GMRC ADVANCED RECIPROCATING COMPRESSOR TECHNOLOGY (ARCT)

Optimizing a Reciprocating-Compressor, Acoustic-Manifold Design Requires a Balance Among Pulsation Control, Compressor Performance and Mechanical Vibration Control

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As part of the ongoing Gas Machinery Research Council (GMRC) Advanced Reciprocating Compressor Technology (ARCT) research project, a joint project with U.S. Department of Energy (DOE) at Southwest Research Institute (SwRI), several new pulsation control technologies have been developed and evaluated to mitigate design constraints for manifold systems and the attached piping systems. These include alternatives to cylinder nozzle orifices and tunable side branch absorbers. This presentation highlights recent results from the ongoing research and development effort.

Optimizing a reciprocating-compressor, acoustic-manifold design requires balancing pulsation control with compressor performance and mechanical vibration control. Although large pulsation filter bottles can minimize residual manifold pulsations and/or internal manifold pulsations, they are generally more expensive to build than smaller bottles and present more of a design challenge for controlling mechanical natural frequencies and operating stress levels. Pressure drop elements (e.g., orifices) used in combination with surge volumes and choke tubes can help control pulsation amplitudes. However, they also reduce compressor capacity and increase horsepower requirements.

For the large number of low-speed compressors installed on gas transmission pipelines, operational changes requiring end deactivation can introduce large-amplitude

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residual pulsation levels. If the original pulsation filter bottle was not initially designed for end deactivation, the residual pulsations could adversely affect adjacent compressor units, flow measurement and compressor performance. For modern, high-speed (and medium-speed) compressors, nozzle resonance and manifold pressure losses reduce unit performance and efficiency and, if not accounted for properly, can result in undersizing of the driver at design time.

Introduction and Background

This research project was initiated in 2004 and is currently funded by GMRC and the industry. SwRI has been tasked with evaluating current acoustic design techniques and addressing areas requiring improvement. DOE co-funded the project in 2005.

SwRI has developed and is currently evaluating new concepts in acoustic design technology that could improve unit operating flexibility, reduce manifold pressure drop, and allow improved control of vibration and operating stress levels.

Four prototype devices have been developed through efforts to improve pulsation control technology, including the tunable side branch absorber (TSBA), the infinite length nozzle (ILN), the tapered cylinder nozzle (TCN) and the virtual orifice (VO). The TSBA addresses the reduction of low frequency piping pulsations and has broader potential. The ILN, TCN and VO address the reduction of cylinder nozzle resonance pulsations. Each of these devices is designed to minimize pulsation without incurring additional horsepower cost. Here, we present the initial results of the project pulsation control development work.

Controlling Piping Resonance — Reduced Vibration

Fixed geometry side branch absorbers (SBA), or Helmholtz resonators, have been industry proven to reduce low frequency pulsations for fixed-speed machines or machines that operate over a small speed range. The physical characteristics of an SBA include the choke tube, which is strategically connected to the main gas piping, and the attached volume. Figure 1 illustrates the physical characteristics of an SBA and provides data showing its effectiveness at controlling 1x pulsations.

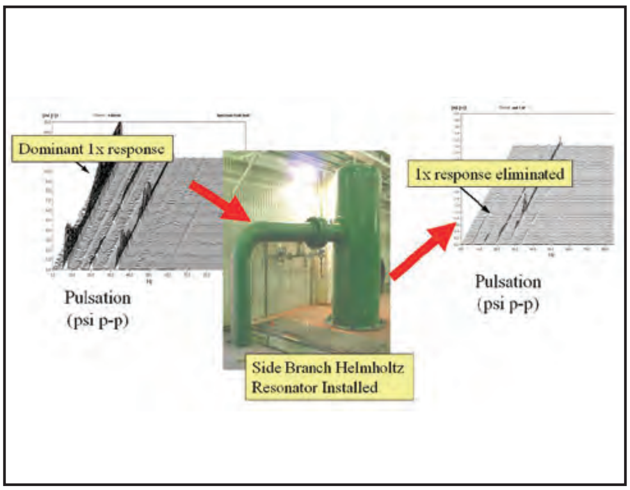


Figure 1. Effectiveness of a side branch absorber (SBA).

As shown, a choke tube connects the volume to the main gas piping. The resonant (Helmholtz) frequency can be estimated from Equation 1. Once initially sized and located, acoustic simulation tools can then be used to verify and/or tune the SBA for optimum attenuation.

Equation 1. $$f_H = \frac{C}{2\pi}\sqrt{\frac{A}{L_e V}}$$

f_H = Helmholtz frequency, Hz
C = Sound velocity, ft./sec.
A = Cross-sectional area of choke tube, ft.^{2}
L_e = Effective length of choke tube, ft.
V = Absorber volume, cf

Although effective pulsation control has been achieved with a fixed geometry SBA, this design has limitations. For optimum reduction at low frequencies, the absorber volume can be relatively large and sometimes difficult to install in existing systems because of space constraints. Also, the fixed geometry SBA is tuned for a single frequency and can achieve satisfactory pulsation reduction only over a narrow bandwidth. For example, fixed geometry SBAs have been implemented successfully to detune first and second order acoustic piping resonances over a 60 rpm speed sweep (270 to 330 rpm).

