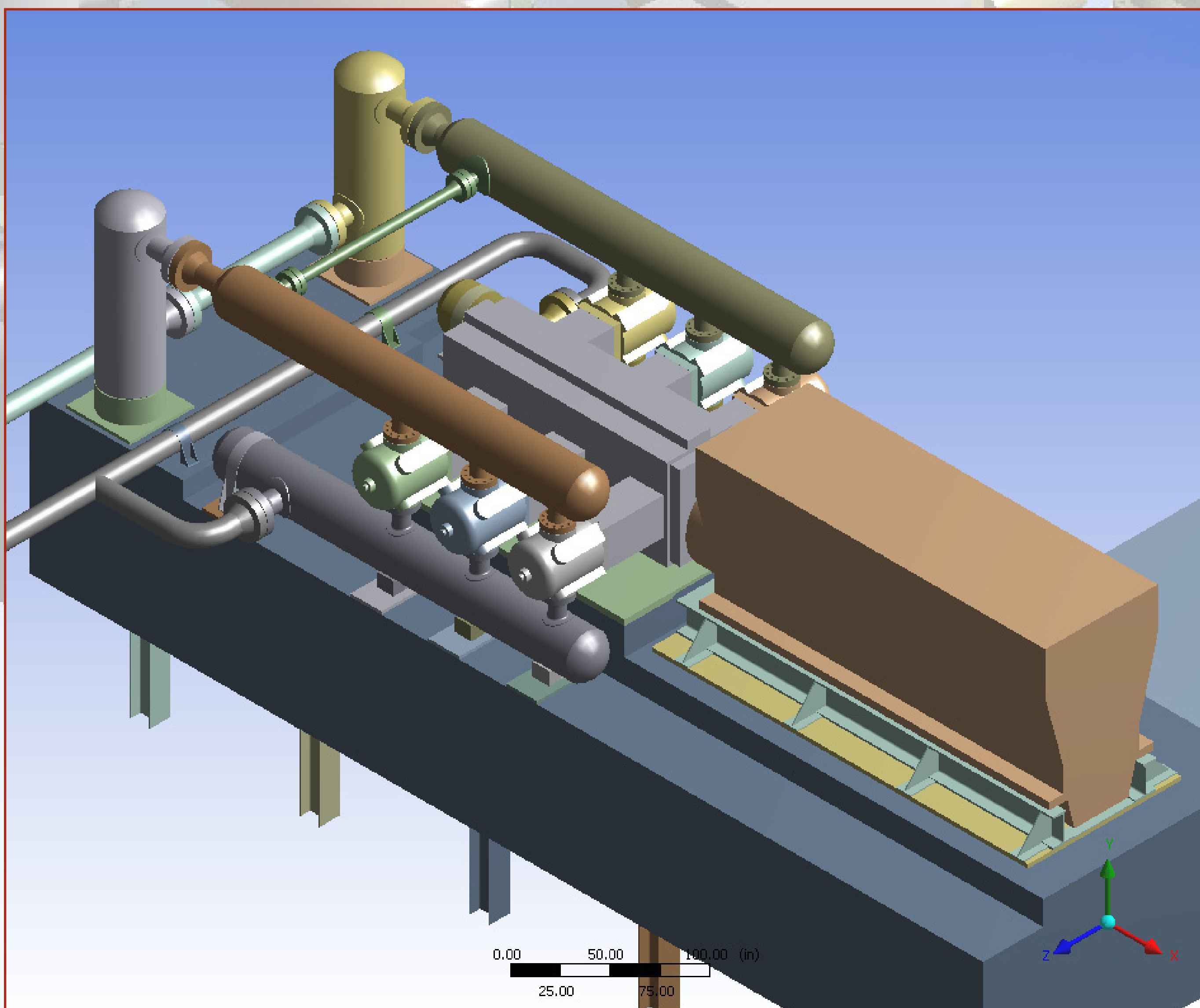
MAGAZINE ARTICLE REPRINTS

MECHANICAL MODELING OF COMPRESSOR MANIFOLD SYSTEMS

Reprinted from Compressor Tech Two Magazine
March, April-May & June 2006 Issues



SOUTHWEST RESEARCH INSTITUTE®

Benjamin A. White, P.E., is a Senior Research Engineer at Southwest Research Institute®, San Antonio, Texas, U.S.A., where he is primarily responsible for the mechanical design and analysis of a variety of compressed fluid systems. He has extensive experience in vibration control and stress analysis for a wide variety of compressor manifold and piping systems at both the design stage and for revamp work on existing installations. He is a Cum Laude graduate of Texas A&M University, College Station, Texas, U.S.A., with a B.S. degree in Mechanical Engineering. He is a registered professional engineer in the State of Texas and a member of the American Society of Mechanical Engineers (ASME).

This paper was originally published in three parts in COMPRESSORTechTwo. Part I (March 2006) includes a basic introduction and discusses current modeling technology. Part II (April-May 2006) is a case study comparing different modeling tools. Part III (June 2006) discusses mechanical manifold modeling as it applies to liquefied natural gas (LNG) systems.

For additional information, questions and references, contact the author at benjamin.white@swri.org.

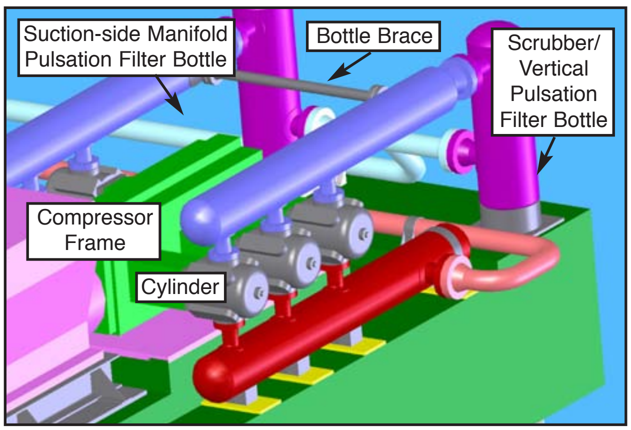
MECHANICAL MODELING OF COMPRESSOR MANIFOLD SYSTEMS

As Reciprocating Compressor Designs Continue to Evolve Over Time, it is Important That the Design Analysis Tools Also Keep up to Date

By Benjamin A. White, P.E.

PART I

For most new reciprocating compressor installations following the API Standard 618 design methodology, a Design Approach 3 pulsation and vibration analysis should be done. (API Standard 618, *Reciprocating Compressors for Petroleum, Chemical and Gas Industry Services, Fourth Edition*, American Petroleum Institute, Washington, D.C., 1995.) Items M5 and M6 in API 618 specifically cover the mechanical compressor manifold system analysis, often simply referred to as a "Mechanical Analysis."



■ **Figure 1. Typical six-cylinder gas compressor manifold system.**

Mechanical modeling, in concert with a pulsation and performance analysis, attempts to balance compressor performance with acceptable operating stress and vibration levels within the installation, which leads to safe and reliable unit operation. A mechanical analysis should identify and position key system resonant frequencies at optimal points, relative to integer orders of compressor running speed. Stress and vibration levels at key locations within the manifold system are predicted based on residual compressor pulsation levels in the system and cylinder vibration levels. Manifold design parameters, including nozzle wall thickness, pulsation filter bottle re-enforcing pad thickness and size, bottle wall thickness and bottle support locations can all be tailored to reduce stress and vibration levels to acceptable values.

