EFFICIENT BOTTLE-LESS COMPRESSOR PULSATION CONTROL

EXPERIMENTAL TEST RESULTS

2009 GMRC Gas Machinery Conference – Atlanta, GA – October 5-7, 2009

W. Norman Shade, PE, ACI Services Inc.
John J. Bazaar, ACI Services Inc.
Glen F. Chatfield, Optimum Power Technology
John G. Crandall, Optimum Power Technology
Dale K. Wells, Optimum Power Technology

ABSTRACT

Traditional compressor pulsation attenuation systems are carefully designed combinations of primary, and sometimes also secondary, volume bottles, often with complex internal choke tubes, baffles, and chambers, as well as various orifice plates installed at specific locations in the system piping. These devices accomplish pulsation control by adding resistance, or damping, to the system; and they result in additional system pressure losses upstream and downstream of the compressor cylinders. These pressure losses reduce the overall system efficiency, but the trade-offs are tolerable for most compressor applications. However, for common pipeline transmission applications having low pressure ratios (in the range of about 1.1 to 1.6), system pressure losses can noticeably degrade the overall operating efficiency, especially with the use of higher speed (>600 rpm) compressors.

Previous GMC papers have reported the results of computer model simulation studies and lab testing of low-pressure reciprocating air compressors with tuned pulsation attenuation networks (PANs). Both the simulations and the lab tests showed that properly configured tuned pulsation attenuation networks are effective means of controlling compressor pulsations with little or no resultant system pressure loss.

Continuing research has explored new proprietary PAN geometries that completely eliminate the need for traditional pulsation bottles, choke tubes and orifices, with the potential for reduced system cost as well as reduced system pressure losses. This paper reports the results of further laboratory testing as well as the first field testing of a discharge PAN system containing no pulsation bottles and no orifices applied to a 750 to 1000 rpm, 6” stroke reciprocating compressor. Field test results are compared with predictions for both the PAN system and the original two-bottle system, showing the pulsation reduction and system pressure drop for each configuration and confirming the significantly lower pressure drop of the PAN system.

INTRODUCTION

Traditional reciprocating compressor pulsation attenuation systems employ combinations of primary and/or secondary volume bottles, often with complex internal choke tubes, baffles, and chambers, as well as various orifice plates installed at specific locations in the system piping. These devices accomplish pulsation control by adding resistance, or damping, to the system; and they result in additional system pressure losses upstream and downstream of the compressor cylinders. Even when the systems are well designed, the resulting pressure losses reduce the overall system efficiency. The trade-offs are usually tolerable for most compressor applications. Yet, for common pipeline transmission applications having low pressure ratios (in the range of about 1.1 to 1.6), system pressure losses can noticeably degrade the overall system operating efficiency, especially when higher speed (>600 rpm) compressors are used.

The excessive system pressure loss problem has become more apparent in recent years as the use of large, high-speed reciprocating compressors has become more prevalent in the USA pipeline industry. Various solutions are being researched in the industry, and although some show promise, they all revolve around the continued use of volume bottles and damping devices as the fundamental pulsation control technologies. With the increasing importance being placed on higher efficiencies and reduced carbon footprints, more efficient approaches are desirable.

For many years the best high performance engine designers have successfully utilized pulsating pressure waves in reciprocating engines to reduce noise and to tune the intake and exhaust systems to dramatically improve performance. However, until recently, little had been done to investigate the application of this proven theory to compressors where pressures, and therefore forces and energy levels, are much higher and challenging to manage. Extensive computer modeling and simulations of tuned pulsation attenuation networks (PANs) applied to reciprocating compressors have been done by the authors, and the results have shown that properly configured tuned pulsation attenuation networks have the potential to be an effective means of controlling compressor pulsations with little or no resultant system pressure loss [1].

First generation PANs were made up of one or more sequential stages called tuned delay loops [3]. Figure 1 illustrates a compressor cylinder with a single tuned loop PAN in the suction line upstream of the cylinder and a single tuned loop PAN in the discharge line downstream of the cylinder. For the suction tuned loop, the length of the long leg (L_2) minus the length of the short leg (L_1) causes a time delay or phase shift. The internal flow area of each leg is approximately one-half of the flow area of the incoming suction line and the compressor suction nozzle pipe at the exit of the loop. For the discharge tuned loop, the length of the long leg (L_4) minus the length of the short leg (L_3) causes a time delay or phase shift. The internal flow area of each discharge loop leg is approximately one-half the flow area of the compressor discharge nozzle pipe at the loop entrance and discharge line at its exit.

The junctions that split the flow equally into two loops and join the flow back together again must be carefully configured to minimize separation, turbulence and pressure losses. An economic, compact, robust junction device, referred to as a Tuning Section Transition, or TST, has been developed [4] to enable the application of the PAN technology to reciprocating compressors in typical natural gas and other compressor applications.

The TST may be in the form of a separate device that joins two separate loops, or it may be a more compact in-line element that integrates the shorter of the two loops completely within the TST itself as shown in Figure 2. Shown are the internal flow channels of a 1500 psig MAWP in-line TST design that joins the long and short loops for an actual field application using standard 8 in. and 6 in. nominal pipe sizes. The difference in lengths of the two flow paths in the PAN loops is "tuned" to the frequency of a pressure wave to dramatically reduce the pulsation in the piping. If the difference in length is properly tuned to compressor RPM, the number of cylinders and the acoustic velocity of the gas, the pulsations will be substantially reduced with no significant pressure loss.

Properly designed PAN loops will cancel the fundamental frequency that is targeted, along with all of the odd harmonics of that frequency. This phenomenon is interesting, but usually not adequate by itself to cancel all the frequencies present in a typical gas compressor system. However, when two or more PAN loops are employed in series, each designed to cancel a different fundamental frequency and its odd harmonics, the resulting bands of cancelled frequencies become much broader. With accurate system modeling and simulations, this characteristic can be carefully optimized to provide a well-designed pulsation cancellation system that does not depend on the traditional dampening devices and techniques that create undesirable pressure losses in the system. In addition to many extensive computer simulations, the application of this technology was successfully validated with laboratory testing with a two-throw double-acting low-pressure air compressor [2].

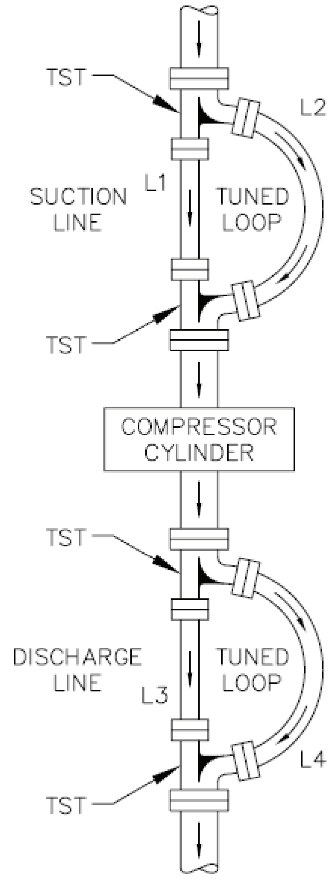

Figure 1: *Pulsation attenuation network (PAN) with tuned loops in the compressor suction & discharge.*

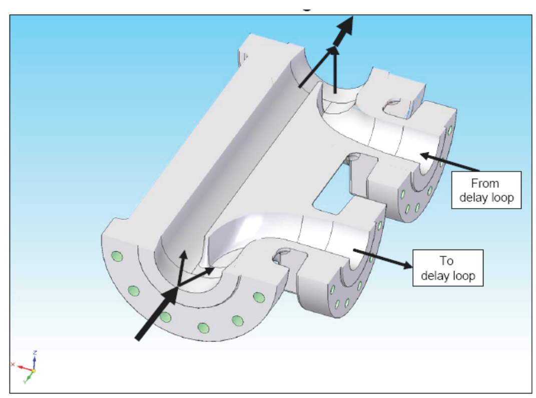

Figure 2: *Cross-section of in-line TST that integrates the short tuned loop internally.*

PROOF OF CONCEPT TEST CONFIGURATION & EVALUATION

With the visionary support of the El Paso Corporation, a full scale proof of concept system was designed, fabricated, installed and tested on a Superior MH64 single-stage reciprocating compressor at the company's Ellisburg, PA station. The existing compressor installation, which included two-bottle (primary and secondary) pulsation control systems on the suction and discharge of each side of the compressor, is shown in Figure 3.

