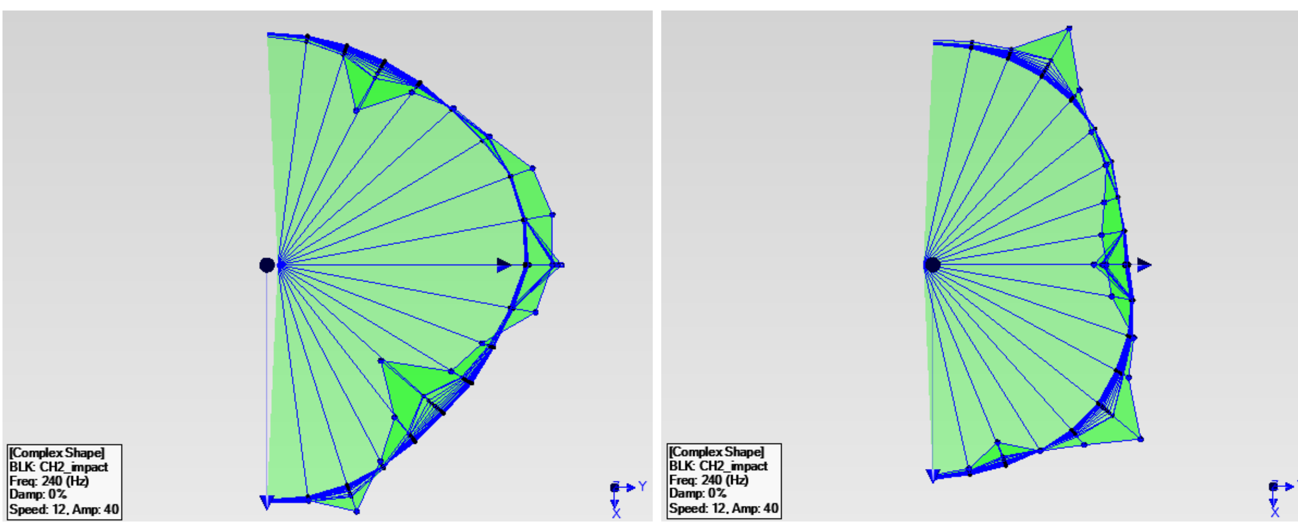


Figure 12. Measured shell mode for unit 301

Dynamic stress analysis

The maximum allowable vibrations on the shell and pipe nozzles were determined using the finite element model. The analysis included modal calculation of the equalizer nozzle on the vessel and the discharge pipe. The mechanical natural frequencies, mode shapes and modal vibration and stress were calculated with this model. Figure 13 shows an element plot of the FE model. It also depicts the location where the vibrations were calculated in the FE model for comparison to the field measurements. Figure 14 shows the results of the FEA.

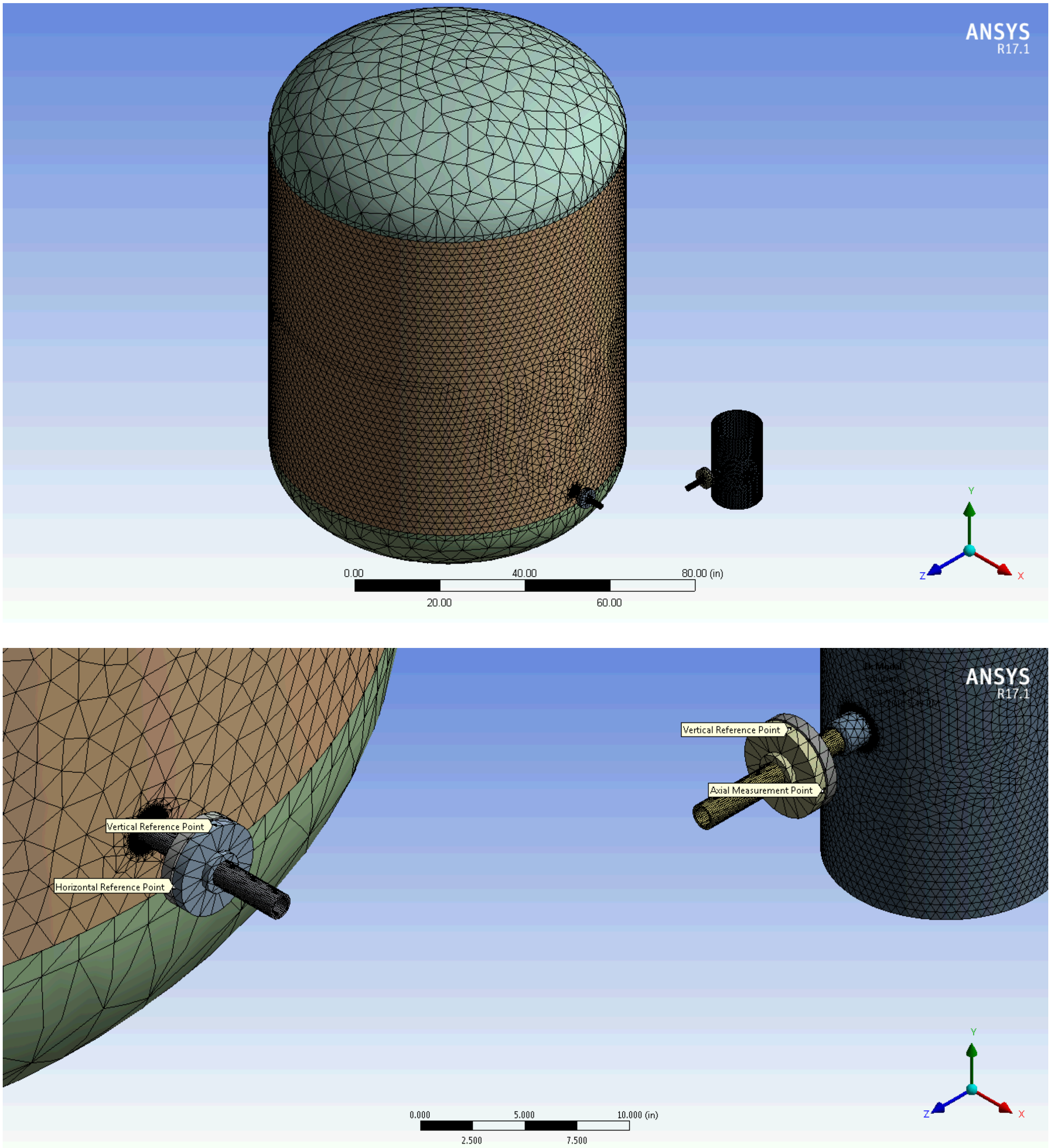


Figure 13. FE model and measurement locations on the nozzles

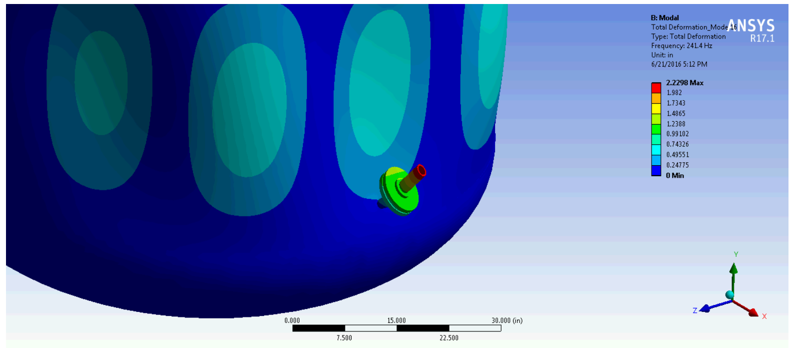
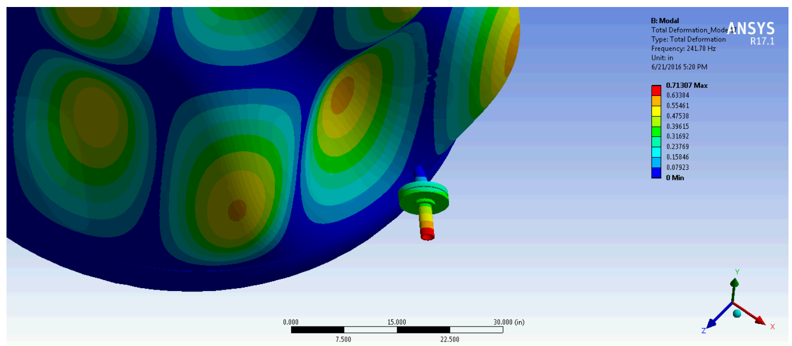
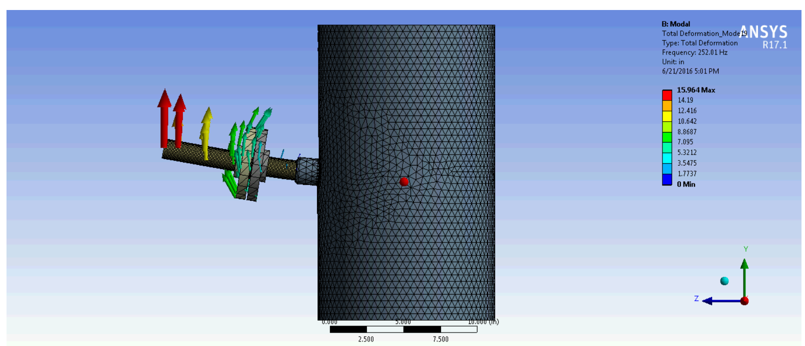
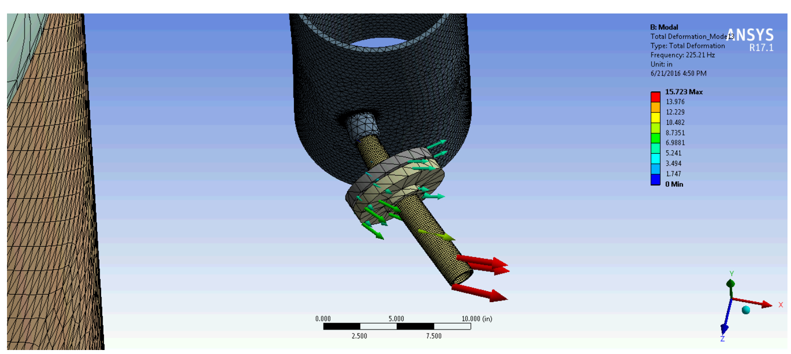
Location	Vibration mode	FEA results
Nozzle on the vessel	Vertical	
	Horizontal	
Nozzle on the pipe	Vertical	
	Axial	

Figure 14. Results of FEA

Based on the FEA results in Figure 14, the maximum allowable vibration on the nozzles was calculated and is summarized in Table 1. The stress per displacement was calculated from the FE model. The maximum allowable dynamic stress of 3,000 psi pk-pk was used to calculate the maximum allowable vibration at the measurement locations on the nozzles. Note that the allowable vibration levels assume vibration is dominant or only high in one direction and low in the other. Maximum vibration that is concurrent in both directions would not be acceptable.

