ADDRESSING VIBRATION ISSUES: A USER'S PERSPECTIVE

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EDITOR'S NOTE

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Proper design of a pulsation control system, along with good mechanical restraint of the pulsation bottles and piping, is essential for minimizing the risk of severe vibration when working with reciprocating compressors. It is important that operators of equipment with vibratory loads have the resources needed to evaluate proposed designs and to assess vibration issues with existing installations. The intent of this article is to provide a user's perspective on analyzing and tackling the vibration issues commonly associated with skid-mounted, high-speed reciprocating compressor packages. A step-by-step comprehensive methodology for evaluating vibrations on the bottles and piping associated with reciprocating compressors is presented. Common issues with on-skid piping arrangements and the lack of effective mechanical restraint are addressed, and recommendations for improving the piping layout and the effectiveness of the mechanical restraint system are made.

For high-speed compressor packages, piping vibration is, more often than not, associated with pulsation energy generated by the compressor. Vibration occurs when a mechanical natural frequency of the piping coincides with a pulsation frequency from the compressor. In many cases, the piping is simply inadequately restrained. Even low levels of pulsation energy can result in unacceptable vibration. Of course, the first step toward mitigating vibration is to determine if vibration levels are problematic.

STEP 1 – Document Vibration Levels

At a minimum, vibration should be measured in the horizontal, vertical, and axial directions on each end of the suction and discharge pulsation bottles, at the top and bottom of the scrubber(s), on piping elbows, and at valve settings (including recycle, blowdown, and relief valves). Vibration should also be measured on the vent piping associated with blowdown and relief valves. For consistency, all vibration data should be measured relative to the crankshaft axis. Ideally, for variable speed compressors, vibration data should be measured over the anticipated operating range to ensure the highest vibration amplitudes are captured.

The compressor package shown in Figure 1 is one of two identical packages currently operating in a single-stage midstream natural gas gathering application. To meet project schedules, midstream companies often use packager-standard, skid-mounted units with generic designs that are not ideally suited for the wide range of operating envelopes they tend to encounter. In addition, to minimize cost and remain competitive, many compressor packages are fabricated with inadequate support of the bottles and piping. It is important that operating companies and packagers fully understand the impact of cost-cutting measures.

Analysis of the example midstream package design depicted in Figure 1 will ultimately show how typical cost cuts associated with prepackaged compressors impact vibration, particularly in the area of mechanical piping restraint. Although the engine is rated for a maximum speed of 1400 rpm, excessive vibrations on the cylinders, bottles, and piping prevented operation at speeds higher than 1300 rpm following installation. Overall vibration amplitudes of approximately 4 ips (101.6 mm/s) were reported when attempting to operate the unit at 1400 rpm. The unacceptable reduction in capacity associated with the speed limitation drove Kinder Morgan to investigate

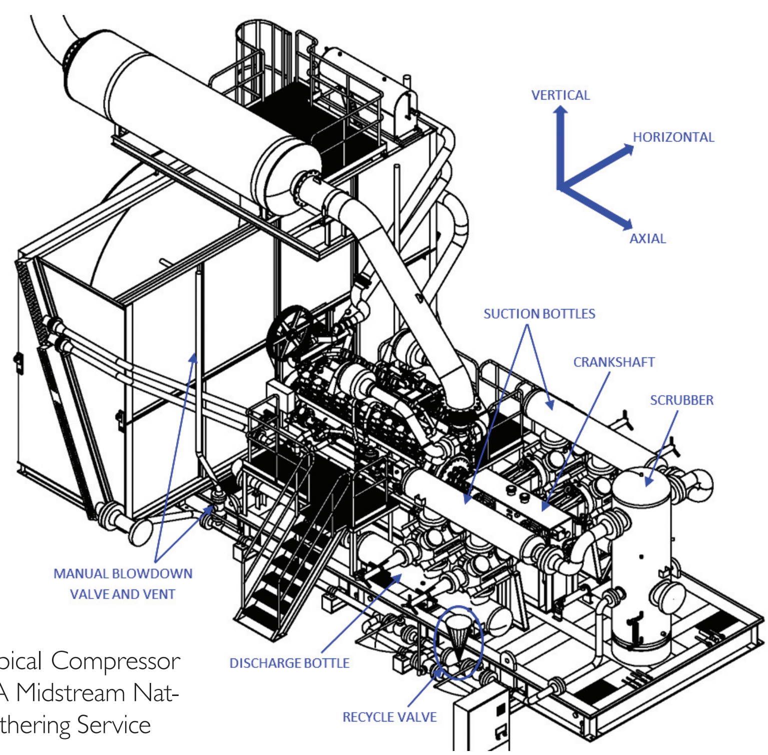

Figure 1. Typical Compressor Package In A Midstream Natural Gas Gathering Service

