Vibration Issues with Engine Driven Reciprocating Compressor Systems

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Abstract

Engine driven reciprocating compressors are commonly used for gas pipeline systems. These units need to be reliable over a range of operating conditions and speeds. This paper presents three case studies detailing issues that were encountered on otherwise standard packages consisting of a 16-cylinder engine driving a six-throw reciprocating compressor in natural gas service.

In the first case study, the engine vibration exceeded the manufacturer's allowable limit when operating near the maximum speed. The engine experienced multiple issues, including high frame vibration, vibration of the control box, and damage to wire harnesses. Mechanical natural frequencies (MNFs) of the engine mounted on the pedestal were found near 1× and 1.5× running speed. Structural modifications were made to detune the MNF and reduce the vibration.

In the second case study, the compressor was experiencing excessive lateral and torsional vibration levels at higher harmonics. The compressor crankshaft was retrofitted with an internal damper. With the damper installed, the measured torsional vibration at the auxiliary end of the compressor crankshaft was approximately half the original amplitude. This eliminated the torsional-lateral interaction (TLI), which in turn reduced the compressor frame and cylinder vibration.

In the third case study, the compressor pulsation bottles had excessive vibration levels. The discharge piping was found to be misaligned and some of the clamps were not fitting properly. Adjustments were made to the piping and supports to correct alignment between the piping, bottles, and compressor cylinders. Straps were also added to the discharge bottles. Once these solutions were implemented, the vibration levels were reduced, and the reliability of the unit improved.

Introduction

Vibration problems with engine driven reciprocating compressor systems can hopefully be avoided in the design stage. API 618 provides standards for design. The manufacturers, packager, and end user may also have criteria that should be followed. Unfortunately, units may still experience issues after commissioning. This can sometimes happen because a similar compressor was successfully deployed previously, but now has slightly larger cylinders, different speed range, and operating conditions.

Some examples of engine vibration problems and solutions:

- Control panels – hard mounting to the engine can damage components inside. Control panels can be mounted with isolators or off mounted to address high vibration.
- Wiring harnesses – must be properly supported and isolated to prevent rubbing and wearing of insulation. High-frequency vibration can damage wiring. Need to use appropriate wrap and clamps.
- Cooling water piping – should be well supported and not resonant. Some units have poor routing of this piping with long unsupported spans.
- Anchor bolts / mounting method – must use proper torque for sufficient bolt stretch to prevent relative slipping of the engine feet to the top of the pedestal. Oil spills will reduce the friction force. Some engines use grout boxes while others use adjustable chocks. These chocks have less surface area than the engine feet but are reportedly easier to align.
- Pedestal design – must support the engine without natural frequencies near running speed or other significant excitation frequencies. Taller pedestals are sometimes needed to accommodate the engine oil pan or discharge bottles under the compressor cylinders. In that case additional gussets or outrigger supports may be required to stiffen the engine pedestal.
- Foundation – for reciprocating equipment it is generally recommended that the foundation be 3 to 5 times the mass of the equipment. Some units, usually rentals, are not mounted on foundations but are placed on a caliche pad often referred to as a “sand box” in the oil field. In this case, the skid must have sufficient mass and rigidity without relying on a foundation.
- Vibration at 1× running speed – the rotating and reciprocating components should be balanced.
- Vibration at 1.5× running speed – related to firing order / balance of four-cycle engine cylinders.
- Engine damper – the viscous fluid in an engine damper can degrade over time. Therefore, the engine damper should be checked and maintained on a regular basis. A failed engine damper can result in a broken engine crankshaft.

Some examples of compressor and piping vibration problems and solutions:

- Skid design must be sufficient to support the frame. The manufacturer will have allowable vibration limits for the compressor frame.
- Crosshead knocking could indicate several problems including high rod load and torsional vibration.
- Cylinder vibration can be due to mechanical resonances and/or acoustic resonances of the nozzles connecting the bottle to the cylinder. Cylinder or crosshead supports can be used to detune mechanical resonances. Nozzle orifice plates can be used to address acoustic resonances.
- Bottle vibration – need to use short nozzles connecting bottle to cylinder, avoid overhung suction bottles, support discharge bottles with straps and wedges.
- Piping strain should be avoided. The bottles need to be properly aligned and leveled.

High torsional vibration may cause problems such as:

- Failed couplings.
- Failed mechanical oil pumps.
- Failed dampers on engine and compressor crankshafts.
- Broken crankshafts.
- Increased vibration of compressor frame, crossheads, pulsation bottle, suction scrubber and attachments due to torsional-lateral interaction.

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Case Study I – Engine Driven Compressor Unit

High engine vibration was found on six dual-drive units located at a gas plant. Two mechanical natural frequencies (MNFs) associated with the engine were excited within the operating speed range. The first was a rocking mode, and the second was a twisting mode of the engine. These natural frequencies were measured during speed sweeps. Amplification from these resonances caused engine vibration that exceeded the manufacturer's allowable limit. Finite element analysis (FEA) was performed to evaluate modifications to improve the support and detune the system.

<u>Unit Description</u>

A general arrangement drawing is shown in Figure 1. These units consisted of:

- 16-Cylinder Engine Running on Natural Gas
  Operating Speed Range = 850 – 1,000 RPM
- Electric Motor, 5000 HP at 900 RPM, 4150 V
- 6-Throw Reciprocating Compressor
  (3) 14-1/8'' Cylinders (14.125'' Bore)
  (3) 9-5/8" Cylinders (9.125'' Bore)

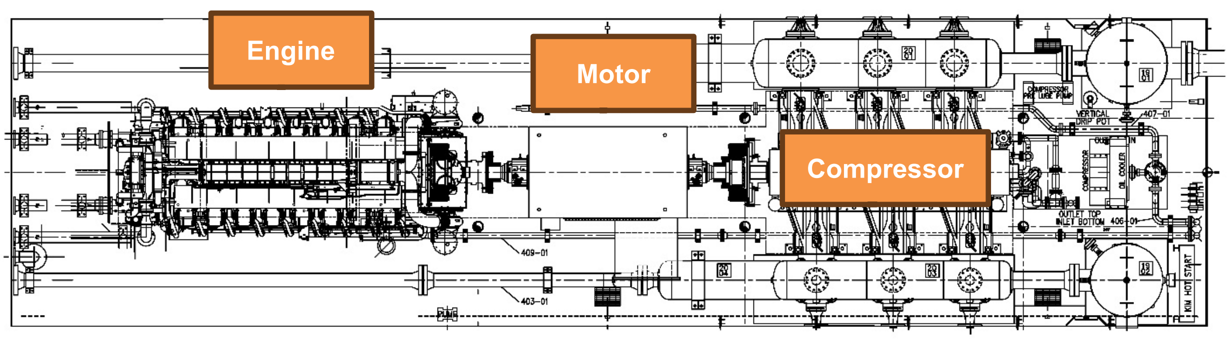

Figure 1. General Arrangement of Unit

<u>Engine Guidelines</u>

Vibration limits provided by the engine manufacturer are listed in Table 1. A limit for the 1.5× is not specified but it should normally be lower amplitude than 1× to meet the overall vibration limit.

Table 1. Allowable Engine Frame Vibration

Engine Order	Allowable Vibration (mils pk-pk)
0.5×	5.0
1×	5.0
Overall	8.5

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Testing

Speed sweep tests (850 – 1,000 RPM) consisted of acquiring vibration at nine points on the engine frame centerline (CL), engine feet, and top of pedestal with the compressor fully loaded. The speed test results help identify natural frequencies (i.e., resonant speeds), and the maximum amplitudes of vibration for comparison to the manufacturer's specification. With the compressor fully loaded, the engine speed was gradually brought to 1,000 RPM and then back down again to 850 RPM. A ramp rate of 60 RPM/minute was programmed into the control panel to give smooth, continuous speed sweeps.

The overall (unfiltered) vibration on the engine frames of all six units exceeded the manufacturer's vibration guideline of 8.5 mils p-p within the speed range. The elevated vibration primarily occurred at the 1× and 1.5× harmonics. Figure 2 shows a comparison plot of each unit for the 1× harmonic (top plot) and the 1.5× harmonic (bottom plot).

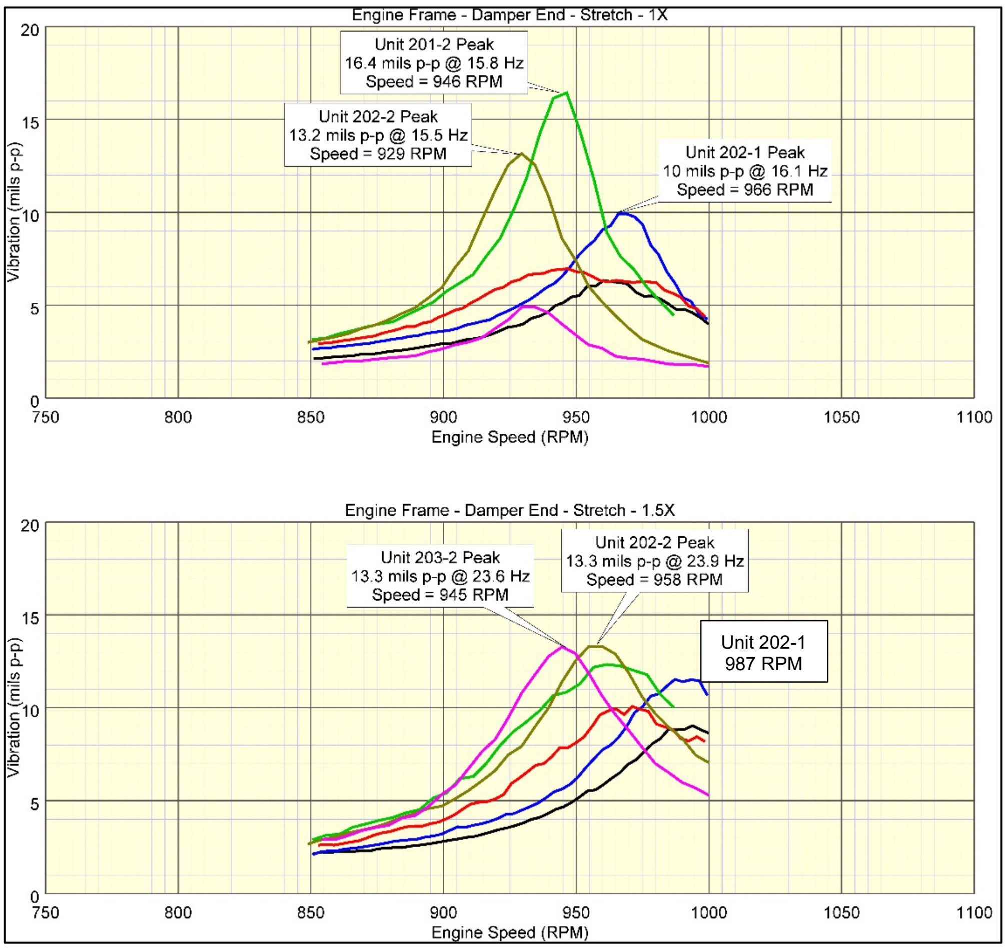

Figure 2. Engine Vibration at 1× and 1.5× Harmonics - Speed Sweep Tests

- Stretch direction vibration (i.e. perpendicular to crankshaft) at the 1× harmonic was high at all locations, while vibration at the 1.5× harmonic in the same direction was only high at the damper and flywheel ends of the engine frame.
- The vibration data indicated two MNFs of the engine, which were resonant with the 1× and 1.5× harmonics.
    - The maximum response due to first MNF occurred between 929 and 966 RPM, which corresponds to 15.5 - 16.1 Hz, with a peak amplitude exceeding 16 mils p-p at 1× running speed. The peak amplitudes at the 1× harmonic varied somewhat between units.

