A PULSATION AND VIBRATION CONTROL METHODOLOGY FOR HIGH-SPEED NATURAL GAS RECIPROCATING COMPRESSORS

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This article discusses the current pulsation and vibration control methodology generally preferred by Southwest Research Institute® (SwRI®) for the design of high-speed natural gas reciprocating compressors. Included in this discussion is the application of a new technology for control of compressor nozzle resonance.

Pulsation Control

Generally, unacceptable compressor or piping vibration levels occur when an acoustic resonance coincides with a mechanical natural frequency (mechanical resonance). The purpose of an acoustic analysis is to minimize the adverse effects of pressure pulsations produced by a compressor or compressors, while working within the operational and site constraints provided.

The most effective method of pulsation control for natural gas compression is the installation of a volume-choke-volume acoustic filter at the compressor connection. A properly designed acoustic filter will create sufficient surge volume to reduce dynamic flow components, control the shape and distribution of the impedance function, such that dynamic pressures are adequately controlled on and off resonance, and limit residual pulsations in the lateral piping to acceptable levels.

Figure 1 shows a simplified spectral plot for an acoustic filter design. The blue lines represent the individual compressor orders generated at a constant speed of 900 rpm, while the shaded areas represent the compressor orders associated with variable speed operation (i.e., 750 to 1,000 rpm). The red plot identifies characteristic frequencies of the acoustic filter design. The lowest response is the volume-choke-volume Helmholtz response frequency (fH) defined by Equation 1. Where,

f_H = Helmholtz frequency, Hz
A = Cross-sectional area of choke, ft2
L = Acoustic length of choke, ft
c = Velocity of sound, ft/sec
V_1 = Volume of cylinder chamber, ft3
V_2 = Volume of filter chamber, ft3

$$f_H = \frac{c}{2\pi}\left(\frac{\mu}{V_1} + \frac{\mu}{V_2}\right)^{\frac{1}{2}} \qquad \mu = \frac{A}{L}$$

Generally speaking, an acoustic filter minimizes pulsation energy above the "cut-off" frequency defined by the filter elements. Residual frequencies associated with the filter elements and the compressor nozzles are not controlled by the filter; rather, they are generally controlled using pressure drop damping. It is important that pressure drop be limited when possible to reduce the adverse effects on compressor station performance and efficiency.

Note that the acoustic filter design generates regions of high amplification that are associated with the filter elements, as well as regions of high attenuation between the filter response frequencies. It is important to place piping response frequencies in the regions of high attenuation to minimize pulsations. Additionally, piping response frequencies should be located such that they do not coincide with compressor orders. Although avoiding compressor orders is possible with the compressor operating at constant speed, it is not usually possible with the compressor operating over a wide speed range.

SwRI's standard practice for a high-speed compressor installation is to develop single bottle, multi-chamber acoustic filter systems, using internal baffles and choke tubes, for installation at the suction and discharge compressor connections. The size of each pulsation bottle is selected to achieve the desired acoustic filter frequency (generally below one times compressor running speed or 1x) with reasonable pressure losses. Compared to a two-bottle acoustic filter design, the single bottle, multi-chamber acoustic filter system generally uses smaller diameter bottles with less overall pressure drop to achieve a lower filter frequency with acceptable pulsation amplitudes. In addition, mechanical analysis indicates that undesirable vibration in the compressor manifold area is less likely with the single bottle design. Figure 2 shows, as an example, the internal design of a single bottle, multi-chamber acoustic filter designed for two cylinders. Each cylinder chamber is connected to the line (filter) chamber using an internal choke tube.

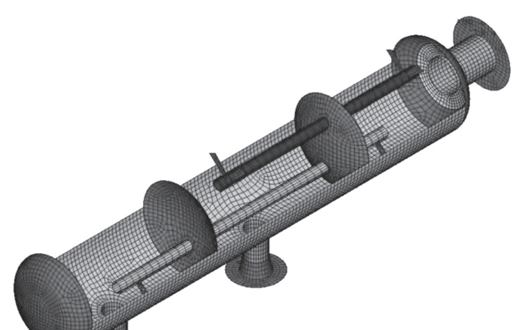

Figure 2. Single Bottle, Multi-chamber Acoustic Filter System

In general, adequate pulsation control for the attached piping can be achieved with a single bottle, multi-chamber acoustic filter system. However, reducing pulsations in the compressor nozzle region, such that compressor performance is not significantly affected, is often much more challenging.