Table 1. Maximum allowable vibration on the nozzles on equalizer line

Result	Nozzle on the vessel		Nozzle on the pipe	
	Vertical	Horizontal	Vertical	Axial
Stress per displacement unit (psi/mils)	450	406	1420	1432
Max displacement (mils pk-pk)	6.7	7.4	2.1	2.1
Max velocity (in/sec pk) at 240 Hz	5.0	5.6	1.6	1.6

Vibration measurements on the separator shell away from the nozzle were as high as 1.5 ips pk at some locations. Given the high frequency of the vibration, 239 Hz, the resulting displacement and dynamic stress in the shell is well below guideline. Failure of the separator shell away from the nozzles is not a concern.

Pulsation measurements

Pulsations were measured on the discharge line of unit 311 at the location shown in Figure 15. Figure 16 shows the pulsation measured at different operating conditions from 0% to 100% slide valve position. Note that the term 'load' is used in the charts of Figure 16 to refer to the slide valve position. The pulsations exceeded API 619 levels at all operating conditions and slide valve positions. The API 619 pulsation guideline is dependent on the absolute line pressure only. The highest pulsation levels were recorded when the unit was running at 40%, 60% and 80% of the load. Testing was done at discharge pressures of 180 psig and 210 psig. The pulsation levels at 180 psig discharge pressure were higher than those at 210 psig discharge pressure.

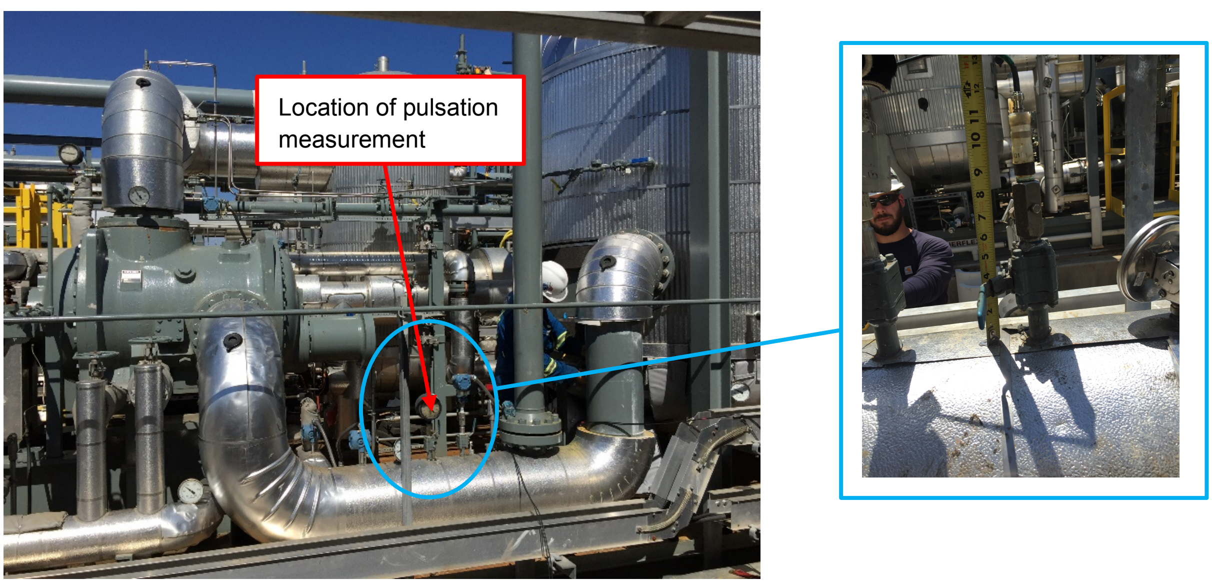
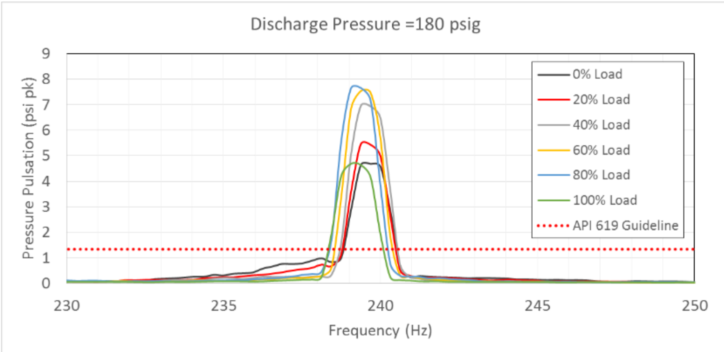


Figure 15. Location of pulsation measurements



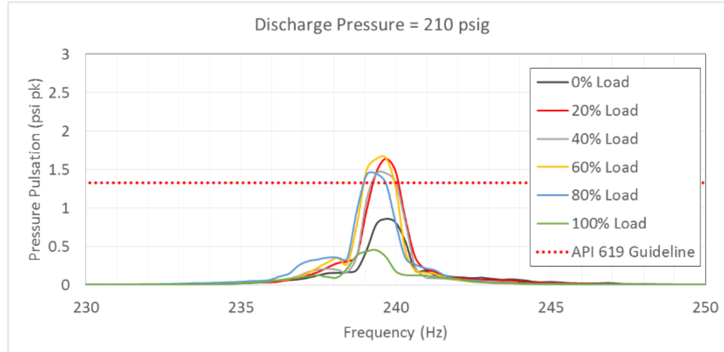


Figure 16. Measured pressure pulsations

Acoustic model and recommendations

Field measurements showed that high pressure pulsations were a significant contributor to the vibration problem. Pressure pulsations in the current discharge system were two to three times higher than the API 619 guideline. An acoustic (pulsation) model was developed to assist in determining a solution. A baseline model was developed for the current discharge system. A plot of the discharge model shows the calculated pressure-pulsation amplitudes for the 40% compressor load case (Figure 17), which is the worst pulsation case.

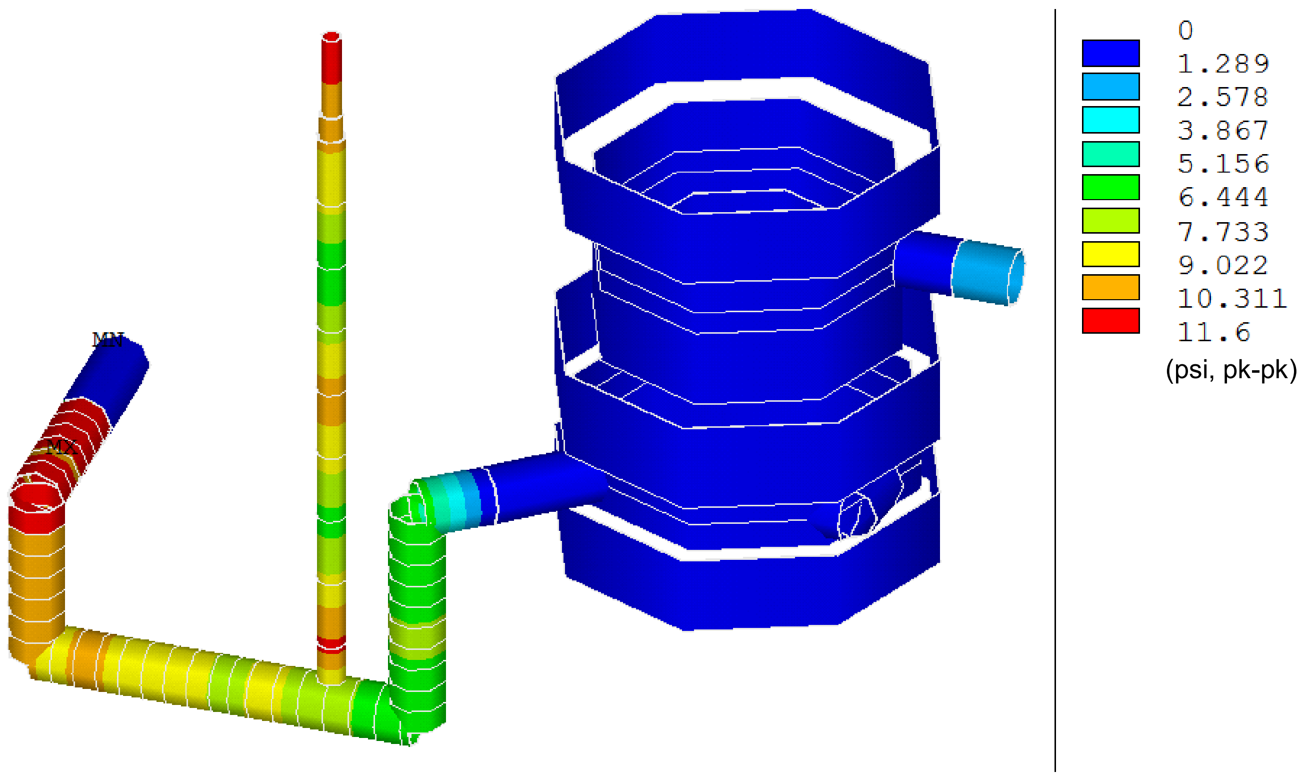


Figure 17. Pressure pulsation in the original (existing) discharge system

Several modifications to the discharge system were presented to reduce pulsations and pulsation-induced shaking forces in the piping between the compressor discharge port and the oil separator inlet. Two design options were presented that included a balance of recommendations that would be simple to implement and would reduce pressure pulsations and shaking forces. The design options included:

- Option 1 – orifice plate solution: the best reduction in pressure pulsations and shaking forces with orifice plate(s) only included an 8.75” ID orifice plate at the oil separator inlet.
- Option 2 – resonator solution: installation of a resonator in the discharge piping to change the acoustical resonance between the compressor and separator

Option 1 – orifice plate solution

The orifice plate solution is shown in Figure 18. The main effect of the orifice plate is to add resistance in order to dampen pulsations. The force that is considered to be the main source of shell vibration, the separator inlet force, is reduced at the compressor operating speed by approximately 40%. This shaking force reduction would translate into a reduction of approximately 40% in vibration, depending on actual operating conditions. A 40% reduction in shaking force and vibration would be a significant change, however, a further reduction in the vibration was desired. A disadvantage of the orifice plate is the pressure drop and power loss. The results of the shaking forces with the orifice plate solution are shown along with the original system results in Figure 19.