In the past, the mechanical modeling of gas compressor manifold systems was done using beam-type structural elements. Such models represent each element in the system as a simple beam. The primary design analysis advantages of beam-type models are their geometric simplicity and the minimal computational time required to obtain results of a stress and vibration analysis. From an engineering standpoint, the primary disadvantage is how the cylinder nozzle to pulsation filter bottle connection is modeled. Beam elements work best when the elements have a relatively high (>3:1) length-to-diameter ratio. When element lengths become short, such as for cylinder nozzles, a significant part of the flexibility exists in the deflection of the filter bottle wall. Beam element models have attempted to address this issue by assigning an additional estimated flexibility or "Flex Factor" at critical junctions. This is a necessary but imperfect solution.

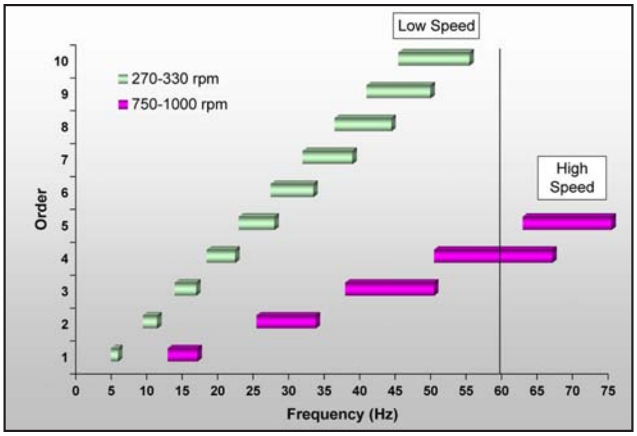
Over the years, there have been evolutionary changes in basic compressor design. For instance, pulsation filter bottles have become larger in diameter, resulting in (relatively) more flexible bottle walls. Compressor cylinder nozzles have become shorter, resulting in higher stresses caused by cylinder stretch. Peak operating speeds have gotten higher and the units now tend to be operated over wide speed ranges, rather than at a fixed speed. High-horsepower units on separable skid systems are more common. Cost-saving efforts have resulted in more flexible compressor skids, smaller reinforcing pads at filter bottle connections, and fewer outboard cylinder supports. As reciprocating compressor designs continue to evolve over time, it is important that the design analysis tools also keep up to date.

One of the most important compressor system mechanical responses to predict is the primary response of the suction-side pulsation filter bottle. That particular mechanical response is primarily controlled by the mass of the bottle, the stiffness of the cylinder nozzle(s) and the stiffness of the bottle wall. As bottle diameters increase and cylinder nozzles get shorter, the flexibility of the shell wall dominates the flexibility of the cylinder nozzles. This is a non-ideal application of beam-element structural analysis.

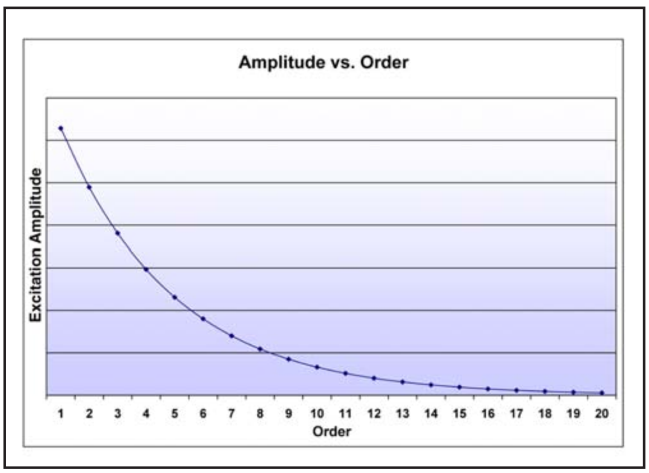
Ideally, the resonant frequencies of a mechanical system, such as a gas compressor installation, should not coincide with the frequencies of excitation. If a mechanical response must exist within the operating speed range of a compressor, accurate stress and vibration predictions are necessary. The most critical area for stresses is in cylinder nozzle to pulsation filter bottle connections. While beam models can be used to help predict dynamic stress and vibration amplitudes, they lack sufficient detail in the most critical areas. Estimated geometrical stress concentration factors must be used to overcome this analysis model shortcoming.

Most compressor units can generate significant excitation energy at frequencies up to the 10th order of running speed — or, sometimes, even higher. As compressor operating speed increases, accurate modal predictions at the higher frequencies become more important. A typical suction-side pulsation filter bottle of today could easily have two or even three major mechanical responses within the first four compressor running orders. See Figures 2 and 3, for example. The estimated "flex factors" used in a beam model might do an adequate job predicting the lowest mechanical response of a suction-side pulsation filter bottle. However, such flex factors are very mode-shape dependent. Therefore, they may not be sufficiently accurate at predicting the higher frequency mechanical responses of the same suction bottle.

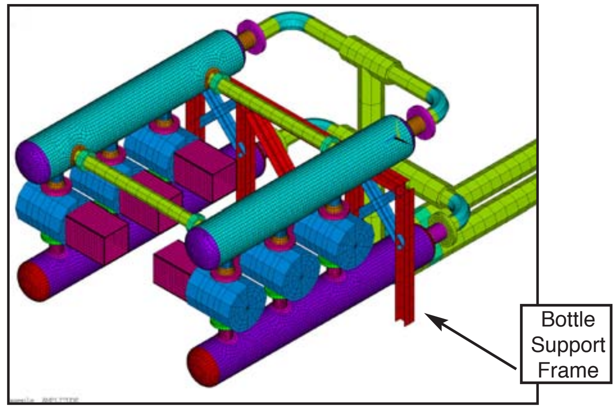
Reviewing Figure 2 in more detail, at a given frequency (e.g., 60 Hz), a low-speed compressor unit (denoted in



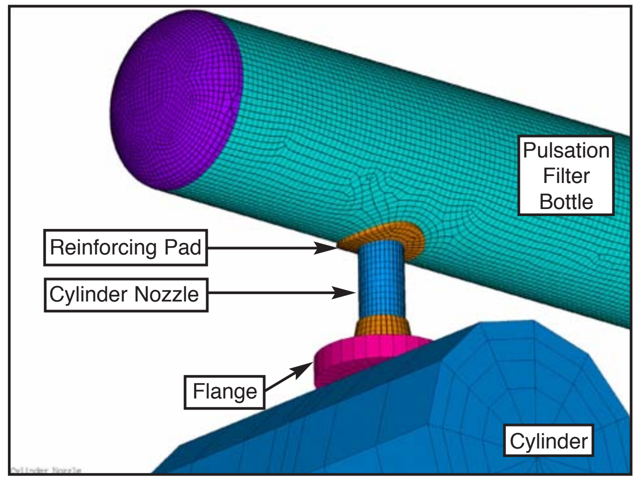
■ **Figure 2. Compressor running order comparison.**



■ **Figure 3. Excitation amplitude vs. compressor running order.**



■ **Figure 4. Typical model layout.**



■ **Figure 5. Nozzle close-up.**

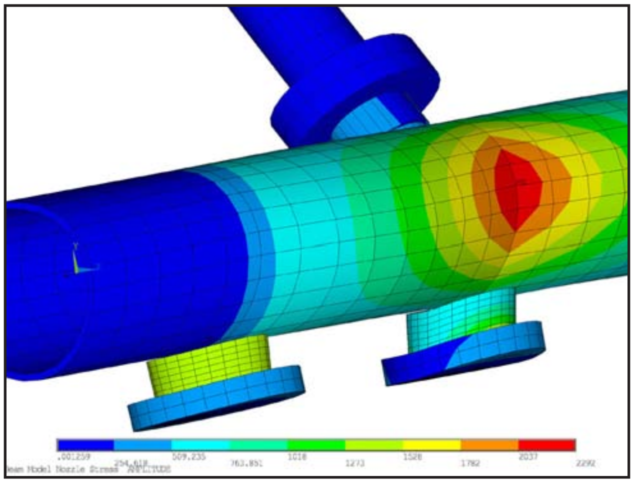
green) is already past the 10th compressor running order (at which point excitation amplitudes would tend to be relatively low), while the high-speed compressor is only at the fourth order (at which point excitation amplitudes would tend to be relatively high). Additionally, the low-speed compressor has more potential gaps between running orders (depending on how wide the speed range is).