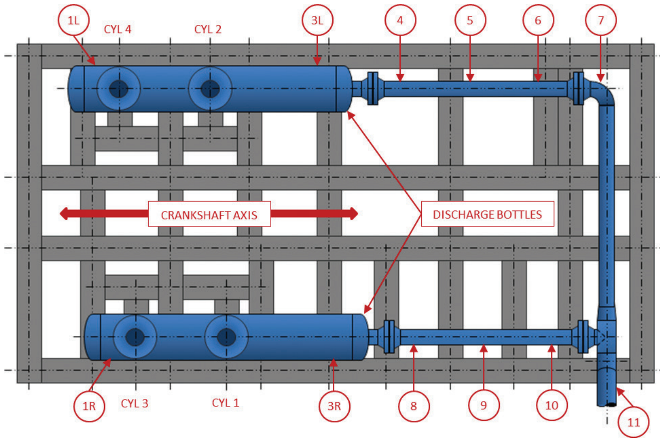

Figure 2. Original Vibration Measurement Locations And Overall Amplitudes

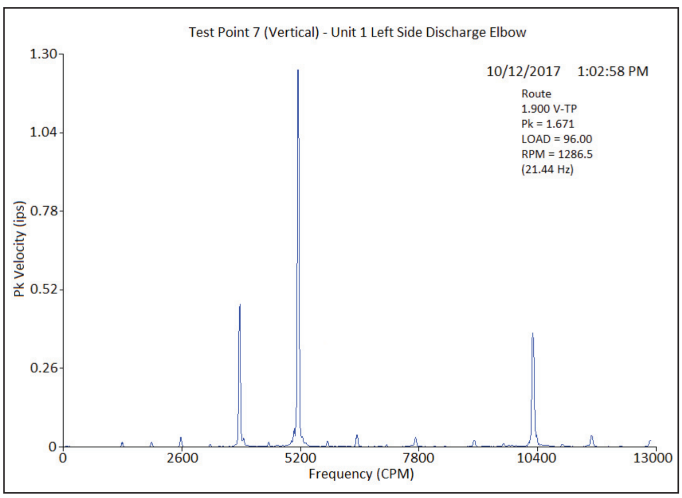

Figure 3. Frequency Plot Of Vibration Amplitudes At Test Point 7 On Unit 1

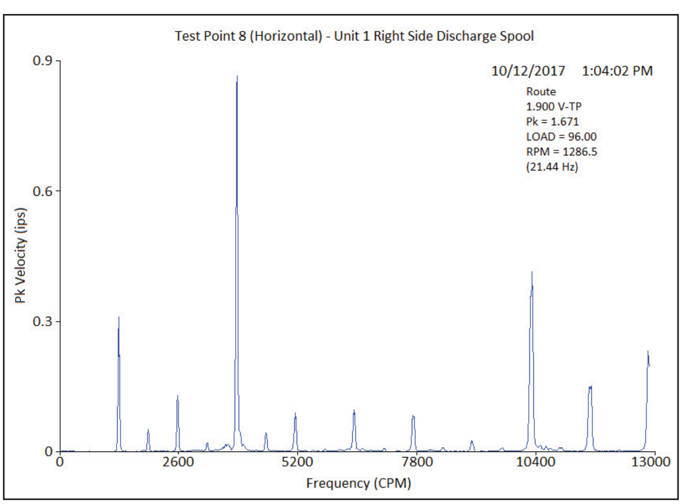

Figure 4. Frequency Plot Of Vibration Amplitudes At Test Point 8 On Unit 1

the problem further. Vibration data for the two identical units were obtained for evaluation. Vibration measurement locations relevant to this evaluation are identified in Figure 2, along with the measured overall vibration amplitudes. Because of the speed limitation, vibration amplitudes were only obtained over a speed range of 1280 to 1300 rpm. Note that axial vibration is measured parallel to the crankshaft and horizontal vibration is measured perpendicular to the crankshaft. Vertical vibration (up and down) should be self-explanatory.

Although vibration amplitudes were relatively high on all suction and discharge bottles and piping, overall vibration amplitudes exceeding 1.5 ips (38.1 mm/s) on the cylinders, discharge bottles, and piping were of primary concern. For this reason, initial efforts to reduce vibrations were focused on the discharge side. The table in Figure 2 highlights the highest vibrations on the cylinders and on the discharge bottles and piping. To determine if vibrations are pulsation-induced and to identify the excitation source, it is helpful to understand the frequency content of the measured vibrations. If acoustic responses of the bottles and piping do not coincide with the measured vibration frequencies, it is likely the vibrations are primarily due to mechanical responses that may benefit from improved mechanical restraint rather than the installation of orifice restrictions or other acoustic solutions.

For example, the frequency plot depicted in Figure 3 shows a vibration amplitude of approximately 1.3 ips (33.02 mm/s) at 85.7 Hz (5146 cpm), or 4X running speed, in the vertical direction on the Unit 1 discharge elbow. Other notable vibration amplitudes occurred in the horizontal direction on the right-side discharge spool piece. In other words, the pipe is moving perpendicular to the crankshaft, or in the direction of cylinder stretch. Figure 4 shows a vibration amplitude of approximately 0.9 ips (22.86 mm/s) at 64.3 Hz (3858 cpm), or 3X running speed, and Figure 5 shows a vibration amplitude of approximately 1.1 ips (27.94 mm/s) at 128 Hz, or 6X running speed.

There are many publications addressing the acceptability of vibration levels on the compressor and associated piping. For high-speed compressors, the authors' preference is to keep the overall vibration levels on the outer end of the cylinder below 0.8 ips (20.32 mm/s), 0-peak. Vibration levels on the bottles, scrubber(s), and piping should be evaluated using the severity chart shown in Figure 6. The vibration severity chart typically used by Kinder Morgan for evaluating vibration was produced by Southwest Research Institute. It is important that the maximum vibration frequencies in problematic areas be documented to determine the source of any pulsation energy that may be exciting the vibration. The maximum vibration amplitudes are noted on the severity chart to better illustrate any vibration amplitudes that may require correction.

From Figure 6, we see that the highest individual peak vibration amplitudes are in the marginal-to-correction range. Although we were unable to document the levels due to safety concerns, we know that vibration is even more excessive at higher speeds. Understanding which frequencies are contributing most to the overall vibration amplitudes helps to determine what might be causing the vibration. The next few steps will help us determine if the vibrations of concern are likely pulsation-induced.