- The maximum response due to the second MNF occurred between 945 and 987 RPM, which corresponds to 23.6– 24.7 Hz, with a peak amplitude of 13.3 mils p-p at 1.5× running speed. The amplitude of this harmonic was more consistent between units than the 1× harmonic.
- A slight variation in natural frequencies was observed among the units and could be due to differences in underlying soil, grout contact, skid weldments, adjustable chock bolt preload and auxiliary equipment weight.

An operating deflection shape (ODS) test was performed on each unit while operating at constant speeds near the previously identified resonances so that the mode shapes of the natural frequencies would be measured. ODS is a method of visualizing vibration mode shape at a specific frequency. A geometric representation (model) was created from physical measurements as shown in Figure 3.

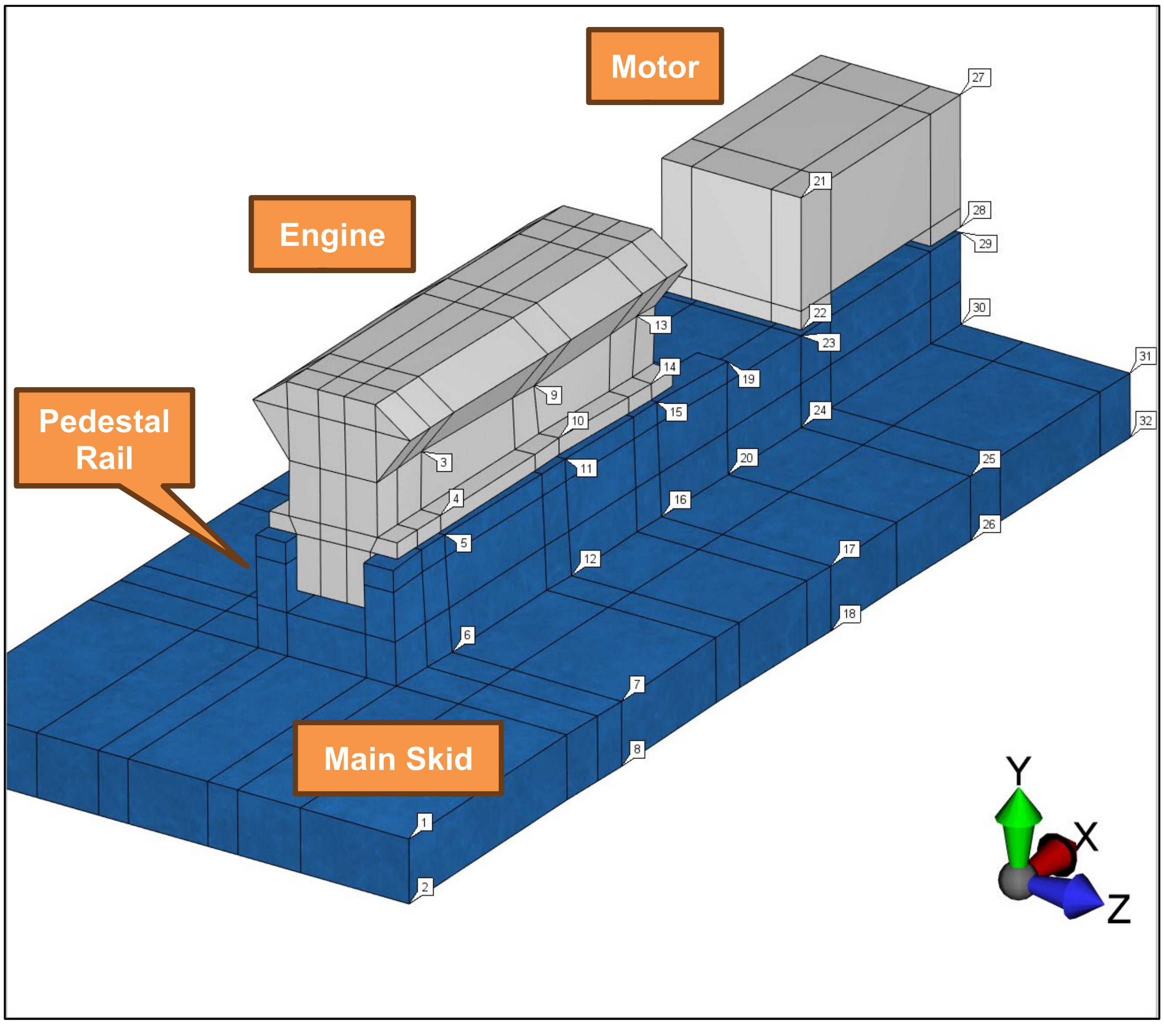

Figure 3. Operating Deflection Shape Test Locations

A roving tri-axial accelerometer was used to measure vibration in the horizontal (X), vertical (Y), and stretch (Z) directions at various locations on the engine, motor, pedestal and skid. The data were phase-referenced to a stationary accelerometer located on the engine frame so that it was also measuring the engine firing half-order. Animations were then generated to visualize the motion and help identify any areas of structural flexibility or looseness.

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Response at 1× Harmonic

As shown in the deflection still plots in Figure 4, the response at 1× running speed involved the engine frame moving in the stretch direction. The skid base remained relatively stationary while the pedestal rails under the engine were bending. The difference in motion between the engine mounting feet and the top of the pedestal was relatively small (less than 1 mil p-p) and therefore was not indicative of slippage of the chock mounts.

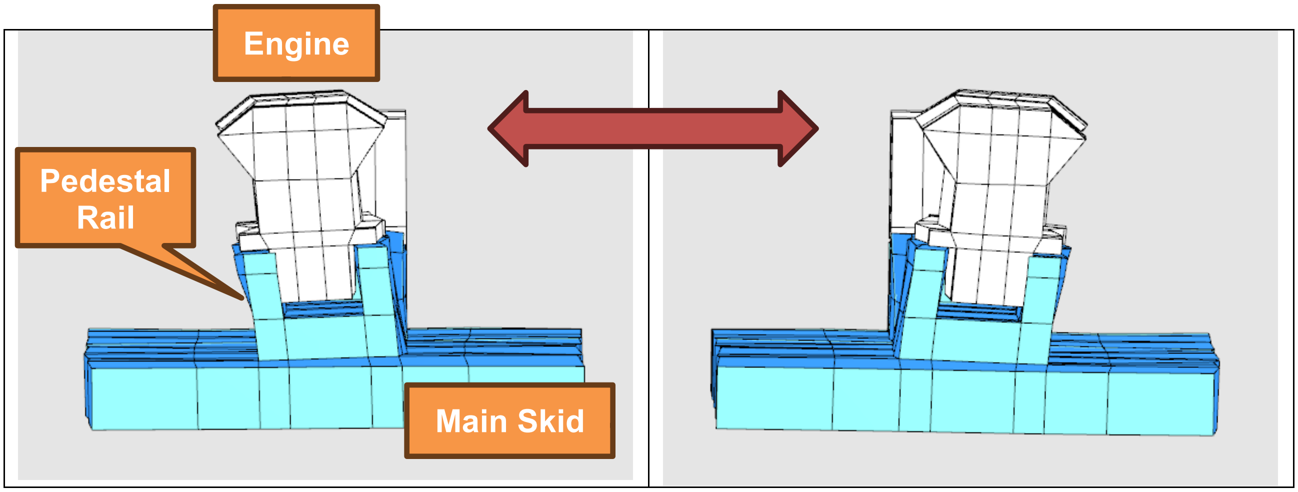

Figure 4. ODS of Engine 1× Run Speed

The ODS also showed that the vibration at the center of the engine was about a quarter of a cycle out-of-phase with the vibration at the ends of the engine frame. This behavior was due to the phasing of the engine throws and their individual unbalanced loads, which is normal for an engine.

This rocking mode of the engine is typically the first MNF of an engine supported on rails, which acts to amplify normal levels of engine unbalance. Pedestals should normally be designed so that they have sufficient stiffness to raise the frequency of this mode to a safe margin above the maximum speed. An ideal separation margin would be 10% or more, but is not always possible.

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Response at 1.5× Harmonic

Figure 5 shows the deflection still plots for the 1.5× engine harmonic. This is the engine yaw mode, with twist about the vertical axis. The skid base was relatively stationary with the pedestal rails bending, but with the damper and flywheel ends out-of-phase. Motion at the center of the engine was minimal for this mode.

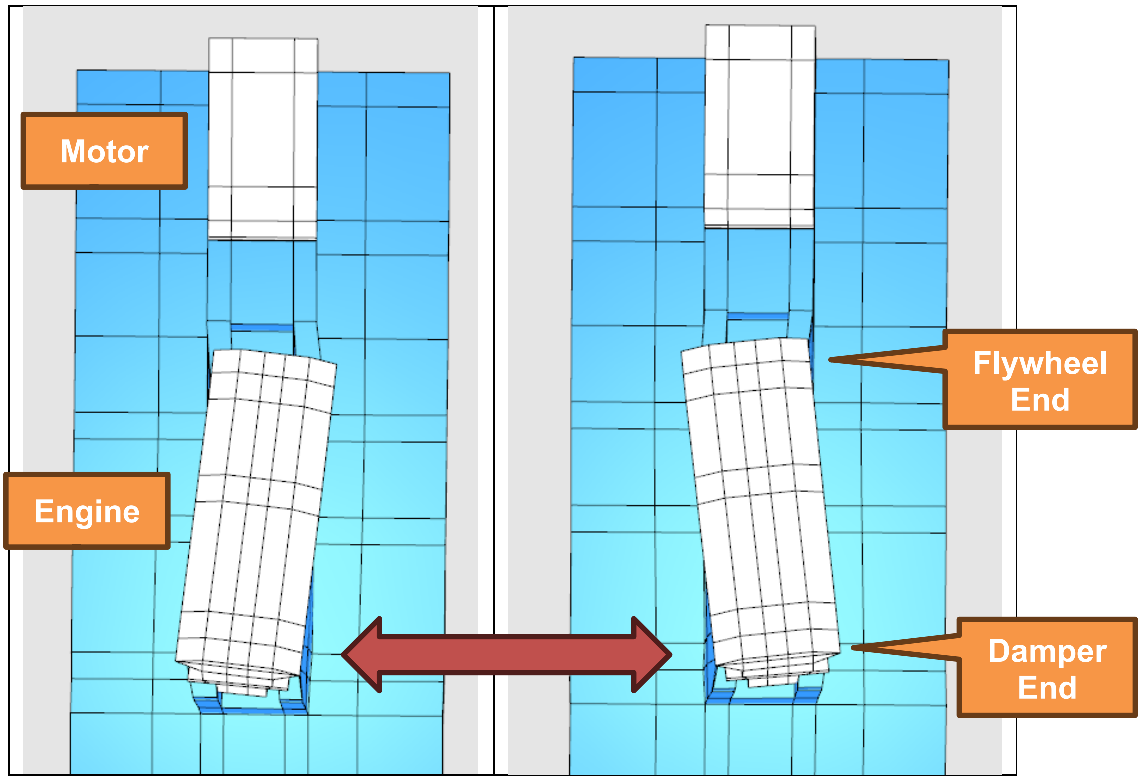

Figure 5. ODS of Engine 1.5× Run Speed

This yaw motion is typically the second or third MNF of an engine supported by rails. It is important that this resonant frequency also has a sufficient separation margin from the operating speed range to avoid amplified vibration of the engine. Again, an ideal separation margin would be 10% or more, but is not always possible. From a practical standpoint, the minimum required separation margin is what is needed to reduce the vibration to an acceptable level.

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Engine Vibration at Minimum and Maximum Speed

Engine vibration was measured while operating at minimum and maximum speeds with the compressor loaded. Figure 6 shows that all individual harmonic peaks were below 5 mils p-p at 850 RPM (minimum speed).