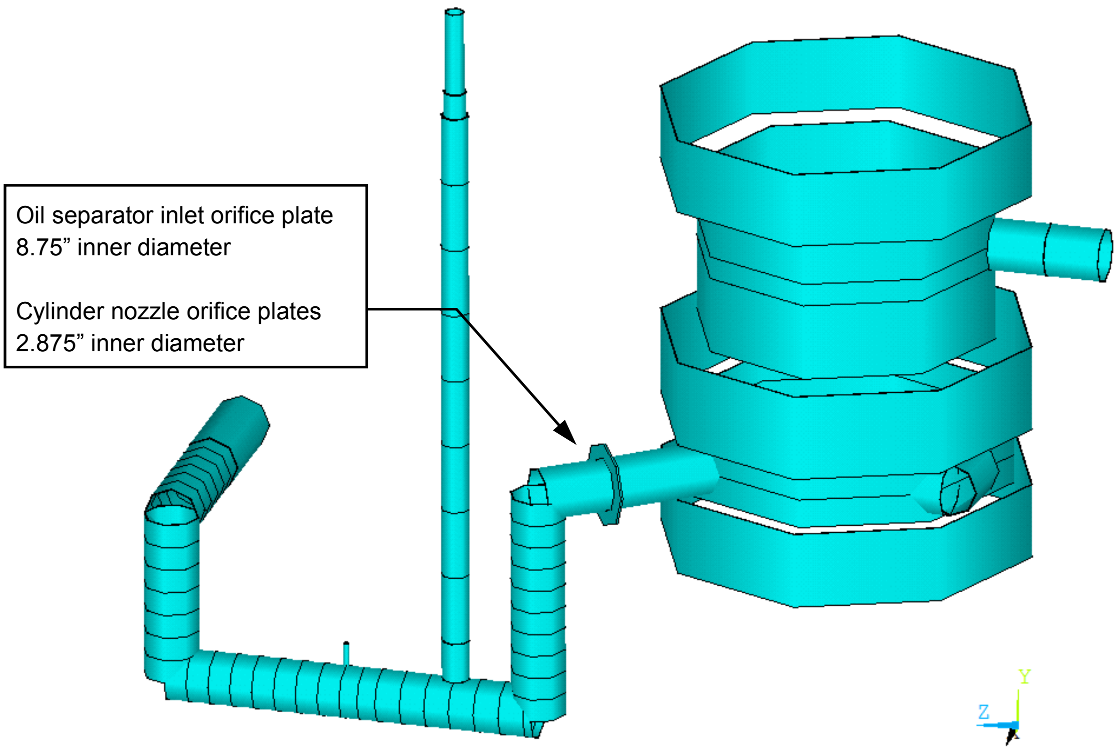


Figure 18. Option 1 – orifice plate solution

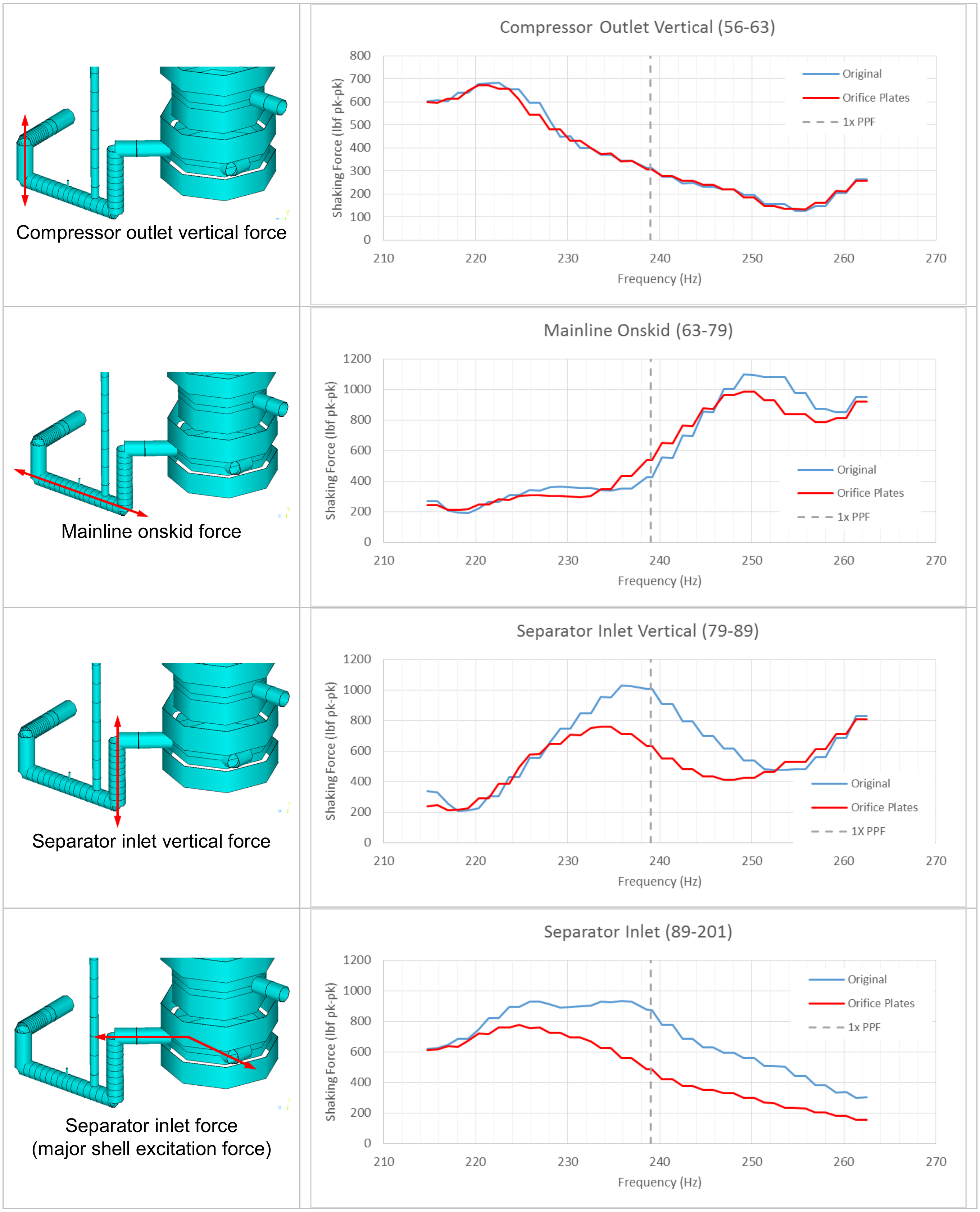


Figure 19. Discharge system shaking forces comparison – orifice plate solution vs original system

Option 2 – resonator solution

A resonator added in the discharge piping gave the greatest reduction in pulsations and shaking forces for the simple design modifications that were evaluated. The resonator, in this case, was a quarter wave resonator which means the resonator length corresponds to ¼ of the wavelength at the frequency of the pressure pulsations to be controlled. The resonator disrupts the pulsation resonance in the discharge system with the ¼ wave resonance generated within it. An image of the pulsation model of the resonator is shown in Figure 20.

The resonator is designed with two sections of 6” pipe of equal length. High pulsations and forces can be generated in an individual quarter wave resonator. The recommended design includes two resonators which are installed at the same location in opposite orientations so there are no net-shaking forces on the resonators and discharge piping.

With the resonator installed, the force considered to be the main mode of shell excitation, the separator inlet force, is reduced at operating speed by approximately 90%. A plot of the pulsation in the pressure pulsation at pocket passing frequency with the resonator is shown in Figure 21. The shaking forces in the discharge piping for the resonator design are compared to the original design in Figure 22. All plots are shown for the 40% compressor load case, which is the worst pulsation case.