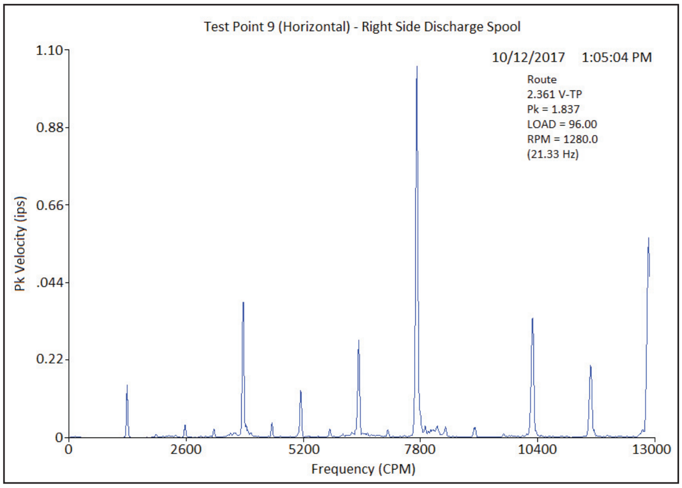

Figure 5. Frequency Plot Of Vibration Amplitudes At Test Point 9 On Unit 1

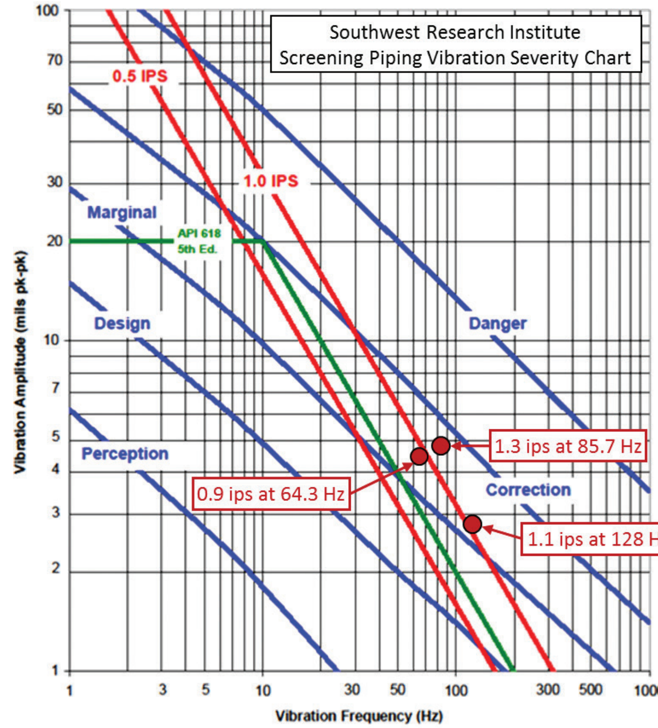

Figure 6. Vibration Severity Chart With Critical Vibration Amplitudes Identified

STEP 2 – Determine Speed Of Sound

To evaluate the impact of reciprocating compressor-generated pulsations on the vibration characteristics of the packaged compressor piping system, it is necessary to know the speed of sound in the medium. There are multiple tools available for calculating the speed of sound. For this paper, a free software tool available from ACI Services Inc., eRCM Gas Modeling Software, was used. To calculate the speed of sound, the user would input the mole percent of each gas constituent, along with the atmospheric pressure, suction pressure, discharge pressure, and suction temperature. Any significant changes to these conditions in the future would affect the speed of sound and the acoustic responses. For example, a heavier gas (higher Btu content) would lower the speed of sound and the acoustic response frequencies.

The eRCM Gas Modeling Software speed of sound tool in Figure 7 was used to calculate the speeds of sound for the suction and discharge piping systems, which are shown to be 1430 and 1521 ft/s (435.9 and 463.6 m/s), respectively.

STEP 3 – Estimate Acoustic Response Frequencies

Once the speed of sound is known, it is time to determine if the pulsation might be contributing to the vibration problems. Dimensioned drawings of the pulsation bottles and packaged piping are necessary to complete this task. Because the vibration levels were highest on the discharge side, the estimated acoustic response frequencies will be presented only for the discharge bottles and piping at this point. There are two identical discharge pulsation bottles on each compressor skid, one on each side of the frame. As shown in Figure 8, each discharge bottle has three chambers separated by internal dished head baffles, two cylinder chambers, and one line chamber. Each cylinder chamber has its own choke tube that carries gas from the cylinder to the line chamber.

The acoustic response frequencies associated with the cylinder nozzle and the internal gas passages of the compressor cylinder are difficult to estimate due to the complexity of the passages. If the compressor operates over a wide speed range, it is not possible to avoid placing this response somewhere in the speed range. Because of this, on high-speed units, orifice plates are often located at the compressor cylinder flange connections to reduce pulsations at the acoustic response frequencies associated with the cylinder nozzle and the internal gas passages. If orifices are present at the cylinder flange connections, then what can be done with relative ease to minimize pulsations at these response frequencies has been done. If orifices are not installed at the cylinder flanges connections, it is reasonable to consider the acoustic response frequencies associated with the cylinder nozzle and gas passages as a potential source of vibration. At a minimum, it is good practice to install full bore spacer plates at the cylinder flange connections if orifices are not initially installed so they can be installed later if needed. In this case, orifices are located at the cylinder flanges.

Next, we look at excitation frequencies for which we are able to better estimate. To do this, it is important to understand that all finite pipe spans will have acoustic response frequencies associated with the length of the span and the speed of sound of the gas. The points of maximum pressure are dependent on the boundary conditions at each end of the pipe (open or closed) and the harmonic. Figure 9 shows the first two harmonics associated with the various boundary conditions.

Notice that an open-closed pipe (such as a line to a closed bypass or recycle valve) will always have a pressure maximum at the closed end; so, the highest pulsation amplitudes associated with this line will be measured at the closed valve. The quarter-wave response frequencies associated with an open-closed pipe are defined by Equation (1).

$$f = (2n - 1)\frac{a}{4L} \qquad (1)$$

where f is frequency (Hz); a is speed of sound (ft/s); L is equivalent acoustic length (ft); and n is harmonic (1, 2, 3, …).