The effectiveness of the fixed geometry SBA inspired development of the TSBA. Similar to the SBA, the TSBA is composed of a choke tube that strategically connects a volume to the main gas piping. A conceptual image of the prototype is shown in Figure 2. It is equipped with a piston, which allows variability of the volume, thus making the TSBA tunable. Equation 1 shows that reducing the volume of the TSBA increases the associated resonant frequency, while increasing the volume decreases the resonant frequency.

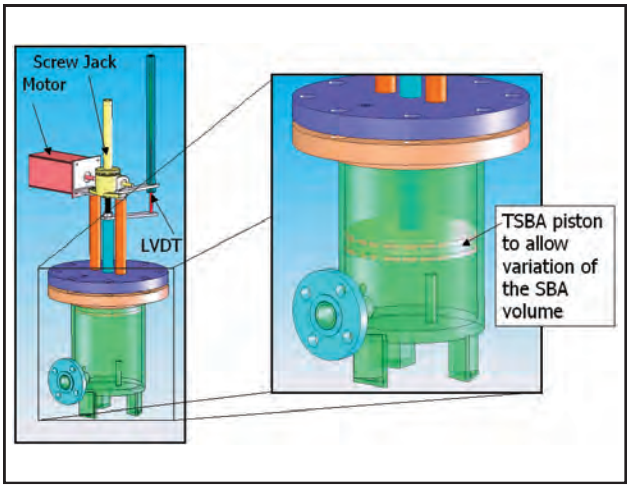


Figure 2. Conceptual 3-D drawing of the tunable side branch absorber (TSBA).

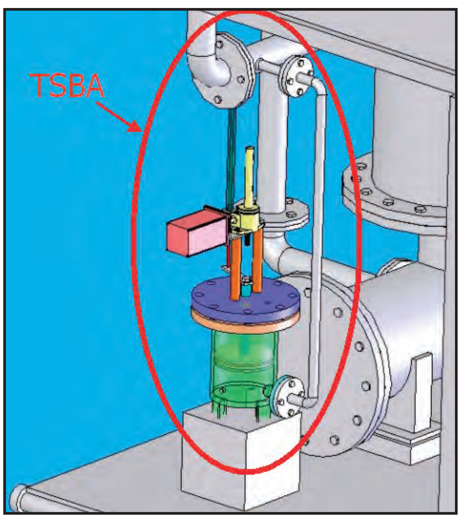
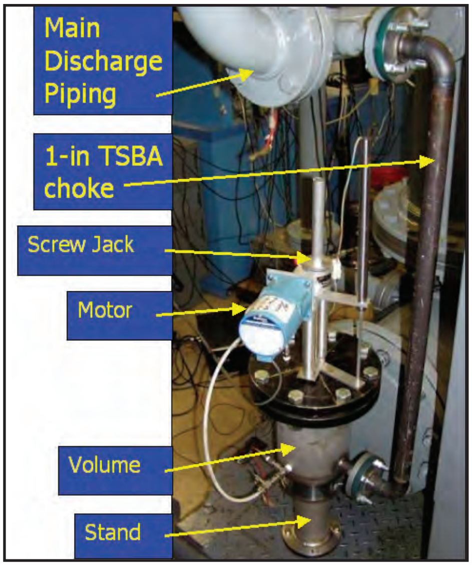


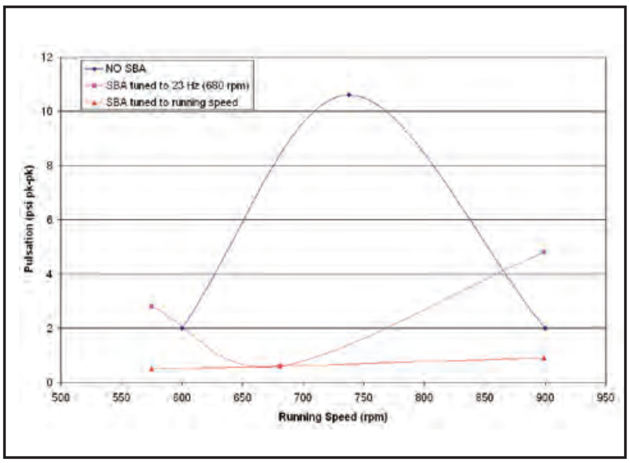
Figure 3. Conceptual 3-D drawing of the TSBA installed in the RCTF.

Conceptual drawings of the prototype TSBA installation at the SwRI reciprocating compressor test facility (RCTF) are presented in Figure 3. A span of 1 in. (2.5 cm) diameter piping connects the variable volume to the 3 in. (7.6 cm) diameter main gas piping. The RCTF compressor piping system had a lateral resonance at two times the compressor running speed (2x), which was the target issue of the prototype TSBA.

The prototype TSBA was installed in the RCTF, as shown in Figure 4. As previously described, the choke tube strategically attaches the variable volume to the main gas piping. Figure 5 summarizes measured pulsations over a compressor speed range of 600 to 900 rpm with and without the TSBA. The line marked with diamonds shows the piping resonance without the TSBA installed. The line marked with squares shows the characteristic pulsation spectrum (a valley with two peaks) of a TSBA tuned to a single frequency. The line marked with triangles shows the measured pressure pulsations with the TSBA manually tuned to track compressor running speed. The ability to tune the SBA enables successful operation over a wider speed range. The valley between the two peaks created by installing the TSBA is varied as the operating speed varies.