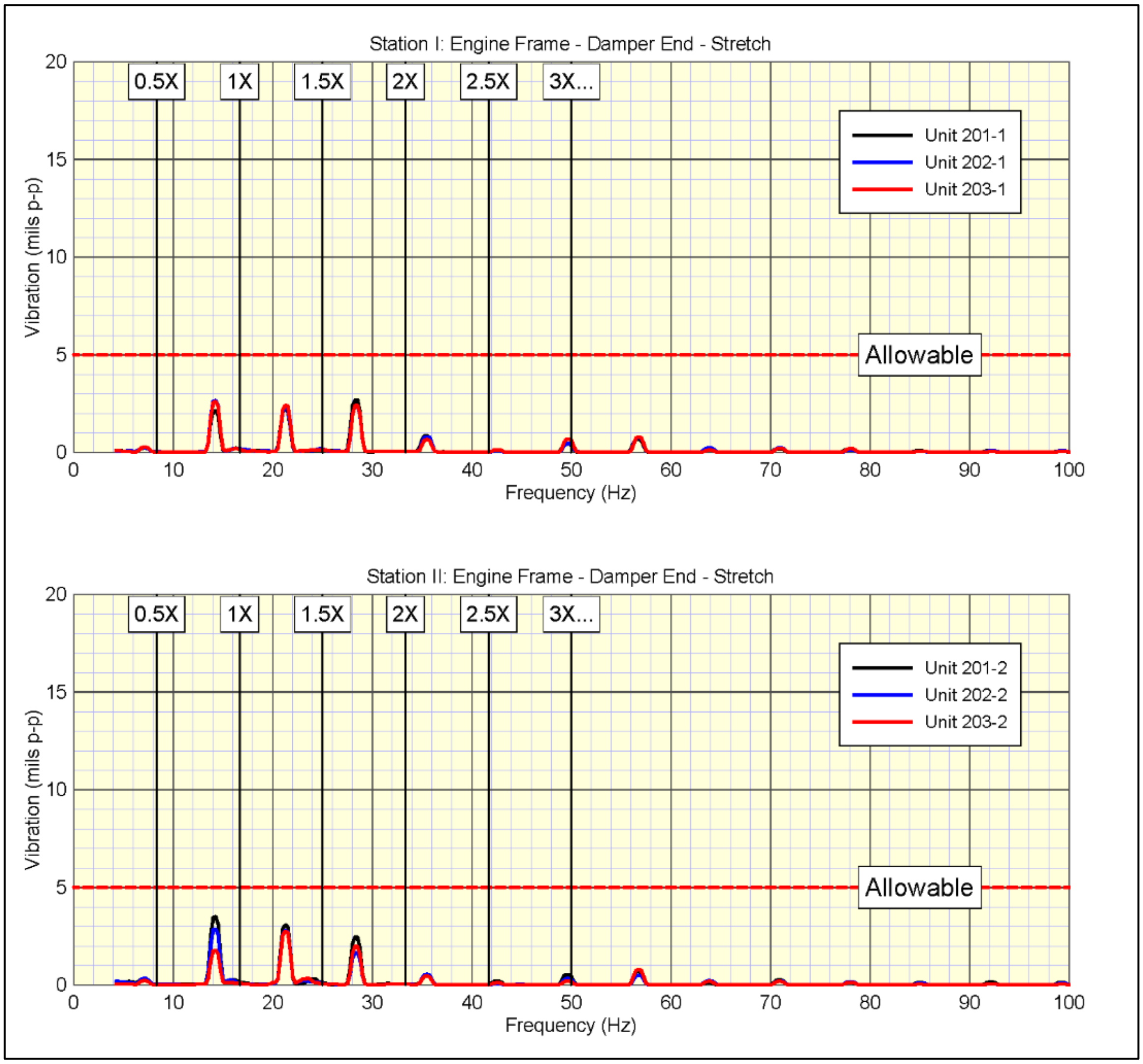

Figure 6. Engine Vibration at 850 RPM

Figure 7 shows that the 1.5× harmonic on all units exceeded 5 mils p-p when operating at 1,000 RPM. Engine vibration met the allowable at all other harmonics.

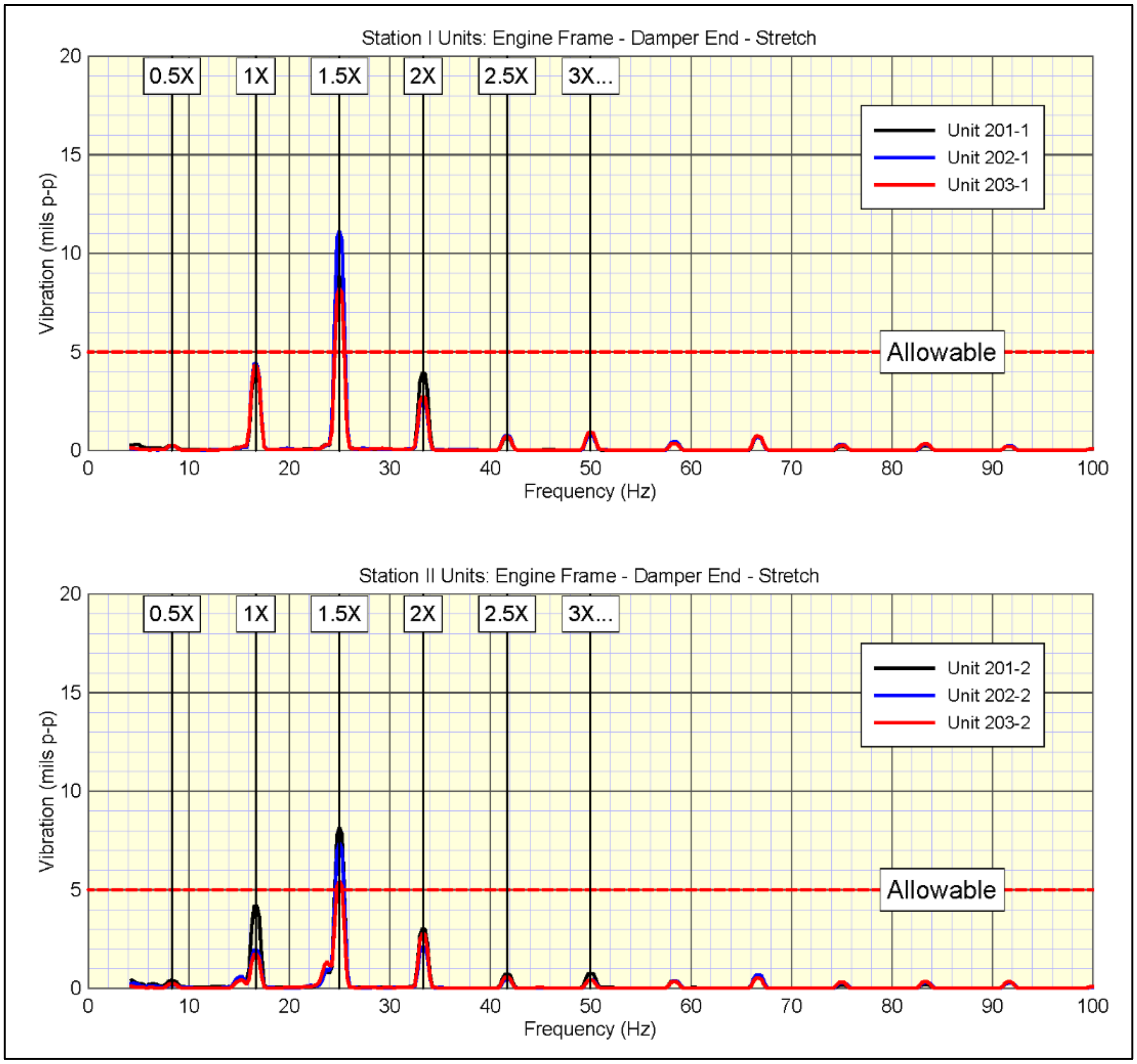

Figure 7. Engine Vibration at 1,000 RPM

Conclusions

The root cause for the high engine vibration was excitation of two MNFs within the middle portion of the operating speed range. When operating at resonance, normal engine excitation at 1× and 1.5× was greatly amplified. Steel structures normally provide minimal damping. For example, the amplification factor (AF) was estimated to be 33 or 1.5% critical damping for the first mode. This means the vibration was 33 times higher than if running well away from a resonance (with sufficient separation margin).

The mounting height of the engine was dictated by the oil pan and the diameter of the discharge bottles under the compressor. In general, it is best to minimize the height of the rails under the engine to keep the center-of-gravity as low as possible. Stiffer rails would mean that the MNFs are as high as possible. Since the existing rails under the engine could not be shortened, additional supports were necessary to increase stiffness and therefore raise the two MNFs so there would be sufficient separation margins.

Unbalanced engine forces can cause vibration at 1× running speed. Sources of unbalance could include engine

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components, flywheel, coupling, and clutch. The amount of unbalance varies between units within some tolerance. The MNF of the engine is excited by the sum of the unbalance from the engine damper, crank assembly, flywheel, coupling and clutch.

The response amplitude due to the 1× harmonic was observed to have variation among the units. Every time the engine-to-motor clutch is re-engaged, it will do so at a random relative angle. Therefore, the combined unbalance could change each time as will the vibration at 1× speed. This would make it difficult to balance the engine in the field. Therefore, it is important to detune the resonance by adding support stiffness.

Firing order of the four-cycle engine can cause vibration at 1.5× running speed. The excitation at 1.5× was consistent among the units when the compressor was fully loaded, and the vibration exceeded the guideline. It was previously demonstrated that the vibration amplitude at the 1.5× harmonic varied with load and is thus related to gas forces. Therefore, the 1.5× harmonic cannot be "balanced out" like the rotating and reciprocating parts of the engine.

<u>Recommendations</u>

- The short-term recommendation was to operate the units at minimum or maximum speed (preferably at minimum speed) until the resonances were corrected. For these two operating speeds, the engine vibration was much lower than when operating at resonance within the middle of the speed range. However, the plant really needed to operate at maximum speed to make production.

- Install vibration sensors on the engine frame and trend the overall vibration levels.

- Stiffen the engine pedestals in the stretch direction. Figure 8 shows some examples of various modifications from other projects.

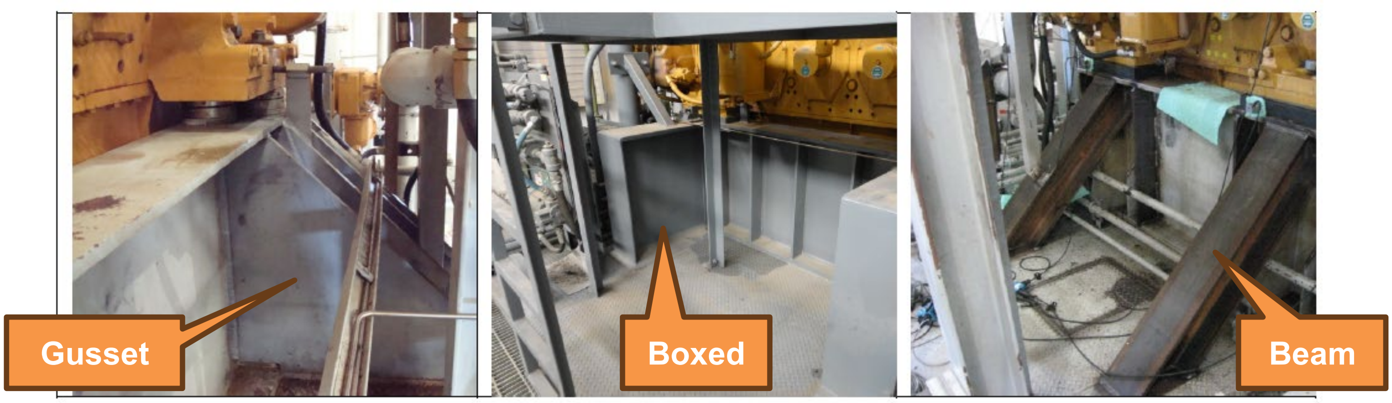

Figure 8. Examples of Various Skid Modifications

- Perform a finite element analysis (FEA) to evaluate possible modifications before field implementation.
  - The model should be normalized to match the two natural frequencies corresponding to the resonant speeds found from the vibration tests.
  - The mode shapes of the calculated MNFs should match the ODS animations.
  - The analysis should include expected values of engine shaking forces, per the manufacturer's catalog. The analysis can then be used to determine if these forces are sufficient to cause the measured vibration levels.