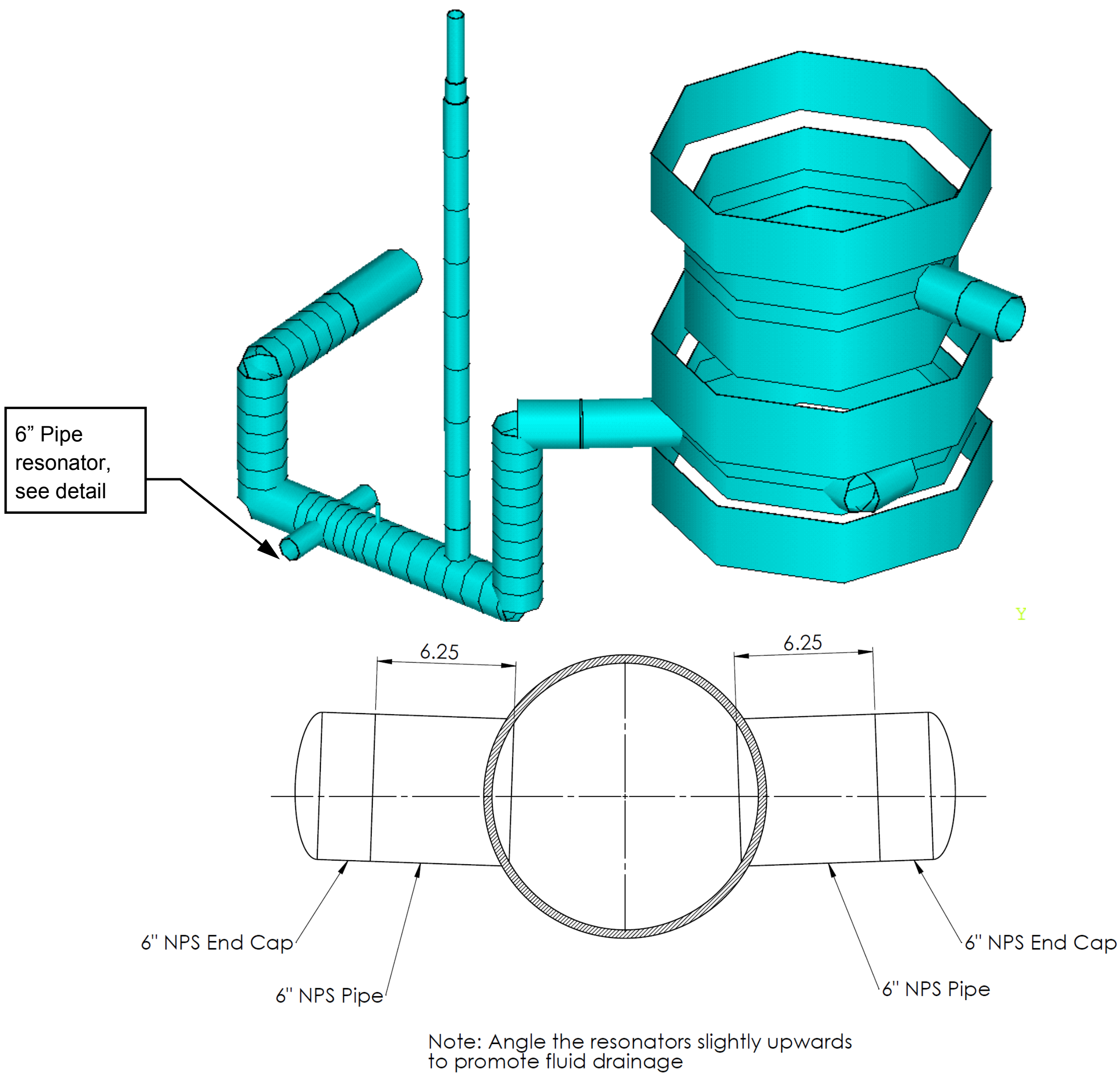


Figure 20. Option 2 – resonator added to discharge piping

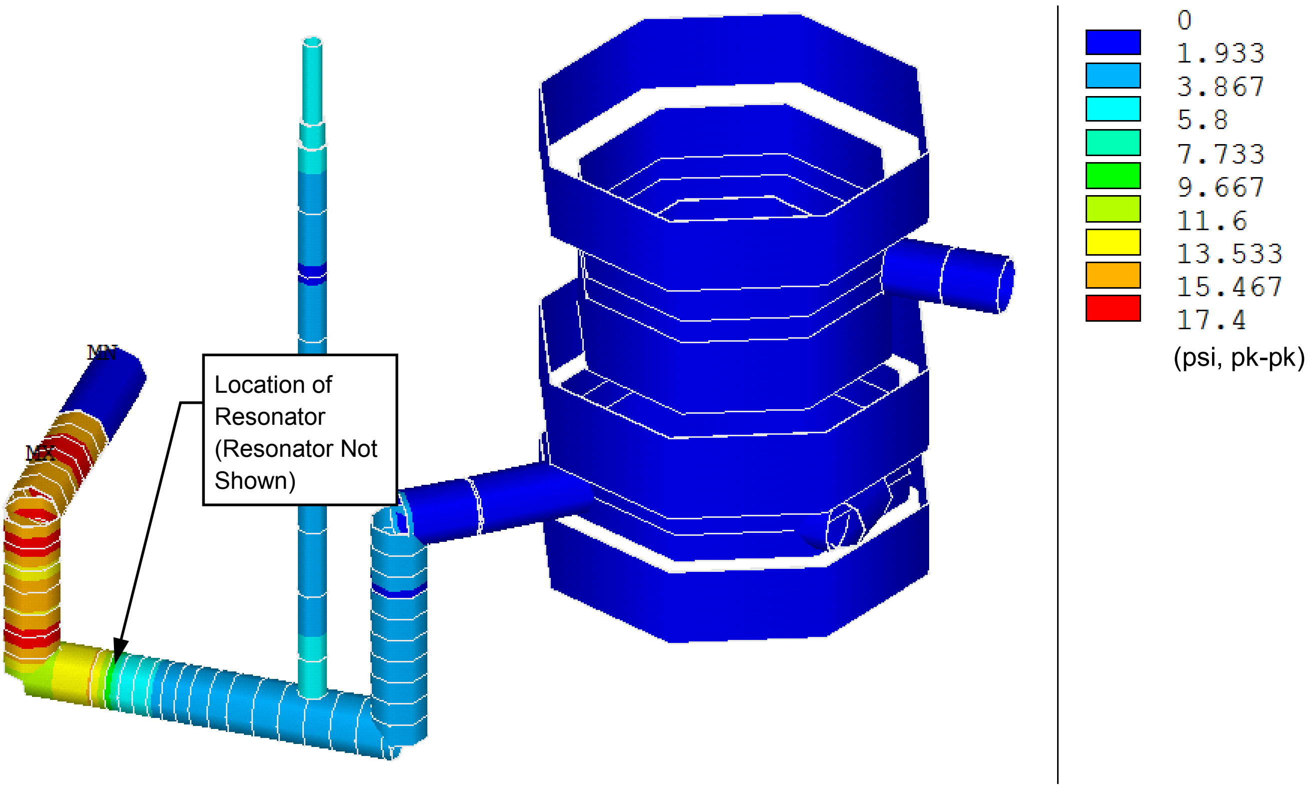


Figure 21. Overall pressure pulsation with the resonator (option 2)

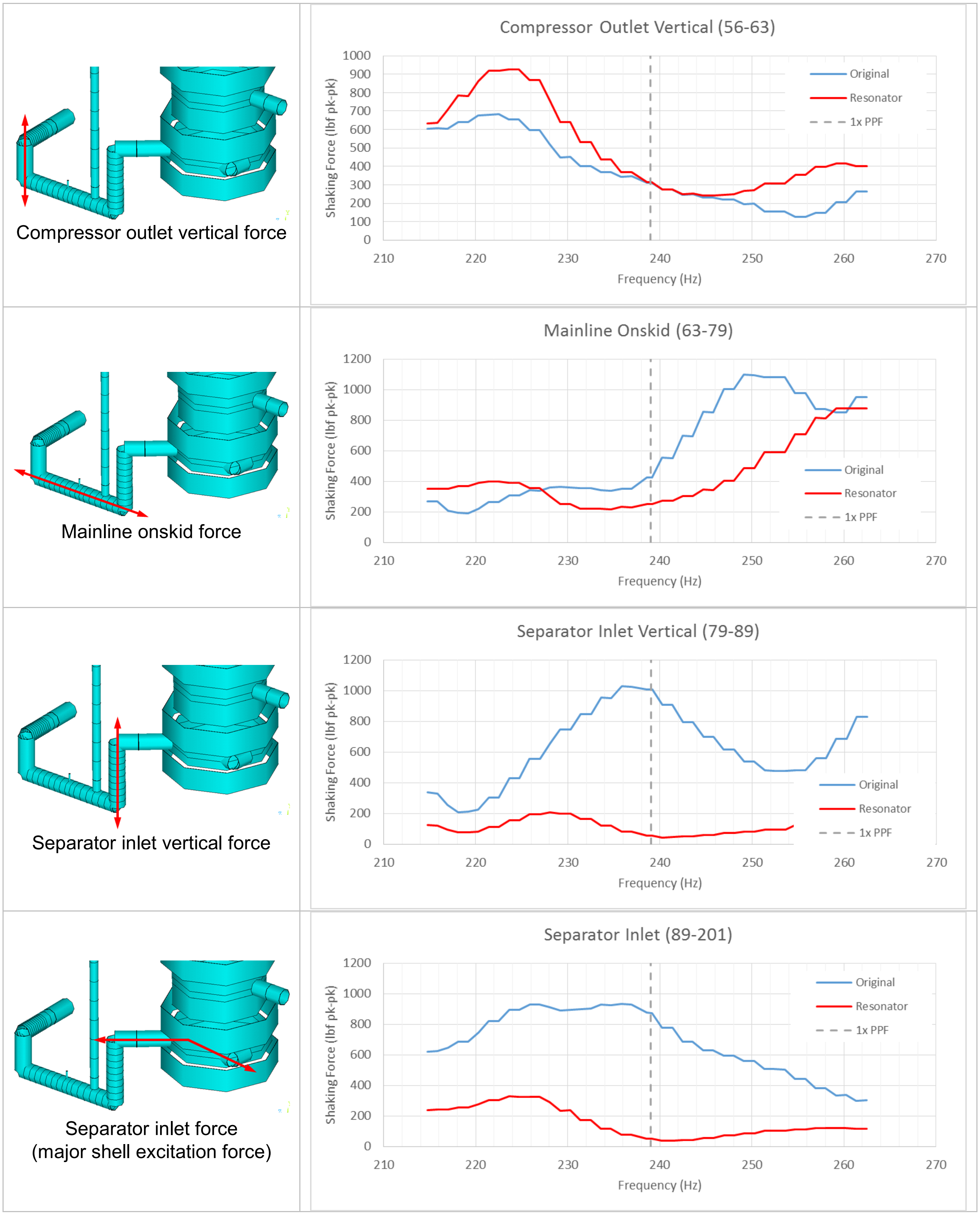


Figure 22. Discharge system shaking forces comparison – resonator solution vs original systems

Field verification of resonator

Williams decided to proceed with the resonator installation. Advantageously, the resonator would not cause additional pressure drops which the orifice plate would do. Also, the resonator resulted in greater reduction of the pressure pulsations and shaking forces than the orifice plate solution.

