ATYPICAL RESULTS FROM IMPROPERLY SIZED & CHARGED PULSATION DAMPENERS

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ABSTRACT

At a facility with four quintuplex pumps, pulsation dampeners were being used to control pulsations in the discharge line. However, the initial dampeners were both undersized and undercharged for the application. As a result, in addition to some typically expected results, including high pulsations and frequent failures of the internal bladders, the effective volumes of the dampeners and lengths of piping in the system set up an acoustic natural frequency that caused significant safety concern and limited system operability. This natural frequency was in a range that could be excited by the pumps such that the presence of the dampeners in the system was actually causing even higher pulsation levels. This paper will look at the troubleshooting efforts including field testing and acoustic simulations. Results from the modified system will also be discussed.

INTRODUCTION

A discharge piping system, as shown in Figure 1, being supplied by four (4) positive displacement (PD) quintuplex pumps operating in parallel at Liberty Resources', end-user, site was experiencing excessively high vibrations and numerous failures.

Initially, the end-user worked with the pump and pulsation dampener vendors to troubleshoot the root cause of the vibration. First, the suction header was redesigned to allow for a reduction in the speed of the fluid entering the inlet of the pumps. This was based on the recommendation of vendors and personnel with 20 plus years of experience with PD pumps. The geometry and construction of the suction header allowed for gas to break out of solution and be removed from the working fluid before entering the pumps as an attempt to eliminate “free gas” induced cavitation.

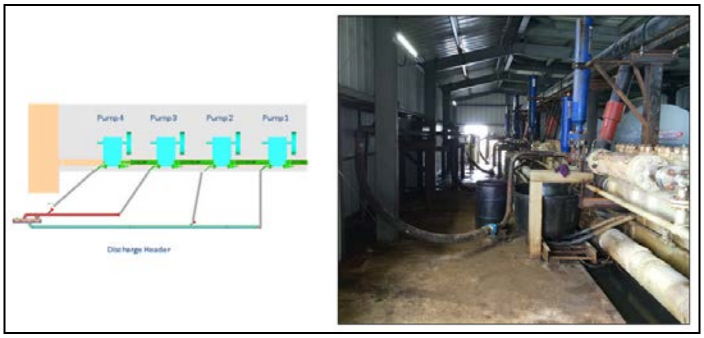

Figure 1: Facility Layout

The discharge header was also redesigned in an attempt to better control the fluid introduction from each pump into the piping. The discharge header was built by a well-known and reputable fabrication firm. A material change was made to more ductile steel as well as increasing the piping wall thickness. A large emphasis on quality control and inspection was placed on the welding and fabrication of the discharge header as the end-user was first told that improper welding was a contributing factor to the original failures. The constraint points of the discharge header were also modified and the running speeds of the pumps were adjusted to offset the running frequencies. All attempts failed in reducing the vibrations on the discharge piping, and as such failures still occurred, as shown in the examples of Figure 2.

Of note, there was a similar installation that existed at another nearby facility of the end-user. This location, however, has six (6) positive displacement quintuplex pumps operating in parallel, but has not experienced the same severity of vibrations or the failures. Both locations have the capability to operate with either crude oil or water as the process fluid.

Southwest Research Institute® (SwRI®), contractor, was tasked with performing field measurements of vibrations and pulsations at both facilities. The purpose for gathering this data was to help confirm the presence of pulsations, and provide data for a pulsation acoustic simulation analysis that was performed for both facilities.

Additionally, the contractor was also tasked with performing a pulsation analysis on the existing discharge piping systems for both facilities following the field testing to evaluate the piping system acoustics and pulsations, and to determine acceptable modifications to reduce pulsation energy. Therefore, a pulsation analysis as described by API 674 (C.1.3) was performed by SwRI for the discharge piping system of the set of four pumps at the first facility and the six pumps at the second facility. The majority of work is focused on the first site given the severity of vibrations seen there, so the remainder of this paper will focus on the work done for this facility.

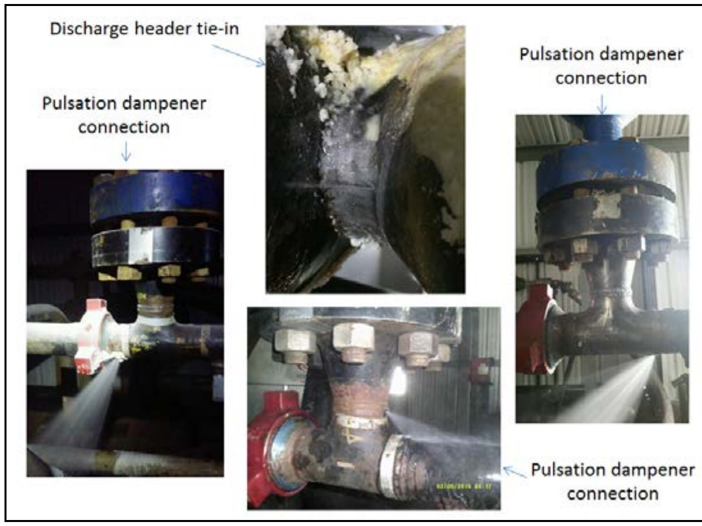

Figure 2: Examples of Recent Failures in Piping

INITIAL FIELD RESULTS

Data was taken in the field using dynamic pressure transducers, velocity probes and a sound level meter. At both facilities, the dynamic pressure transducers were installed at full-port valves that were placed in the lines at available pressure tap locations. These locations were selected by contractor at the time of the visit and the valves were installed by local site personnel. The velocity probes used were magnetically mountable, so a number of test cases were performed where these probes were moved about to get a full coverage of vibrations. The sound level meter was typically left in one location to gather ambient acoustic noise levels during operation of the equipment. Specific details of installation at the individual facilities are provided below.

The facility, as illustrated earlier in Figure 1, consists of four pumps connected to a split discharge header before running underground to two well locations.

Dynamic pressure transducers were installed near the pump on the discharge line of each pump, and on the suction line of pump 1. The velocity probes, as previously mentioned, were roved around the entire facility. Test points on the header are shown in Figure 3. Where possible, vibrations in all three directions were recorded for these points. However, in most locations, vibration measurements were limited to horizontal (perpendicular to the piping run) and vertical measurements.

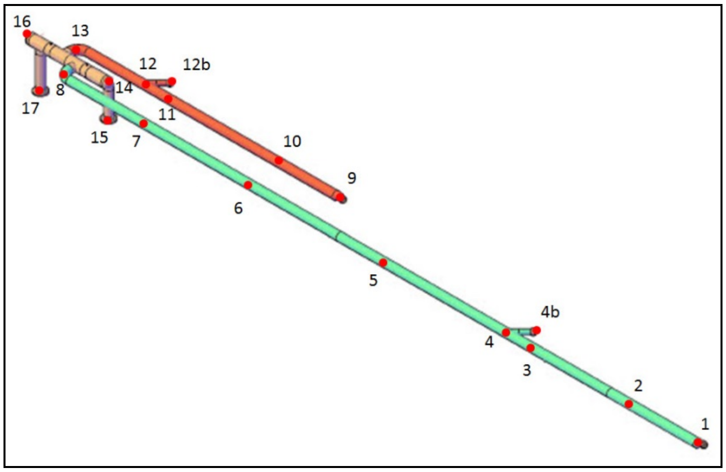

Figure 3: Header Test Points

Vibrations were also measured on each of the pump discharge lines. Given the similarity between pumps 1-3, the test points were identical whereas the points on the pump 4 discharge line are similar but ultimately unique given the different geometry. The test points for all four discharge lines are illustrated in Figure 4.

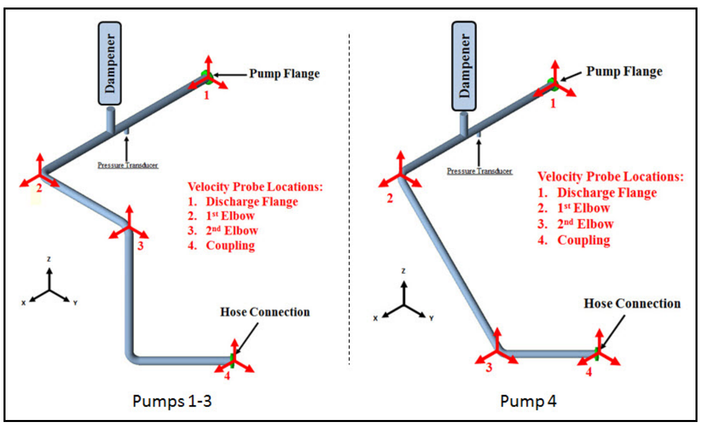

Figure 4: McGregor Pump Discharge Test Points

At both facilities, in order to install the valves and dynamic pressure transducers, it was necessary to shut down all pumps. During the time when the pumps were shut down, the opportunity was taken to perform impact testing on the dampers, the piping and related support structures in order to determine mechanical natural frequencies and mode shapes.

After completion of installation and impact testing, the pumps were brought back online and set to the desired operating conditions as dictated by the current well configuration and operating conditions. The velocity probes were relocated numerous times as described above to gather vibration responses of the entire system.

While measuring the vibrations of the pump discharge piping at each location, the speeds of the units were varied by local personnel between a high-speed and a low-speed by manually changing the speed set point of the motors. This set point was actually a percentage of full speed for the motors. Specific details of the unit speeds for both facilities are described in the tables below, along with the operating pressures for the cases. There was no readout for the specific speeds (rpm) of the pumps as related to these operating cases, so it was necessary to rely on measured data in conjunction with other unit details to determine, and are discussed in the following section.

During this initial testing period, it was only possible to test the units while operating with water as the process fluid.

Table 1: Test Points

Test Case	Motor Set Points (% of max speed)				Suction Pressure (psi)	Discharge Pressure (psi)
	Pump 1	Pump 2	Pump 3	Pump 4		
High-Speed	69	67	78	78	38.9	3902
Low-Speed	62	60	68	68	39.0	2765

Measurements at the facility showed strong dynamic responses in both vibrations and pressure pulsations of the discharge system near the fifth order of running speed for the individual pumps. Measurements for pulsations on the discharge lines for the four units are shown in Figure 5 below for the period of time when the units were changed in speed from the normal operating speed to a lower operating speed. The high peak of the Pump 4 discharge pulsations (blue curve) is indicative of an acoustic resonance in the piping system.