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Finite Element Analysis

Finite element analysis (FEA) was used to evaluate possible structural modifications. The results showed that extending the 0.75" thick middle gussets out to be 45°, as shown in Figure 9, would shift the two offending MNFs to a safe margin above the speed range so that resonance would be avoided, and vibration would be significantly reduced to an acceptable level.

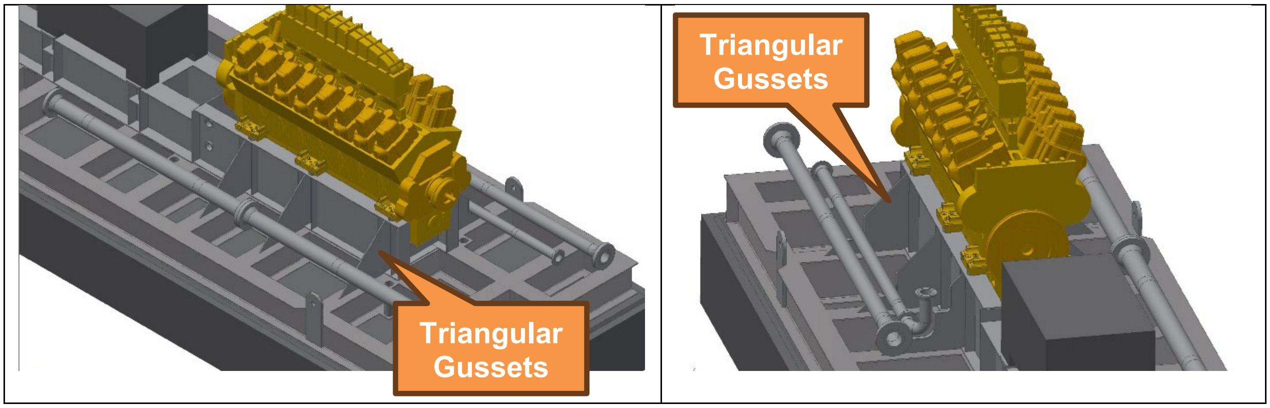

Figure 9. Initially Proposed Skid Modification

The packager initially confirmed that the proposed design modification should clear the process piping as well as the starter exhaust pipe. However, after further evaluation, there were too many interferences with existing piping. Therefore, a simpler approach was requested. Six struts were analyzed and predicted to work. Again, there were interferences with the starter on the side of the engine, so only four struts were initially installed on a single unit and tested.

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Several problems were found with the struts installed. The overall vibration on the flywheel end of the engine frame was still too high at maximum speed. The problem was that the first mode was pushed into the 1.5× range, and the second mode into the 2× range. The four struts were not as effective as predicted by the FEA. Figure 10 shows the vibration response data and slice plots. Five struts were modeled as shown in Figure 11.

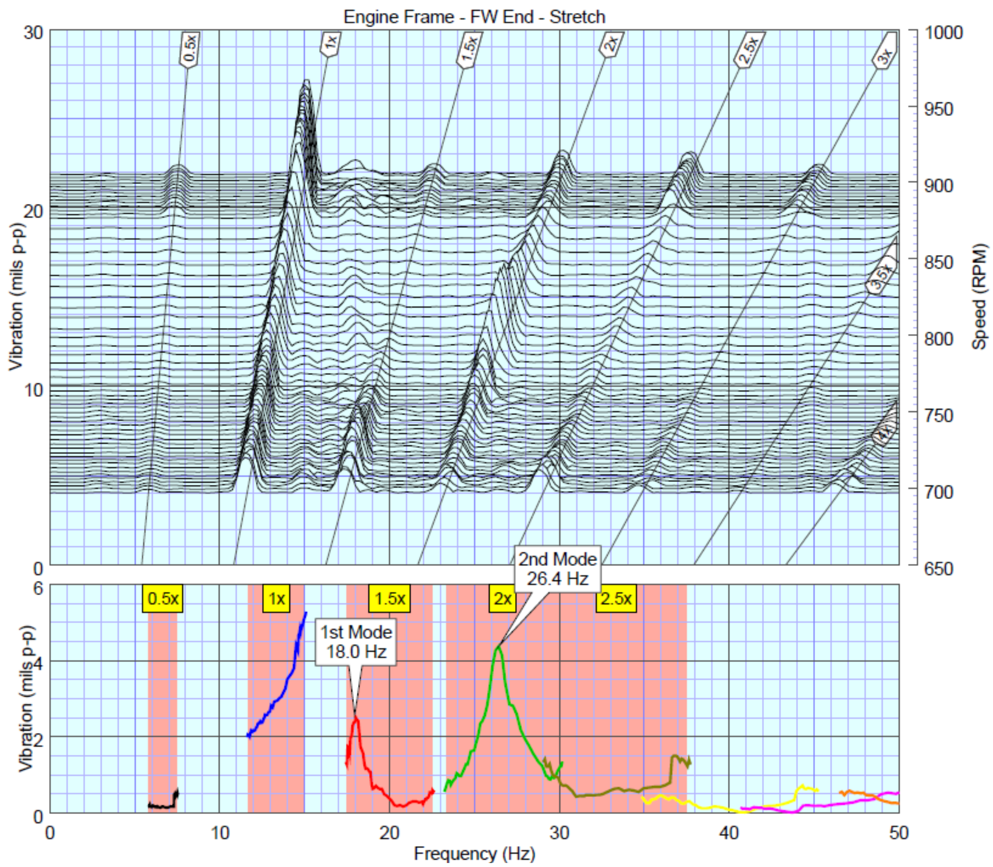

Figure 10. Vibration Response with Four Added Struts

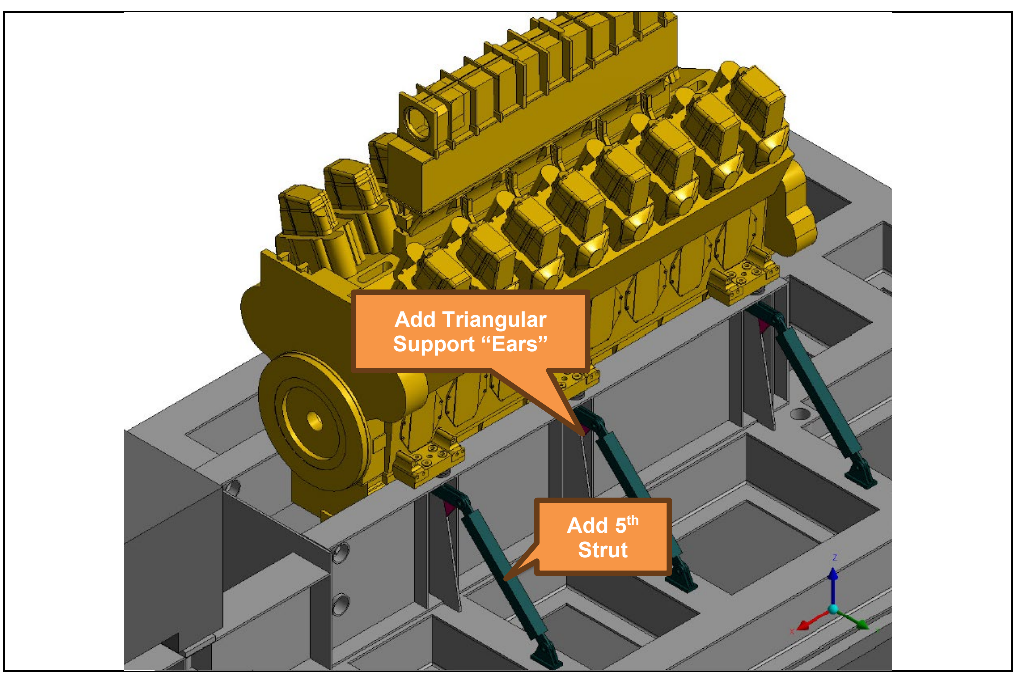

Figure 11. Model with Five Struts

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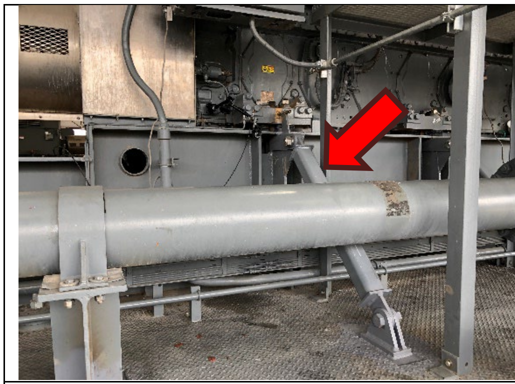
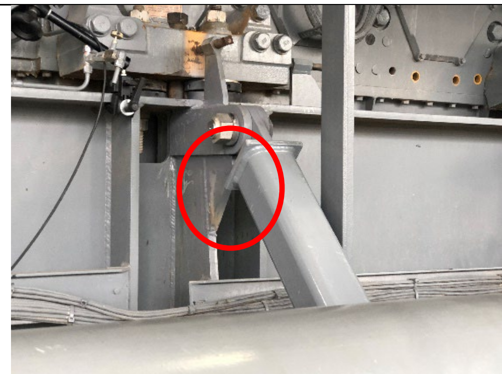
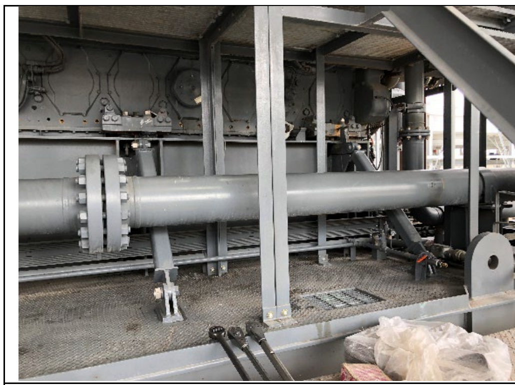
Additional modifications were made in the field. These included: adding triangular plates to stiffen the "ears" and a 5th strut to the engine pedestal as shown in Figure 12.



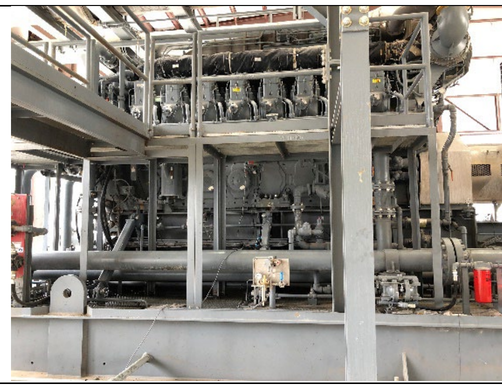
Support #5 Added Near Engine Flywheel



Triangular Pieces Under Attachment Points



Two Supports at Center and Damper End



Two Supports at Center and Damper End

Figure 12. Additional Skid Modifications

Vibration tests were repeated with a total of five struts installed. Triangular pieces were added under all five attachment points. In summary, the engine had acceptable vibration levels throughout the entire operating speed range of 850 – 1,000 RPM. There were no speed restrictions on the unit.