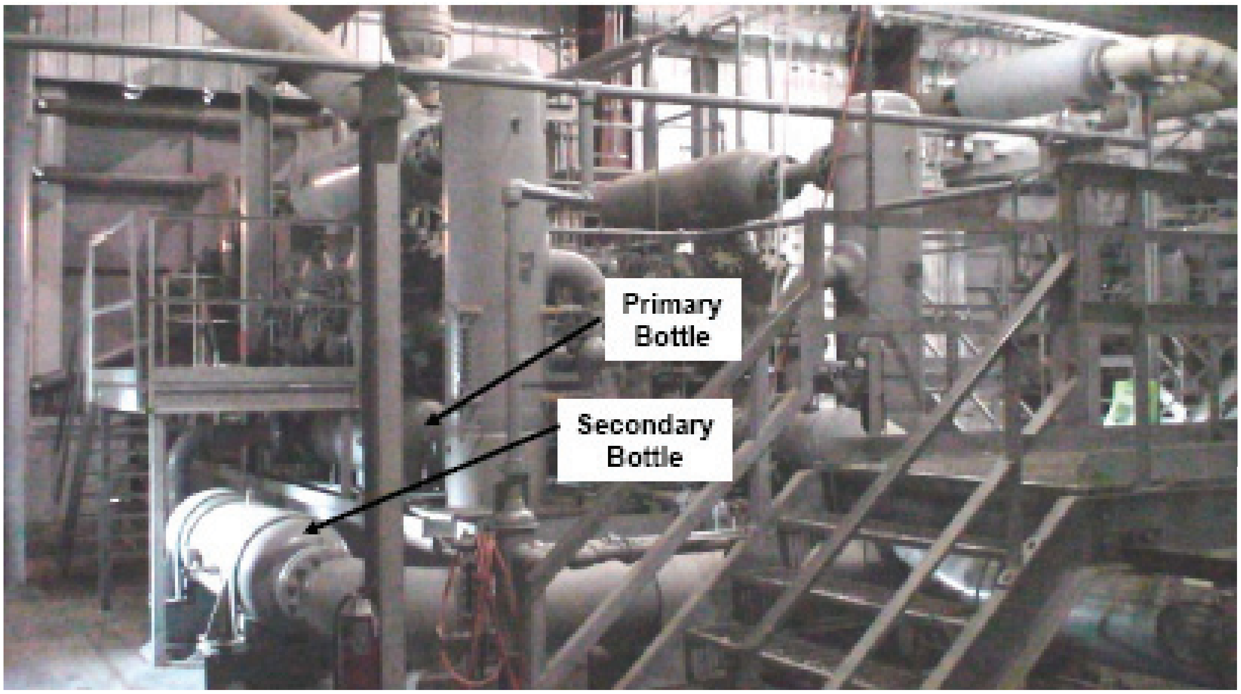

Figure 3: *Existing field compressor installation*

It can be seen that space on the skid is very crowded, and since any field proof-of-concept test has some measure of uncertainty, it is very important to minimize the permanent changes to the existing system, so that the original components can be quickly reinstalled if necessary.

Of course, the application of this technology to a large, high-pressure compressor in the field must address the significant dynamic forces that are present in the piping system prior to the cancellation of the pulsations. This requires that a detailed system mechanical analysis be completed and that the PAN components be robust and safe designs. Many of the same technologies used in the design and manufacture of modern reciprocating compressor cylinders and piping systems can be utilized for the PAN system elements. The resulting field proof-of-concept system that was retrofitted to the discharge of one two-throw side of the four-throw, 750 - 1000 rpm Superior MH64 compressor is shown in Figure 4.

This second generation PAN includes two tuned length discharge pipes connected to a Y-branch TST, followed by two tuned loops in series using in-line TSTs. This design completely eliminates the need for the traditional large pulsation bottles, internal baffles, choke tubes

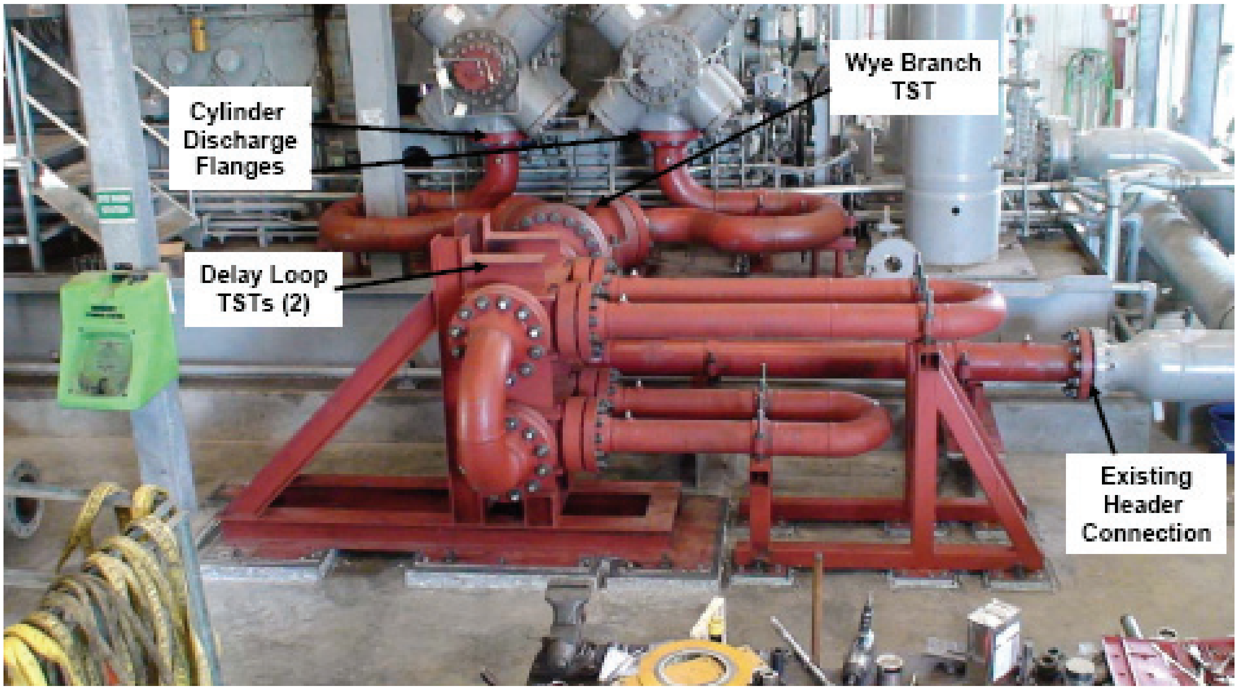

Figure 4: *Proof-of-Concept PAN system installed on field compressor*

and orifice plates. The requirement that the system be removable to enable reinstallation of the existing two-bottle discharge system resulted in more complexity than what would be required with a complete "clean sheet" system design. "Packaging" of the PAN systems remains a work in progress, and as the development progresses, it is anticipated that more compact and less complex designs will result.

Design Iteration Based on Computer Simulations

The design of the first field proof-of-concept PAN system has been an evolutionary process. Figure 5 compares the predicted pulsation control effects and the pressure drop of the field compressor operating at 850 rpm with three alternate pulsation control systems applied to the discharge side of one two-cylinder side of the compressor. The existing baseline system has a header bottle connected to the two parallel 9.5 in. diameter cylinders. The baseline system also employs flange orifices and a secondary pulsation bottle. The bottles contain internal baffles and choke tubes. As verified by simulation and actual field testing, the pulsation control of the baseline system is good, and the pressure drop is not exceptionally high at the conditions shown. However, at lower pressure ratios, the higher system pressure drop adds an appreciable amount to the horsepower that is required to push the flow through the system.