The first mode of an open-open pipe (such as a line between two bottles) will have a pressure maximum at the midpoint of the pipe span and pressure minima at the open ends, while the second mode will have pressure maxima at the 1/4 and 3/4 points with pressure minima at the open ends and at the midpoint. The half-wave response frequencies associated with an open-open pipe are defined by Equation (2).

$$f = n\frac{a}{2L} \qquad (2)$$

where f is frequency (Hz); a is speed of sound (ft/s); L is equivalent acoustic length (ft); and n is harmonic (1, 2, 3, …).

A closed-closed pipe (such as a pulsation bottle chamber or scrubber) will always have pressure maxima at the closed ends. The first mode will have a pressure minimum at the midpoint of the chamber, while the second mode will have pressure minima at the 1/4 and 3/4 points. The half-wave response frequencies associated with a closed-closed pipe are also defined by Equation (2).

For the case under evaluation, we will estimate the acoustic response frequencies associated with the pulsation bottle chamber lengths and the on-skid discharge piping. Because the flow areas of choke tubes and line connections are typically small compared to the bottle flow area, a bottle chamber can be considered a pipe with two closed ends. To determine the acoustic length of a bottle chamber, it is necessary to calculate the volumes of heads and internal baffles. Equivalent lengths for the heads and baffles are estimated by dividing these volumes by the circumferential area of the bottle.

For the discharge pulsation bottle shown in Figure 8, the chamber lengths are about equivalent. Using the calculated discharge speed of sound of 1521 ft/s from Step 2 and an estimated chamber length of 3.5 ft. (1.07 m), Equation (2) estimates the first mode of the chamber length at 217.3 Hz. Since this frequency is much higher than the measured vibration frequencies of concern, no further review is required. It should be noted that, for these bottles, the first mode of the cylinder chamber length would not be excited since the cylinder connection is at the acoustic midpoint of the chamber and the standing wave has a pressure node at the midpoint of the chamber.

Equation (2) can also be used to estimate the acoustic response frequencies associated with the internal choke tubes within the discharge pulsation bottle. The physical length of each choke tube shown in Figure 8 is 64 in. (1625.6 mm). The equivalent acoustic length of a choke tube can be estimated by adding an end correction of approximately 0.3 times the internal diameter at each end, which would make the acoustic length of each choke tube 65.7 in. (1668.8 mm). Using the calculated discharge speed of sound of 1521 ft/s and a choke tube length of 65.7 in., the first mode of the choke tube length would be 138.3 Hz. Again, this frequency is higher than the measured vibration frequencies of concern.

Another length response that can impact vibration of the on-skid discharge piping is associated with the length between the two discharge bottles. Figure 2 shows the configuration of the piping between the Cylinder 1-3 discharge bottle and the Cylinder 2-4 discharge bottle. The physical centerline length of this pipe is about 27.4 ft. (8.35 m). Note that the pipe length is a rough estimate due to the lack of dimensions on the drawings. Using Equation (2) with a speed of sound of 1521 ft/s and a pipe length of 27.4 ft., the first mode of this length response would be 27.8 Hz. The second and third modes of this length response would be 55.5 and 83.3 Hz, respectively. The third mode of the discharge pipe length response is reasonably close to one of the vibration frequencies identified on the severity chart in Figure 6; however, an adequately sized volume-choke-volume acoustic filter would effectively minimize pulsation amplitudes at frequencies associated with piping lengths downstream of the bottles, but further review is required.

A simple volume bottle will attenuate pulsations in the attached piping, but will not effectively eliminate high-frequency pulsations associated with acoustic responses in the piping. An adequately sized volume-choke-volume acoustic filter design will minimize excitation of acoustic response frequencies in the attached piping at frequencies higher than the filter cutoff frequency, with the exception of acoustic response frequencies associated with the acoustic filter elements. The discharge pulsation bottle shown in Figure 8 clearly shows a volume-choke-volume acoustic filter design. The bottle is comprised of three separate volume chambers, two cylinder chambers, and one line chamber. An individual choke tube connects each cylinder chamber to the line chamber. To determine the effectiveness of the filter, it is necessary to calculate the Helmholtz (filter) frequency associated