Initially, the unit speeds were dropped by 5 percentage points; however, this resulted in elevated pressure pulsations and vibrations. Therefore, speeds were dropped an additional 5 percentage points.

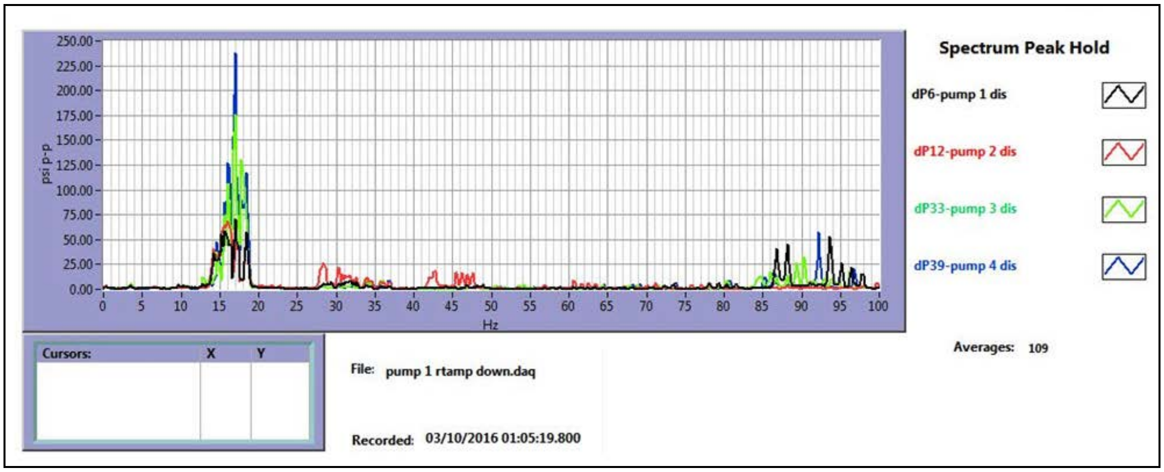

Figure 5: Ramp-Down Pulsations

Analysis of the pressure pulsation spectra during the various speed changes indicates that the speeds of the various pumps ranged from 2.8 Hz to 3.7 Hz, or 168 rpm to 222 rpm. The running speeds for the various pumps are shown in Table 2 below, along with the resulting plunger frequencies (or 5x running speeds). In general, the pumps can be grouped into two sets: pumps 1 and 2, which run at speeds roughly 0.4 Hz lower than pumps 3 and 4. This difference of 0.4 Hz running speed generates a difference of about 2 Hz at the plunger frequency. And it was this difference in the primary pulsation response of the units that was the source behind the “beating” responses that are audibly and visually apparent to on-site personnel. The beating was also apparent in the raw time waves of measured data, as shown in Figure 6, where the time difference between peaks of the vibration data measured on the header (lower graph) was equivalent to a 2 Hz signal.

The peak seen in the ramp-down data (Figure 5) occurs at approximately 17 Hz. Given the speed of the units, the 3rd and 4th pumps were providing excitation nearly exactly at this frequency during their period of running at the medium speed; hence, the large amplitude of measured pulsations. Pulsation measurements made during the period when the pumps were at their medium speeds are shown in Figure 7 below. Pulsations measured on the 1st and 2nd pump discharges do not show amplitudes of those seen by the 3rd and 4th pump because they are running well below this frequency in the low and medium speed cases, and do no run close enough to the acoustic resonant frequency in the high speed case.

Table 2: Pump Speeds

Pump	Pump Speed / Plunger Frequency [Hz]		
	Low Speed	Medium	High
1	2.9 / 14.5	3.1 / 15.5	3.3 / 16.5
2	2.8 / 14.0	3.0 / 15.0	3.2 / 16.0
3	3.2 / 16.0	3.4 / 17.0	3.6 / 18.0
4	3.2 / 16.0	3.4 / 17.0	3.7 / 18.5

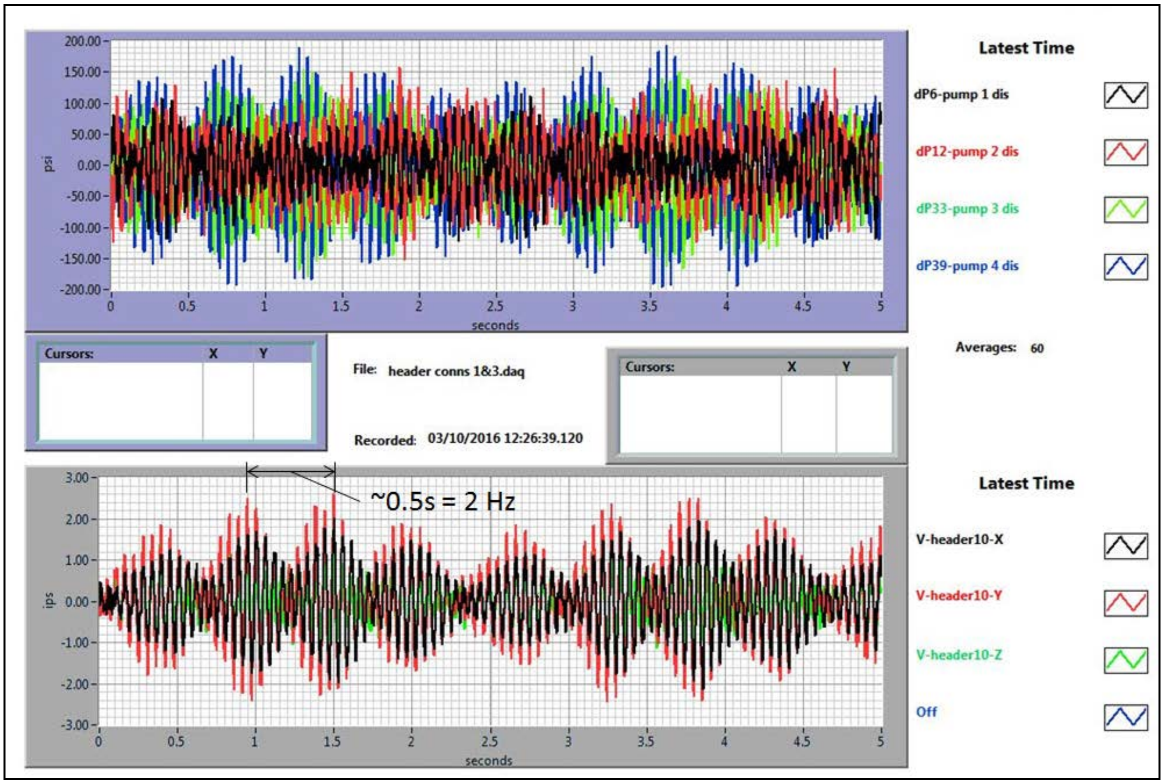

Figure 6: Beating Response in Raw Time Wave Signals

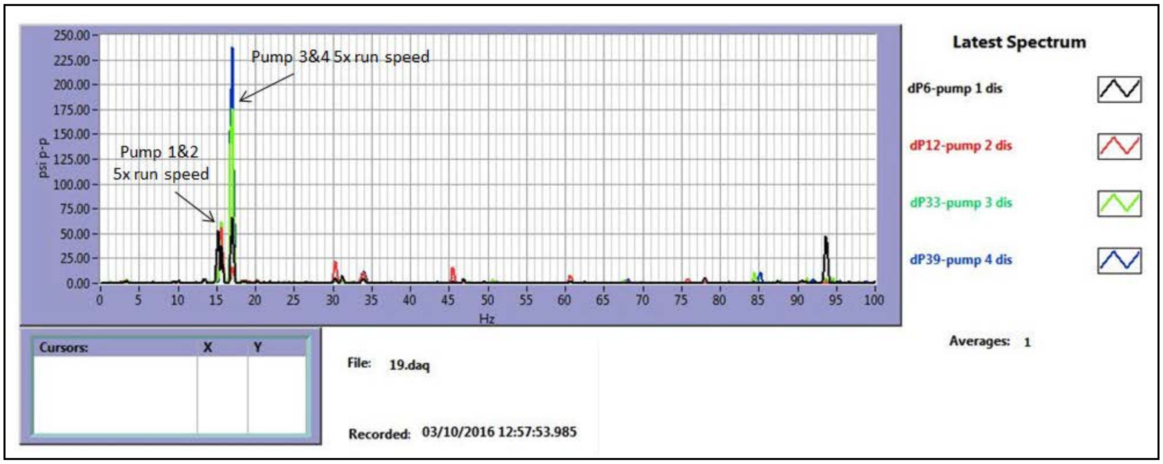

Figure 7: Middle Speed Pulsations

The pulsation measurements also showed that the two sets of pumps (1 & 2 and 3 & 4) were not isolated from each other. As such, there were pulsations that traveled between the two sets so that frequencies of vibrations were present on all pumps at both running speeds. However, the pulsations associated with Pumps 3 & 4 were more prevalent simply due to the higher amplitudes that were being generated.

Additional peaks were also identifiable at higher frequencies. In general, the amplitudes at these higher frequencies were lower in amplitude than those seen at the 5X running speed, but the relative amplitudes of the local peaks do make one peak in particular stand out in the 85-100Hz range. In particular, the response at Pump 1 seemed to be more excitable. The contribution of this peak to the overall pulsation, and subsequent vibration, levels was unlikely to be significant though.

Ultimately, the high levels of pulsations measured in the discharge piping near the 5X running speed frequencies were likely the result of near-coincidence with an acoustic resonance in the system. Additionally, given the number of elbows in the vicinity of the high pulsations, this energy would also translate into mechanical vibrations. Therefore, it was recommended that this pulsation energy be reduced in the system. The aim of the acoustic modeling effort was to address this.