In designing a first PAN system that could be retrofitted to this compressor with minimal cost and disruption, the intent was to replace the complex two-bottle, choke tube and flange orifice system with a simple (empty) plenum header bottle to tie two parallel cylinders together for connection to the two-loop PAN network. However, the analysis of the plenum bottle, which had a fairly large diameter, showed that the high-pressure pulsations resulted in some significant forces that had to be reduced in order to achieve acceptable stresses in the bottle. As a result, an internal baffle and large orifice had to be added within the bottle. Although this controlled the forces to acceptable levels, it also added significant pressure loss, so that the resulting hybrid system was quite far from ideal. As shown in Figure 5, for a pressure ratio of 1.43, the predicted pulsation control was similar to the baseline system (less than 0.2% of the pressure level), but the hybrid system's pressure drop improved by only about 8%, with more than half of the pressure drop occurring inside the baffled bottle.

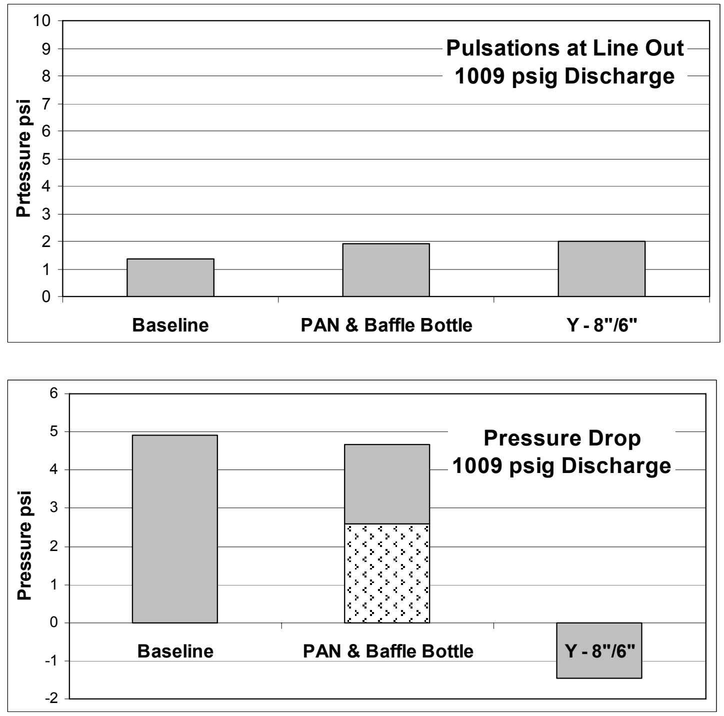

Fig. 5: *Predicted comparison of baseline system with PAN system on discharge side of 2 parallel cylinders.*

With that lesson learned, further design, modeling and simulations were used to create the much more promising configuration shown in Figure 4. Taking advantage of the 90 degree crankshaft phasing between throws, the authors created an innovative manifold Y-branch TST that could be used to connect the two 8 in. diameter cylinder discharge connections with very favorable results. The use of equal lengths of pipe between the cylinders and the Y-TST resulted in the inherent cancellation of much of the pulsation as the two flow streams merged at the Y-TST outlet. The use of 8 in. pipe instead of a 30 in. diameter header bottle made the pulsation induced dynamic forces more manageable. The outlet side of the Y-TST expands into a 12 in. diameter section and then a nozzle is used to gently reduce the diameter back down to 8 in. diameter so that the previously built TSTs and PAN loops could be connected downstream of the Y-TST. Figure 5 shows that this arrangement controls the predicted pulsations to less than 0.2% of the pressure level, yet no pressure drop is predicted. In fact, due to the Bernoulli effect, the simulations predict a slight pressure gain in the discharge piping for some speeds.

The relative pulsation cancellation effects predicted for the initial Y-TST and the tuned loops that make up the discharge PAN system are shown in Figure 6 for a pressure ratio of 1.49. Since the system has to operate over a 750 to 1000 rpm speed range, this multiple loop system is necessary for best results. The manifold Y-TST alone is very effective at speeds up about 950 rpm. The additional PAN loops are effective in reducing pulsations even further across the entire speed range.

Figure 7 compares the predicted pressure drop for the initial prototype of the bottle-less PAN system with the current two-bottle dampener system across the speed range for a 1.49

pressure ratio. The bottle-less system has significantly less pressure loss than the original conventional two-bottle attenuation system. In fact, the simulations predict a modest increase in discharge pressure at the higher speeds due to the Bernoulli effect in the expansion section downstream of the Y-branch.