Engine speed sweeps were performed for each of the four compressor load steps. With the additional modifications (5 struts), the first two natural frequencies increased as intended to above 1× maximum speed (18.8 – 18.9 Hz), and above 1.5× maximum speed (27.2 – 27.7 Hz).

These values compared well to the predicted natural frequencies of 20.0 and 28.2 Hz from FEA. Table 2 summarizes the measured and predicted mechanical natural frequencies for various modifications.

Table 2. MNFs with Modified Pedestal

	Measured (4 struts)	Predicted (4 struts)	Measured (5 struts)	Predicted (5 struts)
1st MNF	18.0 Hz	19.6 Hz	18.8-18.9 Hz	20.0 Hz
2nd MNF	26.4 Hz	27.1 Hz	27.2-27.7 Hz	28.2 Hz

The unit was much improved with five struts on the engine pedestal. The 1× response at maximum speed (1,000 RPM) reached 5.5 mils p-p on the flywheel end of the engine. The maximum response at the 2× harmonic was 4.5 mils p-p at the damper end. Figure 13 compares the "as found" condition on the flywheel end (maximum of 20 mils p-p) with the final condition (maximum of 5.5 mils p-p). Because these modifications were successful, the struts were then used to modify other existing units already installed at this plant.

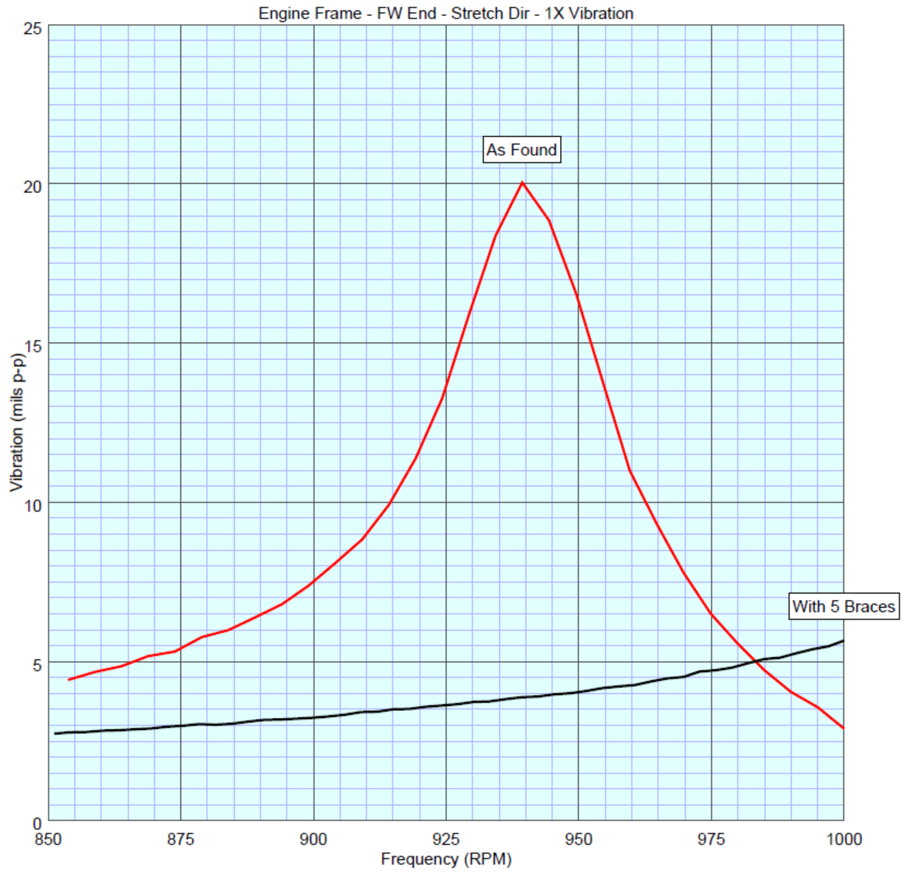


Figure 13. Comparison of 1× Engine Vibration as Found vs. Modified

For new packages, a gusset plate design was implemented as shown in Figure 14. These have been shown to be effective in tuning the structural natural frequencies above 2× running speed and keeping the engine vibration below the manufacturer's allowable limit.

Figure 14. Engine Pedestal Modifications with Support Plates

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Case Study II – Engine Driven Compressor Unit

An engine driven compressor unit at a natural gas pipeline station was experiencing high lateral and torsional vibration. This unit consisted of:

- 16-cylinder natural gas engine
  Operating speed range = 850 to 1,000 RPM
- Disc pack coupling
- 6-throw reciprocating compressor
  Single stage with cylinder bore of 12.25-inch
  Head end value unloaders (HEVU's) on cylinders 1, 3, 5
  Head end fixed volume clearance pockets (HE FVCP's) on all cylinders

Torsional Vibration

Torsional vibration was measured at the auxiliary end of the compressor crankshaft. The encoder location is shown in Figure 15. The system had a torsional natural frequency (TNF) near 80 Hz that was being excited by 5× running speed. This resulted in torsional vibration exceeding the manufacturer's limit, meaning damage to the oil pump and drive chain would likely occur if continuously operated on the resonance.

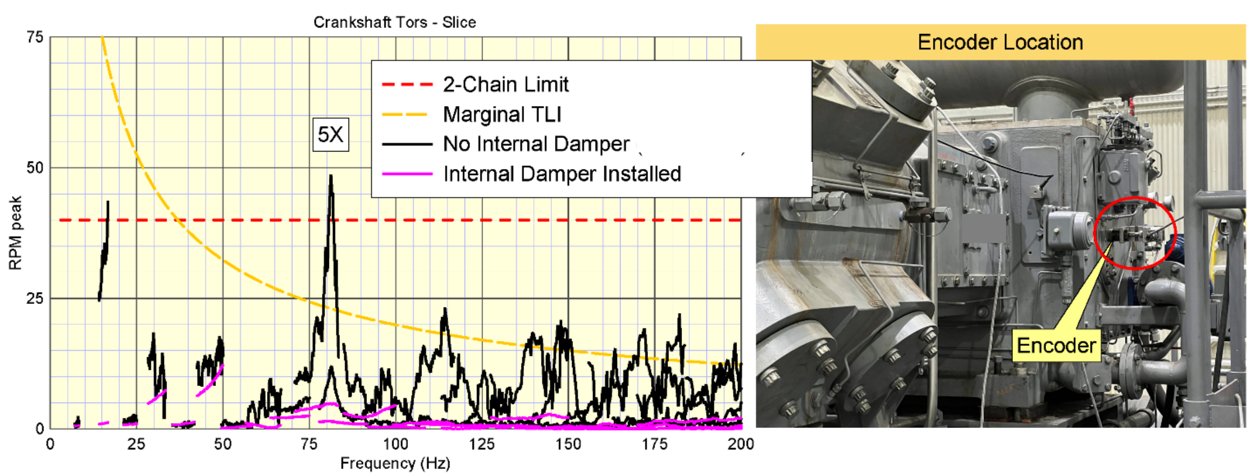

Figure 15. Comparison of Torsional Vibration Levels

The torsional vibration was greatly reduced by installing the compressor manufacturer's internal damper to the auxiliary end of the compressor. The peak amplitude at 5× running speed went from 48 RPM 0-p (black line) to 5 RPM 0-p (magenta line), which is well below the limit (40 RPM 0-p) and below the guideline for torsional-lateral interaction (TLI).

Compressor Frame Vibration

The reduction in torsional vibration at 5× running speed after installation of the internal damper also reduced the lateral vibration of the compressor frame at 6× running speed. This is because TLI can occur at ±1 order of the torsional vibration harmonic. Figure 16 shows spectra slices extracted from the waterfall plots of the compressor frame horizontal vibration with and without the internal damper. The significant reduction in 6× running speed amplitude helped reduce buzzing of some of the small-bore attachments on the compressor.

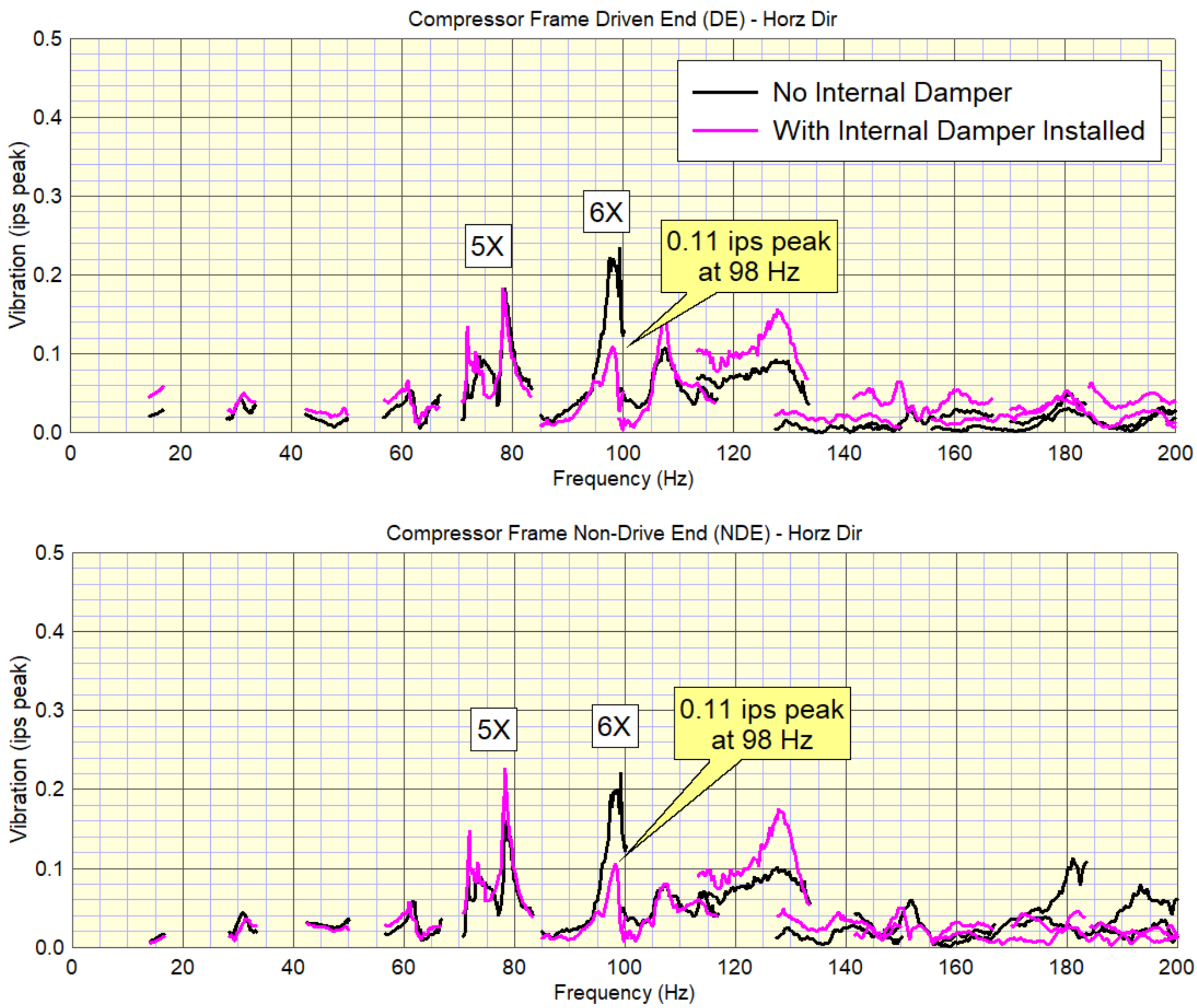

Figure 16. Slice Plots – Lateral Vibration Measured on Compressor Frame

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Compressor Cylinder Vibration

Vibration was also measured on the compressor cylinders. The overall vibration levels on cylinders 2 and 6 exceeded the manufacturer's guideline of 1 ips 0-p at certain speeds in the vertical direction, reaching a maximum amplitude of 1.1 ips 0-p. The vibration primarily occurred at 5× running speed. The amplitudes in the vertical direction were essentially unchanged with and without the internal damper.