In terms of vibrations, amplitudes were generally greater at the high-speed case than the low-speed case due to the amount of energy required to go into running at higher speed. The middle speed case picks up due to the coincidence with the acoustic resonance, but in terms of evaluating the vibrations under normal operating conditions, the following figures are shown for the high speed operating case. Figure 8 through Figure 12 below show vibrations on the first and second elbows of the discharge piping of Pumps 1 through 4 respectively. The highest vibrations recorded were actually on Pump 1, with a peak at 17.8 Hz of 2.2 ips 0-p (inches per second zero-to-peak). Based on the contractor's general piping severity screening chart, this level of vibration fell into what was considered highly likely to fail. Given the limited restraints on this piping, some flexibility can be taken into account, but there was enough restraint and stress concentration along this piping (also considering the large mass of the pulsation dampener), that these vibration levels were considered excessive. And given the history of these units, there was no doubt that these vibrations should be corrected.

The coincidence with frequencies of internal pressure pulsations means the vibrations were likely a result of the pulsation energy coupling to the mechanical piping. In addition, operating deflection shape (ODS) modeling of the vibrations was used, and showed that the major points of motion were at the elbows of the discharge piping; which were ideal locations for pulsation energy to couple to the mechanical system. Therefore, the ideal resolution focused on the removal of the pulsation energy to avoid excitation of the mechanical system altogether. This was actually the primary focus of the second part of this project, the acoustic modeling, which is discussed in later sections.

Pulsations were also measured on the suction side of Pump 1, example shown in Figure 9, to determine the possibility of cavitation occurring on the suction side of the pumps. Overall and spectral pulsation levels in these signals were significantly less than what was seen on the discharge side. The overall timewave pulsations generally ranged from about ±20 psi. Given a static pressure of 39 psi, this meant pressure on the suction side dropped to no more than about 19 psi. This pressure would be sufficiently above vapor pressure for the water that the chance of cavitation occurring in the pumps was not considered significant. Additionally, the low levels and overall appearance of spectral pulsation did not indicate the likelihood of cavitation either.