The once-standard practice of using beam elements to model compressor manifold systems is no longer the best approach for analyzing today's reciprocating compressor designs. To address this concern, Southwest Research Institute now uses the ANSYS finite-element software package to perform all mechanical manifold analyses using three-dimensional, shell-type finite-element models. Figure 4 shows a typical model layout of a reciprocating compressor, including the pulsation control system.

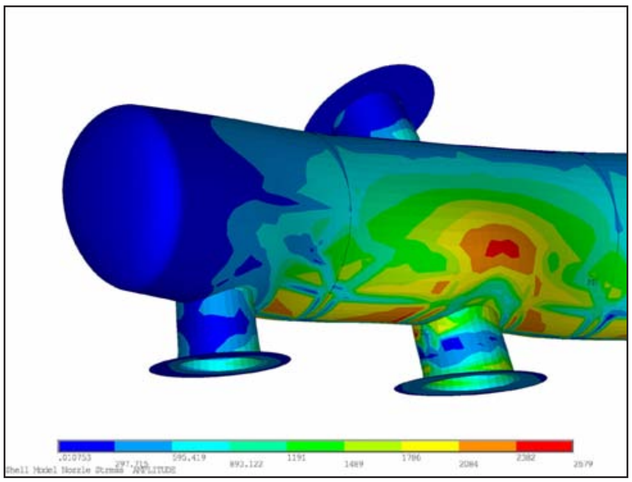
A detailed finite-element model enables a much more accurate prediction of the behavior of the cylinder nozzle to filter bottle junction, as illustrated in Figure 5. This enables accurate estimation of nozzle flexibility (in both bending and shear). It also enables accurate estimation of bottle wall flexibility (that can change as the mode shape changes), thus, providing greater accuracy over a wider frequency range. Predicted dynamic stress amplitudes in the cylinder nozzles will be more accurate, since geometrical stress concentration factors are now inherently built into the model. Figures 6 and 7 show quite different stress predictions (in both location and amplitude) between the beam and shell models.

The more detailed finite-element model also enables an accurate representation of the cylinder flanges and the flexibilities associated with joints between flanges, and between the cylinder and crosshead guide. The less critical areas of the system, such as the line piping attached to each filter bottle, can still be modeled with beam elements to reduce computational time without a significant loss of accuracy.

With today's multicylinder compressors, the compressor manifold pulsation filter bottles often have long overhangs, as a result of the acoustic damping requirements of the pulsation filter bottle design. With wide compressor operating speed ranges, it is often impossible to place the primary mechanical response frequency of such a filter bottle overhang outside the operating speed range. The concept of resonance avoidance must become resonance management, instead. To control filter bottle vibration, supporting the overhanging ends of each bottle with a frame is often necessary. Bracing the suction bottles together across the top of the compressor unit typically also is necessary. See Figure 4. Based on the model predictions of the dynamic characteristics of a compressor pulsation control system, the necessary bracing and support frames for the filter bottles can be fine-tuned.



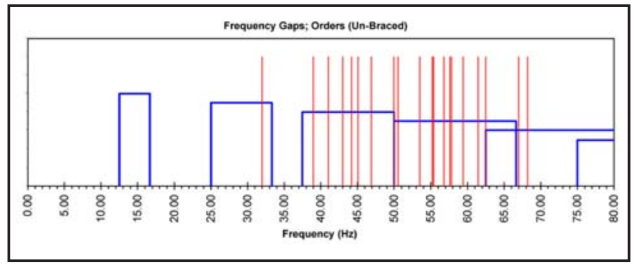
■ **Figure 6. Beam element stress.**



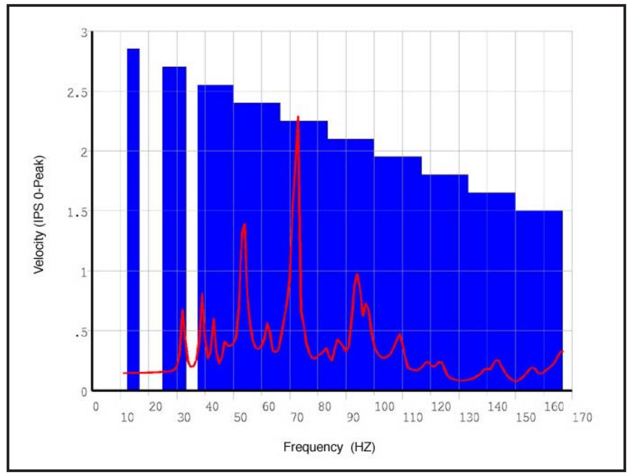
■ **Figure 7. Shell element stress.**

Figure 8 shows the predicated mechanical response frequencies for the compressor manifold system pictured in Figure 4 before the braces between the suction-side pulsation filter bottles were added. The compressor running orders from 750 to 1000 rpm are shown in the background. Figure 9 shows the predicted vibration profile for this system. Figure 10 shows the predicted mechanical response frequencies for this same system with braces added between the suction-side filter bottles. Figure 11 shows the predicted vibration profile for the braced system. The thick horizontal black lines on Figures 9 and 11 indicate a velocity of 1.0 in./s (25.4 mm/s) (zero-to-peak). It is important to note that Figures 8 and 10 look very similar (with lots of mechanical responses predicted within the operating speed range of the compressor unit, primarily coinciding with the third and fourth compressor running orders). However, the resulting vibration profiles (Figures 9 and 11) are very different. The principal difference is that the braced system prevents excitation forces (primarily cylinder stretch, in this case) from exciting the mechanical responses of the suction-side bottles. This example illustrates the concept of resonance management.