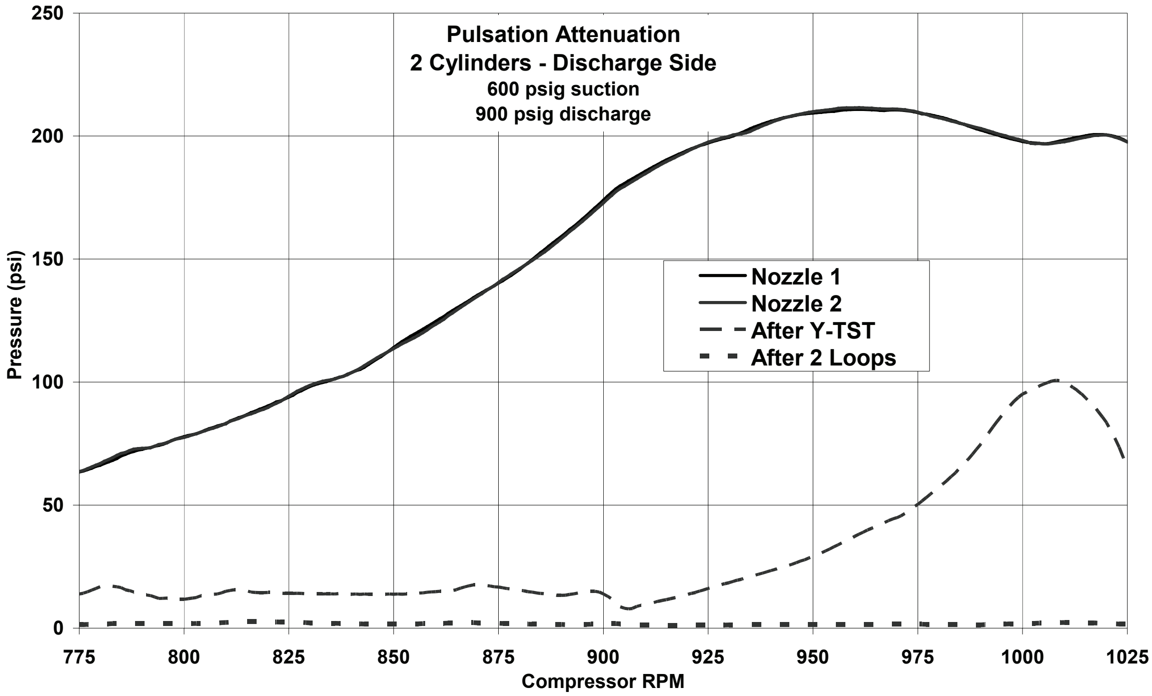

Figure 6: *Predicted pulsation reduction in bottle-less PAN.*

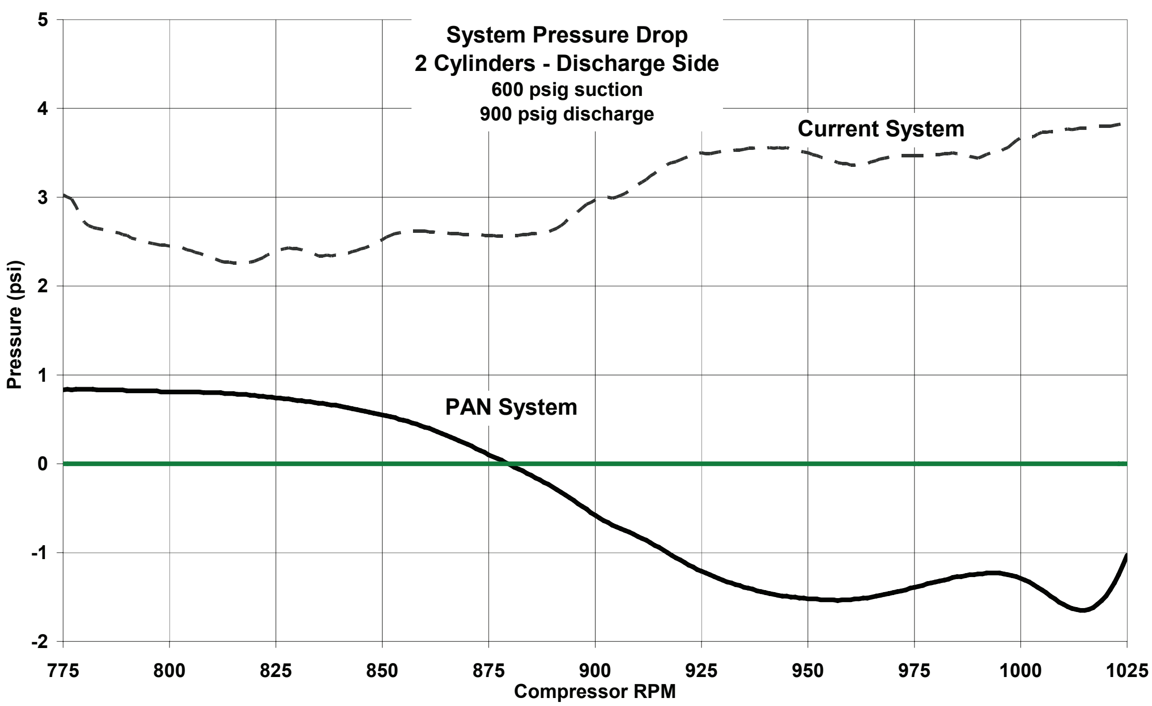

Figure 7: *Comparison of predicted pressure drop in traditional bottle-orifice dampener system and a bottle-less PAN system.*

Laboratory Air Compressor Testing: Proof-of Concept Y-Manifold and Tuned Loop PAN

The system discussed here was designed to be most effective for cases where all cylinders are double acting. A PAN system with different loop lengths would be required for cylinders that can be operated in single acting mode. This variant was successfully addressed in the low-pressure air compressor testing, and it was further evaluated in later field phases. Prior to building the proof-of-concept PAN system for the field evaluation, further lab testing was conducted using a low-pressure reciprocating air compressor set up as explained further in reference [2]. In this testing phase, the test compressors were configured to simulate the crank-throw timing and the volumetric efficiency of the MH64 field compressor. The lab test loop was fabricated from plastic components and pipe having the same lengths as the system designed for the field configuration.

Figure 8 compares pressure traces measured at several points in the low-pressure air compressor test system with the compressor operating at 1000 rpm. The trend shows a very significant pulsation cancellation effect after the manifold Y-TST, which is similar to what was predicted by the earlier simulations shown in Figure 6. The amplitude is further reduced by the first delay loop, and the second delay loop creates further reductions in the pulsation levels.

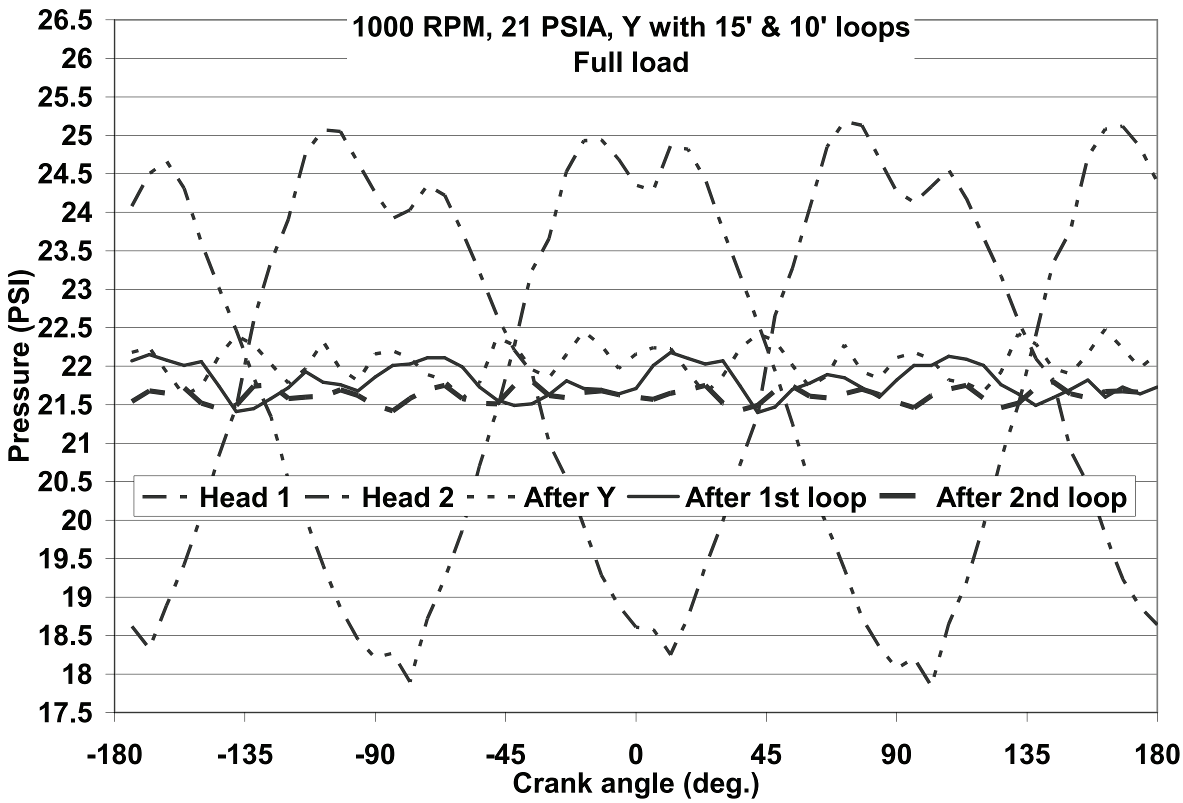

Figure 8: *1000 rpm pulsations in air test PAN system*

Figures 9 and 10 show pulsation levels at these same points in the system over the entire operating speed range. Figure 9 is for the case of both cylinders fully loaded, and Figure 10 is for the case of 2 cylinder ends partially unloaded with added fixed clearance to reduce their volumetric efficiency. As expected, the measured pressure drops from the air test PAN were extremely small throughout the speed range.