For constant speed units, it is often possible to locate the acoustic resonance associated with the internal gas passage-nozzle configuration away from the operating speed by adjusting the nozzle length. In some cases, this can also be done when operating over a narrow speed range. If the speed range is wide, resonance typically cannot be avoided and further modifications are most often required to reduce pulsa-

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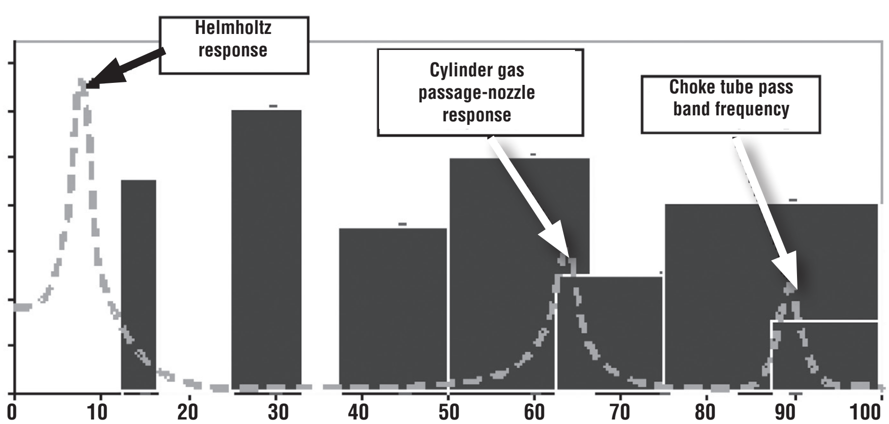

Figure 1. Filter Frequency Spectrum

tion amplitudes at the resonance frequency. It can also be difficult to avoid resonance when operating over a wide range of pressures and temperatures.

A conventional means of reducing pulsations in the cylinder nozzle region is to install orifice plates at the cylinder flanges or inside the nozzle near the pulsation bottle. However, it is necessary to carefully consider the horsepower cost associated with locating pressure drop elements in regions of high dynamic flow. Newer technology developed through the Gas Machinery Research Council's research and development program is also available for pulsation reduction in the cylinder nozzle region. As shown in Figure 3, the "virtual orifice" is basically a small volume that attaches to a valve cap. A small pipe connects the volume to the gas passage. The virtual orifice is designed such that its acoustic response frequency matches the target frequency to be damped.

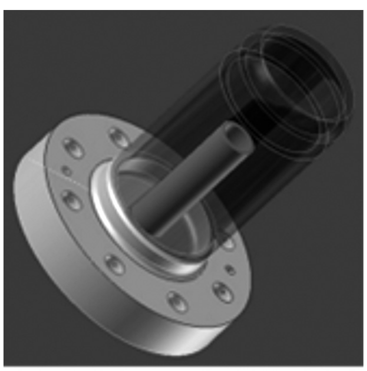

Figure 3. Virtual Orifice Design

Figure 4 describes initial testing of the virtual orifice. Notice that pulsations at the compressor nozzle-gas passage resonance frequency are significantly reduced with the virtual orifice installed. Compared with conventional orifice plates, compressor efficiency was approximately 10% higher at maximum speed with the virtual orifice installed.