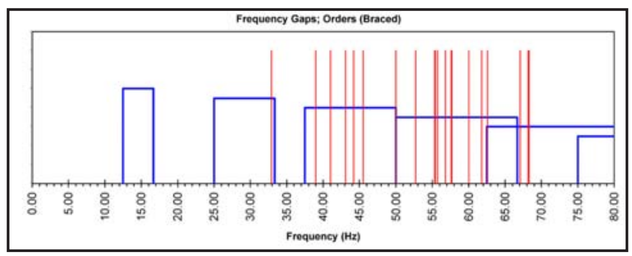
When designing a new compressor system, the specifications of the cylinder nozzles are very important. A cylinder nozzle diameter and flange rating are usually pre-determined by the cylinder selection. However, the nozzle length and wall thickness should be carefully selected. If a cylinder nozzle is too long, it becomes too flexible, thus, lowering the mechanical responses of the suction-side pulsation filter bottle. If it is too short, excessive nozzle stress can be generated because of gas compression loads (cylinder stretch). A thin-wall nozzle will make these stresses worse. Although the exact nozzle specifications need to be fine-tuned during the analysis process, as a general guideline, the length of each cylinder nozzle should be approximately one to two times its outside diameter (measured from the outside of the bottle wall to the face of the cylinder flange). Of course, this can be influenced by such factors as the flange rating, the number of cylinders per bottle, and the nozzle-to-bottle diameter ratio. In general, the cylinder nozzle wall thickness should be as thick as practical (i.e., as a minimum, Schedule XS, and, preferably, Schedule 120).



■ **Figure 8. Mechanical response frequencies (un-braced).**

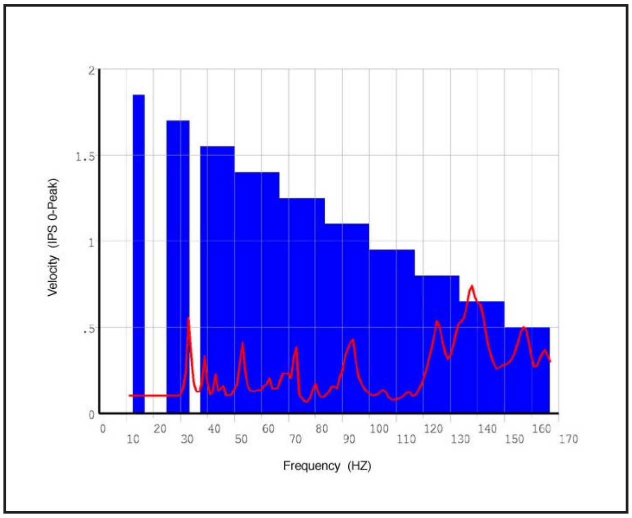


■ **Figure 9. Velocity (un-braced).**



■ **Figure 10. Mechanical response frequencies (braced).**

While finite-element shell models are more computationally intensive, today's computer capabilities enable the benefits of a more detailed model to outweigh the extra modeling effort. Considering the cost of addressing compressor system vibration problems after start-up, mechanical modeling during the design stage is a very cost-effective alternative. Potential problem areas can be identified ahead of time and solutions can be optimized during the design stage.



■ **Figure 11. Velocity (braced).**

Summarizing Some Key Points:

Mechanical modeling of reciprocating compressor systems (using the correct analysis tool) at the design stage is important. Proper mechanical modeling of the compressor pulsation filter systems can reduce the risk of installing a pulsation control system with damaging resonance conditions, high dynamic stress or high vibration conditions that can, at worst, lead to mechanical fatigue failures of system components or, at best, lead to significant maintenance issues.

With today's high-speed reciprocating compressors typically operating over a wide speed range, the concept of resonance avoidance is replaced by the concept of resonance management.

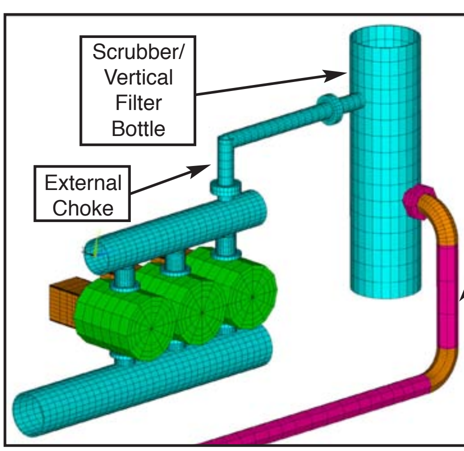
When designing a new compressor pulsation filter system, suction-side pulsation filter structural support needs should be anticipated. A support frame and bracing between the filter bottles (as per Figure 4) may be required. For average-sized suction-side pulsation filter bottles [i.e., between 18 and 36 in. (457 and 914 mm) in diameter], the average support frame would typically be built from 8-, 10- or 12-in. (204-, 250- or 305-mm) wide flanged beams or square tubing (depending on a variety of design factors, such as the exact layout of the frame and the frame height). The brace between the suction-side bottles typically would be fabricated from 6-, 8- or 10-in. (152-, 204- or 254-mm) diameter pipe or square tubing — again, depending on a variety of unit-specific factors.

As noted, cylinder nozzles should be between one and two pipe diameters in length. In general, the cylinder nozzle wall thickness should be as thick as practical (i.e., as a minimum, Schedule XS, preferably, Schedule 120). ■

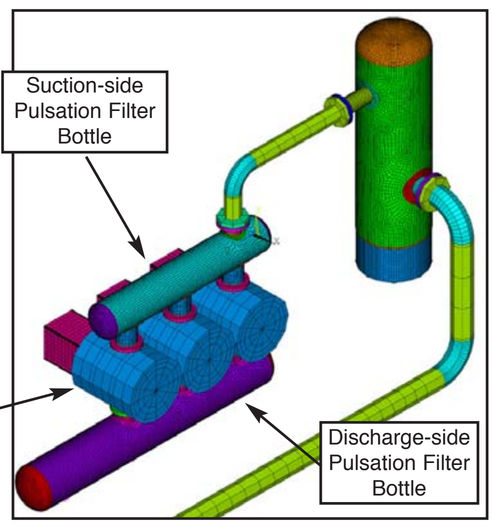
PART II

From Part I of this series, note that a compressor manifold system could be mechanically modeled with either beam-type or shell-type finite elements. Other options are available, of course, such as using brick-type finite elements. However, brick element models provide only very small accuracy improvements over shell-type models, but at a high additional cost in time required to complete the modeling process. Part 1 of this series also concluded that for today's high-horsepower, high-speed reciprocating compressor designs, the use of shell-type finite-element models is a necessity for accurate modal, dynamic stress, and vibration predictions.

In this part, mechanical modeling results of a specific compressor design using a beam element model will be compared with a shell-type finite-element model. The unit in this particular example is a high-speed (i.e., 750 to 1000 rpm), single-stage, six-cylinder, natural gas transmission unit with a vertical scrubber. The piping between the primary suction-side pulsation filter bottle and the vertical scrubber is referred to as the "external choke." Since the unit is symmetrical, only one half of the compressor is shown. Figures 1 and 2 show the basic beam and shell element models, respectively.



■ **Figure 1. Beam elements.**



■ **Figure 2. Shell elements.**

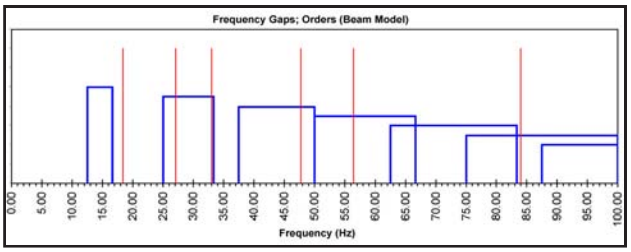
The first step in any mechanical manifold analysis is to predict the mechanical natural frequencies and mode shapes of the compressor system. Any given compressor system will have a few standard mechanical responses (such as for the suction pulsation filter bottle) and several complex responses, in which multiple elements in the system are moving. Table 1 lists some of the major mechanical responses that were predicted in this case using each model.