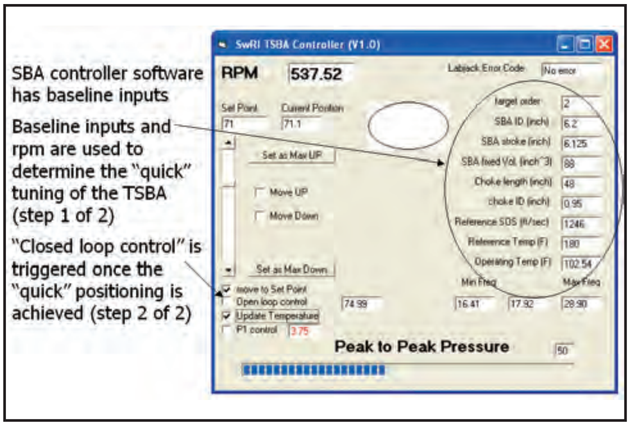


■ **Figure 4. Prototype TSBA installed in the RCTF piping system.**

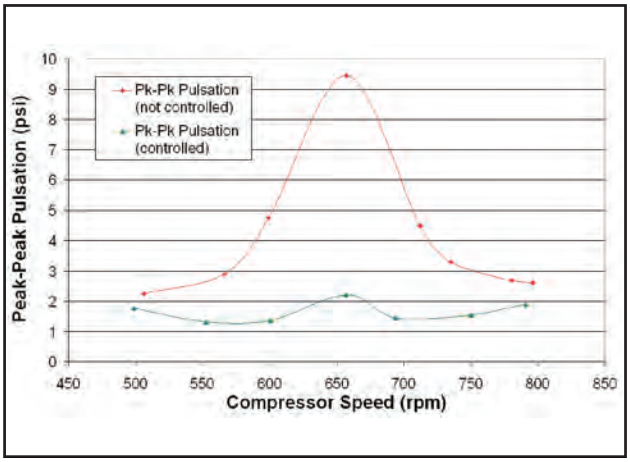


■ **Figure 5. Measured pulsation data with manual control of TSBA frequency.**

Automated control of the TSBA was then implemented. Temperature and compressor speed real-time inputs were



■ **Figure 6. Automation control software for tuning the TSBA.**



■ **Figure 7. Measured data for uncontrolled TSBA vs. TSBA with automated control.**

used to determine the initial quick tuning of the TSBA. Temperature enables a more accurate approximation of the gas sound speed — compressor speed is needed to determine the approximate frequency of 2x. The controller then fine-tunes the TSBA based on the real-time pulsations feedback. A screen shot of the software written to control the TSBA tuning is presented in Figure 6. Data recorded with the TSBA uncontrolled showed a distinct 2x lateral resonance. However, the 2x resonance peak was not noticeable with the TSBA automatically tuned by the controller system. A direct comparison of the data (Figure 7) shows that much lower pulsations are present when the TSBA is automatically controlled than when it is not controlled (and fixed at minimum volume).

Pulsations were about four to 21 times lower with the TSBA installed as an integral part of the acoustically resonant piping system. The TSBA can be used successfully over wide speed ranges and in systems where operating conditions may vary significantly. The relatively simple and rugged design should make it easy to install in an existing piping system. The prototype TSBA was able to significantly reduce pulsations caused by coincidence of compressor-generated pulsations with the piping acoustic resonance.

Although the data presented illustrates the effectiveness of the TSBA over a wide speed range, it is important to note that the TSBA can also be implemented over a variable range of molecular weights. Probably the single most important feature of the TSBA is that it can replace pressure drop damping (orifice installation) for reduction of low-frequency pulsations. Orifices, which are also used to effectively reduce pulsations, add unwanted horsepower cost. Experimental testing has shown that residual 1x and 2x pulsations can be mitigated using a TSBA without added horsepower cost.

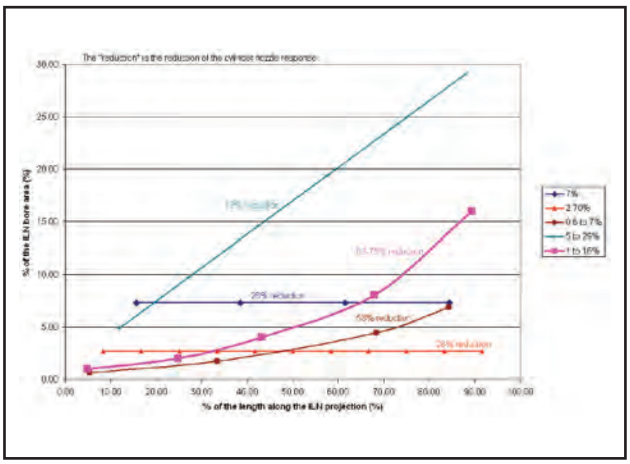
Controlling Cylinder Nozzle Resonance — Increased Efficiency

Because they occur at a relatively low harmonic, pulsations at the cylinder nozzle resonance frequency on a variable high-speed compressor are generally high amplitude. Acoustically, it would be advantageous to shorten the compressor nozzle length and raise the resonance frequency. However, shortening the nozzle increases the mechanical coupling in the nozzle region, resulting in higher stresses and the transfer of more vibration to the attached piping system. Other means of reducing nozzle pulsations are needed.