Figure 17 shows the waterfall plot obtained on compressor cylinder 2. The peak response was 0.95 ips 0-p at 77 Hz (5× at 924 RPM) and indicates a resonance. Elevated 5× vibration was also observed at other locations on the compressor-manifold system and piping. This could lead to small-bore attachments buzzing and clamps loosening.

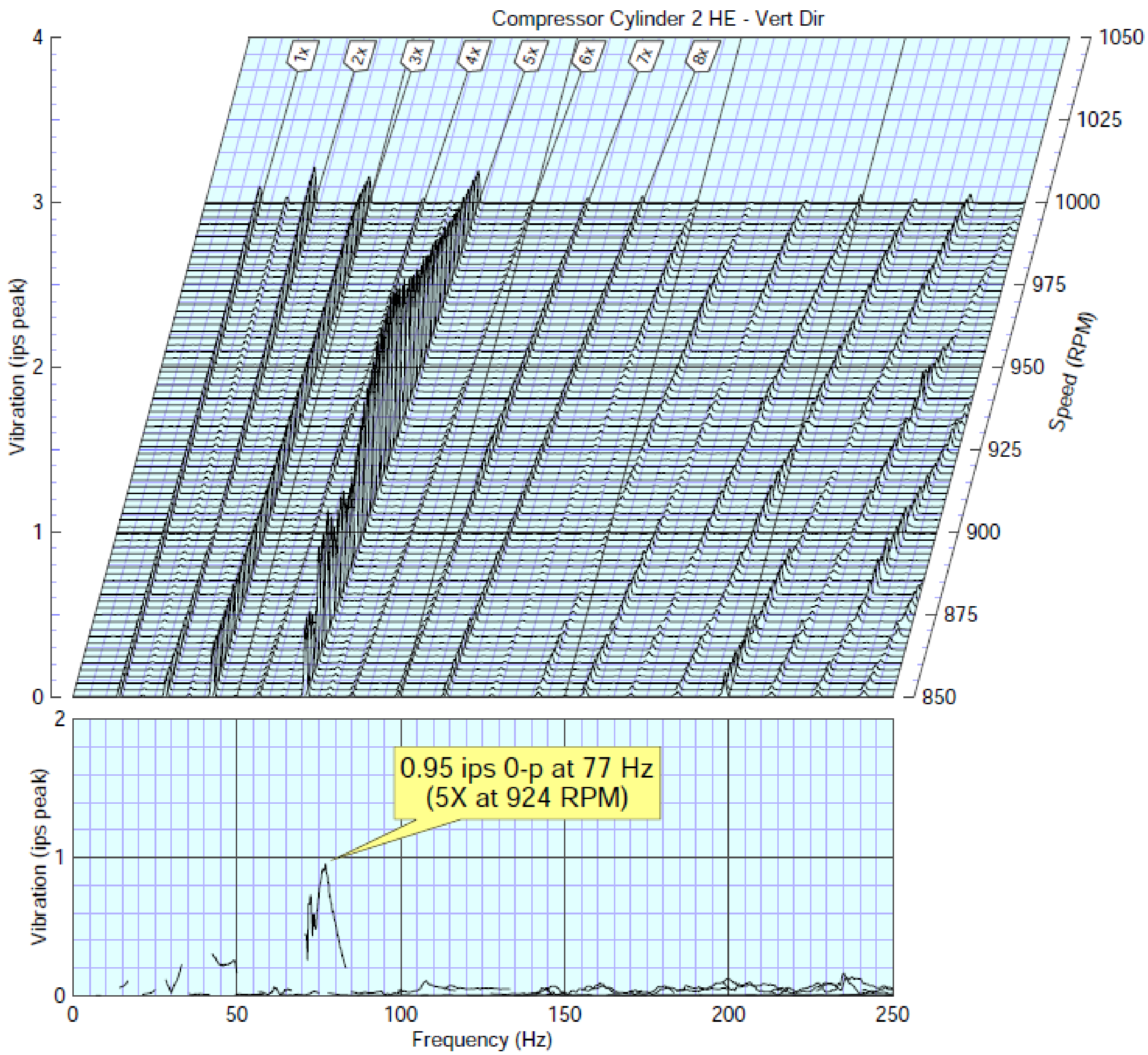

Figure 17. Waterfall Plot - Compressor Cylinder 2

The vibration peak at 77 Hz (5× running speed) was likely due to the cylinder mechanical natural frequency previously measured before the compressor internal damper was installed. While installation of the damper reduced torsional vibration and lateral frame vibration, it did not reduce the cylinder vertical vibration. Cylinder vibration in the vertical direction can also be caused by acoustic resonances in the nozzles as discussed later in this paper.

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Operating Deflection Shape

Further testing was performed to evaluate the vertical motion of the compressor cylinders. Operating deflection shape (ODS) data were obtained on the throw 2/4/6 side of the compressor while operating at a constant speed of 925 RPM, which was the approximate speed that the peak vibration occurred (resonant frequency was 5× running speed). A roving tri-axial accelerometer in conjunction with a phase reference was used to collect vibration amplitude and phase data at multiple points while the operating conditions were held constant. Figure 18 shows a still image of the exaggerated deflected shape at 5× running speed (77 Hz). The deflected shape involves out-of-phase twisting motion of the outer cylinders (throws 2 and 6).

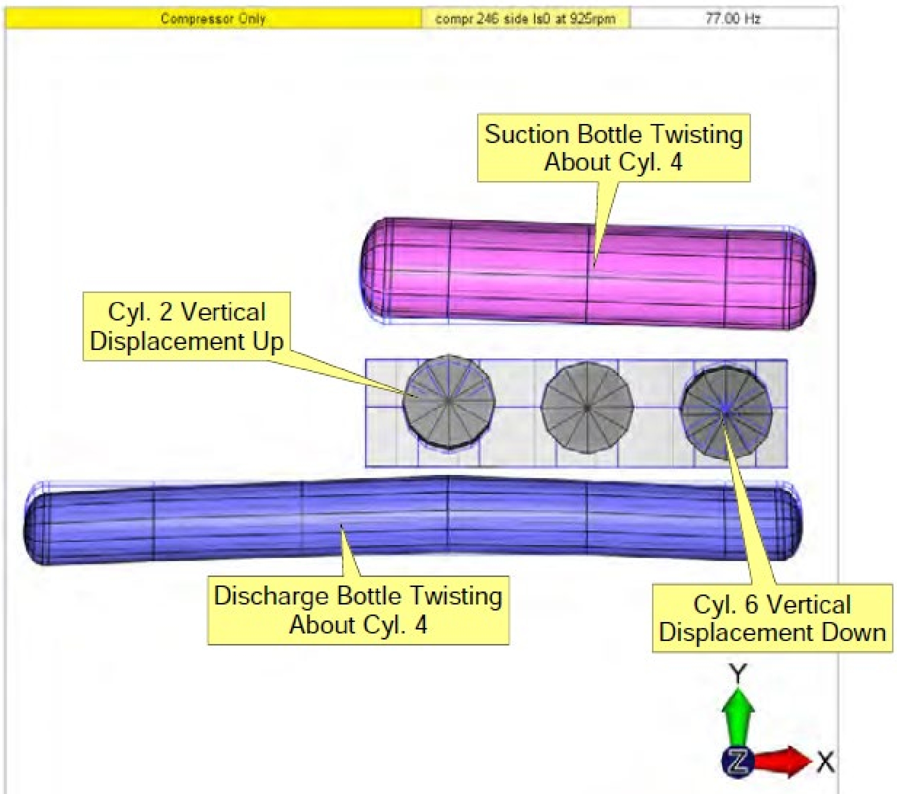

Figure 18. ODS at 5× Running Speed - Twisting of Compressor Manifold

This is a typical mode shape for this cylinder configuration; however, it is normally difficult to excite. Therefore, pressure pulsation induced shaking forces within the discharge nozzles were considered the likely source. This was based on the dimensions and hand calculations.

Conclusions

- With the internal damper installed in the compressor, the auxiliary end of the crankshaft met the torsional oscillation and TLI guideline. The highest speed fluctuation of the compressor crankshaft was reduced from 48 to 5 RPM 0-p at 5× running speed, which was well below the limit.
- Lateral vibration on the compressor frame was reduced from 0.22 to 0.11 ips 0-p at 6× running speed.
- Cylinder vibration in the vertical direction was still a concern at 5× running speed. Discharge nozzle pulsation could be the source of the excitation, not TLI. A more restrictive orifice in the discharge flange was recommended to help reduce the amplitude of the pressure peak. Normally, the orifice plates are sized to provide 0.25% pressure drop at the cylinder flange / discharge nozzle and at the discharge bottle outlet.
- The discharge bottle supports should also be improved by using recommended straps and wedges with a gasket-type material (e.g., Fabreeka) to prevent metal-to-metal contact.

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Case Study III - Eliminating Piping Strain and Aligning Bottles

This engine driven compressor unit had a history of failures caused by vibration and was actually the same unit as discussed in Case Study II. These failures began soon after the unit went into service and included damage to the main wire harness on the engine, the wedge chocks under the discharge bottles becoming loose and damage to three crossheads. These failures necessitated that the unit be analyzed for pulsation and/or vibration.

An optical assessment of the unit elevations at key points on the frame and bottles revealed the alignments of the compressor frame, bottles and lead line piping were inconsistent from point to point and between points at the same position but on opposite sides of the unit. The issues observed were tabulated and shown in Figure 19. The unit was also evaluated for pulsation and vibration after repairs were performed.

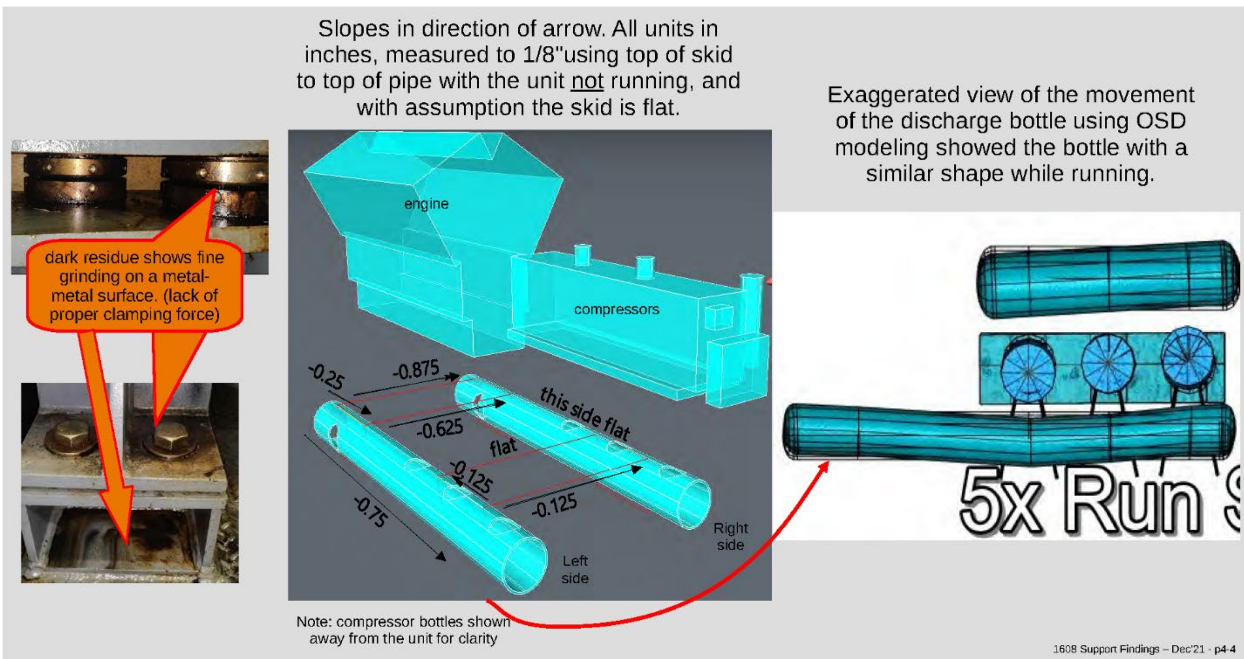

Figure 19. Left Discharge Bottle Warping – Grinding at Connection Points

Optical alignment checks were made on the flange and nozzles flanges of the suction and discharge bottles. A close inspection of the skid and its fabrications revealed issues with the alignment of the chocks under the discharge bottles, the support at the base of the bottle straps and lead-line chocks loosening.