Installation of the resonator required replacing the pipe spool and fabricating the resonator piping per the recommended location. Follow-up measurements were taken by Williams to verify the adequacy of the resonator. Vibration was significantly reduced on unit 312, however, vibration remained relatively high on the other units. Pressure pulsations were measured on all units. The pressure pulsations measured by Williams are shown in Figure 23 for different slide valve positions along with the pressure pulsations calculated from the pulsation model of the ideal resonator design. The compressor was fixed-speed, so the field pulsation measurement was recorded as a single pulsation amplitude. The pulsation amplitude is shown as a horizontal line to visualize the measurement more easily. The pulsation measurement would be more accurately shown as a single point at 239 Hz on this chart, however, this would make it more difficult to interpret the results. In some cases, pressure pulsation varied slightly during the field measurement. This variation is shown as a horizontal band.

The comparison of the measured and calculated pressure pulsations shown in Figure 23 confirms that the pressure pulsations from unit 321 agree well with the pulsation model. Unit 301 and 311 are supposed to be identical to unit 321, however, the results were somewhat different. There may be small differences between the units, either in their operating parameters (pressure, temperature or condition) or in the implementation of the resonator that caused a variation in the pressure pulsations. A variation in the pulsation resonance of ±5% to ±10% is a common approach to consider differences between units. The grey band shown in Figure 23 considers a ±5% variation in the acoustic resonant frequency of 239 Hz. The field pulsation measurements show a closer correlation with the pulsation model results. Additional pulsation simulations were conducted to evaluate the sensitivity of the resonator design to different parameters.

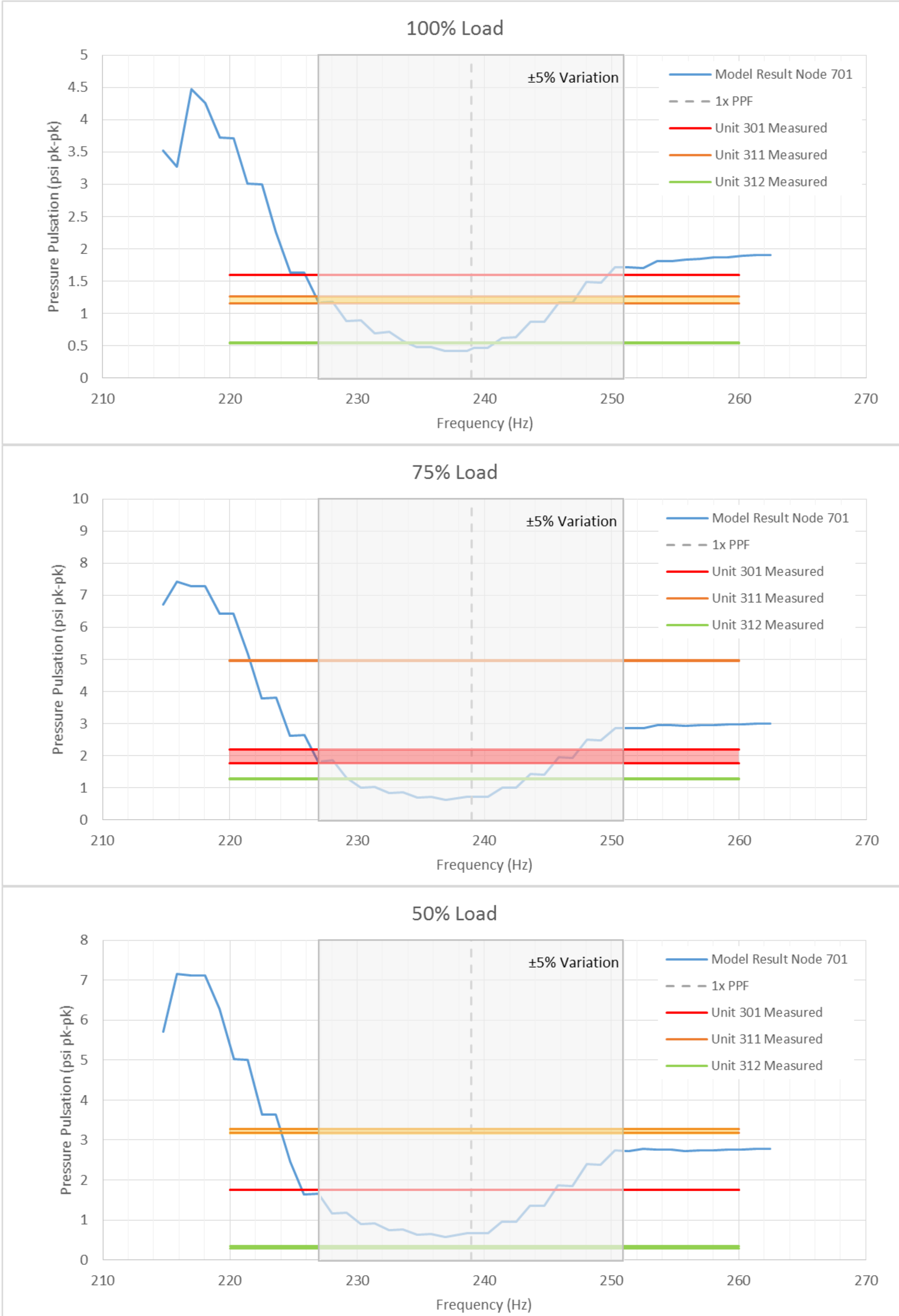


Figure 23. Calculated pressure pulsations versus measured pulsations

Resonator sensitivity analysis

Resonator length

The effect of the resonator on pressure pulsations and shaking forces in the discharge is sensitive to the resonator length since the resonator length defines the resonator frequency. The as-built resonator length may vary from the simulated resonator length due to fabrication tolerances. Another variation in length may come from the shape of the end cap on the resonator. The end cap has an approximately ellipsoidal shape that results in a difference between the physical length and acoustical length. The sensitivity of the model to the resonator length was tested with the pulsation model. The change in the shaking force in the separator inlet is shown in Figure 24 considering a 1" change in the resonator length, about a 15% change in the overall length. The results show that an accurate simulation of the resonator length as well as an accurate construction of the resonator is very important. Shaking-force amplitudes could vary by a factor of three or more with a 15% change in length.

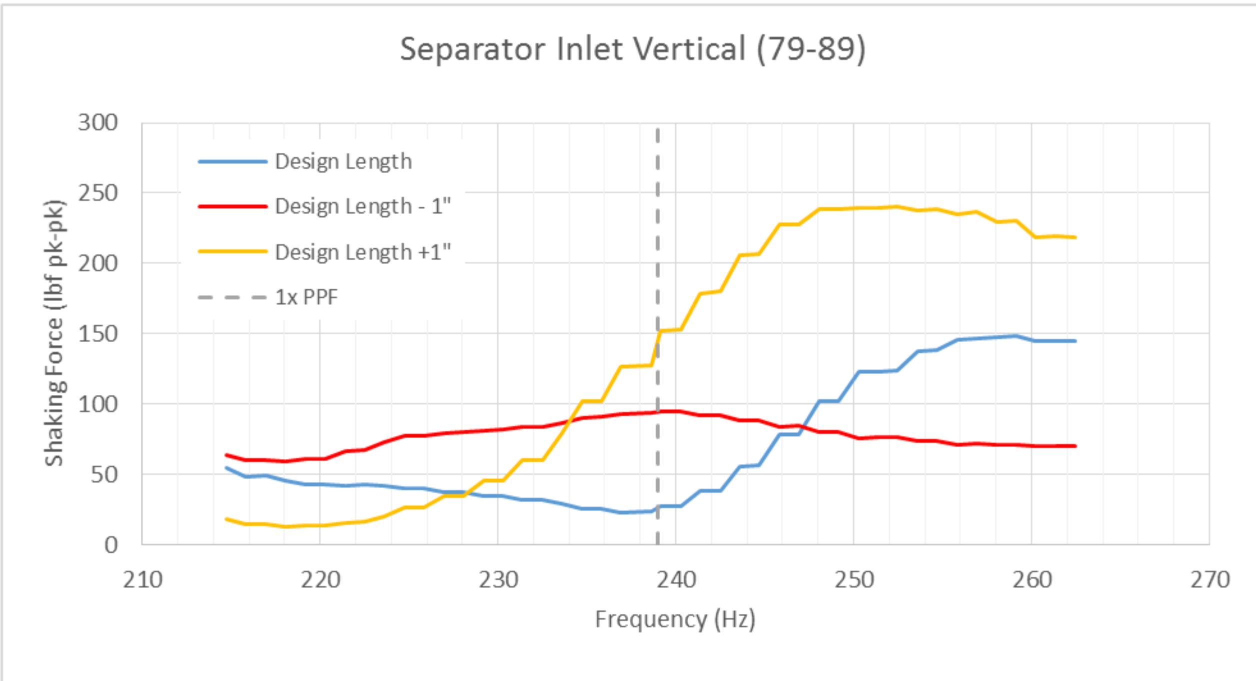


Figure 24. Resonator sensitivity as a function of length

Two-phase flow

The oil in the screw compressor was thought to perhaps change the thermodynamic properties of the flow, which may negatively affect resonator performance. Calculations showed that the oil takes up a negligible portion of the flow and does not significantly change acoustical properties.

Liquid level in separator

The impact of the liquid (oil) level in the separator was evaluated for its possible effect on pulsations in the discharge piping. Pulsation model simulations were conducted to evaluate the impact on shaking forces in the discharge piping, as shown in Figure 25. The pulsation and shaking forces in the inlet line are not significantly affected by the liquid level in the separator. It should be noted that 3D-pulsation effects are not captured in this plane-wave analysis. The pulsation resonance in the piping is calculated to be a plane-wave mode, so the impact of 3D-pulsation resonances is expected to be small.