Currently, variable high-speed machines most often require orifices to dampen nozzle pulsations, thereby reducing the risk of increased vibration and reducing the impact of pulsations on the cylinder performance, but resulting in added horsepower cost. Although maximum attenuation is typically achieved with the orifice located close to the volume, in most cases, the orifice is located at the cylinder flange connection so that it can be easily removed or replaced if necessary. Experimental testing has shown that the Infinite Length Nozzle (ILN), the Tapered Cylinder Nozzle (TCN), and the Virtual Orifice (VO) each provide some degree of pulsation reduction at the cylinder nozzle acoustic resonance frequency with less horsepower cost than an orifice. In some cases, the new prototypes resulted in less horsepower cost than that of the conventional nozzle originally installed.

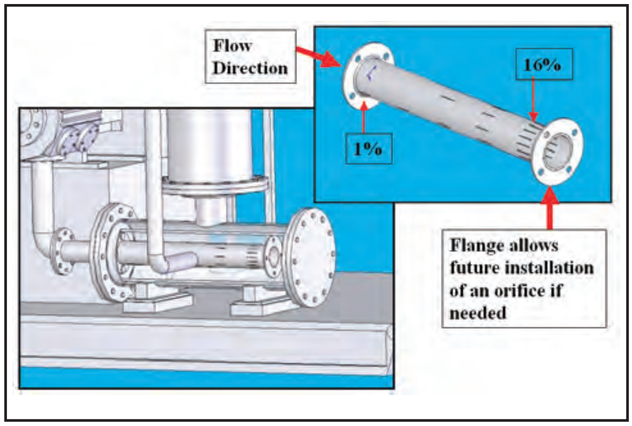
Infinite Length Nozzle

The theory behind the Infinite Length Nozzle (ILN) concept is that multiple reflections (at each perforation) reduce the amplitude of the cylinder nozzle resonance. Simulation data for various perforation distributions are summarized in Figure 8. The squares plot presents the perforation distribution used for the first ILN tested in the RCTF. Notice that predicted pulsations are reduced by 62 to 73% on resonance. The pulsation reduction is stated as a range because of the high and low ratio operating conditions that were modeled.

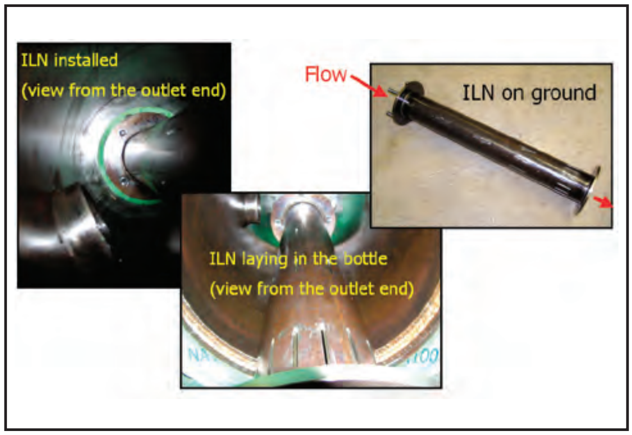


■ **Figure 8. Simulation results based on various ILN configurations.**

The first ILN prototype design installed in the RCTF had a perforation distribution along the projection that ranged from 1 to 16% of the bore area. A conceptual model of the ILN is shown in Figure 9. The length of the projection was based on ILN simulations using a similarly sized cylinder and bottle. Photographs of the installed ILN are shown in Figure 10.

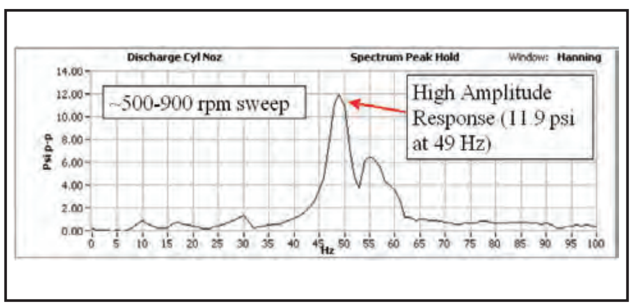


■ **Figure 9. Conceptual installation of the ILN in the RCTF bottle system.**



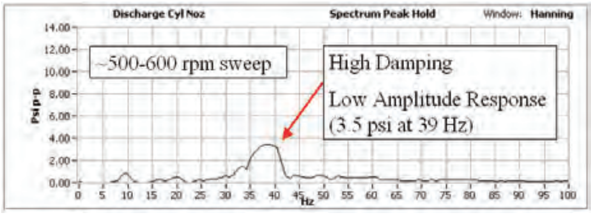
■ **Figure 10. Different views of the ILN tested in the RCTF.**

Experimental data were acquired with and without the ILN installed. Without the ILN installed, measured data for the baseline low ratio case (Figure 11) showed a resonance of 11.9 psi (82 kPa) at 49 Hz. For the low ratio operating conditions with the ILN installed (Figure 12), measured data showed a resonance of 3.5 psi (24 kPa) at 39 Hz. The shape of the fourth order acoustic nozzle resonance (wide response) suggests high damping. Measured data with and without the ILN installed are tabulated for the high and low ratio cases in Figure 13. Note that the amplitude of the cylinder nozzle resonance is much lower with the ILN installed than without.