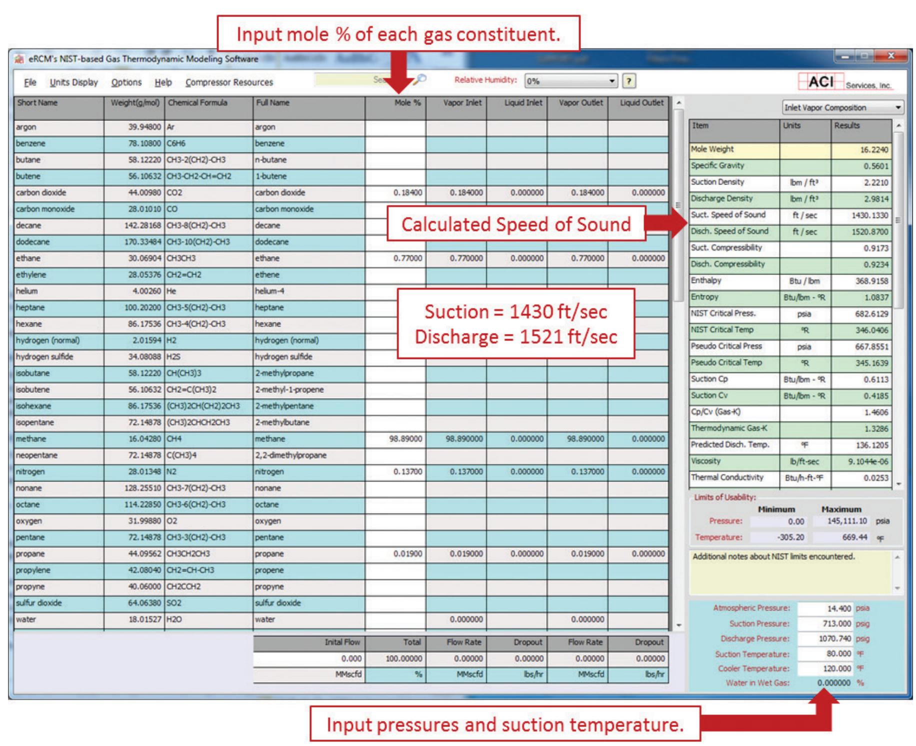

Figure 7. Speed Of Sound Calculation

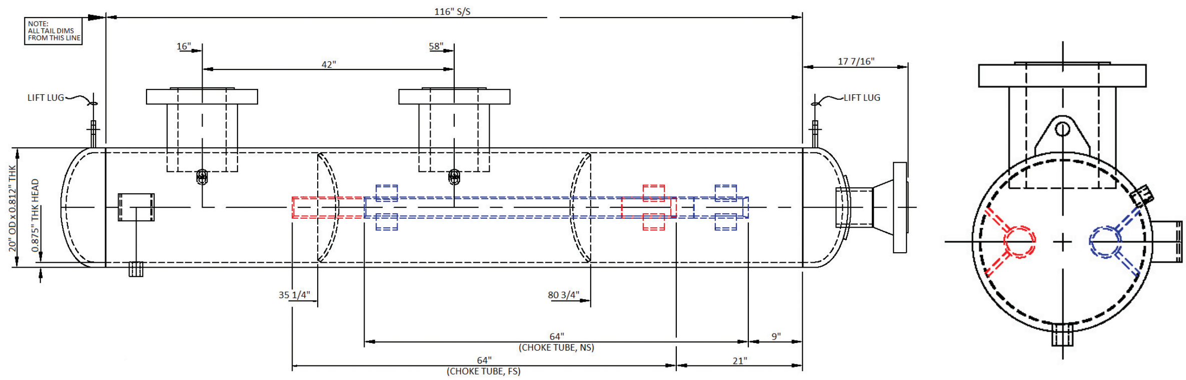

Figure 8. Discharge Pulsation Bottle

with each volume-choke-volume configuration. Equation (3) can be used to calculate the filter frequency and its associated cutoff frequency. In this case, only one filter frequency will be calculated because the bottle chambers are all equal and the choke tubes are all the same length.

A volume-choke-volume filter frequency of about 12.5 Hz is calculated using Equation (3). Pulsations above the cutoff frequency of 17.7 Hz will be effectively minimized in the attached piping, with the exception of pulsations at frequencies associated with the filter elements. It is expected that pulsations at frequencies associated with the filter elements will be controlled through proper placement of the acoustic response frequency, along with the pressure drops through choke tubes and orifices.

(3) $$f_H = \frac{a}{2\pi}\left(\frac{A}{LV_1}+\frac{A}{LV_2}\right)^{\frac{1}{2}},\ f_{co} = \sqrt{2}\, f_H$$

where f_H is Helmholtz (filter) frequency (Hz); f_{co} is cutoff frequency (Hz); a is speed of sound (ft/s); A is the cross-sectional area of choke (ft^2); L is the acoustic length of choke (ft); V_1 is the volume of the cylinder char-ber (ft^3); and V_2 is the volume of the filter chamber (ft^3).

With an adequately sized acoustic filter, we know that pulsations associated solely with the acoustic length responses of the discharge piping will be minimized. However, pulsations associated with the compressor nozzle and with elements of the acoustic filter (bottle chamber and choke tube lengths) will pass through the filter.

STEP 4 – Create Pulsation And Vibration Frequency Interference Chart

Once the acoustic response frequencies are known, they can be compared with the pulsation frequencies generated by the compressor to determine if pulsation energy is a likely source of excitation for the measured vibration. Pulsation frequencies generated by a reciprocating compressor can be determined from Equation (4).

(4) $$f = n\,\frac{N}{60}$$

where f is frequency (Hz); N is compressor speed (rpm); and n is compressor order or multiple of run speed (1, 2, 3,…).

The reciprocating compressor-generated pulsation frequencies are shown in Figure 10 for a machine operating over a speed range of 1200 to 1400 rpm. The blue bars show the range of frequencies for the first 10 multiples of compressor speed (1X, 2X, 3X, etc.). The acoustic response frequencies estimated in Step 3 are identified by the vertical blue lines, and the vibration frequencies plotted on the vibration severity chart in Figure 6 are identified by the vertical red lines. Because vibration data were only measured in the 1280- to 1300-rpm speed range, Figure 10 shows coincidence between pulsation and vibration frequencies only at these speeds.

To summarize what we have learned so far, while operating over a 1280- to 1300-rpm speed range, there were three vibration frequencies of concern, 64.3 Hz (3X running speed), 85.7 Hz (4X running speed), and 128 Hz (6X running speed). We know the vibration increases at higher speeds, but we were not able to acquire frequency and amplitude data at these speeds due to safety concerns. We do not have impact data and do not know the mechanical natural frequencies of the pulsation bottles and piping. It is not clear at this point if the suction bottle motion may be contributing to vibration of the discharge bottles and piping. We also do not have measured pulsation data and can only determine if pulsation is a potential issue based on the calculated results from Step 3.