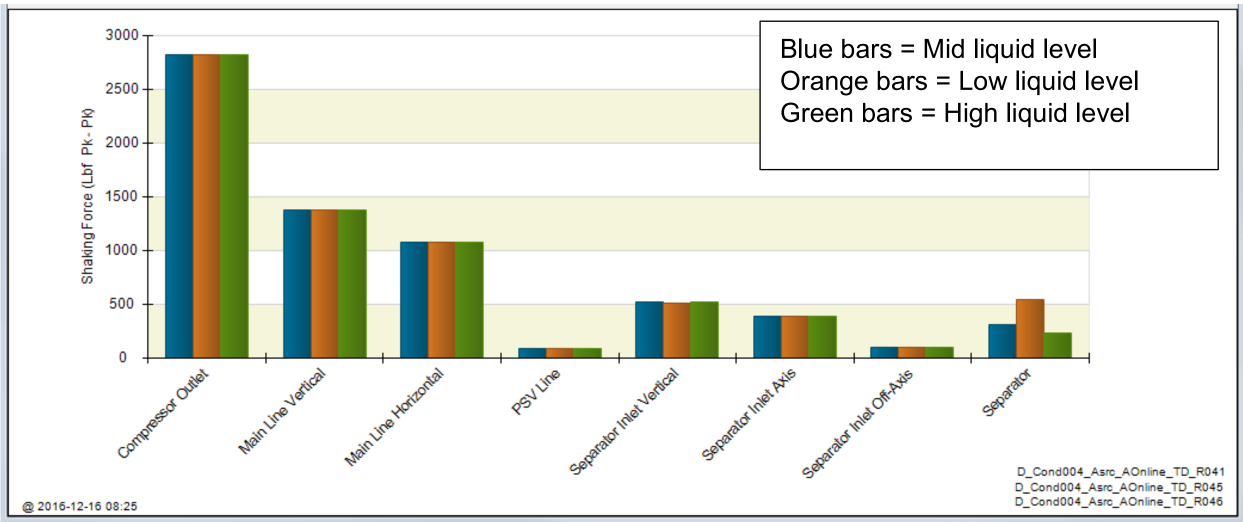


Figure 25. Resonator sensitivity to liquid level at various test points

Discharge pressure

The discharge pressure varies over a small range for the compressor operation. The sensitivity of the model response due to changes in the discharge pressure was evaluated. The discharge pressure is expected to vary between 178 psig and 220 psig. The calculated shaking force for the range of pressures is shown in Figure 26. The shaking force is relatively insensitive to variations in pressure at pocket passing frequency of 240 Hz. If the system response was shifted to be similar to the 250 Hz response, there would be a significant increase in the shaking force (50% higher) when the pressure increases from 178 to 220 psig. Care must be used in documenting the discharge pressure when comparing pulsations from field measurements between units as well as with the model results.

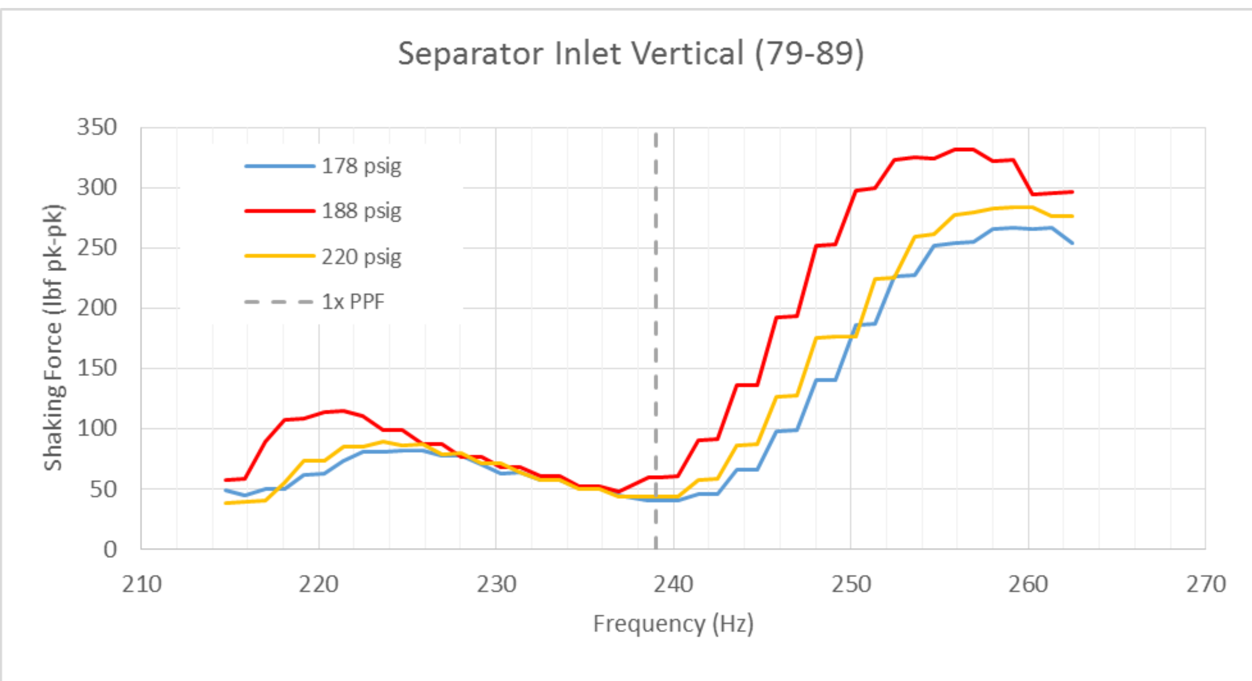


Figure 26. Resonator sensitivity to discharge pressure

Discharge temperatures

The effect of discharge temperature was also evaluated with the pulsation model. The temperature had a strong influence on the acoustic velocity or speed of sound. The model results show a significant impact of the temperature on the pulsations and shaking force. The resonant frequency is shifted to a lower frequency as the temperature is lowered. An example plot showing the separator inlet force is shown below in Figure 27. The design study considered a relatively narrow temperature range for normal operation based on operating logs. The normal discharge temperature varied from roughly 144°F (62°C) to 148°F (64°C). The resonator was designed for optimum performance at a discharge temperature of 148°F (64°C). Even though the temperature range is relatively small, a small variation in the resonator performance is to be expected. The discharge temperature must be documented when taking pulsation measurements for comparison to the model results.

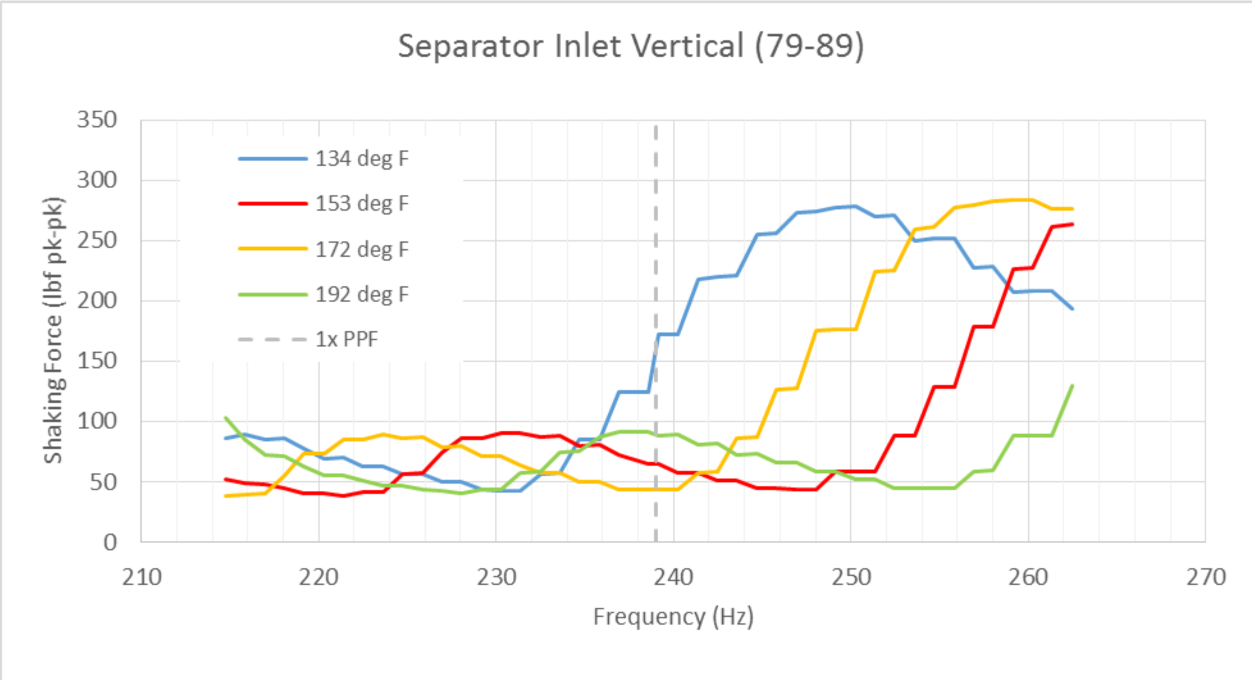


Figure 27. Resonator sensitivity as a function of discharge temperatures

The sensitivity of pressure pulsations in the separator due to higher discharge temperatures was also evaluated. The pressure pulsation in the separator is important as it is a possible excitation source for pulsation resonances in the separator and mechanical shell resonances. Figure 28 shows the predicted pulsation in the separator for three discharge temperatures: 144°F (62°C), 164°F (73°C) and 183°F (84°C). Operating at much higher temperatures of over 160 °F (71°C) to 180 °F (82°C) would explain some of the differences between the field measurements and pulsation model results.