■ **Figure 11. Experimental data without ILN installed.**

A second ILN of similar length, but with a different perforation distribution, was installed in the RCTF. The second ILN prototype design had a perforation distribution

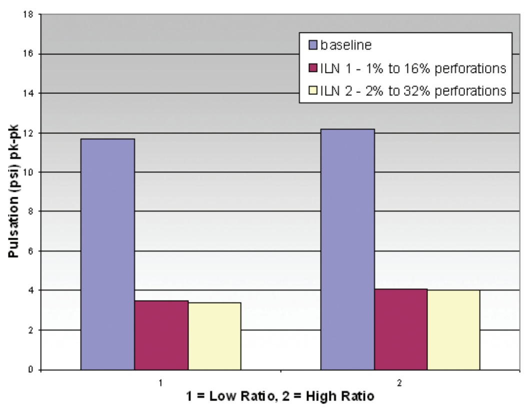


■ **Figure 12. Experimental data with ILN installed.**

Compression Ratio	Without ILN		With ILN	
	Frequency (Hz)	Amplitude (psi p-p)	Frequency (Hz)	Amplitude (psi p-p)
1.7	49	11.7	39	3.5
2.7	50	12.2	39	4.1

■ **Figure 13. Correlation data with and without ILN installed.**

along the projection ranging from 2 to 32% of the bore area. Figure 14 compares the maximum pulsations measured for the baseline piping system to those measured with each of the two ILN prototypes installed. Pulsation measurements acquired for the first ILN are similar to those measured for the second ILN. However, what the graph does not show is the frequency shift that occurred with the second ILN. The second ILLN resulted in a 3 Hz increase in the response frequency as compared to that of the first ILN prototype.



■ **Figure 14. ILN performance.**

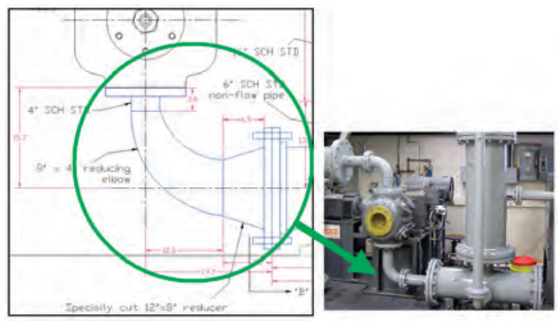
In summary, key observations from evaluation of the two ILN prototypes installed at the discharge cylinder nozzle outlet included: maximum fourth order cylinder nozzle pulsations were reduced by 70% with no orifice; pressure drop was reduced by approximately 72% or more on the fourth order resonance; as a result of installing the second ILN, the pulsation amplitudes remained approximately the same while the resonant frequency increased; nonresonant second order pulsation amplitudes increased because of the distribution of the cylinder nozzle resonance over a large frequency range and the decrease in frequency of the cylinder nozzle resonance.

Although not the most promising of the three concepts evaluated for controlling cylinder nozzle resonance, the ILN demonstrated potential and it was determined that further development is warranted.

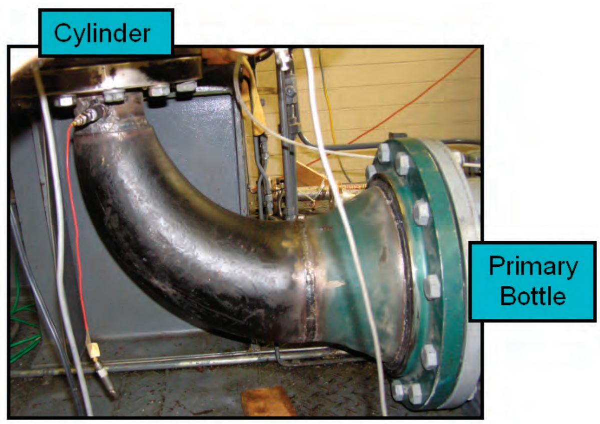
Tapered Cylinder Nozzle

The tapered cylinder nozzle (TCN) concept successfully addresses compressor nozzle resonance issues. The TCN effectively shifts the cylinder nozzle resonance to a higher frequency, reducing mechanical coupling.

The prototype TCN design developed for the reciprocating compressor test facility (RCTF) incorporated an 8 x 4 in. (203 x 102 mm) reducing elbow followed by a 12 x 8 in. (305 x 203 mm) reducer, as shown in Figure 15. The installed TCN configuration is shown in Figure 16. Experimental data with and without the TCN installed are tabulated for the high and low ratio operating conditions in Figure 17. For the low ratio operating conditions, a resonance of 7.7 psi (53 kPa) at 75 Hz was measured in the discharge cylinder nozzle. Without the TCN installed, a resonance of 11.9 psi (82 kPa) at 49 Hz was measured in the discharge cylinder nozzle. Note the much higher cylinder nozzle resonance frequency measured with the TCN installed.