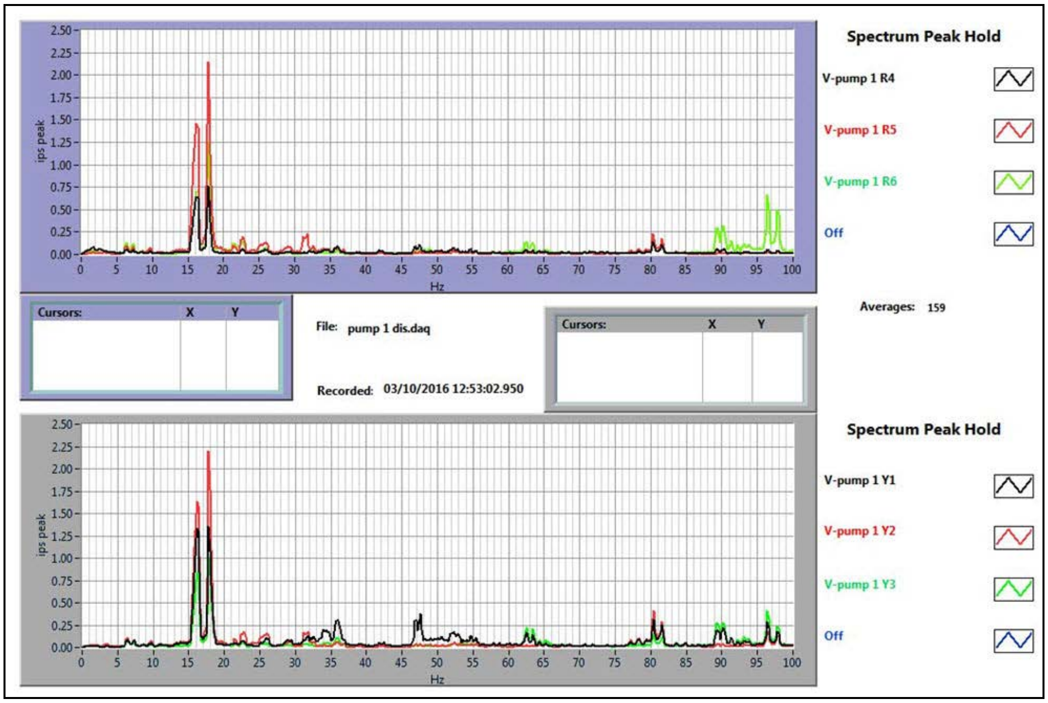

Figure 8: Pump 1 Discharge Piping Vibrations @ High Speed Operation

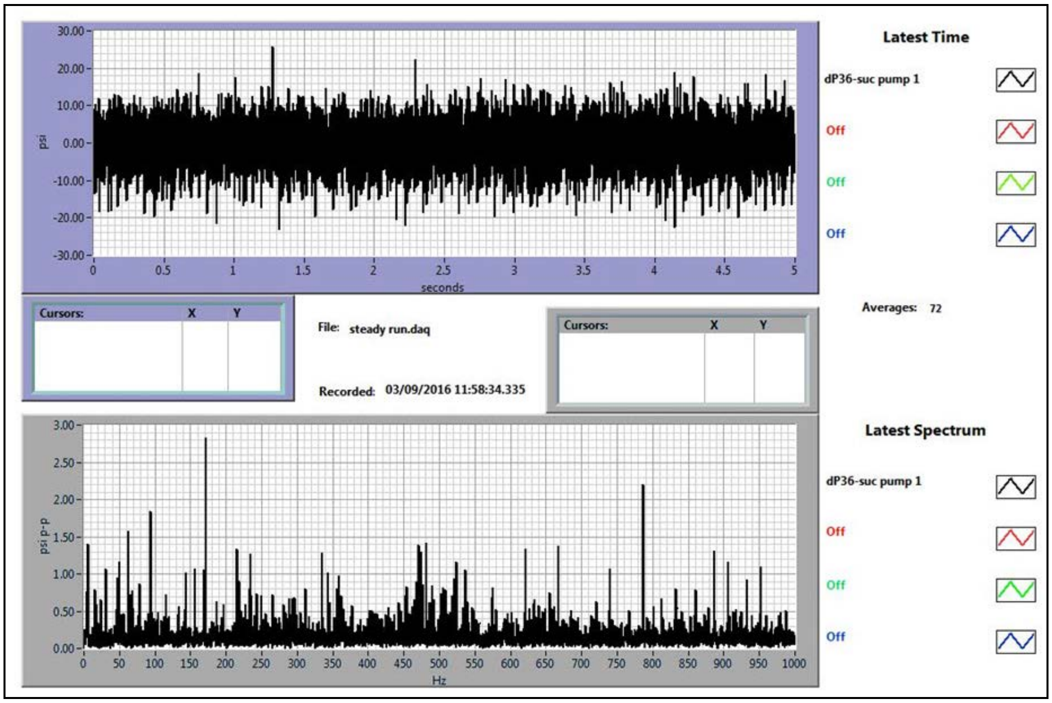

Figure 9: Suction Line Pulsations

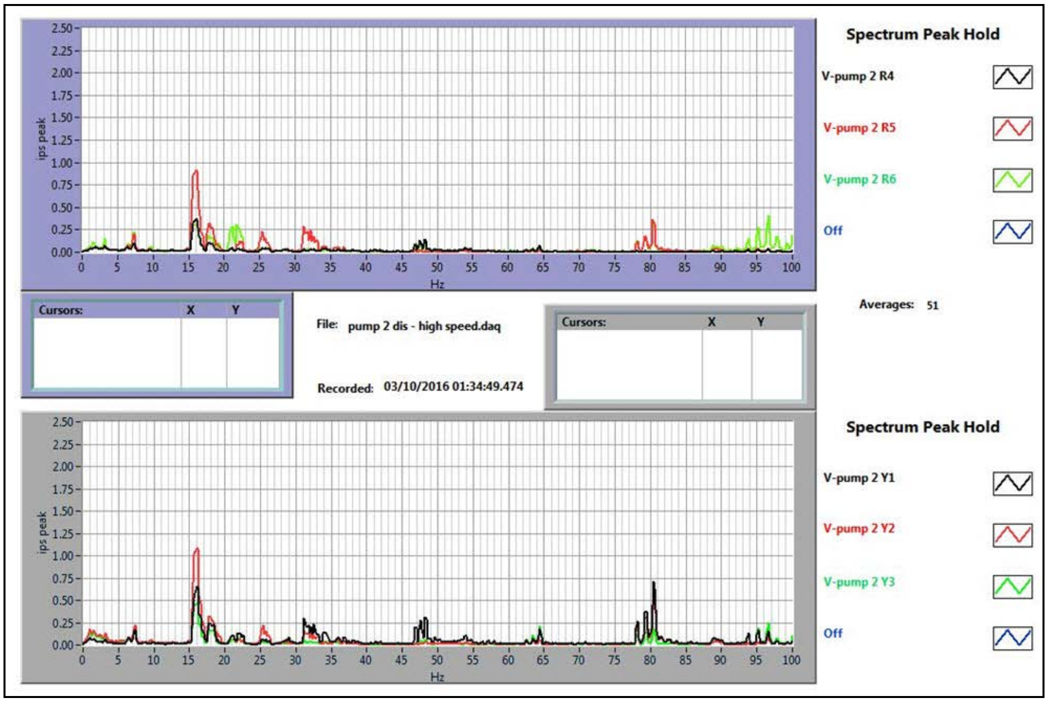

Figure 10: Pump 2 Discharge Piping Vibrations @ High Speed Operation

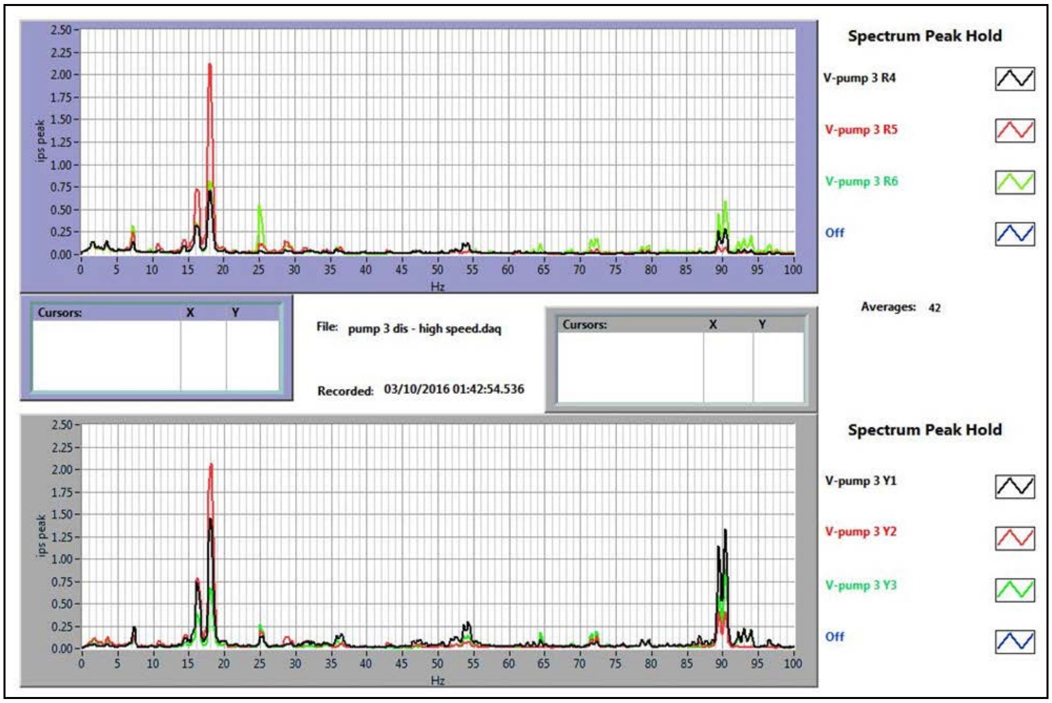

Figure 11: Pump 3 Discharge Piping Vibrations @ High Speed Operation

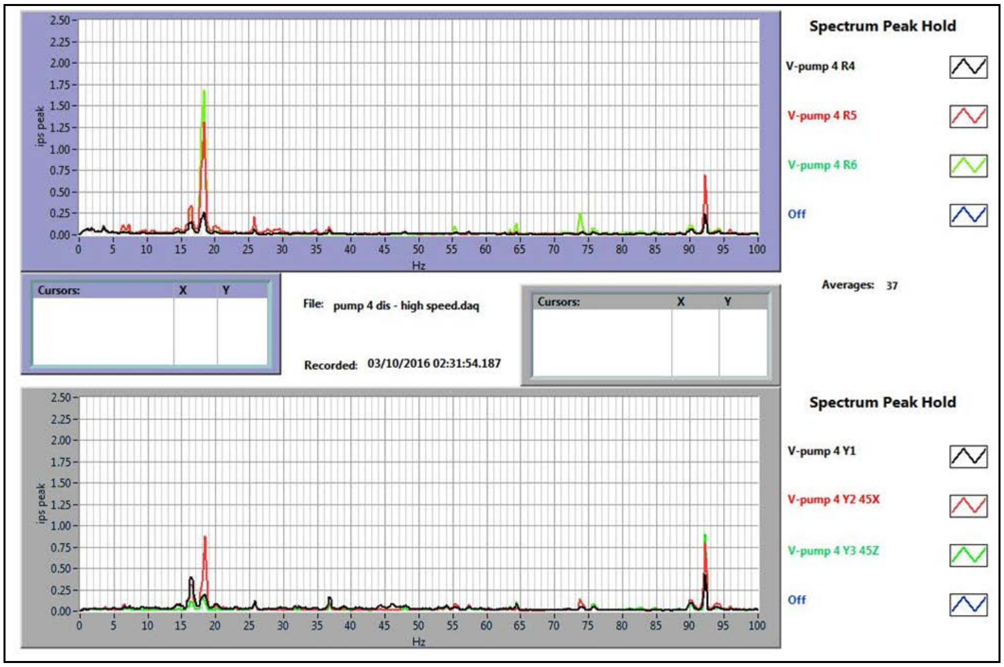

Figure 12: Pump 4 Discharge Piping Vibrations @ High Speed Operation

Based upon the results of the acoustic modeling, the pulsation levels could be eliminated, or at least significantly reduced. However, there was the possibility that some level of pulsation energy would remain in the system, and this could potentially still drive some mechanical vibration. So recommendations for adjustments to the mechanical restraints of the system were still provided.

In general, contractor recommended that any restraints added to the system be of sufficient size and stiffness to provide adequate restraint and damping. Part of these clamp designs call for damping material between the piping and the clamp as well as between the piping and the support under the piping. This material would provide not only good contact between the piping and restraint, but would also provide wear resistance and vibration damping.

Locations and attached support structures for these clamps were also critical to proper performance of the system. Ideal routing of piping would place large headers and long runs of pipes close to grade for easy support. Frames built up to support piping would also be stiff in all three directions. For example, the small supports made from I-beams for some of the discharge pipes with added plates on the ends were much better supports than the I-beams by themselves (see Figure 13); some locations already had this modification, but this modification should be applied to all instances.

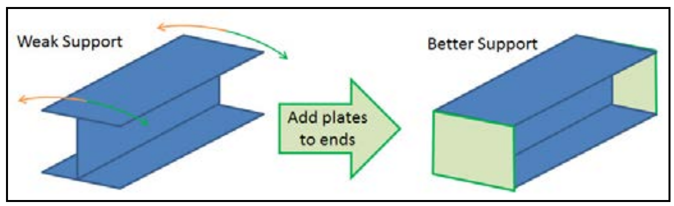

Figure 13: Supports for Small Elevated Piping

Some specific locations that might benefit from added supports were the various pump discharge pipes. Especially those on Pumps 1 through 3; the number of elbows and lack of supports mad this run of piping very loose with numerous mechanical resonance modes that could be excited. In comparison, the discharge piping on Pump 4 is shorter, with one fewer elbow, so the piping would be stiffer and less prone to excitation at lower frequencies. In addition, the orientation of the elbows on Pump 4 meant there were fewer planes of excitation (see Figure 14 for an illustration of this concept), so there was less chance of exciting multiple modes.

The ideal installation would place the header near the ground using strap-type clamps with adjustable wedge blocks to restrain header motion, and using well-supported lateral piping (or possibly flexible hosing) from the pumps with limited use of elbows. In lieu of this type of major modifications to the header design, it would be desirable to replace all discharge pipes with a design similar to Pump 4, with some added restraints, as shown in Figure 15. Alternatively, if the piping design is kept the same for Pumps 1-3, restraints should be added as indicated in Figure 16.