Looking at the pulsation and vibration frequency interference chart in Figure 10, we see that the third mode of the acoustic

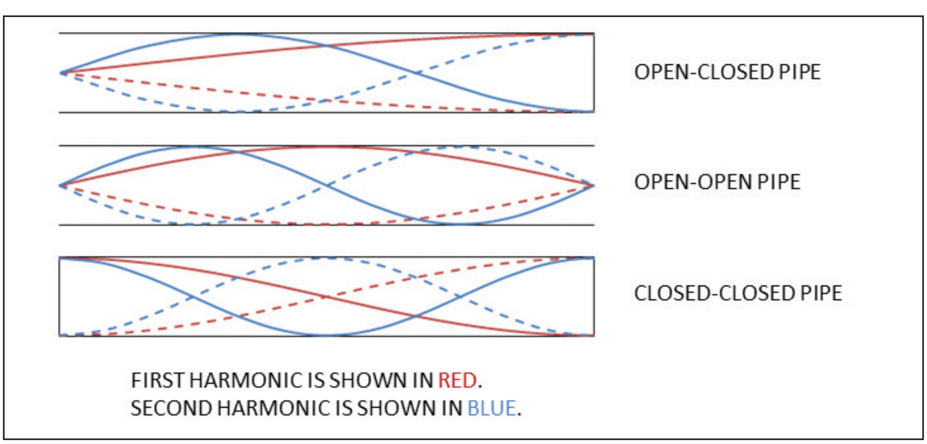

Figure 9. Standing Wave Patterns

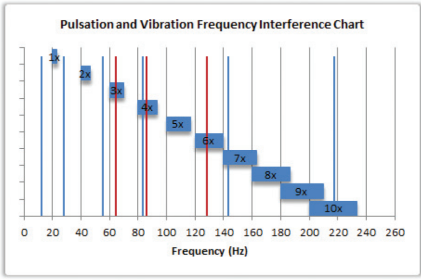

Figure 10. Pulsation And Vibration Frequency Interference Chart For 1200 To 1400 rpm

Figure 11. Discharge Pulsation Bottle Supports As Initially Installed

Figure 12. Discharge Pipe Supports As Initially Installed

response frequency associated with the pipe between the two discharge bottles can be excited by pulsation from the compressor at 4X running speed. We also see a vibration frequency of concern very near the acoustic response frequency; so, there is likely some amplitude of pulsation that is exciting the vibration. Since we know from Step 3 that the volume-choke-volume acoustic filter cutoff frequency is below 1X minimum running speed (20 Hz at 1200 rpm), which is sufficiently low to minimize pulsation energy at frequencies associated with the discharge piping downstream of the pulsation bottles, we will need to evaluate design and installation of the bottle and pipe supports.

STEP 5 – Evaluate The Bottle And Pipe Supports

Although pulsation energy is often a primary source of excitation, vibration is often significantly reduced by simply ensuring adequate restraint of the bottles and piping. In this case, because there were no obvious sources of excessive pulsation energy, it was determined that the high vibrations were likely due to mechanical responses of the bottles and piping acting together. Repair or replacement of bottle and pipe supports is typically the most cost-effective solution as well, provided there are no obvious issues with the design of the pulsation bottles.

Figure 13. Discharge Bottle Support Replacement

The bottle and pipe supports on the compressor package under investigation were found to be inadequate for the application. As shown in Figure 11, the initially installed shim blocks under the discharge pulsation bottles were very narrow and there were no clamps on the bottles.

Pipe supports on the piping downstream of the bottles were also found to be inadequate. The straps were not gusseted, limiting lateral stiffness, and there was no liner material between the inside surface of the strap and the pipe to improve contact and reduce the risk of wear. As shown in Figure 12, vibration had already loosened the strap nearest the discharge bottle. The pipe straps were attached to a piece of channel that also appeared to have inadequate stiffness.

The lack of good mechanical support on standard high-speed compressor packages has become all too common. A typical high-speed package will have an adequately designed pulsation filter, but will often have insufficient mechanical support. With the exception of the pipe supports located on the pipe running parallel to the cooler, which were not easily accessible, all bottle and pipe supports on the two identical packages evaluated were replaced with E-Z Line clamp and shim block assemblies to add mechanical stiffness. Figure 13 illustrates the difference in the quality of support for the discharge pulsation bottle. Although we would have preferred to have a clamp near the weld seam on the line end of the discharge bottle, it was not possible in this case without modifying the skid structure below the bottle. It is important that supports be located directly over skid beams to achieve adequate stiffness. There should always be skid beams located under the discharge bottle beneath the cylinders, between the cylinders, and inside the weld seam on the long end of the bottle.

Figure 14 shows the difference in the quality of support for the on-skid discharge piping. Notice that the channel to which the original pipe straps were attached was replaced with a wide flange beam that spans the existing skid beams. The surface of the shim blocks and the inside of the clamp have a Viblon liner to eliminate metal-to-metal contact, which can cause wear. The original straps did not have any type of liner material. It should be noted that the support modifications required relocation of existing conduit and auxiliary water lines.


Figure 14. Discharge Pipe Supports Replacement

It would have been much easier to make these changes when the packages were initially fabricated.