■ **Figure 15. Tapered cylinder nozzle (TCN) concept for the reciprocating compressor test facility (RCTF) discharge cylinder nozzle.**

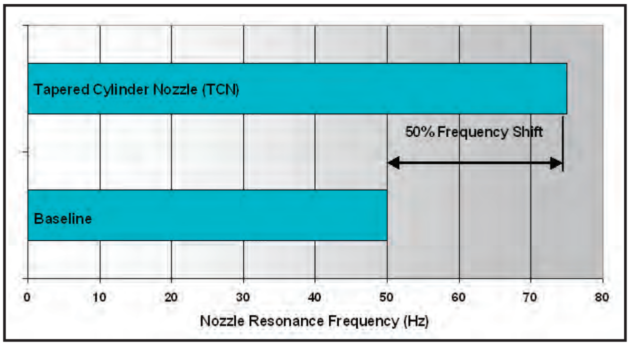


■ **Figure 16. TCN installed in place of the original RCTF discharge cylinder nozzle.**

Compression Ratio	Without TCN		With TCN	
	Frequency (Hz)	Amplitude (psi p-p)	Frequency (Hz)	Amplitude (psi p-p)
1.7	49	11.7	75	7.7
2.7	50	12.2	74	7.9

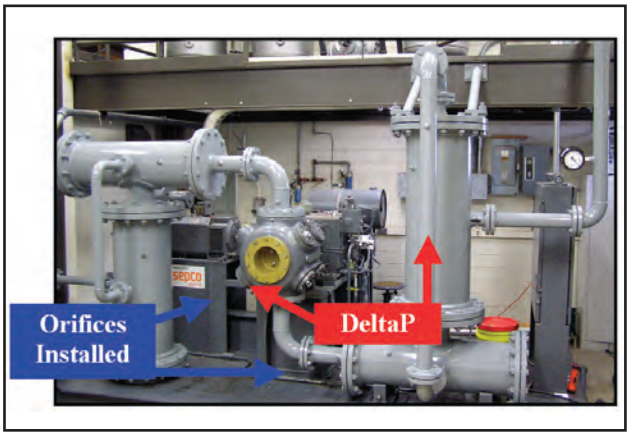
■ **Figure 17. RCTF correlation data with and without the TCN installed.**

Figure 18 illustrates the 25 Hz frequency shift in the cylinder nozzle resonance (50% increase) achieved with the TCN installed. The significant frequency shift observed in this case placed the cylinder nozzle resonance on a higher, weaker compressor harmonic.



■ **Figure 18. Large frequency shift observed with the TCN installed.**

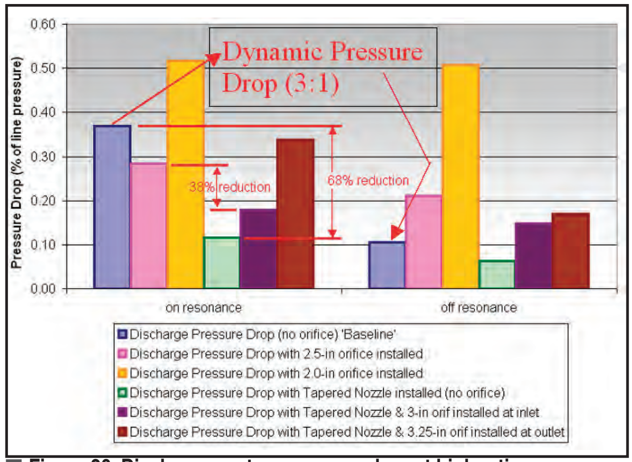
Relative pressure drops were evaluated with and without orifice restrictions for the baseline and TCN installations. Pressure differential data were measured across the cylinder nozzle, primary bottle, choke piping and secondary bottle using the DeltaP measurement locations noted in Figure 19. Various orifice sizes were evaluated in the discharge compressor nozzle at the flange locations identified in Figure 19. Orifices of 2.0 and 2.5 in. (51 and 64 mm) I.D. were evaluated at the compressor cylinder flange in the baseline installation. With the TCN installed, orifices of 3.0 and 3.25 in. (76 and 83 mm) I.D. were evaluated at the compressor cylinder flange and at the compressor nozzle to bottle connecting flanges, respectively.



■ **Figure 19. Orifice locations and pressure drop (DeltaP) measurement locations.**

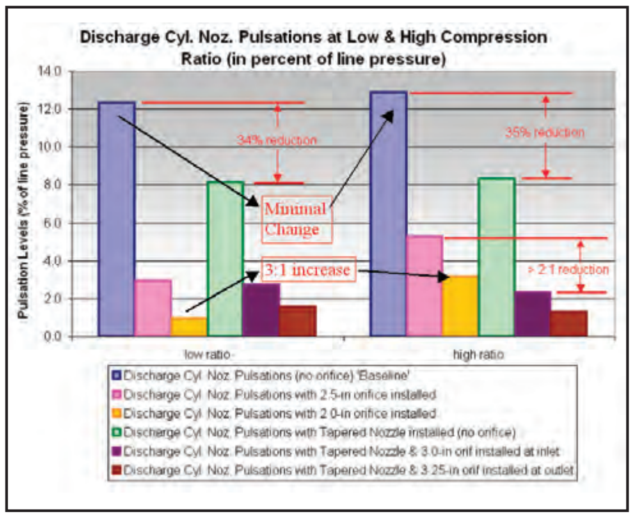
Pressure drop data with and without orifices installed were acquired with the compressor operating on and off the discharge cylinder nozzle resonance. As shown in Figure 20, pressure drop was about a factor of three lower off resonance without the TCN installed due to the dynamic effects of pulsations. In addition, a 68% discharge pressure drop reduction was observed on resonance with the TCN installed (no orifices). Pressure drop was reduced by about 38% with a large-bore orifice installed in conjunction with the TCN, when compared to a small-bore orifice installed in conjunction with a typical cylinder nozzle.