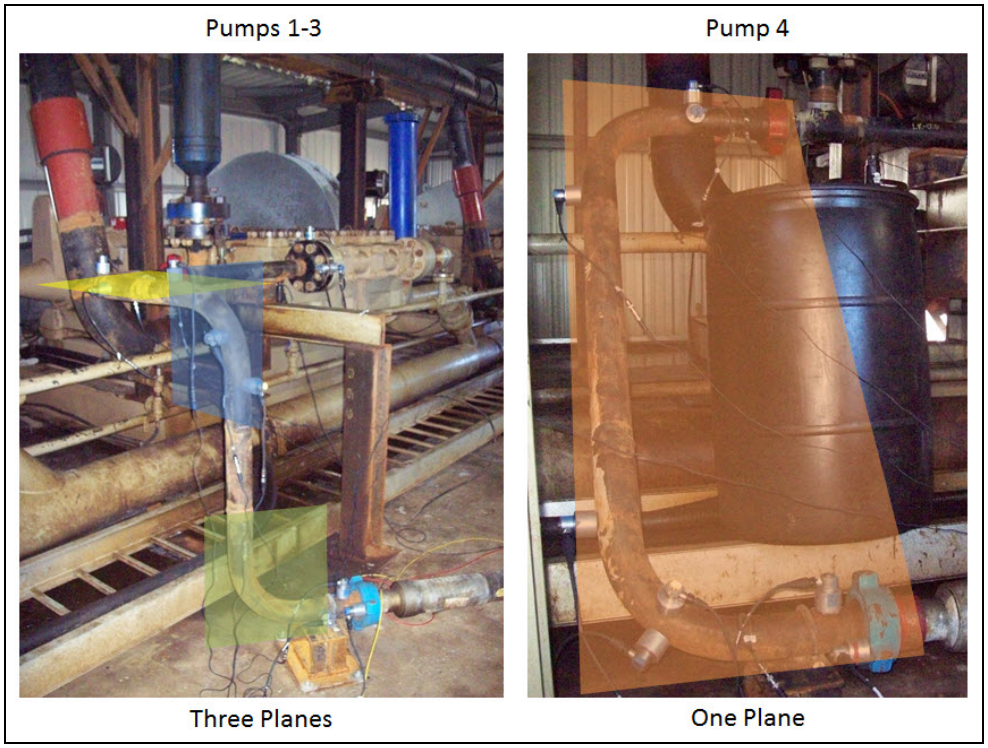

Figure 14: Planes of Excitation

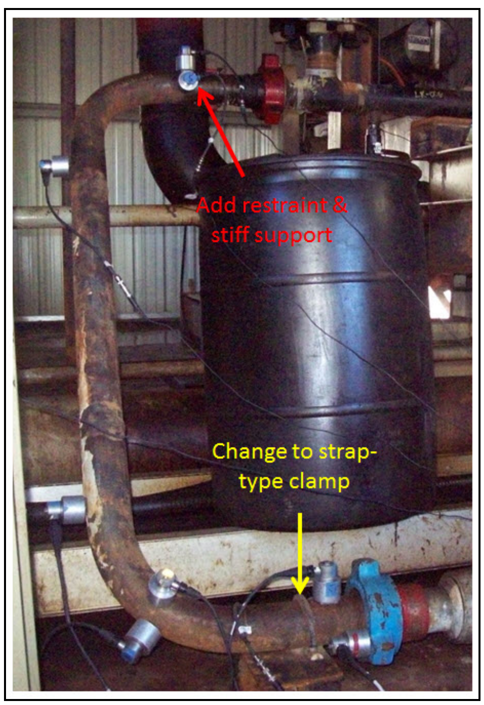

Figure 15: Restraint Recommendation for Pump 4 Discharge Piping

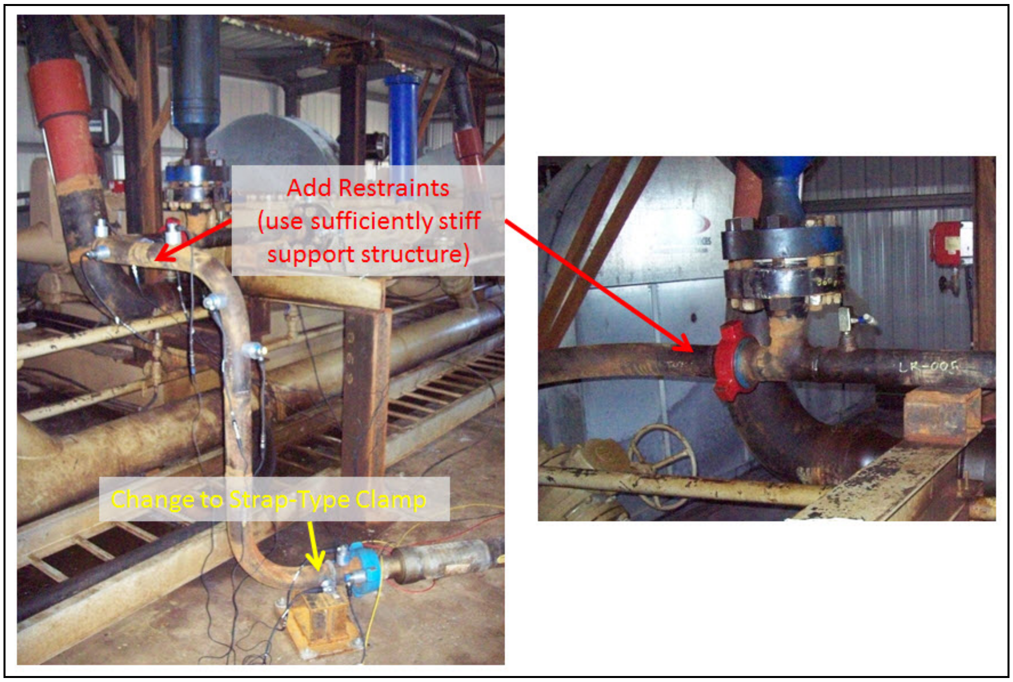

Figure 16: Restraint Recommendation for Pumps 1-3 Discharge Piping

Besides the discharge piping, a number of supports on the header could also be altered to improve vibration response. In particular, there are four supports (identified in Figure 17) on the header with similar designs that are prone to mechanical resonance excitation. The very first one in this image was particularly susceptible to vibration, with a resonance at 17 Hz and mode shape as indicated in Figure 18.

Figure 17: Supports on Header

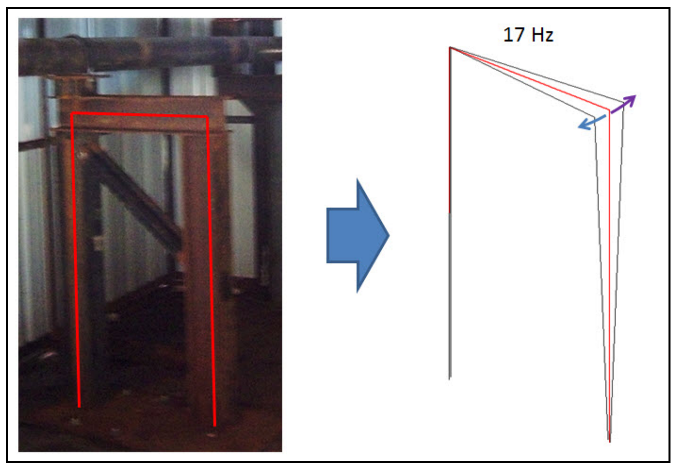

Figure 18: Modal Response of Support

Ideally, as previously mentioned, the header would be lowered closer to grade so that it could have shorter supports. Given the necessity of remaining at this elevation, however, the next best option is to provide additional bracing to stiffen the support in the direction of the pipe axial direction. At the very least, a rough FEA model of this support shows that by simply moving the support over, so that the pipe is supported in the middle of the cross-beam instead of at the corner, the first natural frequency could be increased as illustrated in Figure 19. (Note: this is not an exact model of the support, but only a conceptual illustration of the possible support; therefore, natural frequencies were not expected to be an exact match to field data.)

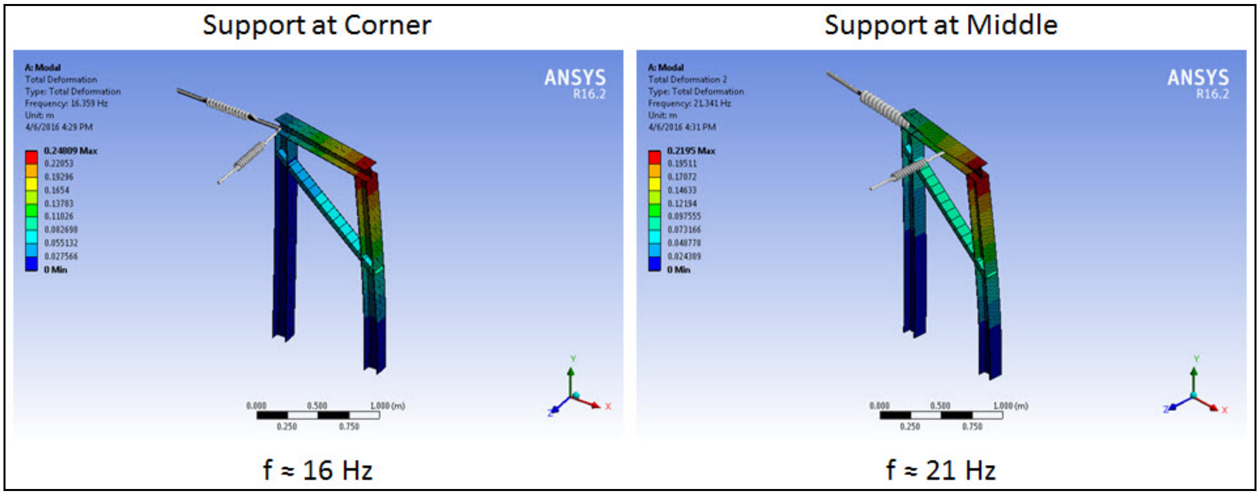

Figure 19: FEA Results for Support Location

As with the supports on the discharge piping, changing the simple U-bolt restraints for strap-type clamps was also recommended wherever the pipe was restrained. Other restraints also existed on the header, as shown in Figure 20, that were in essence only weight supports given the high length-to-diameter ratio of the supporting pipes (yellow arrows). As such, the U-bolt restraints would do little more than possibly wear on the pipe. Replacing the U-bolt restraints with a pair of supporting wedge blocks with proper low friction material would be more adequate. This would also allow for adjustments so that the header piping would be supported given any thermal expansion that could lift the pipe off the support. It was noticed while on-site that the support for the connection of Pump #1 (shown in the upper right image of Figure 20) was only providing limited, if any, weight support as the pipe was lifted off its ground support and could be moved by hand.

If these supports were, however, meant to provide more than just weight support, the piping should be replaced with an appropriate framed structure using a strap-type clamp restraint as discussed earlier.

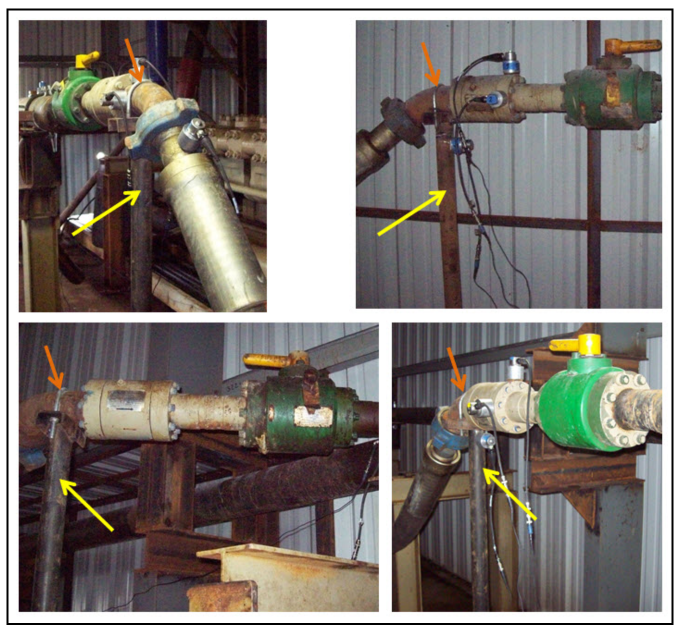

Figure 20: Weight Supports on Header

Additional Field Data

After initial acoustic analysis efforts, it was determined that further field data of the two stations would be beneficial; specifically, pulsation data taken during operation of the units with oil as the working fluid instead of water. As such, contractor shipped a digital recorder and two dynamic pressure transducers to end-user so that this additional data could be obtained.

Data files captured during these additional runs were sent electronically back to contractor for analysis. This data was recorded at the facility at a range of different operating conditions. In the first set of data (recorded on April 13, 2016), most of the pulsation dampeners at the facility were effectively not present as most of the internal bladders had failed and so the gas charge was not present. During this test, 6-9 wells were active with discharge pressures ranging from 2450-3900 psi, and the units were run in speeds ranging from 40-73% of full speed. During a second set of data, the number of wells online increased to as many as 13 with discharge pressures consistently near 3850 psi, and the pumps running at speeds from 42-92%. Additionally, during this second round of testing, the bladders in the pulsation dampeners were replaced part way through the testing, so the effect of the dampeners on the system response was recorded.

Based on the first set of data collected by this means, it was determined that a resonance in the system with oil as the working fluid existed at about 18 Hz, as shown in Figure 21. This resonance was seen as the 5x order of running speed neared the resonance (especially during the 13th record), and also when the 10x order of running speed was near the resonance (as seen during the 6th record).