Analysis of the bottle straps showed that the lengths vary from 60 inches to 62.75 inches, and the lead-line strap lengths vary from 46 inches to 49.5 inches. Bottle straps in excess of 60 inches and lead-line straps in excess of 46 inches are too long to be installed properly. For example, when the straps are not of the correct length, it does not press against the top of the pipe or bottle to apply clamping force and does not leave a gap between the flats when torqued. When the straps are at their optimal length, there should be no gap at the top of the pipe or bottle and only a small gap at the flats. See Figures 20 and 21 for additional findings.

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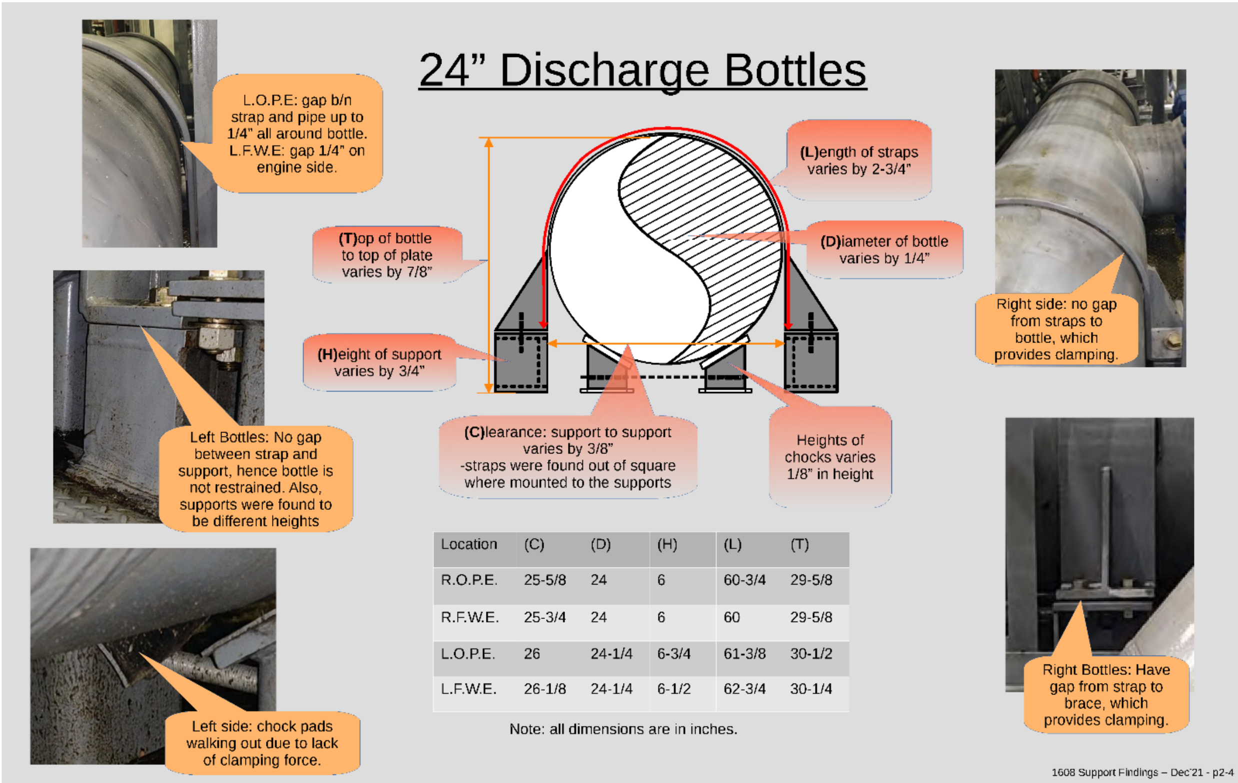

Figure 20. Straps Fit on Discharge Bottles

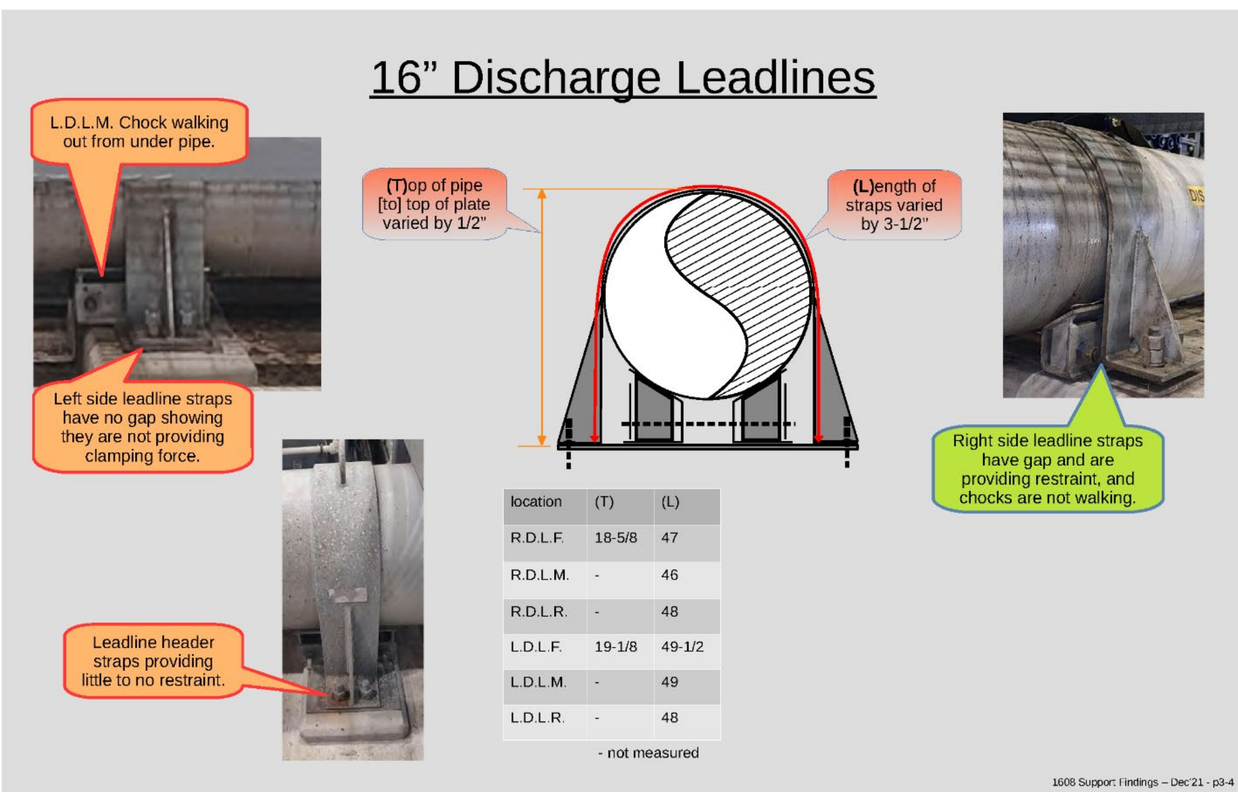

Figure 21. Straps Fit on Discharge Piping

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Optical alignment analysis was performed on the unit at the top of the compressor, at the crosshead guides, compressor cylinders, suction and discharge bottles, and suction and discharge lead-lines. These checks showed the unit was not sitting in a singular plane. The elevations of the discharge bottle on the right side varied significantly when compared to the elevation of the left side. There were also significant differences in elevation at comparable locations between the left and right sides of the compressor frame, the lead lines and the bottles (see Figure 22).

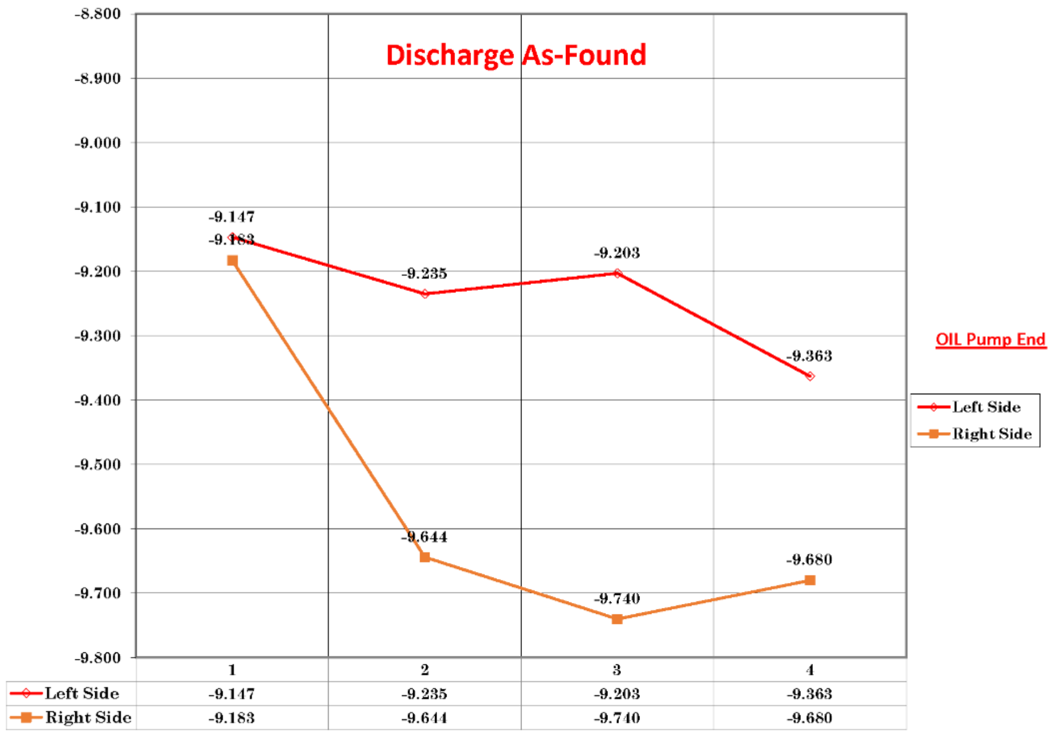

Figure 22. As-Found Elevations on Discharge Bottles

Elevation readings at the top of the compressor frame showed significant variation in the reading along each side (left to right and front to back). These findings were tabulated and plotted on graphs to give an idea of how the unit sat. The analysis showed the compressor frame to be twisted, with the elevation of the cylinders on the “even numbered side” cylinders (2, 4 and 6) running uphill from the flywheel to the oil pump end. On the “odd numbered side” (cylinders 1, 3 and 5) running uphill from the oil pump end to the flywheel end, hence the compressor frame was twisted across its centerline (see Figure 23).