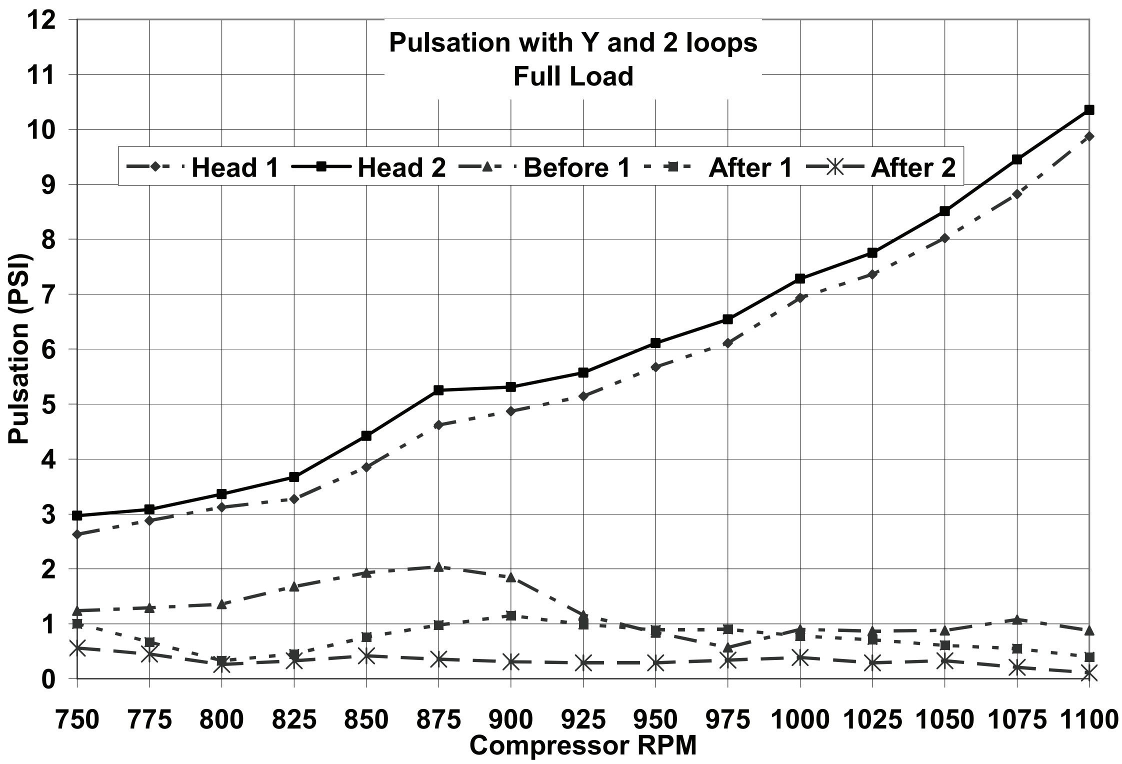

Figure 9: *Air test PAN pulsation measurements.*

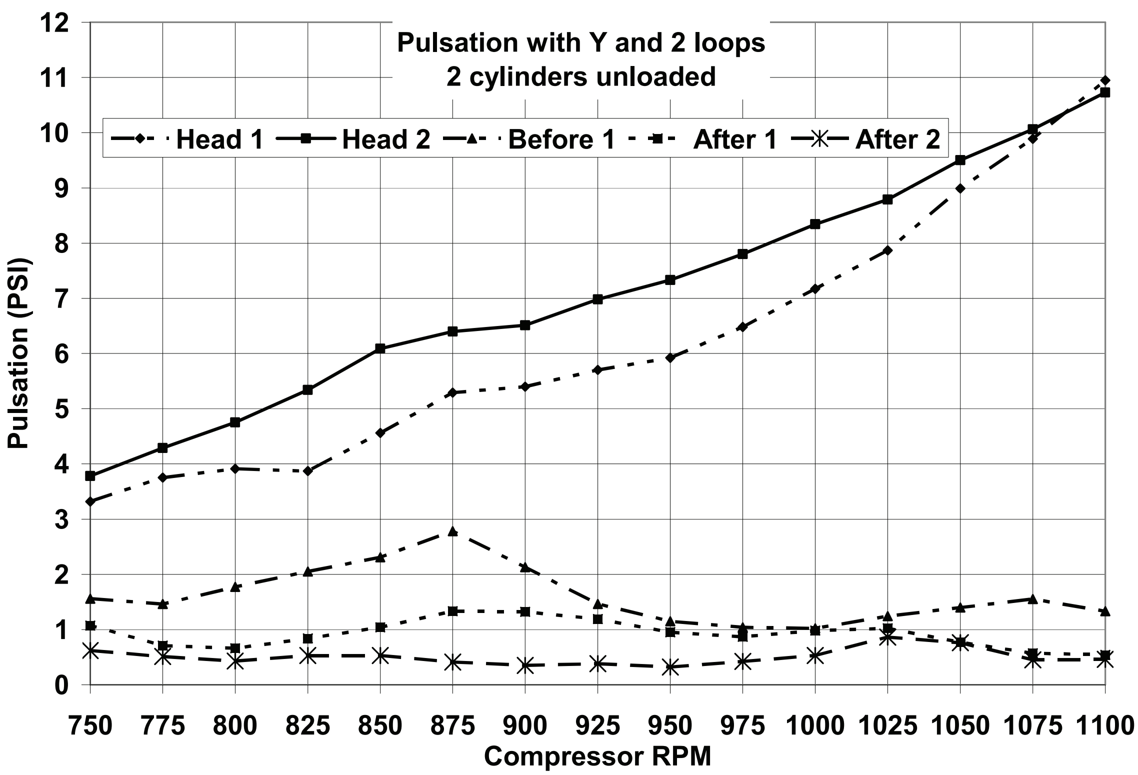

Figure 10: *Air test PAN pulsation measurements.*

Field Testing: Existing Configuration

Operating Condition	Speed Rpm	Suct. (psig)	Disch. (psig)	Pressure Ratio	Measured ΔP (psi)	Predicted ΔP (psi)	Error (psi)
Both cylinders fully loaded	1000	600	900	1.488	4.90	4.26	0.64
Both cylinders fully loaded	1000	665	810	1.213	4.58	4.43	0.15
Both cylinders with FVCP open	1000	600	900	1.488	4.19	3.46	0.73
Both cylinders with FVCP open + other side SACE	1000	665	810	1.213	4.58	4.53	0.05
Both cylinders with FVCP open + other side SACE	800	600	800	1.326	2.71	2.38	0.33

Figure 11: *Field test pressure drop results for existing discharge two-bottle system.*

After installing additional pressure measurement connections into the piping, baseline testing of the existing Ellisburg two-bottle field system was conducted in 2008. A full series of pulsation and pressure drop data were obtained over a range of operating speeds at various unloading steps.

Figure 11 compares some of the measured pressure drops with predicted values, from the simulation, on the discharge side of the compressor system. The comparison between measured and predicted varies from 0.05 to 0.73 psi for the various operating conditions and load steps shown.

Field Testing: Proof-of-Concept Y-Manifold and Tuned Loop PAN Configuration

The previously described proof-of-concept manifold and tuned loop PAN system was installed on one of the Ellisburg compressors, as shown in Figure 4, in June 2009, with testing taking place through the month of July. Fitting this first PAN design into the existing station with minimal changes posed a major design challenge. It was necessary to avoid interference with conduit, utility piping, structural supports, etc. It was also necessary to preserve mounting pads for the existing two-bottle system, so that it could be reinstalled after the PAN proof-of-concept testing was completed. Despite extensive site measurements, 3-D CAD modeling and mechanical analyses, a last minute piping change was required to avoid interference with conduit and utility piping located along the side of the compressor skid and which could not be easily moved for the proof-of-concept PAN installation. This resulting PAN piping change added length to the pipes between the cylinder discharge flanges and the Y-branch TST. Although the change was necessary and believed to have minimal consequences at the time it was made, operating behavior proved otherwise, as the pressure amplitude increased and the longer pipes provided more area on which the higher pressure amplitudes acted.

Figure 12: *View of longer than optimal PAN pipes before Y-branch.*

Of course, the pulsations in this section of piping are not yet attenuated in any way, so they tend to be significant in amplitude as are the resulting forces. In addition, the geometric arrangement that was required to fit to the previously designed system components made it impossible to adequately clamp the piping to the main structural sections of the skid without major rework. A decision had been made not to install concrete piers and to instead tie the PAN mounting stand to the 4 to 5 in. thick concrete floor using a broad mounting foot, grouted to the floor and held down with multiple foundation bolts.

A better support system was designed and could have been fabricated for adequate clamping of the pipes, however, in consideration of the fixed project budget and limited availability time for the compressor, it was decided to make the PAN proof-of-concept system a temporary change and to accept tolerable, but higher than optimal, piping vibration during the testing and evaluation phase, followed by reinstallation of the original bottle system before the critical winter operating period.