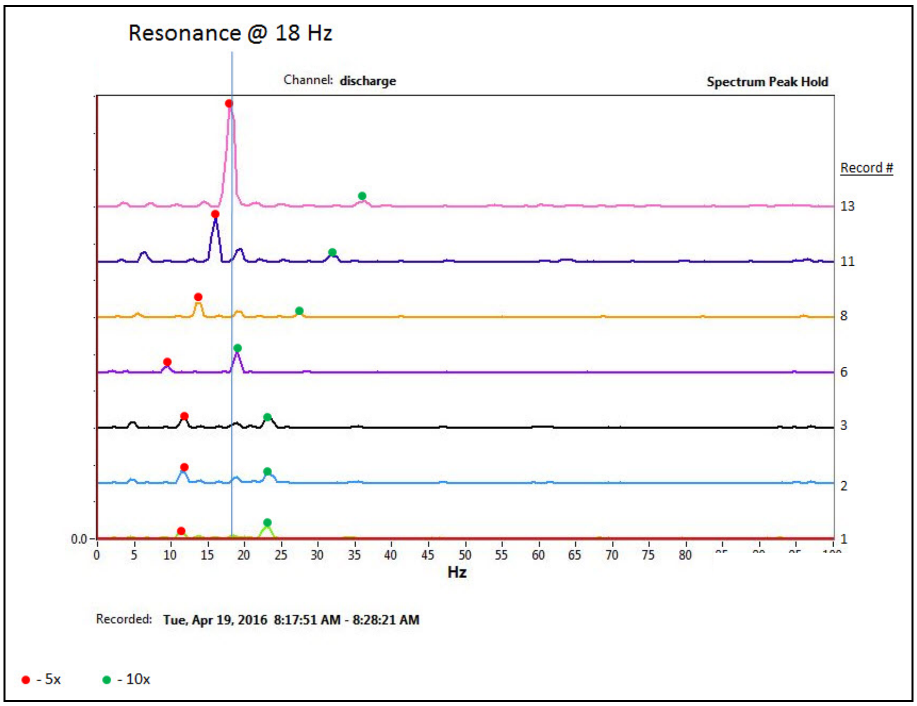

Figure 21: Oil Pulsation Data - Set 1 Discharge Spectra

As far as the possibility of cavitation occurring on the suction side of the pumps, there were minor indications of cavitation-type responses, but since the suction pressure, about 100 psi, was already well below the vapor pressure of the oil, (provided as 2,505 psi) there really was no chance for the classical cavitation response to occur. The more likely scenario was that vapor in the liquid stream was causing sporadic fluctuations of the compressor valves leading to spikes in the pressure pulsation time waves.

Data in the second set of post-field data indicated additional possible responses. Initially, with the bladders blown and the dampeners fully deactivated, possible acoustic resonances existed at 13 Hz and 16 Hz as shown in Figure 22. There was also the chance of higher frequency modes being excited by higher orders of running speed (in particular the 15x response seemed to be elevated in the 40-55 Hz range). After the bladders were replaced the 13 Hz mode appeared unaffected, but the 16 Hz mode increased to approximately 18 Hz. Additionally, the higher order mode seemed to be more prominent near 77 Hz. These are shown in Figure 23. Also, besides the shift in possible acoustic resonance frequency with the bladders being replaced, a noticeable decrease in the amplitudes of pulsations was visible as shown in Figure 24.

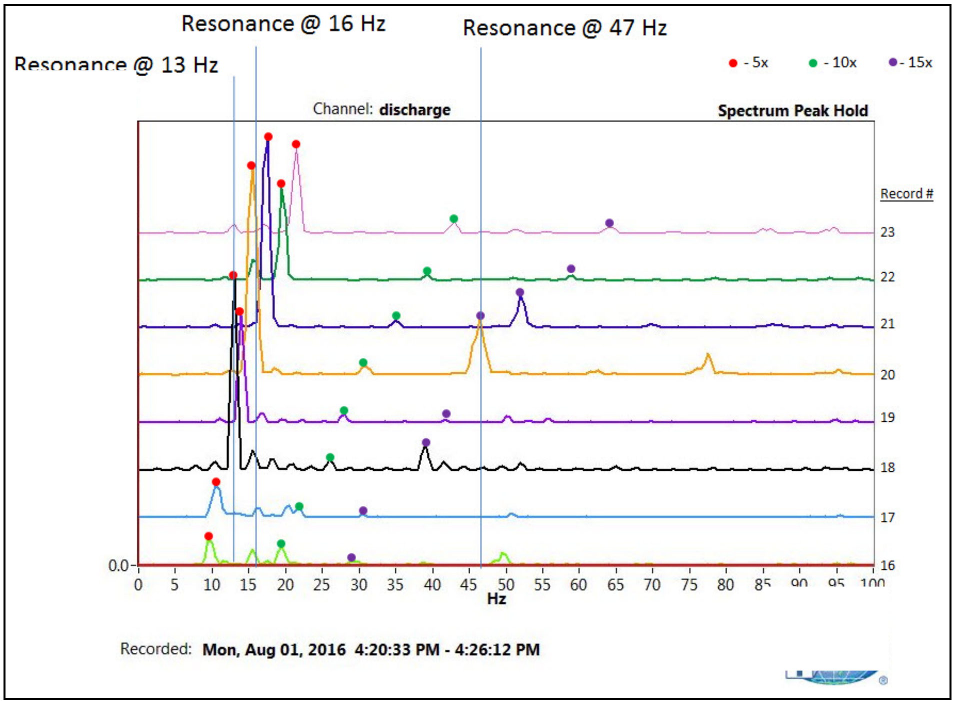

Figure 22: Oil Pulsation Data - Second Set with Blown Bladders

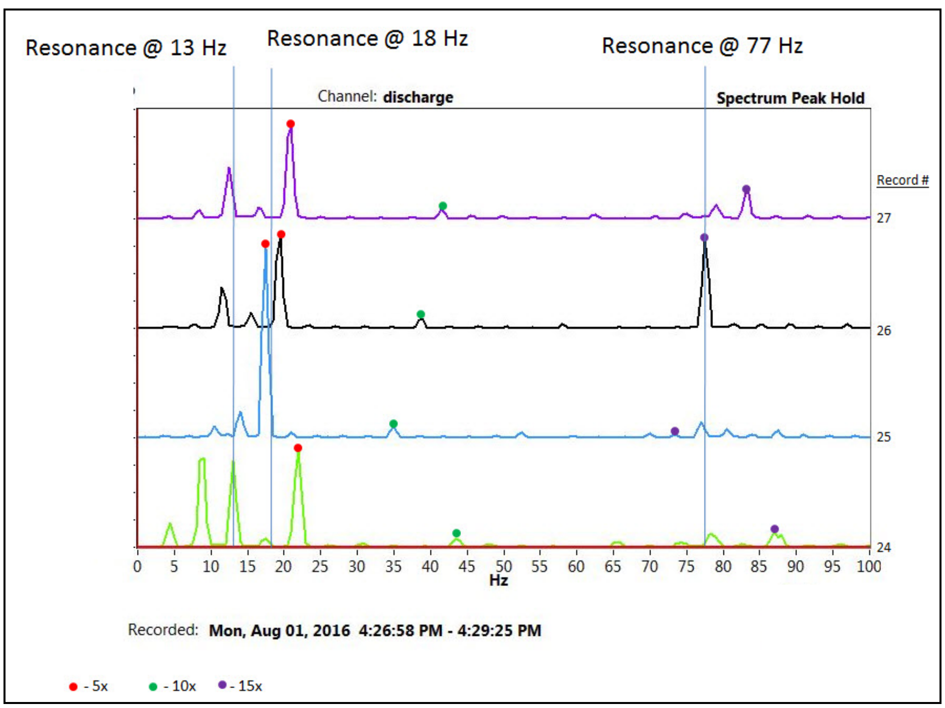

Figure 23: Oil Pulsation Data - Second Set with Replaced Bladders

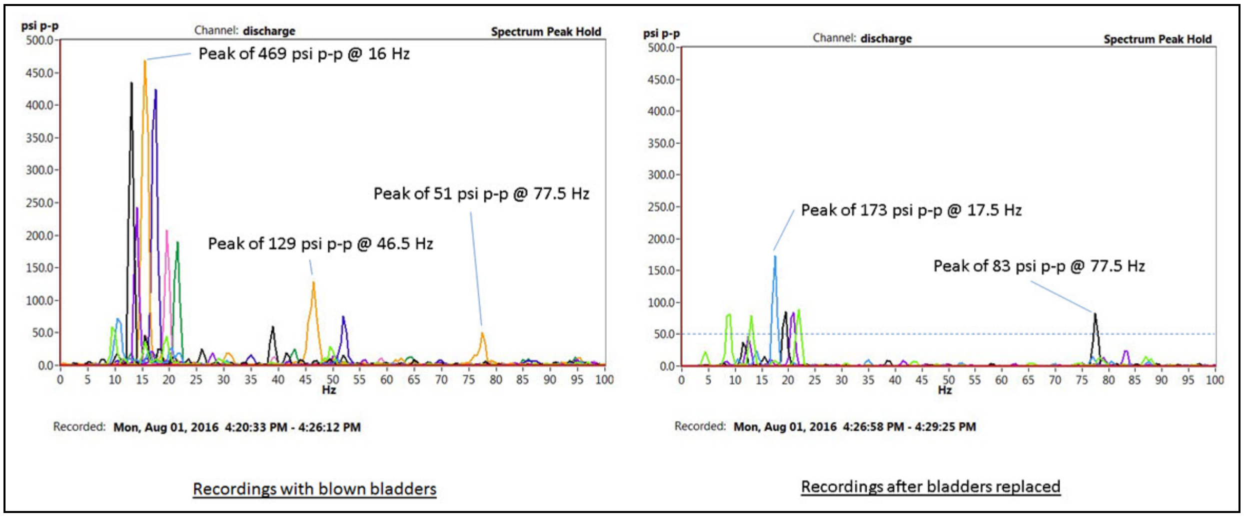

Figure 24: Post-Data - Influence of Dampeners

Results from analysis of the second set of post-field data supported the original field testing conclusions. More importantly, the sets of data with oil as the working fluid provided key data for validating speed of sound and other fluid properties for comparison with the acoustic analysis effort.

ACOUSTIC PULSATION MODELING

Using contractor's proprietary acoustical simulation software, pulsation amplitudes throughout the piping network were predicted for relevant operating conditions to ensure that the recommended pulsation control systems were effective.

Specific operating conditions were selected for analysis from the contractor's field evaluation and information provided by the end-user. These conditions are summarized in Table 3. The mean sound velocities were calculated from the provided operating conditions and liquid properties. Variations in the sound velocities were included in the analysis to account for changes in the operating pressures and temperatures. This was accomplished by varying the pump speed 10% above and below the normal operating speed.

