THE EFFECT OF COMPRESSOR CYLINDER DESIGN ON NOZZLE PULSATIONS — FIELD TEST RESULTS OF THE DRESSER-RAND DDV PIPELINE CYLINDER

In Conjunction with the GMRC 2008 Pulsation Research, the Effect of Compressor Cylinder Geometry Has Been Investigated to Determine When Additional Cylinder Nozzle Controls are Required for the Reduction of Pulsations

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Recent acoustic modeling has indicated relatively low pulsation levels in nozzles of a Dresser-Rand Dynamic Discharge Valve (DDV) pipeline cylinder, compared to conventional high-speed machines. The DDV pipeline cylinder was designed with larger volumes and effective flow areas in the gas passages than is typical of conventional high-speed pipeline cylinders. The DDV cylinder's continuous inlet gas passage design and straight flow discharge gas passage makes this cylinder unique. A reduction in horsepower losses through the nozzle was also expected due to the improved design for low pulsations.

To correlate model results with actual cylinder nozzle pulsation levels and frequencies, field testing of a recently installed DDV pipeline cylinder on a Dresser-Rand 6HOS4 compressor was performed. The testing determined the actual pulsation levels, nozzle acoustic response frequencies and performance of the compressor. This research provided meaningful information on the attenuation of nozzle pulsations for high-speed machines. In this presentation, the DDV pipeline cylinder pulsation levels and field test conclusions related to the reduction of pulsations in the compressor cylinder nozzle area are provided.

The design of a reciprocating compressor cylinder can be in several forms. For the most common gas compressor cylinder design, the upper half of the cylinder is the suction or inlet half and the lower portion of the cylinder is the discharge or outlet half. Suction and discharge gas passages are nearly identical mirror images of each other, and acoustically, each passage has a finite termination point. The suction and discharge valves are the acoustic termination of a quarter wave response typically called the cylinder nozzle response. Dresser-Rand's DDV cylinder design is not a common cylinder design. The cylinder is not split into two equal and opposite sections. The suction and discharge valves are still the acoustic termination points of the cylinder nozzle response. However, the distance from each suction valve to the compressor bottle is not the same, and the discharge valve is a moving boundary condition.

Here, we describe how the DDV cylinder works and why the acoustic cylinder nozzle response improves the cylinder pulsation levels. Unique acoustic characteristics of the cylinder design will be discussed, particularly regarding how the unique characteristics result in alterations of the pulsations that are typically associated with the cylinder nozzle response/resonance. Descriptions of improved cylinder performance characteristics that were designed into the cylinder are discussed. Field measurements taken in February 2008 regarding cylinder nozzle pulsations and cylinder performance are presented. The DDV pulsation and performance characteristics are unlike most common cylinder designs.

The DDV cylinder looks very similar to a common cylinder design, as illustrated in Figure 1. Internal passages and valves depicted in Figure 2 are not typical of most cylinder designs. Suction gas passageways still begin at the upper cylinder flange, but the suction gas passage extends all the way around the head end and crank end of the cylinder. Extending the suction gas passage in this fashion makes space for the installation of eight suction valves. Four of the valves are on the head end of the cylinder and four of the valves are on the crank end. Discharge gas passageways still exit at the lower cylinder flange, however, the discharge gas passage has been significantly reduced. The double-deck discharge valves and the piston have been combined (see Figures 2, 3 and 4) such that the discharge gas flows through the piston/valve when the piston extends to either end of the cylinder. Gas flowing through the discharge valve is depicted in Figure 4. Discharging gas flows from the discharge valve just a short distance out to the discharge flange and connecting bottle.

The DDV cylinder was designed for low compression ratio service typical of the gas transmission market and was introduced in 2005. The Dynamic Discharge Valve concept was introduced over 10 years ago as the Valve In Piston (VIP) cylinder. The DDV differs from the VIP in that it has multiple radial inlet valves in the barrel whereas the VIP has a single axial inlet valve at each end.