Mechanical Response Mode Description	Beam Model	Shell Model
Vertical Scrubber Mode	18 & 33 Hz	11 & 20 Hz
External Choke Mode	27 & 47 Hz	30, 38, 45 & 77 Hz
Suction Bottle Mode	27 & 112 Hz	30, 55 & 75 Hz
Suction (Scrubber Inlet) Piping Mode	56 Hz	38, 51 & 65 Hz
Cylinder Mode	84 Hz	45, 55, & 63 Hz

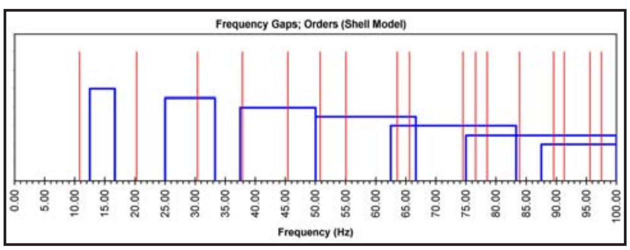
■ **Table 1. Summary of predicted mechanical response frequencies.**

Table 1 shows that some predicted mechanical response frequencies were similar for each model and some frequencies were significantly different. Certain frequencies appear more than once in each column because they involve displacement on more than one component in the system. Figures 3 and 4 show the predicted mechanical responses (shown in red) graphed with the compressor running orders (shown in blue).

The quantity of modes predicted by the shell element model in the 0-100 Hz range is significantly greater than the beam element model. This is because of the greater detail of the shell element model allowing for more flexibility and more complex displacements.



■ **Figure 3. Beam model – predicted mechanical responses.**

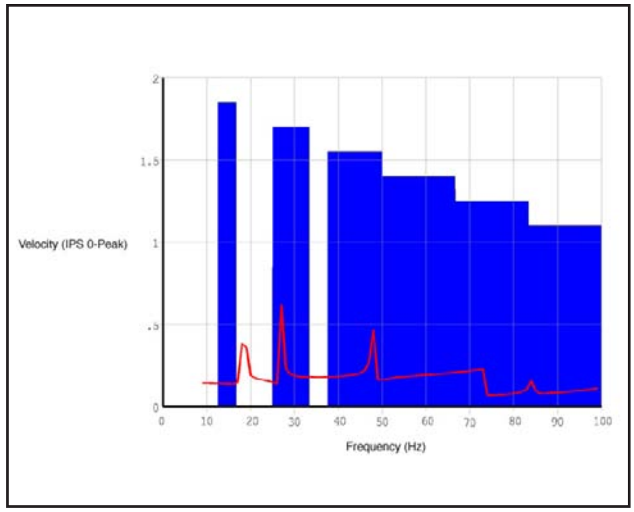


■ **Figure 4. Shell model – predicted mechanical responses.**

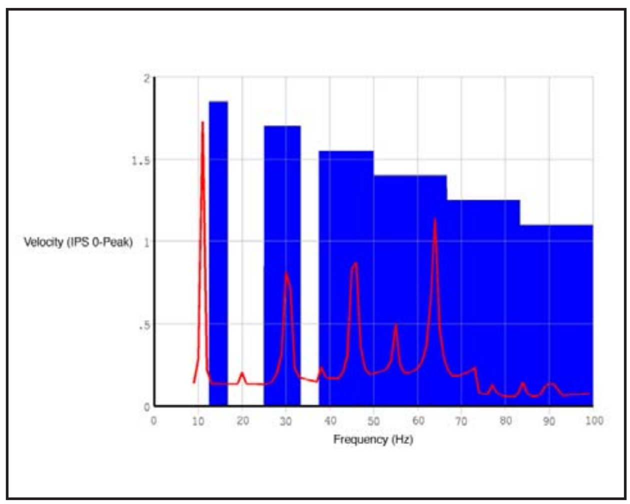
Given the wide speed range of this particular compressor unit, both models predicted mechanical responses to be within the operating speed range. If this were a fixed-speed compressor, the exact frequencies of the mechanical responses would be more critical, since they could possibly be shifted to be between compressor running order frequencies.

Since some mechanical responses show up in the operating speed range, forced response calculations are necessary to predict dynamic stress and vibration amplitudes for the compressor system components. In a typical high-speed, high-horsepower compressor system with good acoustical characteristics (i.e., effective compressor pulsation suppression), the greatest source of mechanical excitation is the gas compression load (resulting in stretch along the axis of the cylinder). This excitation is greatest at the first order of compressor running speed, but can exist in gradually reducing amplitudes to the 10th order or even higher. Figures 5 and 6 show the predicted vibration profiles for the beam and shell models, respectively. Vibration velocities are shown in engineering units of inches per second (zero-to-peak).

For the beam element model (Figure 5), the highest predicted peak velocities occurred at the elbow in the external suction-side choke tube (i.e., between the suction-side manifold filter bottle and the vertical scrubber) near 27 and 47 Hz. The peak scrubber vibration amplitude was predicted near 18 Hz. For the shell element model (Figure 6), peak choke tube elbow vibration velocities occurred near 30 and 45 Hz. Peak scrubber vibration amplitudes were predicted near 11 and 20 Hz. Peak cylinder vibrations were predicted near 63 Hz. Peak suction manifold bottle vibrations were predicted near 55 Hz.



■ **Figure 5. Beam element model predicted velocities.**



■ **Figure 6. Shell element model predicted velocities.**

Initial observations of these two graphs show that the peak velocities for the beam element model are significantly lower than those predicted for the shell element model. For a different compressor layout, the predicted vibration amplitudes of the beam element model might be higher than those of the shell element model. The main point is that the calculated vibration amplitudes are significantly different for each model type.

Using an allowable vibration criterion of 1.0 in./s (25.4 mm/s) at any given frequency, the beam element model meets this criterion, whereas the shell model is near or in excess of this criterion at several different frequencies. Use of the beam element model predictions indicates that this will be a relatively smooth running compressor unit without further system modifications. Use of the shell element model predictions would indicate a much rougher running compressor unit and further system modifications should be pursued.

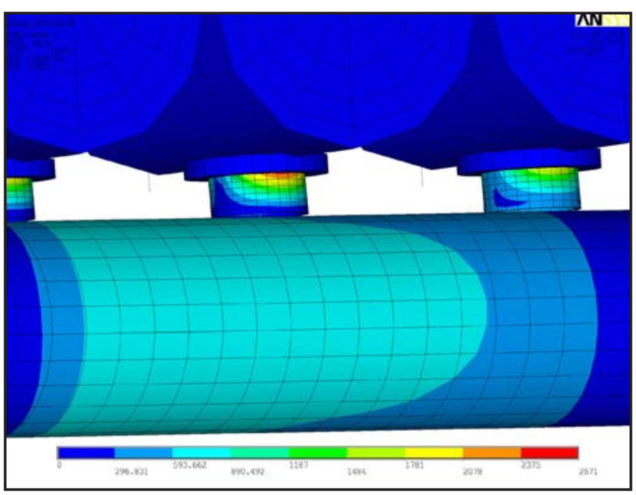
Exploring the details of such system modifications is beyond the scope of this article. However, the most critical areas of this system to address are the flexible choke tube (the elbow should be eliminated or at least moved closer to the bottle) and the flexible cylinders (with the addition of outboard cylinder supports).