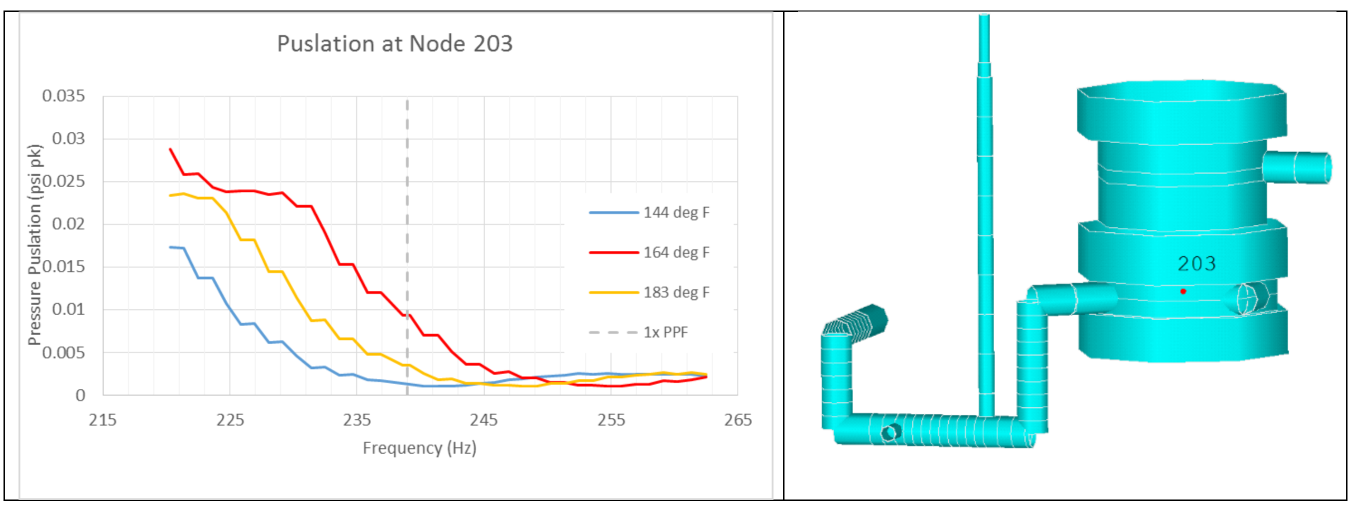


Figure 28. Optimal discharge temperature and resonator performance

The resonator sensitivity analysis showed that the main factors influencing the effectiveness of the resonator are length and temperature. The compressors operate in an essentially closed loop, so the temperatures are not expected to vary much. Variations in the length are likely to be a factor in the differences between measurements and the model results as well as differences between units. A resonator design that had some adjustability in length to tune the design for an installation could be considered. One approach would be to use a flange blind on the end of the resonator rather than an end cap. The length could be adjusted with a spacer in the flange connection.

Mechanical solutions for controlling vibration

The best strategy to lower shell vibration is to reduce the excitation force in the system. Installing pulsation (acoustical) controls are the preferred approach. The simple pulsation control solutions such as orifice plates and resonators that were investigated in this project have limitations as noted in the previous section. Alternative methods of pulsation control such as pulsation filters (bottles) can be used (1). Pulsation filters are less sensitive to many of the factors described in the previous sections. The drawbacks to a pulsation filter are the expense for the filter and space limitations. The space constraints for this project would make the installation of a pulsation filter a difficult task. There is also a risk of vibration on the pulsation filter itself that must be carefully considered in the design. The relatively high power needed for these compressors would result in the generation of significant pressure pulsations. It would be prudent to consider a pulsation bottle design in new packages.

An alternative approach to minimizing vibration on the vessel and the equalizer line is to control or attenuate the structural response rather than the excitation force. Shell vibration of the vessel wall is a significant concern in terms of resulting vibration on the equalizer nozzle as well as other nozzles and internal connections to the shell. Several methods are available for stiffening the shell to reduce vibration. A solution that does not require welding on the vessel or otherwise modifying the vessel is preferred as this will avoid the need to recertification of the vessel. One proposed solution is adding a band clamp around the vessel similar to that shown in Figure 29. The band clamp is located on the vessel in the area where the highest shell vibration was measured.

The proposed band clamp design includes a damping material between the band clamp and the vessel. Adding stiffness alone with the band clamp will not be sufficient. Adding more stiffness increases the current problematic mechanical response above pocket passing frequency, but other mechanical responses are also raised and would then be coincident with pocket passing frequency. The addition of damping will reduce the mechanical response of these new resonant modes.

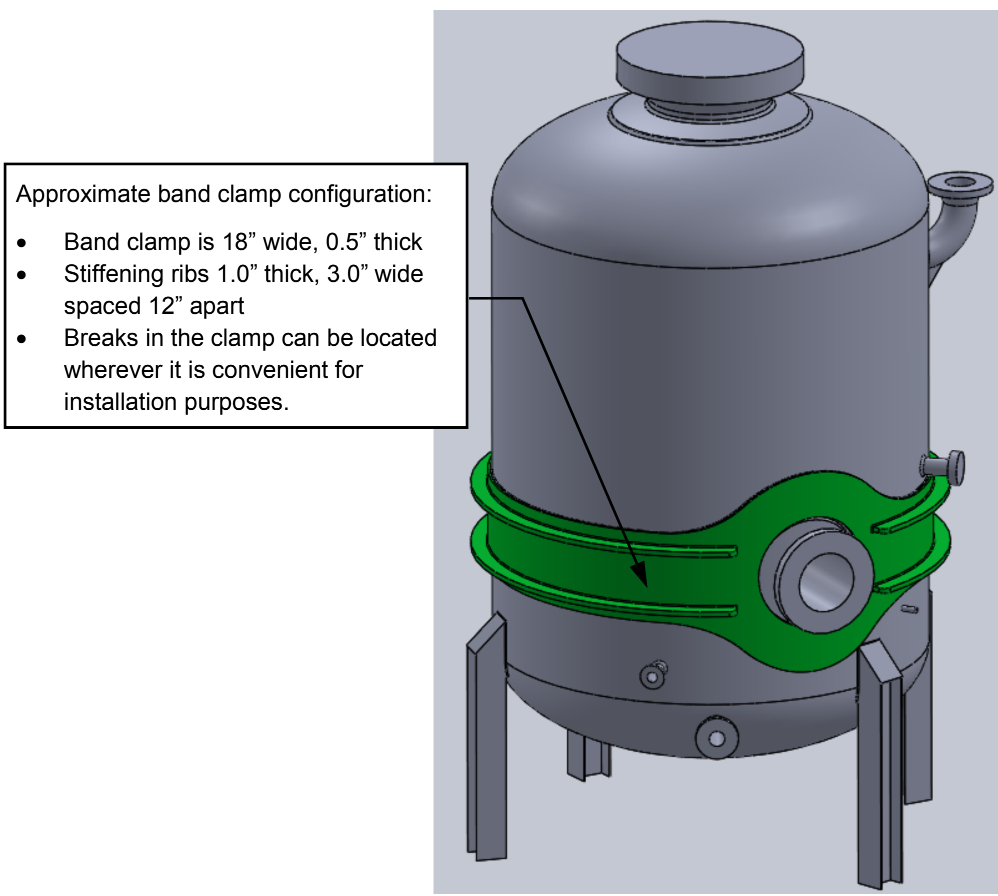


Figure 29. Proposed band clamp arrangement

Additional FEA was conducted to evaluate the potential reduction in vibration due to a damped band clamp recommendation. A harmonic analysis was conducted to calculate the vibration and dynamic stress on the separator due to a dynamic forces acting on the inlet nozzle. Vibration on the shell was reduced by 55% in some areas. See Figure 30 for a comparison of existing separator results and the damped band clamp results. Note that the absolute magnitude of the vibration is not of direct concern in the comparison, only the relative reduction.

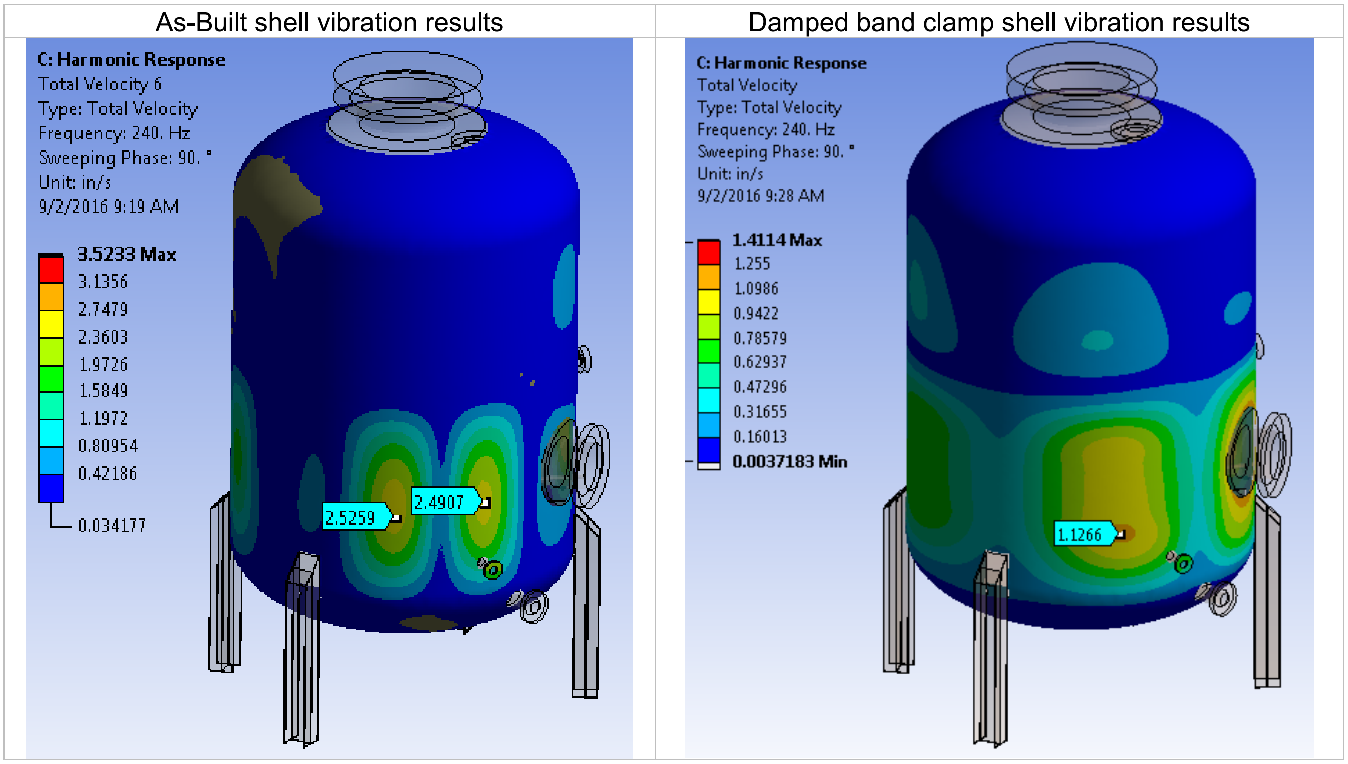


Figure 30. Oil separator shell vibration comparison – existing vs damped band clamp

The damped band clamp would be effective, but it is a large device. Other approaches of adding localized shell stiffening and damping (near the oil equalizing nozzle) were initially attempted to lower overall shell vibration. Results for this strategy showed lower shell vibration in the area that had been stiffened or damped, but shell vibration would be amplified on a different part of the shell, which only endangered the other nozzles and small-bore attachments on the shell.