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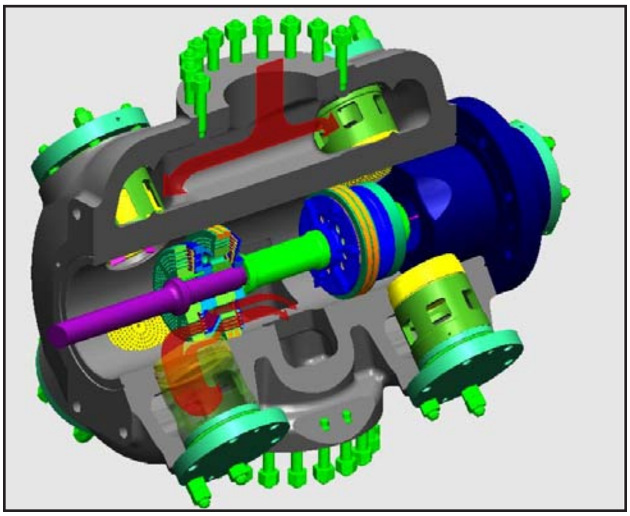
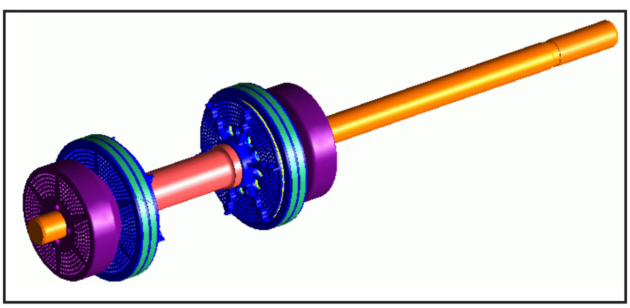
■ Figure 1: The Dynamic Discharge Valve (DDV) pipeline cylinder.

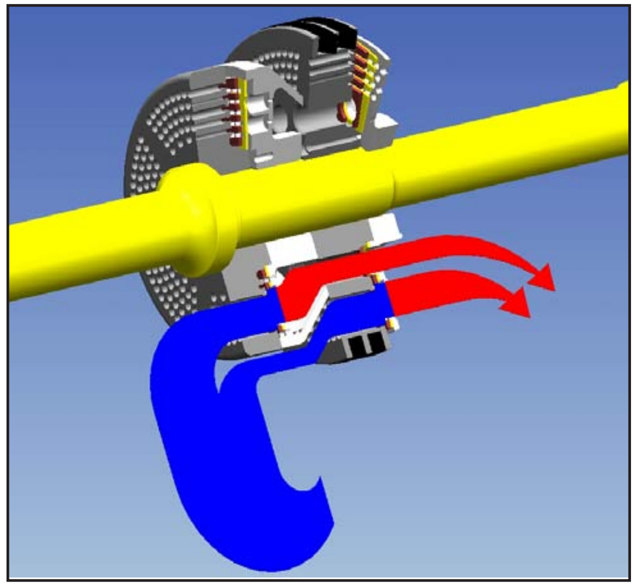


■ Figure 2: Cutaway of the DDV pipeline cylinder.



■ Figure 3: Piston, piston rod and DDVs.

Test Results — Typically, the internal gas passages and cylinder nozzles should be as short as is reasonable such that the cylinder nozzle resonant frequency is coincident with a higher, lower-energy compressor order. The DDV cylinder design does not make the length from the compressor bottle to the suction valves shorter. In fact, the length from the suction bottle to four of the eight suction valves is longer than that of the typical cylinder design. However, having the circular shape of the passageways and the non-uniform lengths to each suction valve creates a poorly defined acoustic end condition. Suction end conditions are no longer well defined because the distance from four of the suction valves to the suction bottle is different than that of the other four suction valves. A pure quarter wave response is difficult to sustain in the uniquely shaped cylinder internals. The additional volume of the internal gas passage is less conducive to a pure quarter wave response. From an efficiency viewpoint, additional suction valves result in additional flow area in each compression chamber.



■ Figure 4: Cutaway of the piston/dynamic double-deck discharge valves.

Design characteristics of the discharge portion of the cylinder are more unique than the suction design characteristics. The cylinder discharge passageways use conventional lengths but the valve location uses a less conventional design, which alters the dynamics (acoustic end condition) significantly. The physical distance from the cylinder discharge valves to the cylinder flange is relatively short; resulting in a high-frequency cylinder nozzle response — assuming the cylinder-to-bottle nozzle is kept reasonably short. Higher natural frequencies are generally preferred (corresponding to less compressor excitation and improved acoustic attenuation). Having the discharge valves in the cylinder piston creates a dynamic and unique end condition. This characteristic, similar to the suction design, results in an environment where it is difficult to sustain a pure quarter wave response such as a cylinder nozzle resonance.