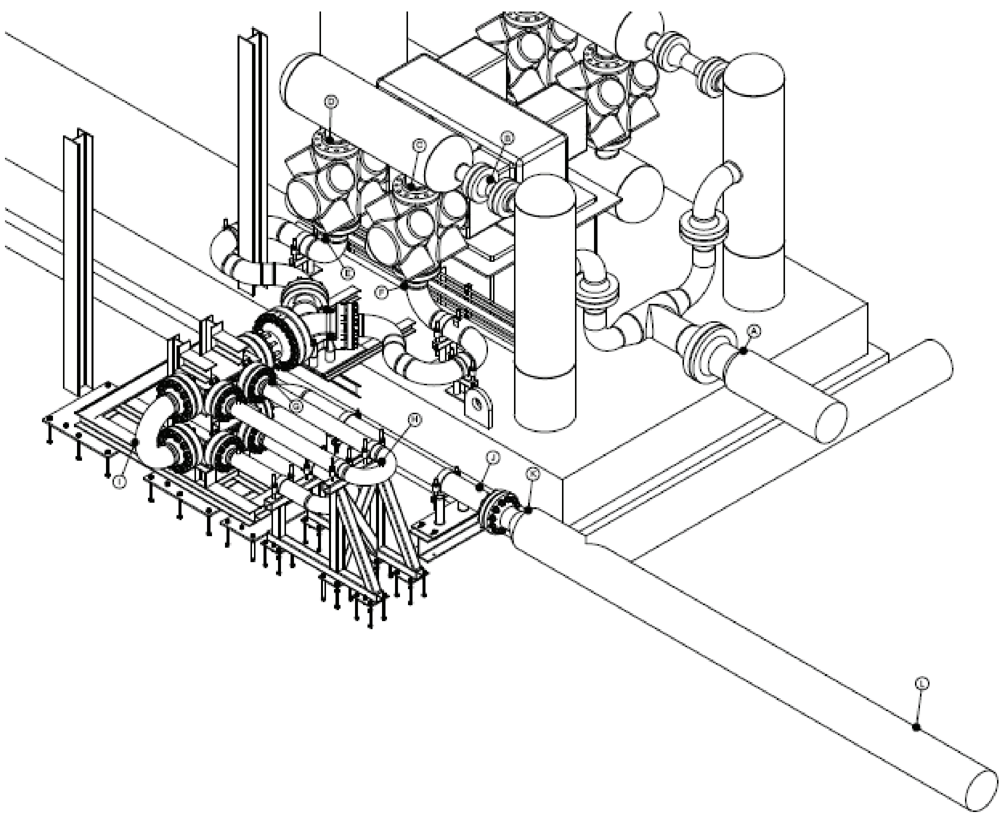

Figure 13: *Field test pressure measurement locations.*

Instrumentation points for the field tests are shown in Figure 13. Pressure transducers with extremely high accuracy were chosen and a rigorous calibration process was followed to ensure that all measurements were obtained with the best possible precision and accuracy. Test data was acquired over a wide range of operating speeds, pressure ratios and unloading steps.

Examples of pressure waves from the discharge PAN system are shown at 860 and 1000 rpm in Figures 14 and 15, respectively. First looking at the 860 rpm test data in Figure 14, the average discharge pressure was 912 psig for this fully loaded run. The pressure wave in the piping immediately after the discharge flanges of each compressor cylinder had a peak-to-peak amplitude of 46.4 psi, with frequencies that were predominantly the typical 4X harmonics of running speed. It can be seen that the pulsations were reduced by 79% to 9.8 psi peak-to-peak after the manifold Y-TST, showing that this first PAN section is very effective. The two delay loops further reduced the peak-to-peak pulsation amplitude to 4.8 psi, which is only 0.5% of the discharge line pressure.

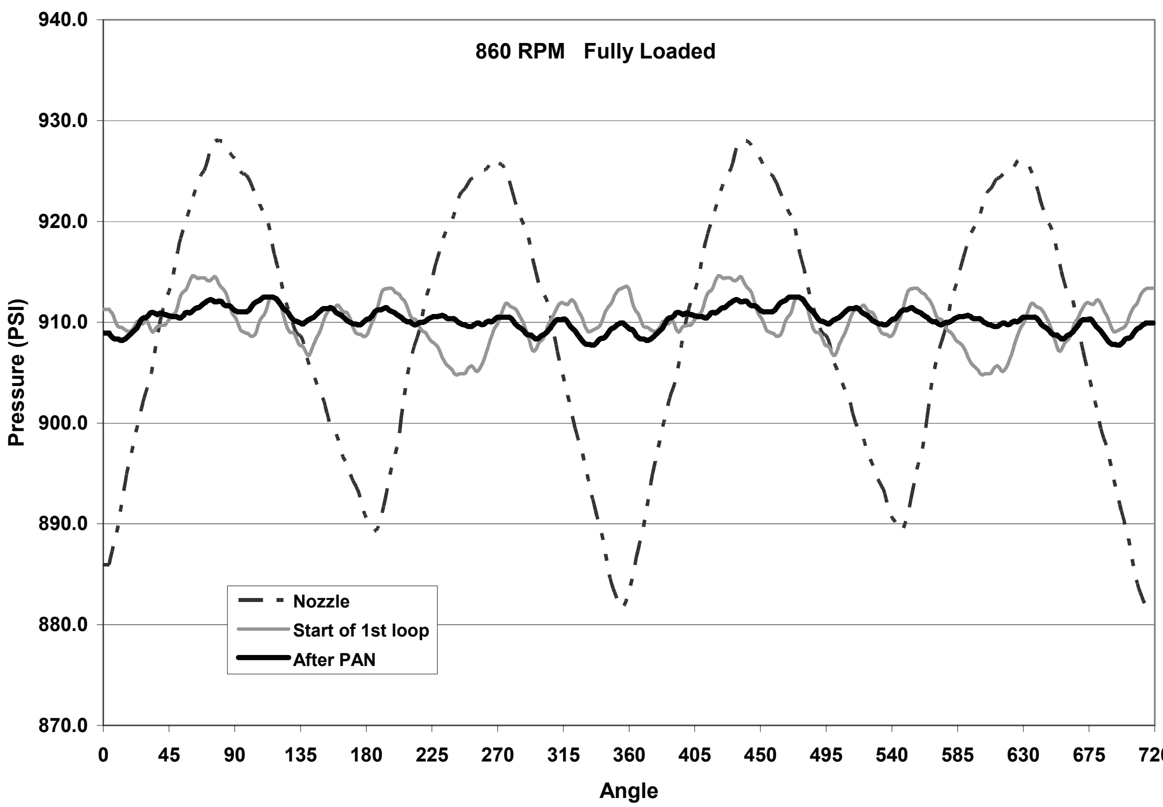

Figure 14: *Field test discharge PAN* pulsation *measurements at 860 rpm – fully loaded.*

For the 1000 rpm, fully loaded run, shown in Figure 15, the average discharge pressure level was 914 psig. The pressure wave in the piping immediately downstream of the compressor discharge flanges had peak-to-peak amplitude of 44.5 psi and contained more frequency components, but mostly the typical 2X and 4X harmonics of the running speed. The pulsations were reduced by 53% to 20.8 psi peak-to-peak after the manifold Y-TST, which is a significant level of pulsation reduction, although not quite as dramatic as found at the lower speed in Figure 14 due to the proximity of a 990 rpm acoustic resonance. The 990 rpm resonance resulted from the longer than optimal pipes required between the cylinders and the Y-TST to avoid interference with obstacles on the existing compressor skid. As before, the two delay loops further reduced the peak-to-peak pulsation amplitude to a low level of 5.3 psi, which is only 0.6% of the discharge line pressure.