The acoustic piping model was constructed using the dampener and pump manifold piping drawings provided by the end-user. A schematic showing a representation of the piping system under investigation is included in Figure 25. Acoustical data was recorded for the operating conditions described in Table 3.

Table 3: Description of Reciprocating Quintuplex Plunger Pumps

Facility	McGregor
Pump Unit No.	Units 1-4
Pump Manufacturer	Accelerated
Service - Type of Fluid	Well Injection Oil
Number of Stages	1
Fluid Temperature (°F)	110
Specific Gravity	~2.8
Pump Speed (rpm)	143-286
Plunger Diameter (in)	2.625
Stroke (in)	7
Number of Plungers	5
Suction Pressure (psig)	40
Discharge Pressure (psig)	3900
Capacity at max speed (gpm)	235
Rated Horsepower (BHP)	625

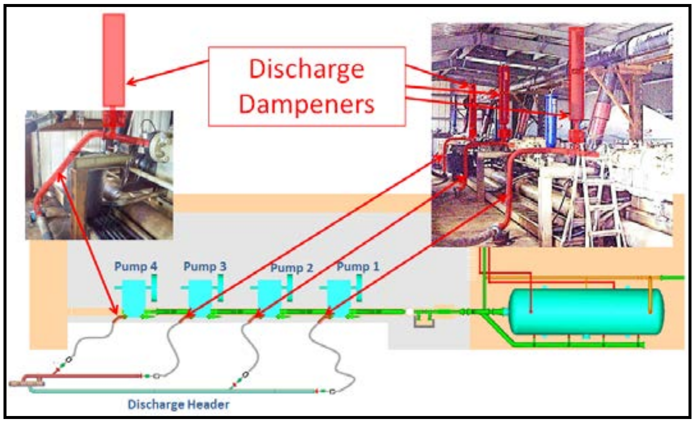

Figure 25: Schematic of Discharge Piping Layout during Field Testing – "Existing System"

Efforts were made to tune the prediction model to the field measured data. Results of the correlation efforts are summarized in Table 4. It was determined that the model predictions were reasonably correlated with field data. Therefore, it was possible to proceed with confidence that the modeled modifications would result in reasonable predictions.

It was determined by analytical calculations and confirmed with software modeling predictions that a piping system acoustic resonance existed in the original piping system. The acoustic resonance went from unit-to-unit. A Helmholtz-type resonance between the units was being excited by the pump pulsations that were inherently generated at 5 times pump running speed because of the quintuplex arrangement of the pumps. A pressure profile, as noted in red and fuchsia sinusoidal-shaped lines in Figure 26, for a Helmholtz-type response has high amplitude pressure fluctuations at and near the volumes, which was what was observed and predicted for the facility. The frequency of the resonance was set by the effective volume of the dampeners, diameter of the connecting piping, and length of the connecting piping. Altering the connecting piping diameter and/or lengths could be implemented to shift the resonant frequency. However, to minimize piping changes, the dampener volumes were modified. By increasing the effective volume of the dampeners, the acoustic resonance could be shifted to a much lower frequency at which the system acoustic response would not be excited or amplified to unacceptable levels.

Table 4: Summary of Correlation Efforts

System Description	Maximum Field Measured Pulsation Amplitude on 5x (psi pk-pk)	Maximum Predicted Pulsation Amplitude on 5x (psi pk-pk)*
Pumping water (≈14-19 Hz)	237	28-346
Pumping oil (≈18-22 Hz)	469	383-603

* A range is given for the above noted "Maximum Predicted Pulsation Amplitude on 5x" because the unit-to-unit phasing is not known for the field measured data, which means the exact field conditions cannot be simulated

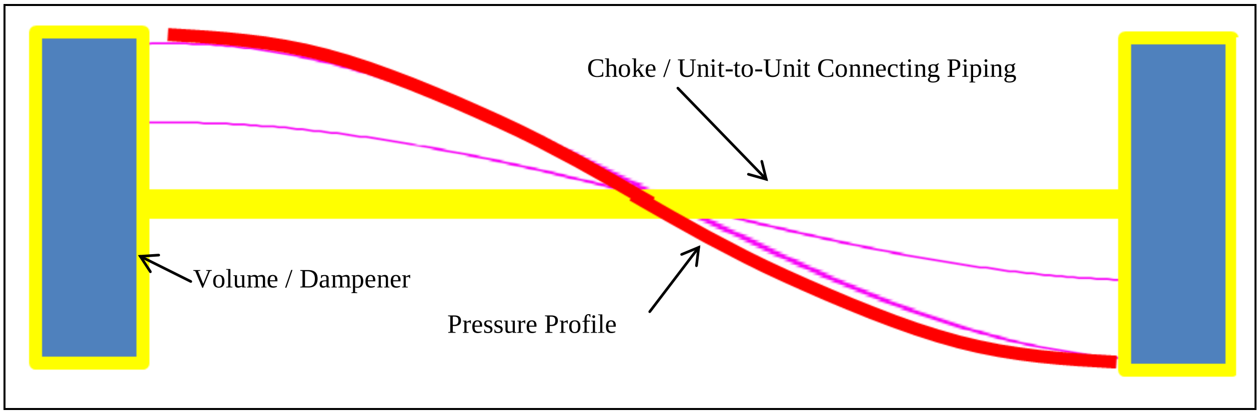

Figure 26: Pressure Profile Plot for a Volume-Choke-Volume Piping Configuration

After reviewing the original dampeners, it was determined, per the dampener datasheets, that the maximum allowable pre-charge of the dampeners was 500 psig. As noted in Table 1, the maximum operating pressure of the system was 3,900 psig. Therefore, the original dampeners could only be pre-charged to approximately 13% of the line pressure; significantly below what is typically recommended, which is usually around 80%. The reason for this is that gas volumes decrease linearly per the ideal gas law. For example, a volume of 1.74 gallons of gas pre-charged to 500 psig will only be 0.21 gallons of gas when pressurized to 3,900 psig. That was the case with the existing dampeners. When a dampener bladder is compressed and stressed as much as the bladders were being stressed in the original dampeners, it will typically lead to failures of the bladders. As noted previously, end-user was experiencing dampener failures. This finding was another reason to replace the dampeners with more appropriately-sized dampeners that could be charged to a pressure closer to the maximum operating pressure of the system. An appropriately-sized or reasonably-sized dampener can be sized using essentially any reputable dampener manufacturer's sizing calculations/programs; however, that should simply be the starting point for the sizing. The ultimate determination of the acceptability of the dampener size should be based on the predictions from trustworthy pulsation modeling tools and experienced designers. Even a well-sized dampener can sometimes be installed in a piping system in such a way that a system resonance or resonances can be excited and result in a vibration problem.

Several piping modifications and layout changes for shifting the acoustic resonance were discussed with end-user, and the system that provided the best balance between their needs, cost, and vibration reduction is presented in Figure 27. It was not necessary that the Pumps 1 and 2 and Pumps 3 and 4 lines cross each other. That was a piping layout modification that end-user chose to implement as an early field attempt to mitigate piping vibrations prior to the final recommended system modifications. The final system included the already crossed flex hoses and the recommended 9.5L dampener installed in place of each of the four existing/original discharge dampeners.

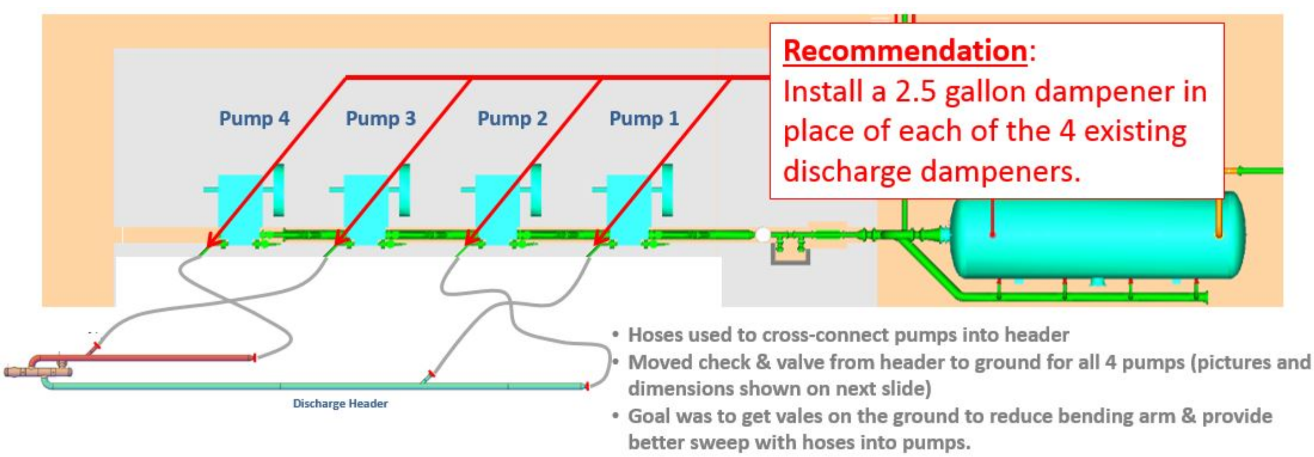

Figure 27: Schematic of Discharge Piping Layout Shortly After Field Testing

The worst-case (assuming worst case phasing between units) predicted pulsations for the existing/original system and modified system, while pumping oil, are summarized in Table 5 and Figure 28. It was predicted that a pulsation amplitude reduction of 97% would be achieved for the worst case 5x pulsations. A pulsation amplitude increase was predicted at orders above 5x; however, the overall reduction in the piping system pulsation amplitudes was considered significant and acceptable. It was predicted that shifting the piping system acoustic resonance to a frequency well below the primary pump excitation order (5x) would result in significantly reduced piping system pulsation amplitudes.