Unique characteristics of the DDV cylinder were measured in a field study of a recent installation (see Figure 5). In the design study for this installation, the acoustic model of the DDV cylinder suggested that cylinder nozzle orifices were not needed to damp the cylinder nozzle resonance, which is typically a high-amplitude response for common cylinder designs. Since the DDV cylinder was relatively new, it was desirable to obtain correlation data for comparison with the prediction data. Pulsation data were obtained for the four-cylinder unit (750 to 1000 rpm speed range) in February 2008.

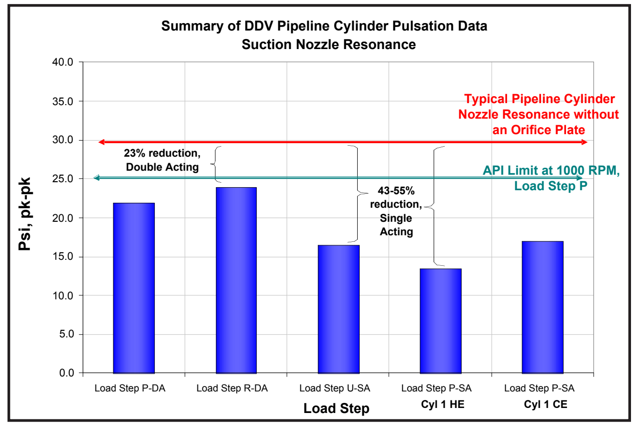
Cylinder valve caps were tapped such that suction system pulsation data could be measured at the cylinder valve, which is the location of the pulsation maximum for a typical quarter wave response. The preferred location available and accessible for measuring the discharge cylinder nozzle resonant pulsations was in the discharge cylinder nozzle. Note that in Figure 6, the API pulsation limit (API 618 paragraphs 7.9.4.2.5.2.1) applies to the suction cylinder flange, whereas the measurement point is at the valve cover. The pulsation levels at the cylinder flange would be expected to be less than at the valve cover. The discharge measurement point is at the same location (cylinder flange) as specified for the API 618 limits.

Data taken at each one of those test point locations are depicted in Figures 6 and 7. The suction cylinder valve maximum pulsation amplitude was approximately 23 psi (159 kPa) at 52 Hz during the double-acting operation of the cylinder. The discharge cylinder nozzle maximum pulsation amplitude was approximately 12 psi (83 kPa) at 94 Hz during the single-acting operation of the cylinder. Maximum cylinder nozzle pulsation amplitudes taken for various operating conditions and loading cases are summarized in Figures 8 and 9. Suction cylinder valve maximum pulsation amplitudes ranged from 13 to 24 psi (90 to 166 kPa), depending on the load case. Discharge cylinder nozzle maximum pulsation amplitudes ranged from 5 to 12 psi (35 to 83 kPa).

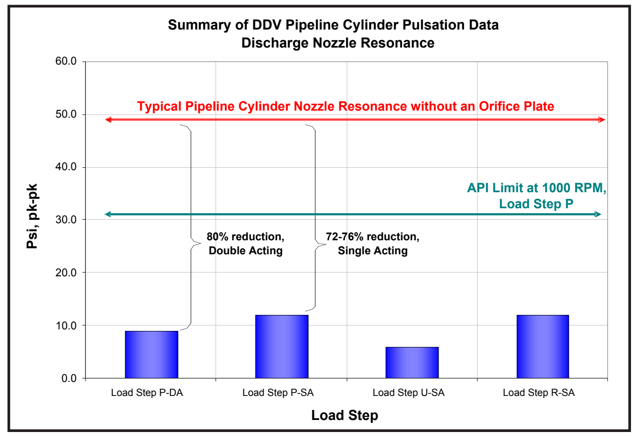
As noted in Figures 8 and 9, these amplitudes are significantly lower than those that are typically observed when testing more common cylinders. Discharge cylinder nozzle pulsations were 72 to 80% lower than that associated with a common cylinder design. Overall, the pulsations associated with the DDV cylinder design and appropriately designed compressor bottles and nozzles are lower than that of a common cylinder design. Unique cylinder characteristics resulted in reduced cylinder nozzle pulsations and improved cylinder efficiency.