Based on practical field experience, this particular type of compressor layout (with a flexible and highly elevated suction choke tube) is very often susceptible to vibration problems (although it makes for a good example). So, for this particular case, the predicted results of the shell-type model are much more agreeable to our experience base.

In general, this type of compressor manifold system modeling has always had some uncertainty (because of a variety of factors such as installation tolerances). However, our overall experience indicates better matching between field and modeling data with the more detailed shell type models than the simpler beam-type models.

A very important observation is that a general change in mechanical analysis philosophy has occurred over the years. With low-speed compressors operating at a fixed running speed, the design approach has historically been to shift the frequencies of mechanical response away from the compressor frequencies of excitation (i.e., integer multiples of compressor running speed). With today's high-speed compressors typically operating over wide speed ranges, that design approach is nearly impossible to implement. At some points in the operating envelope, the compressor will be running near or at resonance. Thus, the basic design approach has gone from resonance avoidance to resonance management. Consequently, having the proper modeling tools is very important to predict accurately system behavior over the entire range of operating conditions.

The next step in the mechanical manifold analysis is a calculation of the composite cylinder stretch excitation. The amplitude of this off-resonance excitation is based on the composite displacement at all compressor running orders. Figures 7 and 8 show the predicted stress amplitudes as a result of applying this load on the beam and shell element models, respectively.



■ **Figure 7. Beam element stress.**

The predicted stresses (both in amplitude and location) were very different between the two model types. The peak stresses predicted for the beam and shell element models were 2671 and 4053 psi (18.8 and 29 N/mm^2), respectively. The shell model predicted stresses were approximately 1.5 times higher than the beam model. This difference was attributed to geometry-related stress concentration factors that were inherently built into the shell element model.

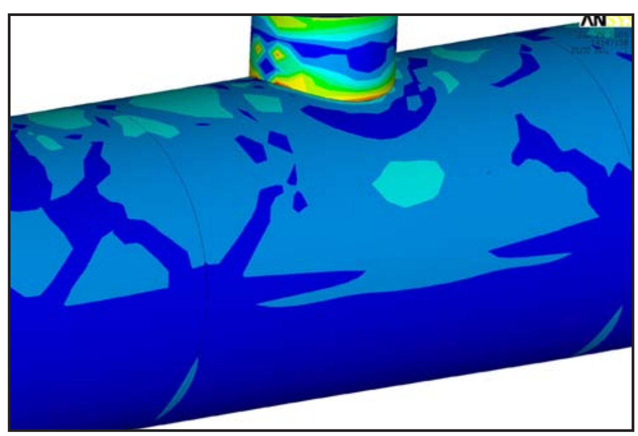
The beam model also lacked the detail to predict accurately the correct location of peak stress. For the beam model, the peak stress was predicted at the discharge cylinder nozzle flange. For the shell model, it was at the discharge cylinder nozzle to bottle connection (a typical failure point).

In general, depending on the size of the nozzle-reinforcing pad used in the model, the stress predictions of the shell model can vary significantly (crossing the border between being acceptable and being excessive). The exact geometry of the reinforcing pad cannot be included in the beam element model. As a result, the beam element model cannot show variations in predicted stress amplitudes as a result of changing the reinforcing pad size.

To help minimize bottle fabrication costs, reinforcing pads have, over the years, generally gotten smaller. They are usually sized based on pressure vessel code calculations using primarily static pressure loads. To minimize stresses caused by dynamic loading, all nozzle reinforcing pads should have a minimum outside diameter of 1.75 times the nozzle outside diameter. For nozzles with nominal diameters of 4 in. (102 mm) or less, the recommended multiplier is 2.0. The repad thickness should be approximately the same as the pulsation filter bottle wall thickness, up to 1.0 in. (25.4 mm) thick.

This article provides a partial summary of the types of loads and stresses that should be evaluated during a typical compressor manifold mechanical analysis. By comparing the two different modeling approaches, beam- and shell-types, the calculated results are significantly different, resulting in different designs. These specific results support the general conclusions that were drawn in Part I of this series.

For this complex mechanical modeling approach, the added details of a shell-type model are necessary to achieve accurate results that agree with measured field data. While shell-type finite-element models are more computationally intensive than beam-type models, current technology already has passed the point where the benefits of a more detailed model outweigh the extra modeling effort. Southwest Research Institute has developed an extensive library of parameterized finite-element compressor designs.



■ **Figure 8. Shell element stress.**

This allows us to provide a detailed finite-element analysis while also keeping costs comparable to beam-type modeling techniques. Considering the costs of addressing vibration problems after compressor start-up, the initial cost of a detailed mechanical model during the design stage is a very cost-effective alternative. Potential problem areas can be identified ahead of time and solutions can be optimized during the design stage.

To Summarize Some Key Points:

Evaluating the mechanical responses of a new compressor manifold system at the design stage is important. Just as important is using the most appropriate modeling technique for the job.

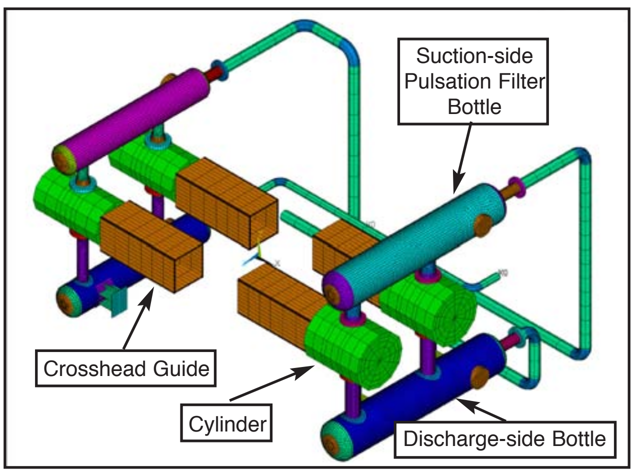
With today's high-speed compressors operating over a wide speed range, the concept of mechanical resonance avoidance must become resonance management instead.

In general, all nozzle reinforcing pads should have a minimum outside diameter of 1.75 times the nozzle outside diameter. For nozzles with nominal diameters of 4 in. (102 mm) or less, the recommended multiplier is 2.0. The repad thickness should be approximately the same as the pulsation filter bottle wall thickness. ■

PART III

For most new reciprocating compressor installations following the API Standard 618 design methodology, Design Approach 3 pulsation and vibration analysis should be used. Items M5 and M6 in API 618 specifically cover the "Mechanical Compressor Manifold System Analysis," often simply referred to as a "Mechanical Analysis."

In today's natural gas industry, one of the fastest growing areas is the use of liquefied natural gas. Currently, LNG accounts for roughly 1% of the U.S. domestic natural gas market. However, in the next 20 years, it is projected to provide as much as 10 to 20%. The compression equipment used to extract the boil-off gas (BOG) is a key component in this process. To ensure safe and reliable operation of such LNG BOG compressors, a very important exercise during the design stage is to analyze the mechanical response characteristics of the compressors.



■ **Figure 1. A typical LNG compressor.**

Part I and Part II of this series established that, in general, mechanical modeling of compressor manifold systems using API Standard 618 design methodology (i.e., items M5 and M6) is very useful during the design stage to minimize the risk of excessive vibration. However, in order to ensure an accurate analysis of the vibration characteristics of a compressor system, the use of a shell-type finite-element model (not beam elements) is required. This is particularly true with LNG compressors.