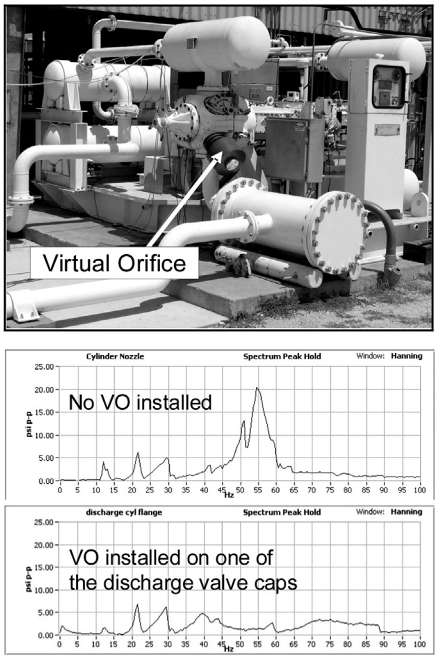

Figure 4. Virtual Orifice Testing

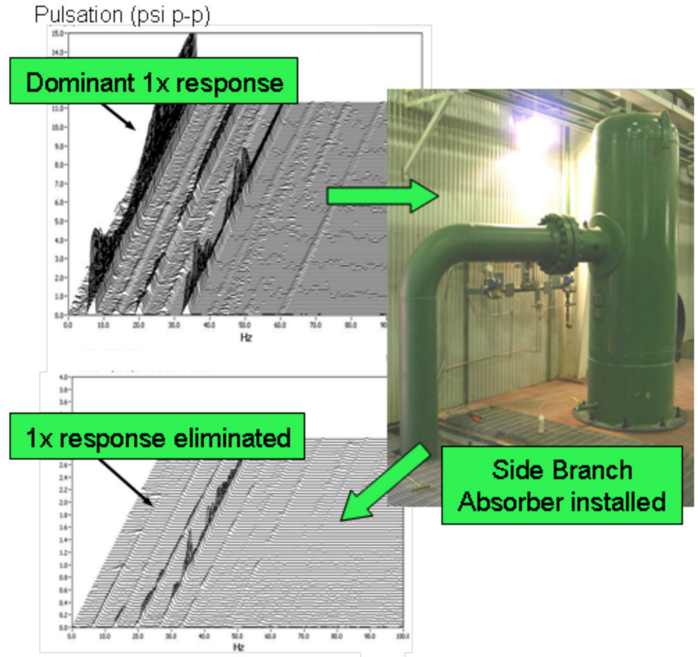

Figure 5. Side Branch Absorber (SBA)

For systems with large operating envelopes, designing an acoustic filter system that will control pulsations over the entire operating range with acceptable losses can be difficult and relatively high first order pulsations can sometimes be problematic. In these cases, further modifications may be required for adequate pulsation control. For improved filtering at the first order of compressor running speed, larger pulsation bottles and/or increased pressure losses are necessary. Larger bottles are often not desirable due to mechanical limitations and/or installation constraints. Increasing the bottle size lowers the mechanical natural frequencies and increases the bottle cost. Smaller choke tubes would increase the system pressure drop, which reduces compressor performance and increases operating costs. In some cases, orifices can be used to effectively attenuate first order pulsation energy; however, orifices that are adequately sized to reduce first order pulsations during single-acting operation can adversely affect compressor performance under the higher flow double-acting operating conditions. In many cases, side branch absorbers (SBAs) can be used to efficiently and effectively reduce first order pulsation energy. Figure 5 shows a typical SBA design that was installed to reduce pulsations at the first compressor order frequency. Notice that the first order pulsations are virtually eliminated with the SBA installed.

A side branch absorber is simply a volume attached to the main line by a connecting choke tube. This type of device is highly effective for reducing pulsation at a unique frequency and has been proven effective over a relatively wide speed range. Because the effectiveness of the SBA is highly dependent on its location along the mode shape of the pressure wave, it is imperative that the piping lengths not change without re-evaluating the design. A tunable SBA (shown in Figure 6) can be used to achieve maximum attenuation over a wide speed range. The tunable SBA technology was also funded through the Gas Machinery Research Council's research and development program. However, in many cases, a fixed volume SBA will achieve sufficient pulsation control.

Vibration Control

Acoustic filter systems generated with combinations of volumes and choke tubes generate mechanical natural frequencies, which must be mismatched with residual acoustic excitation frequencies. Mechanical modeling, in concert with a pulsation and performance analysis, attempts to balance compressor performance with acceptable operating stress and vibration levels, which leads to safe and reliable unit operation. Complex systems have many resonant frequencies for which corresponding mode shapes (deflection patterns) exist. A mechanical analysis identifies and positions key system resonance frequencies at optimal locations relative to integer orders of running speed. Stress and vibration levels within the manifold system are predicted based on residual pulsation levels in the system and cylinder vibrations levels. Mani-

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fold design parameters, including piping wall thickness, reinforcing pad thickness and size, bottle wall thickness, and support locations can all be tailored to reduce stress and vibration to acceptable levels.

Mechanical response analyses are performed to ensure that adequate mechanical restraint exists to control vibration. A detailed finite element model of the compressor manifold system (i.e., pulsation filter bottles, cylinders, and the directly attached piping) is required to predict the mechanical natural frequencies and mode shapes of the compressor manifold system with reasonable accuracy. Key factors for accurate modeling of the compressor manifold include restraint stiffness values, flexibilities in compressor crosshead guides and frames, flexibilities in bottle wall to nozzle junctions, flexibilities in cylinder flanges, and mass and stiffness properties of bottle internals. An ANSYS® plot showing the scope of a typical compressor manifold model for a four-cylinder compressor is shown in Figure 7.