A vibration survey following replacement of the original bottle and pipe supports showed significant improvement. Prior to implementing the support modifications, it was not possible to operate the unit at speeds above 1300 rpm due to safety concerns. Overall, vibration amplitudes of approximately 4 ips had been reported when attempting to operate near 1400 rpm, resulting in the speed reduction. Figure 15 summarizes the overall vibration amplitudes before and after replacement of the supports. Vibrations exceeding 1.5 ips are shaded in red. Notice the overall reduction in horizontal and vertical vibration of the cylinders, bottles, and piping that was achieved simply by improving the mechanical support system. Vibration amplitudes monitored on Unit 1 as the speed ramped up from 1254 to 1400 rpm indicated that the highest vibrations still occurred at maximum speed; so, vibration amplitudes documented at these speeds show the range of vibration amplitudes for these units. A comparison of the vibration data summarized in Figure 15 clearly indicate that vibrations on Unit 1 were reduced to less than 2 ips (50.8 mm/s) over the speed range. In fact, overall vibrations only exceeded 1.5 ips in the horizontal direction on the left-side discharge elbow on Unit 1. Vibrations on Unit 2 were less than 1 ips (25.4 mm/s) in the vertical direction and less than 1.3 ips in the horizontal direction, with maximum vibration again on the elbow. Refer back to Figure 2 for the vibration measurement locations.

The remaining vibration in this area could be due to the fact that the supports on the pipe running parallel to the cooler were not replaced and that it was not possible to locate a support as close to the elbow as would be preferred due to the flange pair welded directly to the elbow. There could also be some pulsation excitation, although the levels should not be excessive. Low-level pulsation energy can excite vibration if the bottles and piping are not adequately restrained. That the units were able to operate at 1400 rpm was, in itself, a significant improvement. However, the high vibration in the area of the left-side discharge elbow remains a concern. The higher vibration on Unit 1 could be due to slightly different loading or there could be some installation differences. Operations personnel continue to monitor this area of the piping, retightening clamps as necessary until we are able to further evaluate our options.

At this point, it would be good to examine more closely the two compressor packages to see if there are any noticeable differences in how the pipe and pipe supports are installed that might explain the higher vibration on Unit 1. An impact test could be performed to document the mechanical natural frequencies of the bottles and piping. This type of test may require assistance from a vibration consultant with better tools for this type of analysis.

Other options to consider are how we might replace or otherwise improve the supports on the pipe running parallel to the cooler or, perhaps, to obtain pulsation data so that we can determine if either reducing the bore of the cylinder nozzle orifices or adding orifices at the discharge pulsation bottle outlets would be beneficial. Figure 16 shows the elbow where vibration remains relatively high. As can be seen from the picture, there is little room to install gusseted supports and Figure 2 shows limited access to skid beams toward the middle of the pipe span. More effective restraint could be achieved with gusseted clamps located near the elbow and near the tee on the other side of the skid.

If pulsation energy in the piping corresponds to pulsation energy in the cylinder nozzles, adjusting the cylinder nozzle orifice size might be beneficial. If pulsation energy in the piping corresponds to a mode of the pipe length between the discharge bottles, it might be beneficial to install orifices at the bottle-line flanges. We have had success at other locations with the dynamic variable orifice (DVO), which allows us to vary the orifice flow area with the operating speed or some other parameter, which, in turn, helps to minimize the pressure drop. The orifice flow area can be tuned such that the flow area is no smaller than necessary to control the vibration. The variability of the DVO also allows us to install the orifice without any significant modeling requirements. Figure 17 shows two configurations for the DVO. The flat DVO allows a variation in beta ratio (orifice inner diameter [ID]/pipe ID) of 0.4 to 0.7, while the conical DVO allows a variation in beta ratio of 0.4 to 0.9.[1] It should be noted that a DVO is thicker than a conventional orifice plate and would require more spacing between the flange faces. In addition, a conical DVO would need to be inserted in a pipe spool.

From an operator's perspective, it is important that the pulsation bottles are properly designed to minimize pulsations in the attached piping. For high-speed units, acoustic filtering techniques are typically the best option for pulsation control, although volume and/or orifices may be sufficient for low-pressure, low-flow applications. It is important that orifices be installed as needed at the compressor cylinder flanges to minimize pulsations associated with the cylinder nozzle and gas passage configuration. At a minimum, full bore spacer plates should be located at the compressor cylinder flanges to allow for future installation of orifices if necessary. It is also recommended that full bore spacer plates be located at the inlet flanges to any on-skid scrubbers and at the discharge bottle outlet flanges. If there are no on-skid scrubbers, full bore spacer plates should be located at the suction bottle inlet flanges. Another option would be to install DVOs at these locations. For conditions at which the DVO is not needed, the flow area can be maximized to reduce pressure drop.

Horizontal Vibration (ips, 0-peak)				
	Unit 1			Unit 2
	Original	After Support Modifications		
Test Point Location	1300 rpm	1254 rpm	1400 rpm	1400 rpm
Cylinder 1	2.21	0.43	NA	NA
Cylinder 2	0.64	NA	NA	NA
Cylinder 3	0.77	0.4	NA	NA
Cylinder 4	0.48	NA	NA	NA
Test Point Location	1285 rpm	1254 rpm	1400 rpm	1400 rpm
1L-Left Side Discharge Bottle	1	0.55	0.84	0.91
3L-Left Side Discharge Bottle	0.75	0.45	0.46	0.46
1R-Right Side Discharge Bottle	1.14	0.43	0.75	0.85
3R-Right Side Discharge Bottle	1.82	0.76	1.19	0.64
4-Left Side Discharge Pipe Spool	1.02	0.87	1.17	1.13
5-Left Side Discharge Pipe Spool	1.32	0.57	0.72	0.62
6-Left Side Discharge Pipe Spool	1.36	1.22	1.82	0.92
7-Left Side Discharge Elbow	0.98	1.35	1.96	1.28
8-Right Side Discharge Pipe Spool	2.05	0.69	1.08	0.64
9-Right Side Discharge Pipe Spool	2.36	0.43	0.33	0.42
10-Right Side Discharge Pipe Spool	1.62	0.45	0.53	0.4
11-Discharge Pipe To Cooler Inlet	NA	NA	NA	NA