Table 5: Summary of Worst-Case Predicted Pulsation Amplitudes

	Worst-Case Pulsation Amplitudes Predicted on 5x (psi pk-pk)		Worst-Case Pulsation Amplitudes Predicted > 5x (psi pk-pk)	
Test Point #	Existing System	Modified System	Existing System	Modified System
Pump Manifold	603	19	41	71
Pump Outlet	597	20	16	40
Pumps 1&2 Header	91	6	21	6
Pumps 3&4 Header	53	6	10	6

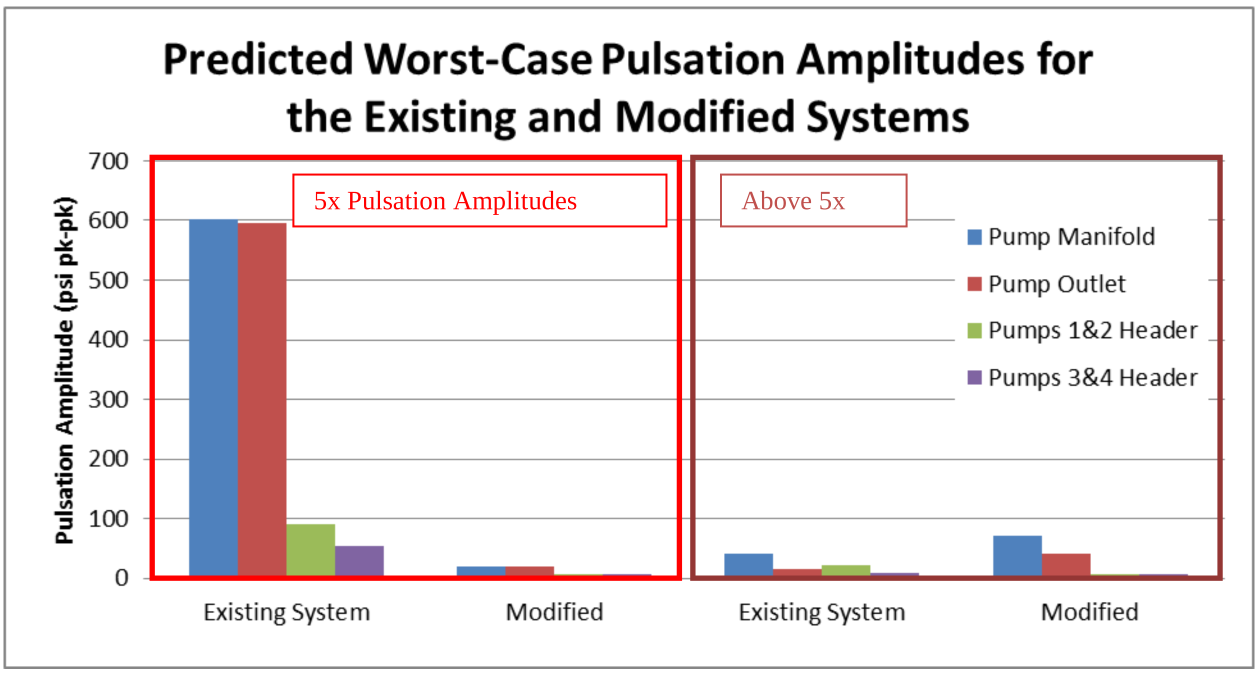

Figure 28: Summary Graph of Worst-Case Predicted Pulsation Amplitudes

Based on the results of the pulsation analysis, it was recommended that 2.5 gallon gas-liquid dampeners be installed in place of the existing dampeners as noted on Figure 27, and that the pre-charge on these new dampeners be set to about 80% of line pressure per standard recommendations for gas-liquid dampeners.

POST-MODIFICATION TESTING

Final measurements made at the facility after the modifications recommended by contractor were implemented still show some characteristics of possible acoustic resonances, as shown in Figure 29. However, the overall pulsation amplitudes indicate these responses are well controlled and as such result in significantly reduced pulsations, as shown in Figure 30. Based on these results, it was evident that the modifications to the facility were successful in addressing the pulsation responses.

Feedback from local site personnel also indicated that mechanical responses of the system have been drastically reduced in a lack of audible response as well as visual indications of vibration. Since the modifications have been applied, the system has been running for nine months without any additional problems.

Two additional units have also been added to the system to increase capacity, and these have been running without incident for three weeks. An acoustic pulsation study was performed prior to installation of the additional units to ensure proper placement and design.

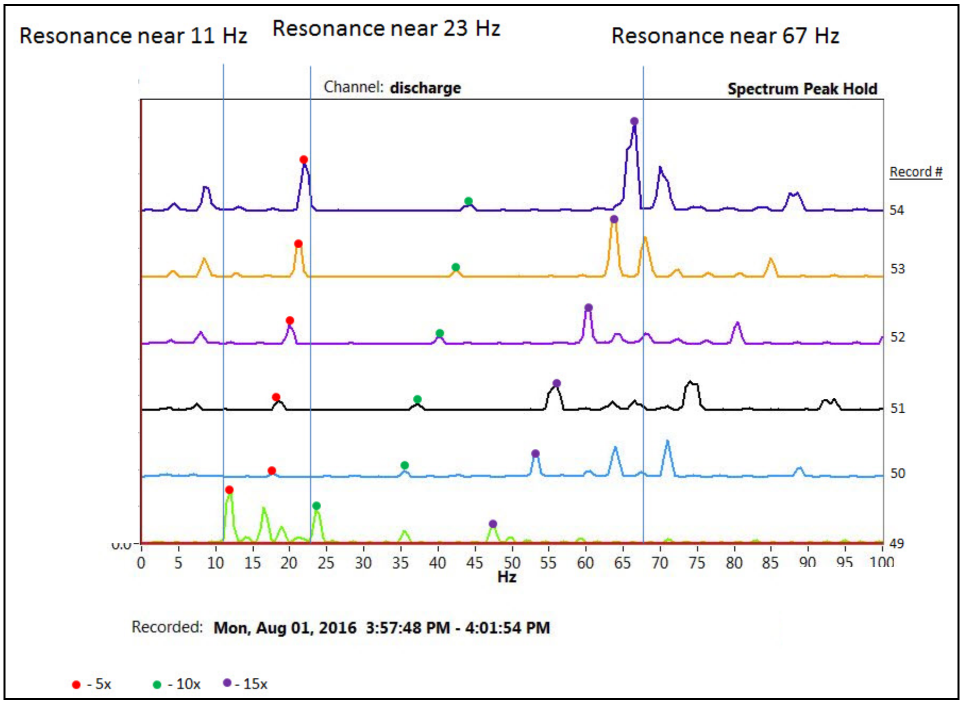

Figure 29: Post-Data – Third Set with Modified System

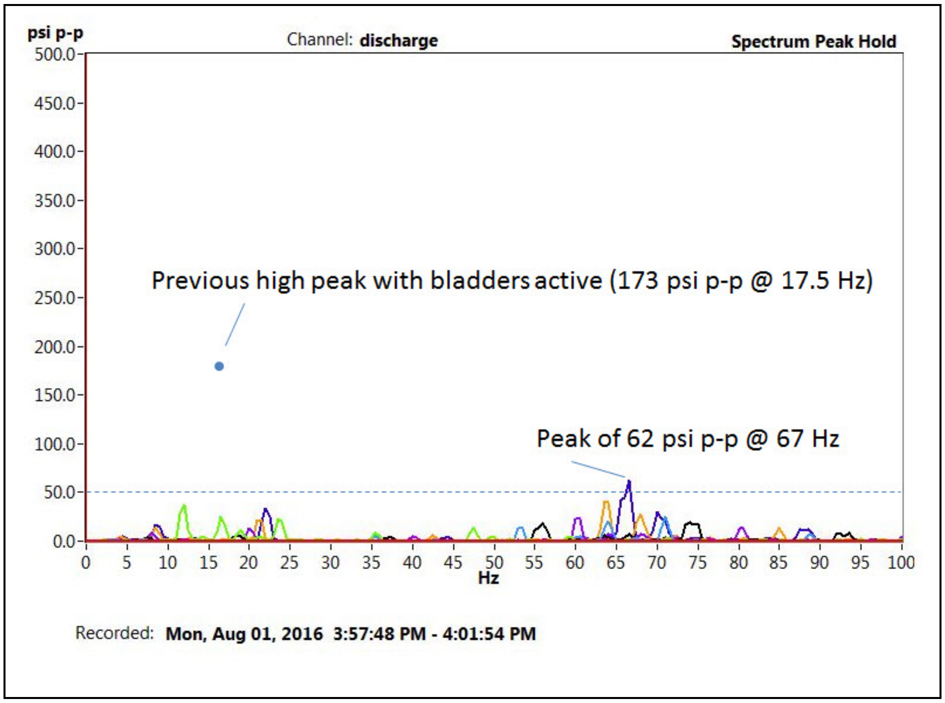

Figure 30: Pulsation Amplitudes of Modified System

CONCLUSIONS

Based on the measurements taken for the original system, it was proved that an acoustic resonance in the system was causing significant amplification of the plunger frequencies for the quintuplex pumps at the facility in question. This was in spite of the fact that pulsation dampeners were installed on the discharge line to reduce such pulsations. It was found during the acoustic modeling effort that the undersized and undercharged dampeners ended up creating a volume-to-volume acoustic resonance in the system with a frequency matching the plunger frequencies.

For typical applications, the wrong size and charge of dampener would lead to ineffective pulsation reduction and failed bladders. However, because of the unique situation of the acoustic resonance that was generated because of these dampeners, the pulsation energy in the system was actually amplified. Combined with some of the original piping design decisions, this energy was allowed to couple to the mechanical system and cause excessive vibrations in the system leading to failures in the piping as well as the dampeners.

With the proper selection and application of dampeners, as well as some improved mechanical piping design and support, pulsations and vibrations at the facilities in question were able to be reduced to acceptable levels for continued operation.

Some of the key take-aways from this project include:

- Proper selection of pulsation dampeners, need to consider the full range of operating pressures, pump size (bore and stroke), pre-charge capabilities of the dampener, number of pump plungers (duplex, triplex, quintuplex, etc.), and acoustics of the attached piping system.
- Even systems with pulsation dampeners can exhibit acoustic natural responses that can amplify pressure pulsations. A pulsation analysis can identify these even in the early stages of design and help to provide solutions to avoid additional complications.

When designing piping layouts, consider how internal pulsation energy can couple to the mechanical system to cause vibrations. Reducing coupling points and limiting planes of excitation make it easier to control system vibrational responses.

REFERENCES

McKee, R., “Acoustics in Pumping Systems,” Proceedings of the Twenty-Fifth International Pump Users Symposium, George R. Brown Convention Center, Houston, Texas, 2009.

API Standard 674, Positive Displacement Pumps—Reciprocating, Third Edition, December 2010.