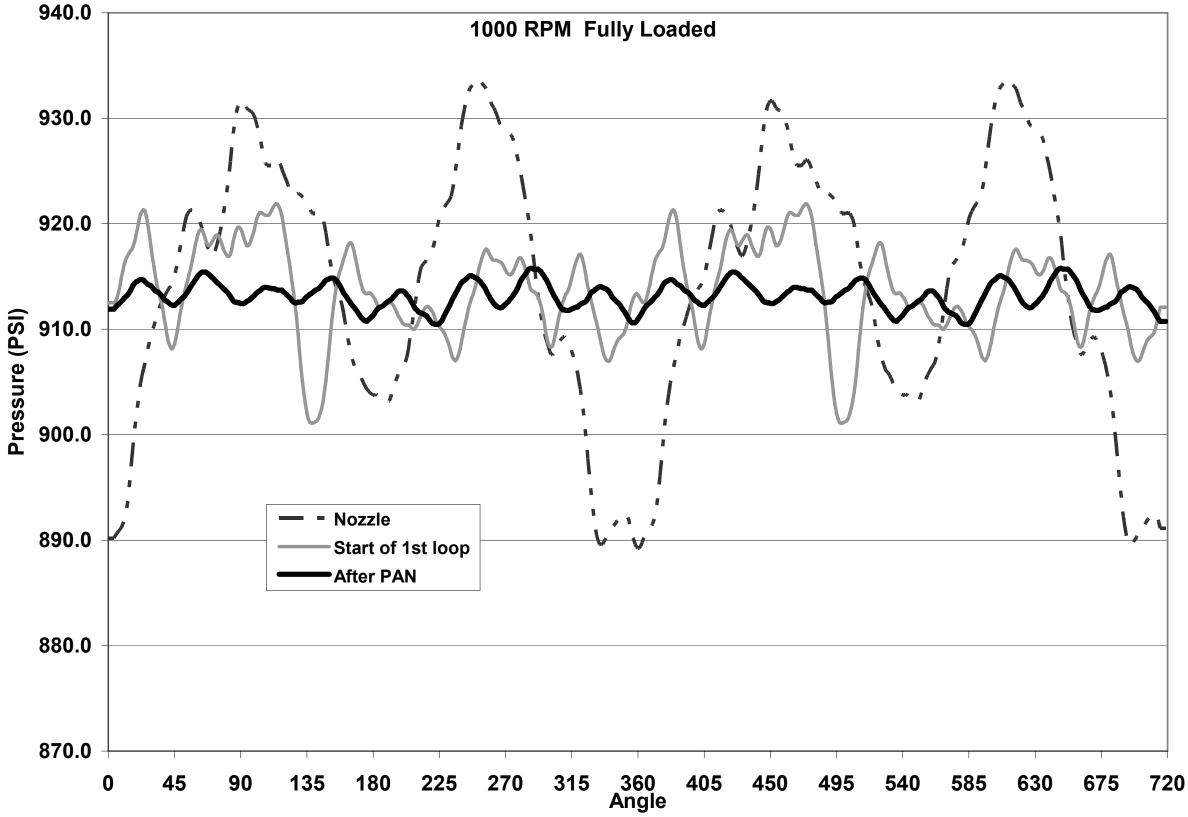

Figure 15: *Field test discharge PAN pulsation measurements at 1000 rpm –fully loaded.*

Overall pulsations from the PAN system are plotted in Figure 16. The data were obtained with an average discharge pressure of 886 psig and a constant pressure ratio of 1.35 over the speed range from 860 to 1000 rpm. At this high-flow, low-ratio condition, the line side pressure pulsations were controlled to between 2 and 5 psi peak-to-peak, which corresponds to 0.2 to 0.6% of the line pressure.

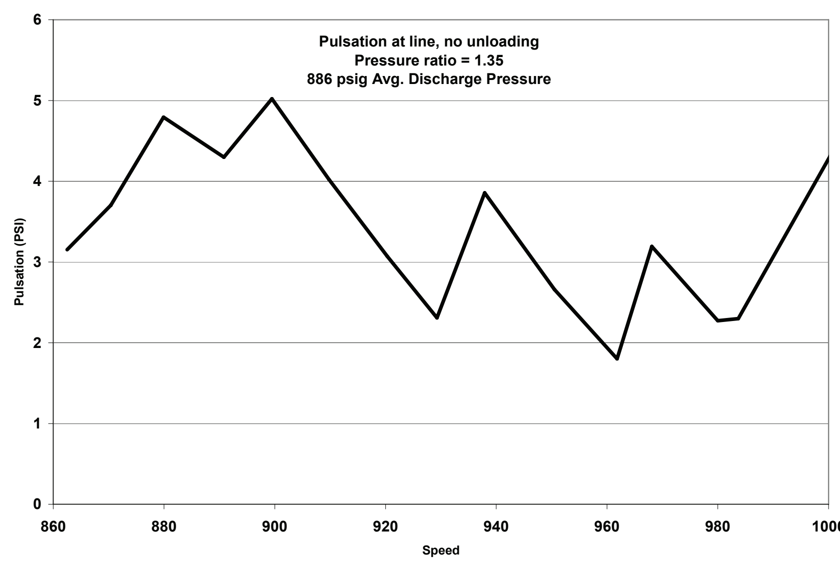

Figure 16: *Field test discharge PAN pulsation measurements at various operating speeds – 1.35 pressure ratio.*

A summary of the pulsation cancellation at various speeds is shown in Figure 17. The reductions after the manifold Y-TST range from 47 to 79%. The total pulsation reductions after the two in-series delay loops range from 88 to 93% over the speed range, with the final pulsations reduced to 0.4 to 0.6% of the discharge line pressure. Both of these data sets confirm that the PAN system is very effective at controlling pulsations over a wide range of operating speeds.

Speed (rpm)	Pulsation at Disch. Flange (P/P psi)	Pulsation after Y-Branch (P/P psi)	Reduction at Y-Branch (%)	Pulsation after Loops (P/P psi)	Total Reduction (%)	Pulsation % of Line Pressure
860	46.4	9.8	79	4.8	90	0.5
900	50.8	27.0	47	3.6	93	0.4
950	43.9	14.8	66	4.0	91	0.4
980	43.5	20.6	53	4.8	89	0.5
1000	44.5	20.8	53	5.3	88	0.6

Figure 17: *Field test discharge PAN pulsation measurements at various operating speeds.*

Of course, the main reason for investigating PAN technology for reciprocating compressors was to confirm its potential for controlling reciprocating compressor pulsations with little or no system pressure losses. The field test compressor was equipped with the aforementioned PAN system on one two-throw side and retained the original two-bottle pulsation bottle system on the other two-throw side. The two systems were joined together at the 16 in. diameter discharge trunk line at the end of the compressor skid. Suction systems on both sides were the original two-bottle systems. Dynamic data was taken more or less simultaneously (within a very short time difference while operating conditions were stable) on both sides of the compressor, so that the pressure drop could be compared. Figure 18 summarizes the measured discharge pressure losses of the two systems over the range of test speeds from 860 to 1000 rpm. This group of tests was run at a high-flow, 1.1 pressure-ratio condition where the system pressure losses tend to be highest. It can be seen that there were small differences in system pressure losses for the individual cylinders on a particular side. Some of this difference might be attributed to measurement uncertainty, considering the fact that small pressure losses were obtained as a difference between two relatively large measured pressure levels. However, since the trends were reasonably consistent over the speed range, it is quite likely that differences were caused by slightly different volumetric efficiencies in the adjacent cylinders. In actual fact, the volumetric fixed clearances of the head ends were different because of the different unloading devices mounted on the cylinder ends.

What is most remarkable in Figure 18 is the 5 to 6 psig difference in measured pressure losses between the two systems across the entire speed range. At 860 rpm, the PAN system discharge pressure loss is only 19% of the bottle system. At 1000 rpm, the PAN pressure drop is higher, but still only 38% of the pressure loss in the two-bottle system. This reduced discharge pressure loss results in approximately 7% lower pressure differential that the cylinders have to move gas across at this low ratio condition. This significantly reduces the horsepower required to move flow through the system. Cylinder indicator cards were taken