	Vertical Vibration (ips, 0-peak)				
	Unit 1				Unit 2
	Original	After Support Modifications			
Test Point Location	1300 rpm	1254 rpm	1400 rpm	1400 rpm	1400 rpm
Cylinder 1	0.41	0.29	NA	NA	NA
Cylinder 2	0.22	NA	NA	NA	NA
Cylinder 3	2	0.19	NA	NA	NA
Cylinder 4	0.45	NA	NA	NA	NA
Test Point Location	1285 rpm	1254 rpm	1400 rpm	1400 rpm	1400 rpm
1L-Left Side Discharge Bottle	0.81	0.34	0.42	0.37	0.91
3L-Left Side Discharge Bottle	0.44	0.13	0.23	0.24	0.46
1R-Right Side Discharge Bottle	0.78	0.28	0.44	0.37	0.85
3R-Right Side Discharge Bottle	0.75	0.21	0.39	0.35	0.64
4-Left Side Discharge Pipe Spool	0.79	0.24	0.54	0.35	1.13
5-Left Side Discharge Pipe Spool	1.06	0.35	0.43	0.73	0.62
6-Left Side Discharge Pipe Spool	0.61	0.39	0.35	0.26	0.92
7-Left Side Discharge Elbow	1.9	0.98	1.3	0.86	1.28
8-Right Side Discharge Pipe Spool	0.55	0.44	0.71	0.5	0.64
9-Right Side Discharge Pipe Spool	1.12	0.28	0.56	0.32	0.42
10-Right Side Discharge Pipe Spool	0.78	0.27	0.71	0.46	0.4
11-Discharge Pipe To Cooler Inlet	0.64	0.26	0.54	0.42	NA

	Axial Vibration (ips, 0-peak)				
	Unit 1				Unit 2
	Original	After Support Modifications			
Test Point Location	1300 rpm	1254 rpm	1400 rpm	1400 rpm	1400 rpm
Cylinder 1	0.49	0.22	NA	NA	NA
Cylinder 2	0.29	NA	NA	NA	NA
Cylinder 3	0.42	0.28	NA	NA	NA
Cylinder 4	0.28	NA	NA	NA	NA
Test Point Location	1285 rpm	1254 rpm	1400 rpm	1400 rpm	1400 rpm
1L-Left Side Discharge Bottle	0.58	0.23	0.37	0.3	0.91
3L-Left Side Discharge Bottle	NA	NA	NA	NA	0.46
1R-Right Side Discharge Bottle	1.06	0.21	0.59	0.37	0.85
3R-Right Side Discharge Bottle	NA	NA	NA	NA	0.64
4-Left Side Discharge Pipe Spool	1	0.28	0.61	0.47	1.13
5-Left Side Discharge Pipe Spool	NA	NA	NA	0.47	0.62
6-Left Side Discharge Pipe Spool	1.09	0.24	0.59	0.28	0.92
7-Left Side Discharge Elbow	1.18	NA	NA	NA	1.28
8-Right Side Discharge Pipe Spool	1.03	0.47	0.58	0.36	0.64
9-Right Side Discharge Pipe Spool	NA	NA	NA	0.41	0.42
10-Right Side Discharge Pipe Spool	1.34	0.46	0.88	0.66	0.4
11-Discharge Pipe To Cooler Inlet	1.31	0.44	0.52	0.34	NA

Figure 15. Comparison Of Overall Vibration Levels Before And After Support Replacement

Since standard packaged designs often include insufficient mechanical restraints for the bottles and piping supports, it is important that operators specify the preferred restraint types and locations. This should be done before the package is fabricated because it may be necessary to add steel to the skid structure. There should be sufficient steel beneath the discharge bottles so that pipe clamps and shim blocks can be located between each cylinder pair if needed and near the weld seam on the long end of extended pulsation bottles. Pipe clamps should be gusseted to increase lateral stiffness and a liner material should be bonded to all surfaces in contact with the pipe.[2]

In general, supports should be located at all directional changes such as tees and elbows and on both sides of valves. On-skid supports should be mounted over a skid beam. Ideally, the skid structure should have steel adequately placed to accommodate supports at the recommended locations. In cases where skid beams are not available in locations that require a pipe support,

Figure 16. Left-Side Discharge Pipe Elbow

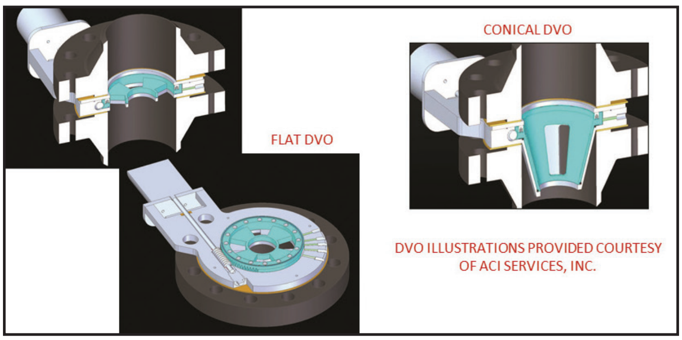

Figure 17. Dynamic Variable Orifice

a long horizontal beam can be installed directly under and in line with the pipe axis. The beam would be welded to the top of two or more existing skid beams, allowing some flexibility in locating necessary supports.

The intent of this paper was to provide equipment operators with a methodology for evaluating vibrations on the bottles and piping associated with high-speed reciprocating compressors and discuss options for cost-effective solutions to vibration problems. It is hoped that the case study presented will help others with similar issues.

REFERENCES

[1]N. Shade, T. Clark, J. Adair, "Evaluation of a Dynamic Variable Orifice for Reciprocating Compressor Pulsation Control," Gas Machinery Conference (2014).

[2]F. Fierro, A. Rivera, "Comparison of Pipe Support Effectiveness and Recommended Applications," Gas Machinery Conference (2018).

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