Pulsation data were also acquired for the various configurations described at low and high compression ratios. Figure 21 summarizes the measured pulsation amplitudes for the various configurations analyzed as a percent of line pressure. Without the TCN, similar pulsations were recorded for both the low and high compression ratios. With a relatively small bore orifice installed, pulsations at high ratio (low flow) conditions were three times the pulsations at low ratio (high flow) conditions. The data show that the acoustic effectiveness of an orifice depends on the flow velocity through the orifice. In addition, a 34 to 35% reduction in cylinder nozzle resonance pulsation amplitudes was observed with the TCN installed (no orifices). Cylinder nozzle resonance pulsations were reduced by a factor of 2:1 with a large bore orifice installed in conjunction with the TCN, when compared to a small bore orifice installed in conjunction with a typical cylinder nozzle.



■ **Figure 20. Discharge system pressure drop at high ratio.**

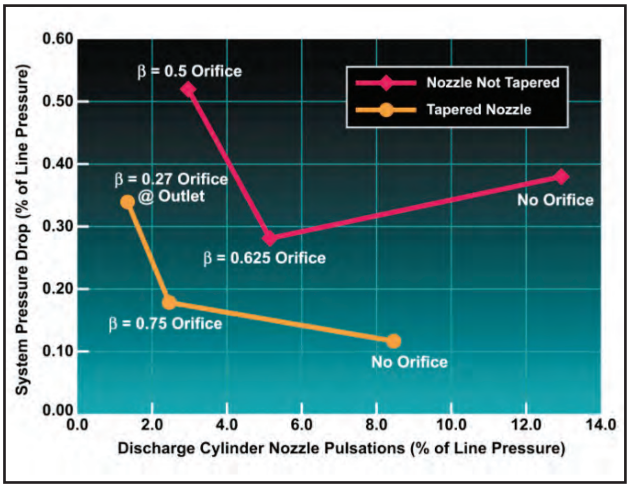
Figure 22 summarizes the high ratio resonance data illustrated in Figures 20 and 21. The x-axis represents the discharge cylinder nozzle pulsations in percent of line pressure and the y-axis represents the system pressure drop in percent of line pressure. It is desirable to be near the origin of the graph such that the system has low pulsations and low pressure drop. The diamonds plot graphically represents the effect of pressure drop on pulsation amplitude without the TCN installed. In a similar manner, the circles plot represents the effect of pressure drop on pulsation amplitude with the TCN installed.



■ **Figure 21. Discharge cylinder nozzle pulsations at low and high ratios.**

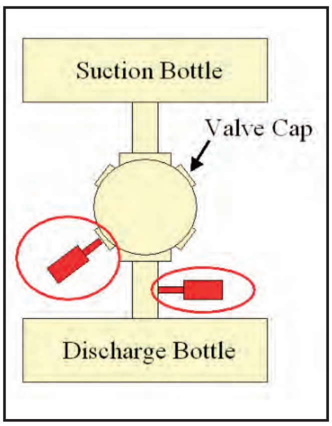
Comparing the diamonds plot from the “no orifice” test point to the “0.625 beta orifice” test point indicates that pulsations decrease significantly while pressure drop decreases only slightly. In addition to that comparison, considering the diamonds “0.5 beta orifice” test point indicates that, pulsations decrease even more; however, pressure drop increases. When comparing the line marked with diamonds to the circles data points, it is important to note that the

0.75 beta orifice circles test point on the orange plot is significantly closer to the origin than the 0.5 beta orifice test point on the diamonds plot. Therefore, the pressure drops and pulsations were lower for a TCN in combination with a 0.75 beta orifice as compared to a typical cylinder nozzle in combination with a 0.625 beta orifice.

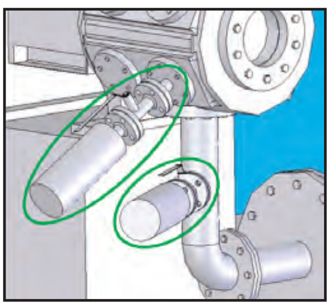


■ **Figure 22. Discharge system cylinder nozzle pulsations and pressure drop.**

A promising feature of the TCN is that it reduces pulsations by shifting the cylinder nozzle resonance to a higher frequency, and most likely reducing mechanical coupling. In summary, key observations from evaluation of the TCN prototype included: the cylinder nozzle resonant frequency shifted above the fourth order with the TCN installed (50% frequency increase); maximum cylinder nozzle pulsation amplitudes were reduced by 34 to 35% with the TCN installed (no orifice); a 3.1:1 pressure drop reduction was observed with the TCN installed (no orifice); the TCN results showed lower pulsations (below 3% of line pressure) with lower pressure drop than would be required in a traditional system without the TCN. (The API 618 pulsation guideline is 7% of line pressure for these operating conditions.)