Adequate restraint of each suction bottle is generally achieved by attaching the long cantilevered end to a frame-type support using a clamp/wedge combination. Closely coupled scrubbers can also be used to support the suction bottles. In some cases, additional restraint is necessary in the direction of cylinder stretch. This additional restraint can be achieved by bracing the suction bottles together across the frame as shown in Figure 7. To accommodate this, suction bottles should have non-penetrating reinforcing pads welded on the frame side such that cross-bracing can be added as needed for vibration control. In multi-stage systems, this would require that the suction bottles have the same centerline elevation. To support the discharge bottles, clamp and wedge-type restraints are typically located near the outlet end of each discharge bottle. In addition, wedge-type restraints should be installed under each cylinder connection and a strap-type clamp should be installed between the cylinder nozzles. To minimize loads due to thermal expansion, the discharge bottle support wedges should not be fully tightened until after the unit reaches operating temperature. The clamps used to restrain the discharge bottles also help to control cylinder vibration.

The predicted mechanical natural frequencies are compared with the frequencies of pulsation and mechanical energy associated with the compressor units. With the single bottle, multi-chamber acoustic filter systems installed, most of the acoustical and mechanical excitation energy in the system occurs at the first and second compressor orders. It is very important to avoid placing any mechanical responses directly coincident with the first compressor order. Ideally, all mechanical responses should also be above the second order; however, this is not always practical when operating at high speeds. Where excitation frequencies coincide with compressor operating frequencies, forced response calculations are necessary to estimate peak stress and vibration levels and to ensure acceptability.

The piping restraint system should also provide sufficient stiffness and mechanical damping to control any residual (non-resonant) vibration. Strap type hold-down clamps (with or without wedges) are best for vibration control. Figure 8 shows a typical strap type clamp. The clamp should be several times stiffer than the pipe and must be attached to a structure that is adequately stiff. A dense fiber-embedded elastomeric pad clamp liner is recommended to ensure full contact with the pipe and to provide damping. A ½-inch gap should be maintained between the bottom of the clamp and the top of the support structure.

Clamps should be located near elbows and tees, where the piping is more flexible and is more susceptible to coupling with excitation frequencies. Concentrated masses (such as valves) should also be well-restrained. Supports should be installed throughout the piping network such that the lowest mechanical natural frequency is above the frequencies of significant pulsation energy, without coinciding with multiples of the normal operating speed. With properly designed acoustic filter systems installed, it is generally sufficient to locate the mechanical natural frequencies above the first compressor order.

This article summarizes SwRI's standard design techniques for the mitigation of pulsation and vibration inherent in high-speed natural gas reciprocating compressor installations. The Gas Machinery Research Council (GMRC) may be contacted for scheduling of detailed short courses on this subject.

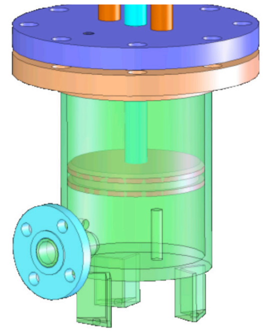

Figure 6. Tunable Side Branch Absorber (TSBA)

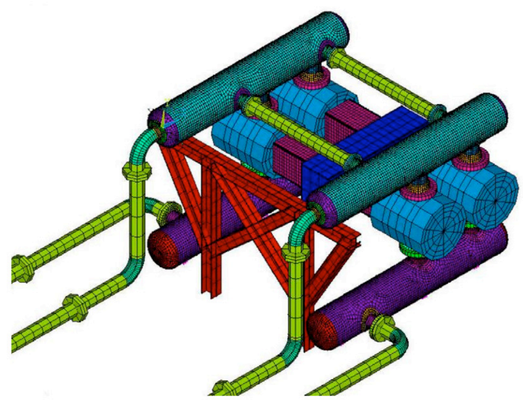

Figure 7. Compressor Manifold Model

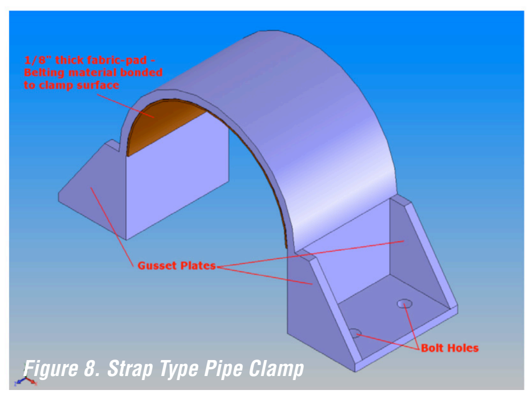

Figure 8. Strap Type Pipe Clamp