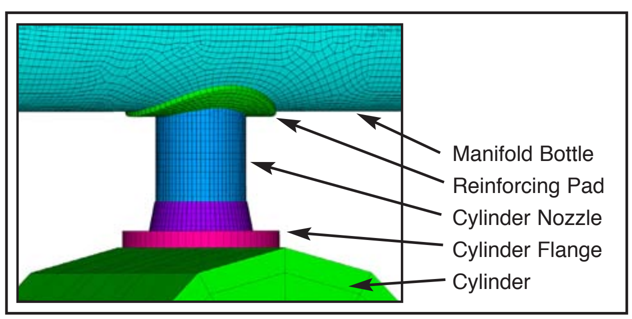
LNG compressor systems have unique characteristics and challenges associated with them. First, the first-stage, suction-side system operates at very low temperatures, 200°F (129°C) below zero. Consequently, the system needs to be designed with thermal contraction and expansion considerations in mind. In some cases, the low stage of compression requires an expansion joint in one of the cylinder nozzles. In general, the added flexibility in the system necessary to enable such thermal contraction also makes the system more susceptible to vibration.

LNG systems also operate at very low gas pressures. This typically results in the use of very thin-walled pressure vessels (i.e., as thin as 0.25 inch). Such thin wall thicknesses may be adequate to contain the low static pressure loads. However, dynamic loads are usually not considered when the initial vessel wall thicknesses are selected. The same dynamic load, which is no problem on a 0.5 in. (12.7 mm) or greater wall thickness pulsation filter bottle, is suddenly very significant.

To accommodate the necessary gas flow rates through the compressor system, the manifold bottles used to filter compressor-induced pressure pulsations are also typically very large in diameter, up to 48 in. (1219 mm). Such large-diameter, thin-walled vessels add to the system flexibility and make modeling and analysis of the nozzle-to-bottle junction(s) very critical.

Given the very high energy density of LNG and the high risks associated with any potential system structural failure, careful evaluation of the mechanical response characteristics of any new compressor unit during the design process is very important. Use of the best modeling tools and techniques is equally important to ensure accuracy in such an analysis.

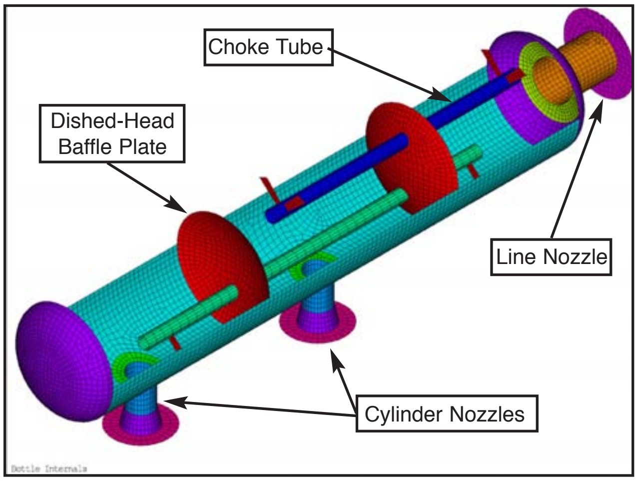
Southwest Research Institute (SwRI) currently performs all mechanical manifold analyses using shell-type finite-element models. The detail used in such a modeling approach is well suited to the application of LNG compressors.



■ **Figure 2. Cylinder nozzle close-up.**

Realistically modeling the bottle flexibility at the cylinder nozzle connection to achieve accurate mechanical natural frequency predictions is very important. This area of the model must be developed with sufficient detail as shown on Figure 2.

This same area, where the nozzle joins the pulsation filter bottle, is also critical for stress calculations. In a typical compressor system, the highest dynamic stresses occur in the cylinder nozzle and in the bottle wall at the cylinder nozzle junction.



■ **Figure 3. Inside a typical manifold bottle.**

Figure 3 shows the internal elements (baffles and choke tubes) inside a typical compressor manifold pulsation filter bottle. Including these elements in the modeling process is important because they can add both mass and bottle shell wall stiffness.

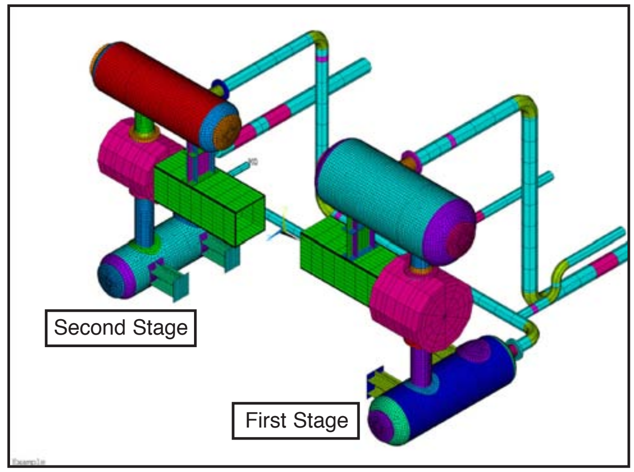
LNG compressor systems are often very flexible to enable thermal contraction at very low operating temperatures. In general, the more flexible a mechanical system is, the higher the vibration amplitudes will be for a given excitation force. Consequently, with any flexible LNG system, accurately predicting the natural frequency is very important. This is necessary so that mechanical responses in the system can be shifted, if necessary, away from potential sources of excitation. Also very important is that the calculated vibration amplitudes be as accurate as possible.

Another possible concern with LNG systems is the excitation of bottle shell modes. This is a possibility any time there are large diameter vessels with relatively thin walls. A detailed finite-element shell model will be able to predict such shell deflection modes. A simpler beam element model would not predict such a mechanical response.

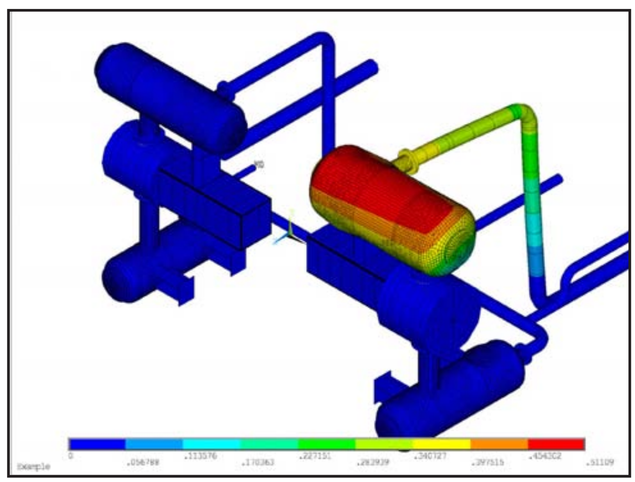
Most bottles used for compressor pulsation control will have more than one internal chamber. Pressure vessel design code requirements often specify one inspection port per chamber. Most inspection ports are fabricated like a typical nozzle, but are sealed off with a blind flange. This type of inspection port can add a significant amount of concentrated mass at several locations along the bottle. Such inspection ports can be vulnerable to localized vibration problems. Such vibration problems can oftentimes result in structural fatigue cracks at or near the inspection ports. This can happen on any type of compressor pulsation filter bottle, but large-diameter, thin-walled bottles (common on LNG compressor units) are more susceptible. In general, it is recommended that such inspection ports be minimized in size (diameter) and quantity. In some instances, existing process gas line nozzles can also be used as inspection ports, thus, minimizing the need for additional inspection ports.