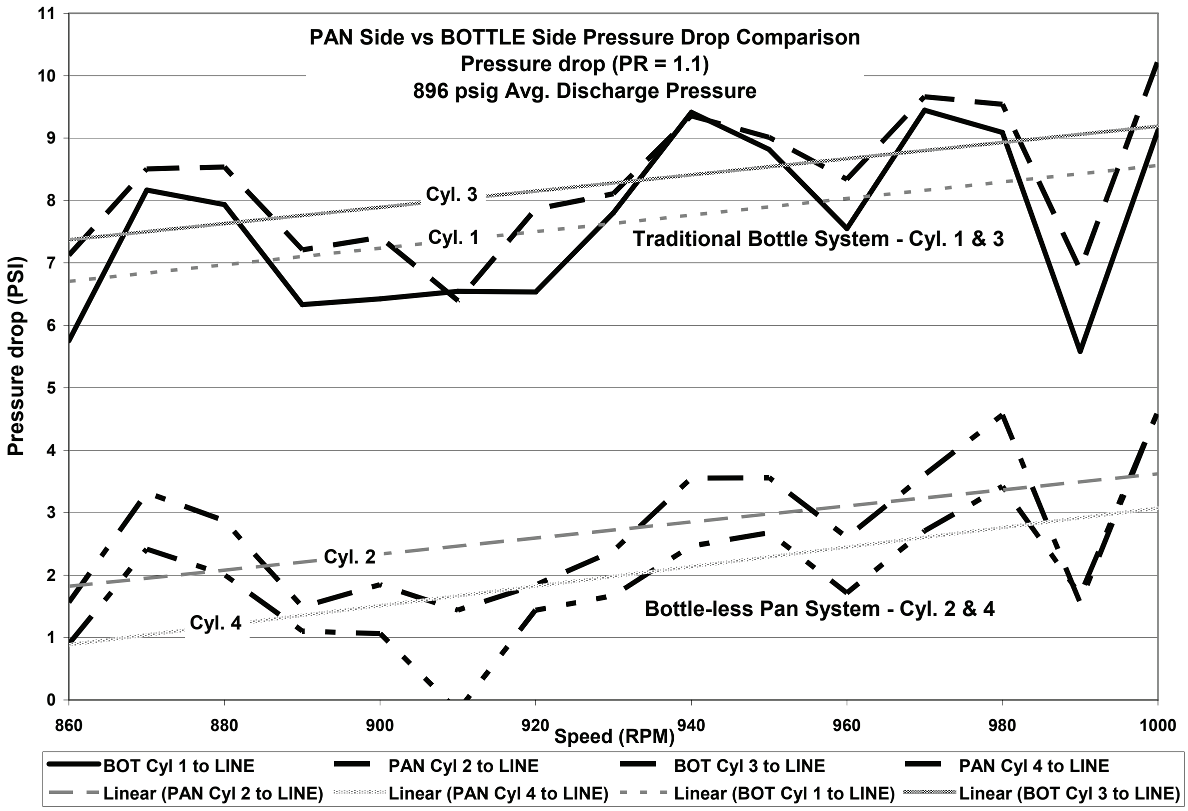

Figure 18: *Field test pressure loss comparison at various operating speeds at 1.1 pressure ratio for PAN and traditional bottle systems*.

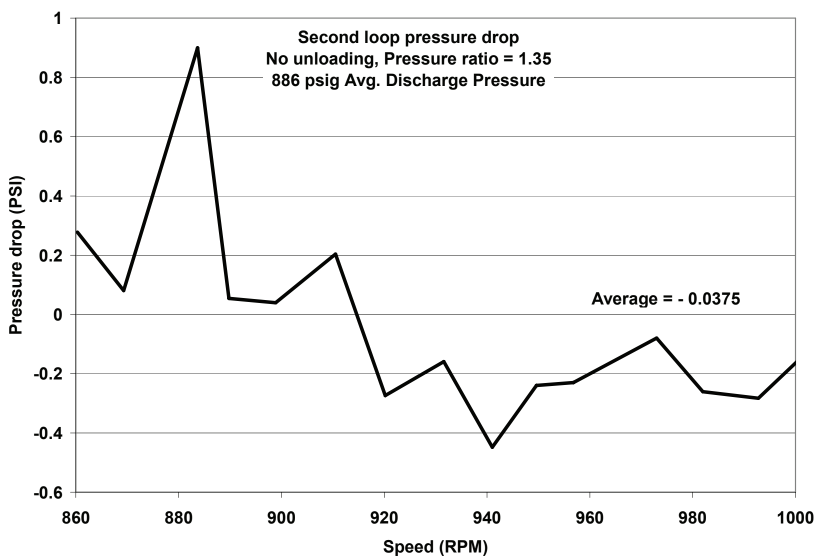

Figure 19: *Measured pressure loss across the 2nd PAN Loop.*

for many of the operating conditions, however the number of pressures that could be read concurrently required that system pressure losses were the only truly simultaneous measurements.

Figure 19 is a representative sample of the measurements obtained for just the 2nd PAN loop in the discharge system. It can be seen that the pressure losses ranged from -0.4 to +0.9 psig across the speed range. With the exception of the single data point at 880 rpm, the pressure loss across the 2nd PAN was close to zero on average. This is comparable to what was consistently predicted in the simulations that preceded the testing. It was not practical to obtain pressure loss measurements across each PAN section at all conditions over the entire speed range, however pressure measurements were obtained across the 2nd PAN and across the entire PAN system, so that the losses in the other sections can be reasonable inferred from the test data. Much more test data was obtained than what can be reasonably presented in this paper, so the content has been limited to a representative sampling to demonstrate the most significant results.

CONCLUSIONS

Over the past three years, a progressive development effort involving computer modeling and simulation, low pressure air compressor system testing and, finally, full-pressure field testing, has confirmed the potential for reducing reciprocating compressor pulsations without the use of traditional attenuation devices that cause excessive pressure losses. Properly designed Pulsation Attenuation Networks (PANs) can reduce pulsations to very low levels, while causing very minimal pressure losses. This is especially important on higher speed (>600 rpm) compressors, where dynamic pressure drop is more of a factor than on low-speed compressors. Pulsation control can be achieved over a reasonable range of operating speeds and operating conditions.

While many best practices have been learned during the developments to date, evolution of the PAN design technology will continue as more applications are encountered. The first field system was not entirely optimized because of space and foot-print constraints, and it is expected that a clean sheet approach on a new unit will provide more opportunity for optimization. Of course, the results presented herein are for the discharge side of the cylinders only. Simulations have predicted similar benefits for the suction side as well, making this a technology with potential for reducing the power required for high-flow, low-ratio pipeline compressor systems by 15% or more in many instances.

Using proven fluid dynamics and compressor cylinder design experience, robust tuning section transitions (TSTs) have been developed to facilitate the reasonable design and packaging of compact PAN systems using standard commercial pipe, fittings and flanges. In these bottle-less systems, the pressure pulsations within the first section of piping from the cylinder flange to the first Y-branch TST can create significant dynamic forces that must be properly predicted and carefully constrained. For sound mechanical design, a complete 3-D model and FEA analysis is recommended for the compressor frame and cylinders, the skid, and the entire piping system in order to provide a well-supported system that operates with acceptable vibration levels at all conditions. This will be especially important when PAN systems are applied to the suction sides of the cylinders. As experience is gained, other configurations may be investigated, such as turning the cylinder suction and discharge connections to the side to keep the suction piping closer to the skid for mounting and support. In any event, it is important to consider each configuration of cylinders and piping as a complete system for optimal tuning of performance and pulsation control.

ACKNOWLEDGEMENTS

The authors wish to acknowledge the El Paso Corporation for providing significant funding and other support that has enabled this research to progress over the past three years. Special thanks are acknowledged to Randy Raymer of El Paso for his encouragement and support for the development and practical adaptation of this technology to the gas compression industry. Finally, the management and owners of Optimum Power Technology and ACI Services Inc. are recognized for their ongoing funding, support and dedication of key technical personnel and other resources to this research.

REFERENCES

1. Brahler, C.; Chatfield, G.; Crandall, J; and Shade, W.; *An Investigation of the Application of Finite Amplitude Wave Simulation with a New Technology for Controlling reciprocating Compressor Pulsations*, 2007 GMRC Gas Machinery Conference – Dallas, TX.
2. Chatfield, G.; Crandall, J.; Shade, W.; and Wells, D.; *Demonstration of Efficient Compressor Control Using Tuned Loop Networks*, 2008 GMRC Gas Machinery Conference – Albuquerque, NM.
3. US Patent Application Serial No. 60/954,914, "Pulsation Attenuation".
4. US Patent Application Serial No. 60/976,075, "Branching Devices For A Pulsation Attenuation Network".
5. Chatfield, G. and Shade, W.; *New Technology for the Efficient Cancellation of Compressor Pulsations*, GMC Journal, February 2009.