The results also showed that the oil separator shell might suffer from non-resonant vibration even after the modifications to stiffen the shell and move the shell MNFs were implemented. The reason for this result is that the oil separator inlet nozzle internal projection is welded to the inside of the shell. The support forces the shell and nozzle to move in unison when acted on by an excitation force. The vessel shell must deform to compensate. A modification that affects a larger area of the vessel shell such as the damped band clamp is required.

Measures to control vibration on the equalizer line were also investigated during field testing. The equalizer line had a small angle support near the discharge line connection. The support was connected to the line by a u-bolt clamp as shown in Figure 31. An initial test was done by site personnel to loosen the u-bolt clamp. Vibration on the piping was observed to be lower than when the clamp was installed. Closer inspection found a gap between the equalizer line and the support. The clamp, when installed, was pulling the piping down to the support adding a large amount of static strain on the piping. High static strain because of the unshimmed support caused high stress in the line and nozzles which may have been a contributing factor to the failure. Also, pipe strain has been observed to cause higher-amplitude vibration than piping that is not strained. A shim was installed between the pipe and support. This reduced vibration on the line and nozzle by 30%, however, vibration remained high, over 2.6 ips pk. Additional tests were done by adding a detuning mass which reduced vibration by a further 10%. A test was done by replacing a section of the equalizer line with a steel braid hose. Vibration on the nozzles connection on the separator and discharge line remained high. There is sufficient excitation from pressure pulsations in the discharge piping and separator to cause high vibration even though the nozzles were somewhat decoupled mechanically with the flexible hose. Simple changes to the support and layout of the equalizer line were not effective in reducing vibration to acceptable levels. Design modifications to more than just the equalizer support were required.

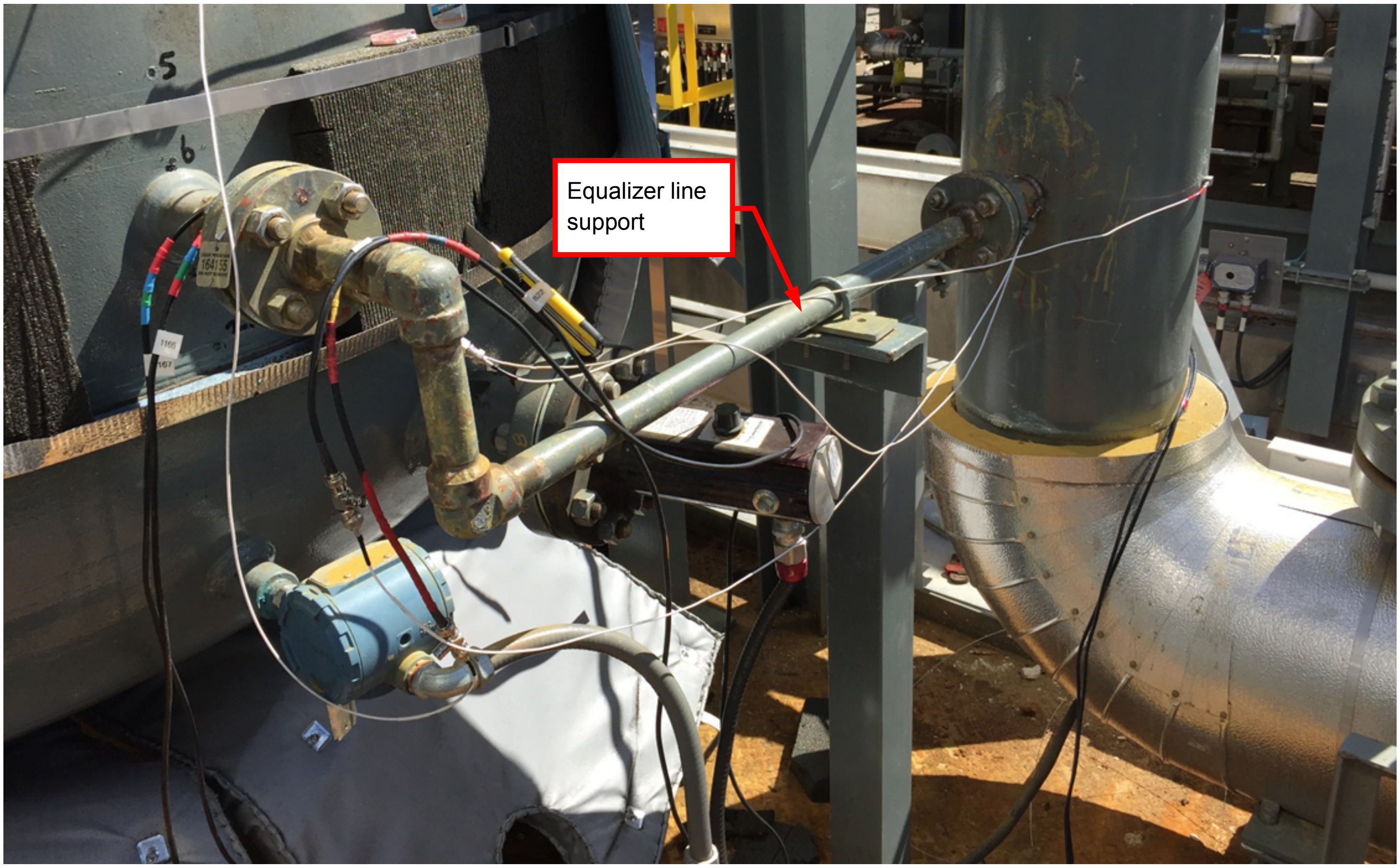


Figure 31 - Equalizer line

Early in the project, there were discussions regarding adding grout to the skid to control the vibration. An improved grout connection between the package skid and foundation or grout added within the skid can often be effective in controlling vibration on packaged compressors. The vibration problems encountered on this package are the result of a pressure pulsation resonance in the discharge system and mechanical resonance of the separator and oil equalizer line. Adding grout to the skid or skid-to-foundation connection would not reduce the vibrations encountered on this package.

Conclusions and recommendations

1. The root cause of the high vibration and failure on the equalizer line is excitation of mechanical resonance of the equalizer line and separator shell by pressure pulsations generated by the screw compressor.
2. Field measurements and pulsation simulations of the compressor discharge system found that pulsations were above the API 619 pulsation guideline, in some cases more than five times higher than the guideline. The high pulsation in the piping was found to be due in large part to a plane-wave acoustical resonance between the compressor and the oil separator at pocket pass frequency. This resonance manifests itself as pulsation-induced shaking forces in the piping which causes high shell vibration of the oil separator. Shell vibration results in high vibration on the equalizer nozzle and piping.
3. The pulsation-induced shaking force in the oil separator inlet line is one of the main excitation sources of the oil separator shell excitation, and thus it is expected that lowering this force, as well as the other forces in the immediate area of the oil separator, will result in reduced shell vibration. The plane-wave pulsations could also excite transverse pulsation modes inside the separator that could excite the shell mechanical modes. This mode of excitation was not considered in this analysis. However, reducing pressure pulsations to reduce the shaking forces in the oil separator inlet line also has the benefit of reducing pressure pulsations in the separator, so excitation of transverse pulsation modes will also be reduced.

4. The pulsation analysis determined that pressure pulsations and shaking forces can be reduced with either an orifice plate or resonator in the pipe spool between the compressor and separator. The resonator performance is sensitive to the resonator length. Accurate construction is important to achieve the maximum benefit of the resonator. A resonator design that has the ability for adjustment in the field may be beneficial. The influence of changing the operating temperature on the resonator performance needs to be considered, but may not have been a major factor for this particular application.
5. The oil separator mechanical analysis found that multiple shell modes existed within the range of the pocket passing frequency. These modes involve the axial movement of the inlet nozzle of the separator, which, when the results of the plane wave analysis are considered, serves as a strong mode of excitation for the shell and nozzle. The analysis determined that a band clamp with damping material between the band and vessel would be effective in reducing the shell and nozzle vibration by more than 50%.
6. Following are recommendations to minimize fatigue failures from vibration for future wet screw compressor packages, based on the work done on this project:
    a. Conduct a pulsation design study on large screw compressor packages. Pulsation controls such as orifice plates or resonators may be required. A pulsation filter (bottle) will likely be required for larger compressors (more than 1,000 HP).
    b. Evaluate the small-bore connections on the discharge piping and oil separator. The evaluation may include design work to assess the components and/or shop testing to measure mechanical natural frequencies before the equipment leaves the fabricator's shop.
    c. Conduct vibration testing on the screw compressor package after starting to assess mainline and small-bore piping and instrumentation. Testing must consider a range of operating parameters.
    d. Check pipe supports to ensure pipe strain has been minimized. The pipe and small-bore lines must be installed so they are held in a neutral position when at hot operating conditions to minimize static loads.

References

1. Nored, Marybeth et al., *Screw Compressor Acoustic Issues and Silencer Design*, Gas Machinery Conference 2011.
2. API 619 Rotary Type Positive Displacement Compressor for Petroleum, Petrochemical and Natural Gas Industries, Fifth Edition: December 2010, American Petroleum Institute