■ **Figure 23. Diagram showing VO concept, which essentially is a side branch absorber in the cylinder nozzle region.**



■ **Figure 24. A 3-D model of the prototype VOs at the cylinder nozzle and valve cap.**

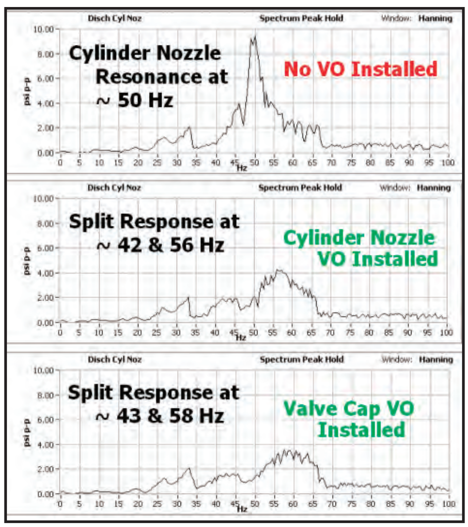
Virtual Orifice

The Virtual Orifice (VO) is a non-intrusive, low pressure drop mechanism that successfully addresses cylinder nozzle resonance. The VO concept, illustrated in Figure 23, is essentially the installation of a side branch absorber in the cylinder nozzle region. The circled objects in Figure 23 represent two possible locations for the VO on the discharge side of the cylinder. In this case, a choke tube connects a volume to the cylinder nozzle or to the valve cap. The choke tube and volume are sized such that the frequency of the VO coincides with the cylinder nozzle resonant frequency.

Two prototypes (Figure 24) were developed, each connecting a volume to either the cylinder nozzle or valve cap using a short span of 1 in. (25 mm) diameter piping. A 1 in. (25 mm) full-bore valve was included in each 1 in. (25 mm) diameter pipe span, allowing data acquisition with and without each of the VOs installed. Opening or closing each valve effectively installs or uninstalls each VO. The VOs were installed in the RCTF, as shown in Figure 25.

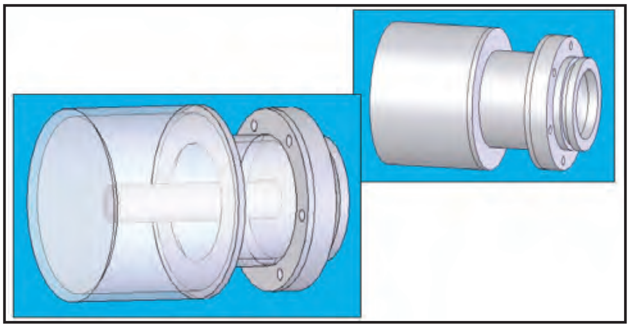
■ **Figure 25. Prototype VOs installed at the cylinder nozzle and valve cap.**

Figure 26 shows measured data without the VO installed, with the VO installed along the cylinder nozzle, and with the VO installed at the valve cap. Since a side branch absorber is known to be more effective when located at a pulsation maximum, lower pulsations were expected with the VO installed at the valve cap. In this case, measured data showed only slightly lower pulsations with the valve cap VO installed, as compared to the cylinder nozzle VO. In general, pulsations at the 50 Hz resonance frequency were reduced by a factor of 4.5 or better with a VO installed.

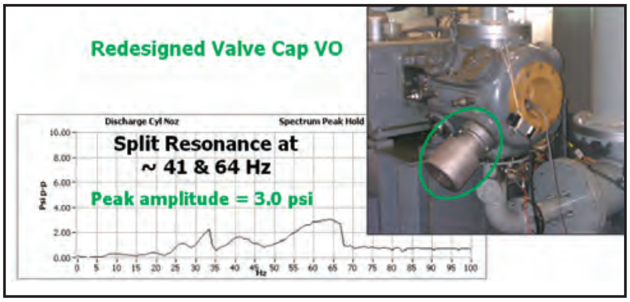


■ **Figure 26. Measured data with and without the VO installed.**

After testing the valved prototypes, a more compact and practical design was built. A 3-D model of the design is illustrated in Figure 27 and the installed VO redesign is shown in Figure 28. Also shown in Figure 28 is the measured pulsation data with the redesigned VO installed. When that data is compared with the data in Figure 26, the redesigned VO reduced the pulsations even more than that of the original VO design.



■ Figure 27. A 3-D model of the redesigned VO.



■ Figure 28. Measured data with the redesigned VO installed.

Because the VO is relatively non-intrusive, it could eliminate the need for an orifice, and because it offers an effective means of reducing the cylinder nozzle resonance, it was determined that further development is warranted. Key features of the VO include: the cylinder nozzle resonance frequency was altered without increasing the pulsations at other orders; the VO design controlled the cylinder nozzle pulsations; reductions of 53 to 61% were observed for the peak pulsation amplitudes; reductions of 78 to 83% were observed for pulsations at the frequency of the original cylinder nozzle resonance; the VO is a relatively simple design that can be installed easily on existing low-speed and high-speed compressors.

Summary

Compressor manifold design must balance several competing design objectives including reducing bottle and attached piping vibration to acceptable levels suitable for long life (low stress) and reliable operation, and reducing system pressure drop thereby increasing overall unit performance. Controlling pulsation levels throughout the system (nozzle, bottle, lateral and header) to acceptable levels while maintaining the broader objectives of reliable, efficient operation is increasingly more difficult with the higher speed, higher horsepower units currently being installed. The technologies described in this presentation have the potential to be applied to currently installed units (both low and high speed), as well as addressing the needs of future designs.

The virtual orifice may eliminate the need for a pressure drop element in the nozzle region of many future designs. Tunable SBAs (TSBAs) can expand the operating window of existing units, not designed for single-ended operation. Incorporating TSBAs into the design methodology for high-speed units may allow for future manifold designs with improved management of vibration and operating stress levels while still providing for good unit isolation and reduced system losses. ■