Since most LNG systems start at very low pressures (near atmospheric), there are usually two or three stages of compression. With each stage having a different design, the likelihood of one of the stages having a vibration problem increases.

Figure 4 shows a typical LNG reciprocating compressor system. It is a relatively simple, two-cylinder, two-stage BOG compressor. One of the lowest frequency mechanical responses in this particular system was associated with the first-stage, suction-side pulsation filter bottle. That particular mechanical response was at almost the same frequency as the second multiple of the compressor running speed, where it is very likely to be excited because of cylinder stretch and acoustical (pressure pulsation) forces. Figure 5 shows the predicted mode shape plot for this particular mechanical response.



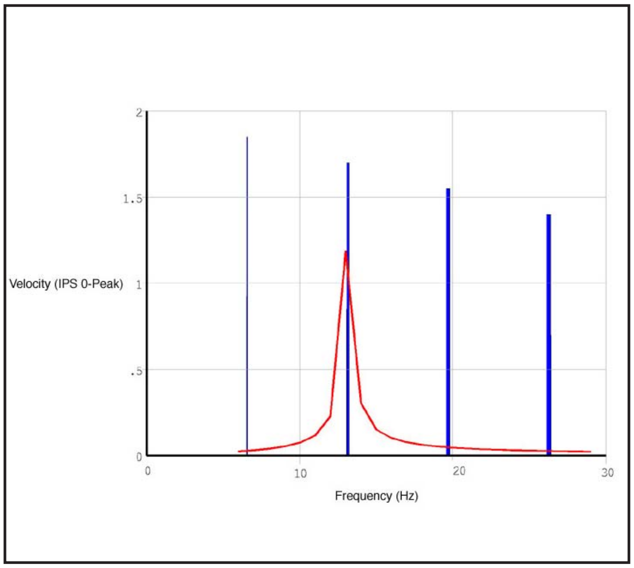
■ **Figure 4. A two-cylinder, two-stage LNG BOG compressor.**



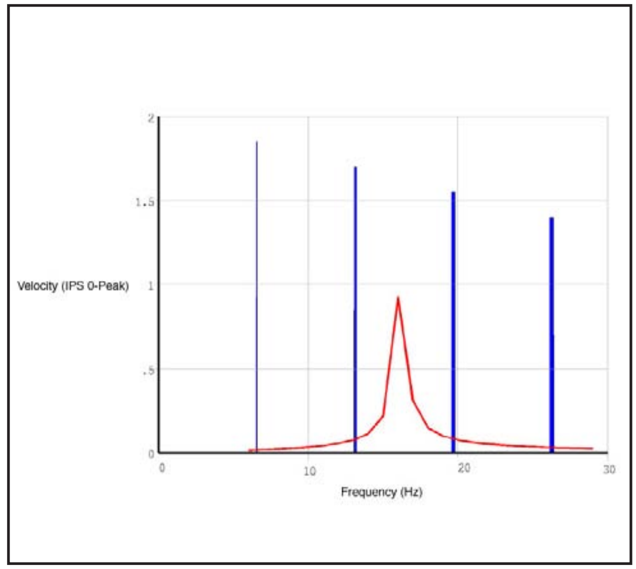
■ **Figure 5. Mode shape plot – first stage suction-side pulsation filter bottle cantilever.**

That particular mechanical response of concern was predicted to be near 13 Hz. That was an unusually low frequency for a primary mechanical response of a suction-side pulsation filter bottle. One of the most significant reasons why that particular mechanical response was so low in frequency was because that particular bottle had such a thin, 0.25 in. (6.35 mm) wall for such a large 48 in. (1219 mm) diameter. Excessive shell deflections were adding a lot of flexibility to the system. As noted previously, this is relatively common in low-pressure LNG systems.

Figure 6 shows a vibration response plot of the first-stage, suction-side pulsation filter bottle. The excitation is



■ **Figure 6. Velocity vs. frequency (original).**



■ **Figure 7. Velocity vs. frequency (modified).**

from unbalanced acoustical (pressure pulsation) forces. The vertical lines (shown in blue) represent the multiples of the compressor operating speed. The peak in vibration at the second compressor running order (near 13 Hz) is very evident and it exceeds our standard allowable vibration criterion. Our initial recommendation was to increase the wall thickness of the bottle. This would have stiffened the entire system around the bottle and would have greatly minimized bottle shell deflections. However, in this particular case, a thicker bottle wall was not practical for various other, non-technical reasons.

As an alternative modification, the support between the filter bottle and the compressor crosshead guide was made larger and stiffer. This raised the primary mechanical response frequency of the first-stage, suction-side filter bottle to approximately 16 Hz. With the mechanical response then placed between the second and third compressor orders, it was much less likely to be excited.

This very brief case study only illustrates one problem and solution associated with a recent mechanical design analysis on one particular LNG compressor unit. However, it does provide an example of the types of problems that can be encountered.

In general, SwRI recommends that a detailed mechanical manifold analysis be performed on all new compressor installations over 500 hp (373 kW). This is particularly true for critical applications, such as LNG units. The potential risks of excessive vibration and overstress failures can be greatly reduced. The cost of analysis during the design stage is relatively small when compared to the cost of trial-and-error repairs in the field, after start-up.

To Summarize Some Key Points:

Mechanical modeling (using the correct tools) of compressor manifold systems at the design stage is important. This is particularly true for multi-stage LNG units, where the likelihood of problems is greater and the consequences of any failures are serious.

All low-pressure compressor manifold pulsation filter bottles should have a wall thickness of at least 0.375 in. (9.53 mm)—higher pressure bottles will be thicker.

Manifold filter bottle overhangs should be properly supported.

The use of pulsation filter bottle inspection ports with blind flanges should be minimized (in diameter and quantity).

As previously noted (in the first two installments of this three-part series), cylinder nozzles should be between 1.0 and 2.0 pipe diameters in length (measured from the outside of the bottle shell to the face of the cylinder flange). In general, the cylinder nozzle wall thickness should be as thick as practical (as a minimum, Schedule XS, preferably Schedule 120). In general, all nozzle reinforcing pads should have a minimum outside diameter of 1.75 times the nozzle outside diameter. For nozzles with nominal diameters of 4 in. (102 mm) or less, the recommended multiplier is 2.0. The repad thickness should be approximately the same as the bottle wall thickness. ■

Southwest Research Institute is an independent, nonprofit, applied engineering and physical sciences research and development organization using multidisciplinary approaches to problem solving. The Institute occupies more than 1,200 acres and provides nearly two million square feet of laboratories, test facilities, workshops, and offices for more than 3,000 employees who perform contract work for industry and government.

We welcome your inquiries.
For additional information, please contact:

Benjamin A. White, P.E., Senior Research Engineer
Machinery Piping Design
Phone (210) 522-2554
Fax (210) 522-4506
E-mail benjamin.white@swri.org

Edgar B. Bowles, Jr., Assistant Director, R&D
Pipeline Technology
Phone (210) 522-2086
Fax (210) 522-4506
E-mail ed.bowles@swri.org

Mechanical and Materials Engineering Division
Southwest Research Institute
6220 Culebra Road • P.O. Drawer 28510
San Antonio, Texas 78228-0510
www.swri.org

18-1160 (JCN: 225,212) jjf