By addressing these alignment anomalies, it was believed it could also improve the vibration of the unit. Compressor cylinders 1, 3, and 5 have head end unloaders and had higher vibration compared to the 2-4-6. At 1,000 RPM and load step 2, compressor cylinders 1-3-5 are moving in the vertical direction. The vibration amplitude was approximately 1 ips 0-p at 3× running speed, but the overall vibration was more than 1 ips 0-p when considering the other harmonics. At this operating speed, the compressor cylinder vibration was higher than the manufacturer’s guideline of 1 ips 0-p overall.

Previously, compressor cylinders 2 and 6 showed vibration levels in the vertical direction exceeding 1 ips 0-p. The highest vibration was occurring at 5× running speed due to a resonance near 77 Hz. After the repairs, the new vibration levels were:

- The vibration peaked at 48 Hz and 78 Hz, but with slightly lower amplitudes of 0.7 ips 0-p.
- The resonance near 48 Hz can be excited by 3× running speed. At 3× running speed, compressor cylinders 2-4-6 were moving in the vertical direction with a maximum amplitude of 0.7 ips 0-p.

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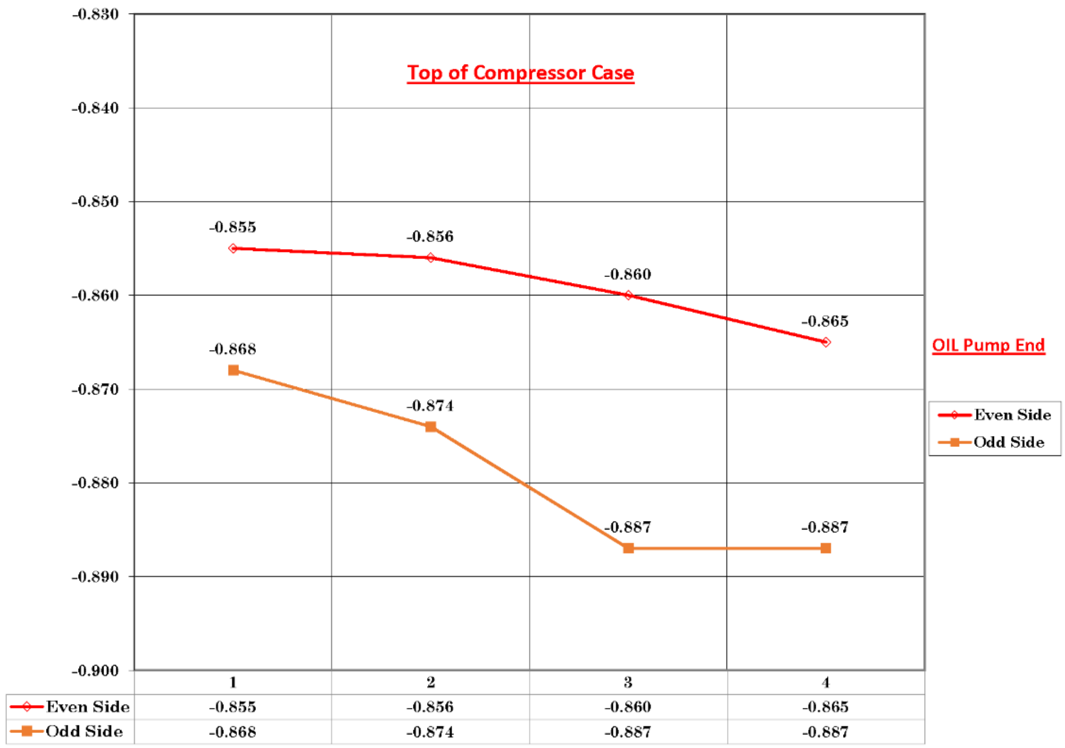

Figure 23. As-Found Elevations on Top of Compressor

A third-party was used to analyze the unit for alignment issues and to perform the repairs needed based on the findings. Their finding concurred with ETC. In addition, they identified that some crosshead guide bolts were too long to be torqued properly and had reached the bottom of the threaded holes before they could be tightened to the required torque values. Hence the crosshead guides were never properly torqued, and this contributed to three crossheads failing over a period of six years.

In addition to the findings, we wanted to analyze the nozzles and flanges on the suction and discharge bottles for alignment. The project included removing the suction and discharge bottles to inspect the nozzle flanges for alignment.

The bottle flange faces were found to be inclined to the nozzle's Z-axis to varying degrees, in both the X-axis and Y-axis. These values ranged between 0.030" to 0.150" across the flange faces. One bottle had all flanges outside the maximum tolerance for flange misalignment. The discharge bottles were sitting on wedge chocks and were not clamped. As part of the proposed solution, two large wedge chocks with straps were installed with ¼' thick Fabreeka lining between the bottle and chock and the straps and bottle. These were installed under each discharge bottle.

Figure 24 shows the location of all existing and proposed chocks with a circled X. A red circled X indicates where a strap was too long, and hence could not be properly installed. Those in green were of adequate length and properly installed.

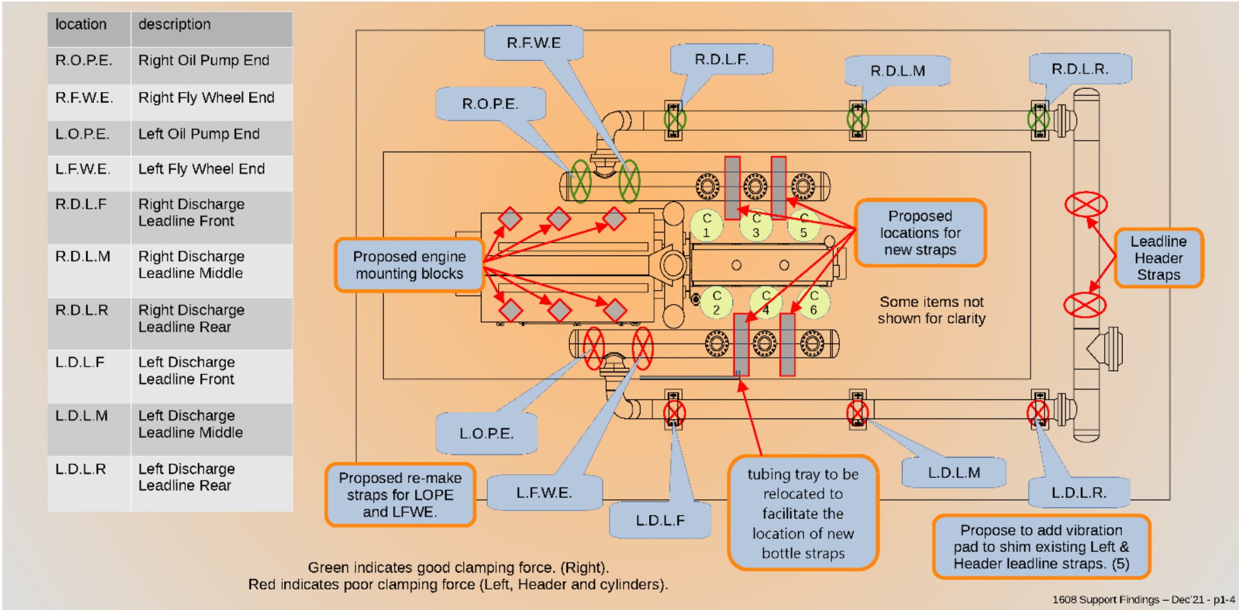

Figure 24. List of Findings and Recommendations to Address Vibration

ETC undertook many of the repairs at the end of 2023. After these repairs were completed, the results of the vibration survey showed improvement. The following items were addressed:

- Removed the bottles to rectify the fit of the flanges on the nozzles. All bottle flanges were brought flat to within 0.030" difference in height across the flange face.
- Installed new chocks and straps between the compressor cylinders as shown in Figure 25.
- Replaced the bolts on the crossheads that were too long, with proper length bolts.
- An upcoming project is planned to address the issues with the straps that were too long.

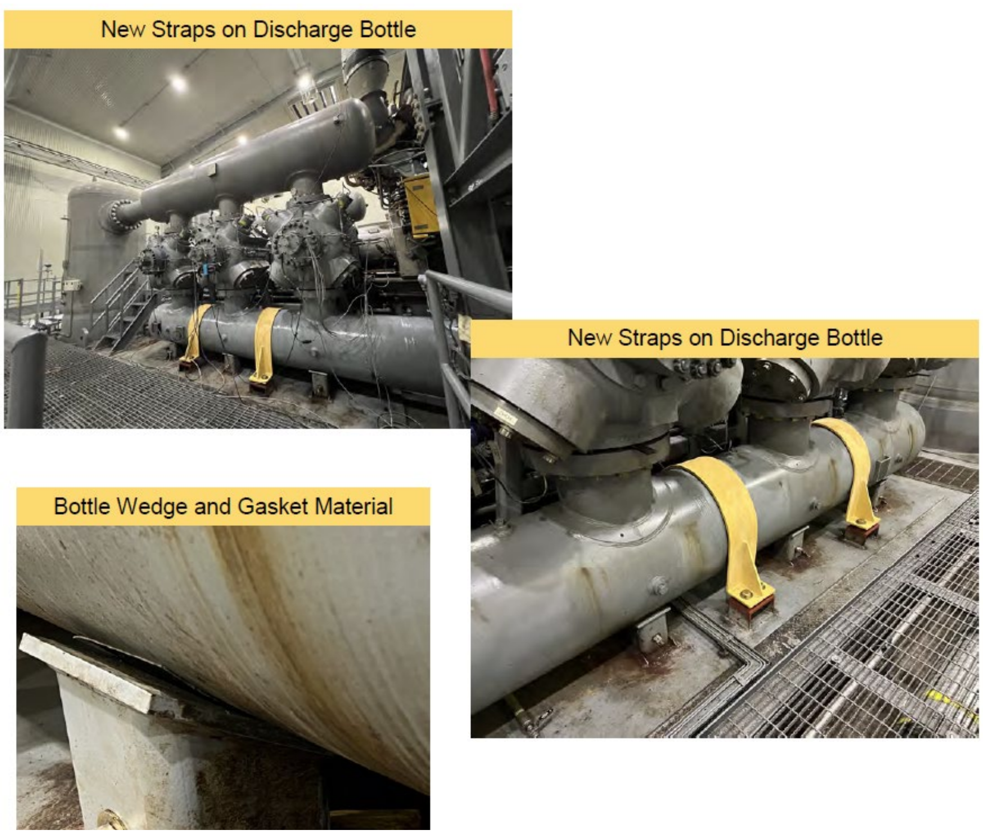

Figure 25. New Chocks and Straps Between the Compressor Cylinders

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With improvements to the piping alignment and the additional straps shown in Figure 25, the vibration of the discharge bottle was greatly reduced. At the outlet end of the discharge bottle, the vibration was previously 12-13 mils p-p at 5× running speed. It is now less than 3 mils p-p, which is considered acceptable for the discharge bottle.

In conclusion, some of the compressor vibration could be attributed to improper alignment to the compressor bottle flanges, improper lead-line and bottle straps lengths, chock strap base misalignment, and the need for additional discharge bottle chocks and straps. There was also a problem found with the stud torque on the crossheads that could have contributed to higher cylinder vibration.

Lessons Learned from the Case Studies

Engine driven reciprocating compressors need to be reliable over a range of speeds and load conditions.

- Mechanical natural frequencies of the engine mounted on the pedestal must have sufficient separation margins. Structural modifications to detune the MNF and reduce the vibration are easier to make during fabrication than after the unit has been installed. Finite element analysis can be used in the design stage to evaluate various options.

- Using an internal damper in the compressor can help control lateral and torsional vibration levels. When available, the internal damper option should be ordered for variable speed applications.

- Pulsation bottles and piping need to be properly supported and aligned. It is recommended to have good contact by using Fabreeka or similar material, making sure the straps are proper length, and wedges are tight when the unit is hot.

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