During the field study, cylinder PV cards were collected to determine the performance of the unit. Pressures and temperatures were taken on the suction, discharge and laterals to determine the overall compressor performance. PV cards were also collected to determine the horsepower and efficiency of selected cylinders. Data were collected on Cylinders 1 and 4 for different load steps. The majority of data and tests were performed at load step P. For this load step, Cylinder 4 remains in double-acting mode while Cylinders 1, 2 and 3 operate in single-acting mode. The PV card based indicated horsepower for Cylinders 1 and 4 are summarized in the tables below. However, the PV card efficiencies are not consistent because of the presence of a large amplitude channel resonance in measurement connection to the compressor cylinder. A channel correction was used to analyze these data but the efficiency results near the 800 rpm resonance and the nearby low-speed conditions are scattered by as much as 5 to 6%. The scatter in efficiency caused by this measurement channel resonance is such that PV card efficiencies do not

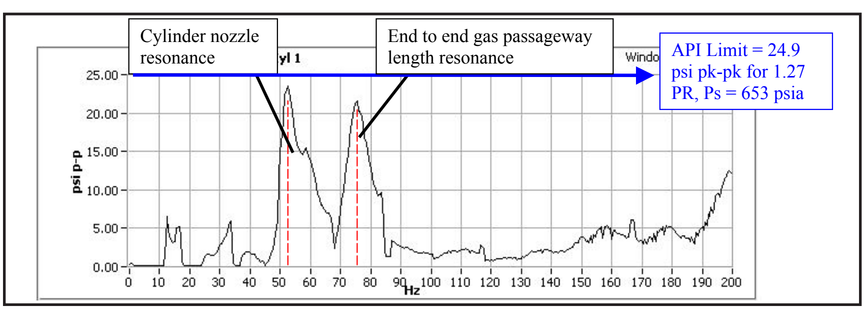
■ Figure 5: Field site for pulsation and performance testing of DDV cylinder.



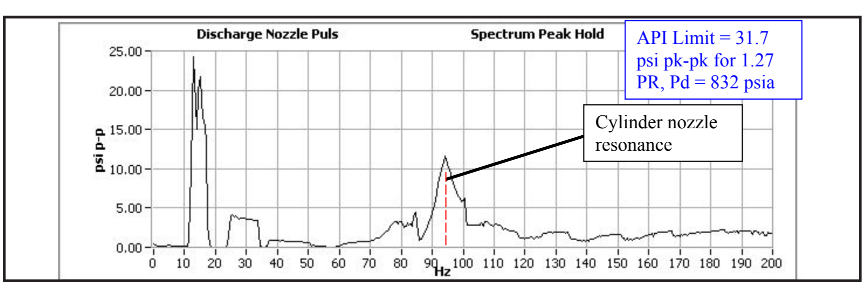
■ Figure 8: Suction pulsation field data summary — DDV pulsations as compared with that of typical cylinder.



■ Figure 9: Discharge pulsation field data summary — DDV pulsations as compared with that of typical cylinder.



■ Figure 6: Suction pulsation field data taken at the crank-end valve cap of a double-acting cylinder.



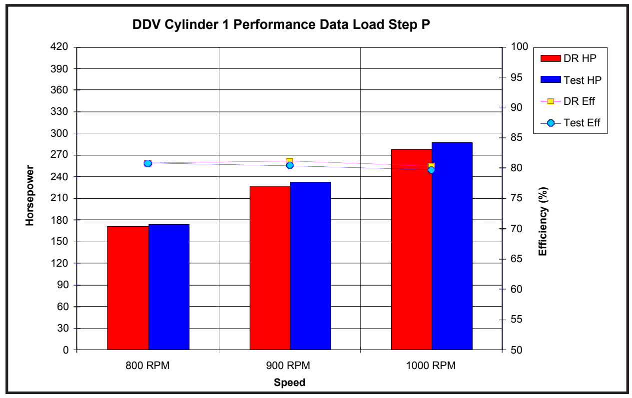
■ Figure 7: Discharge pulsation field data taken at the cylinder nozzle of a single-acting cylinder.

Cylinder 1 - Load Step P					
Speed (RPM)	Ps (psig)	Pd (psig)	Field Test Power (hp)	Test Efficiency (%)	Loading Condition (SA = Single Act, DA = Double Act)
800	642.6	766.3	174.3	80.8	SA
850	636.2	774.1	199.3	80.4	SA
900	632.0	785.2	233.3	80.4	SA
950	629.8	789.4	262.8	78.6	SA
1000	624.8	797.4	287.5	79.7	SA

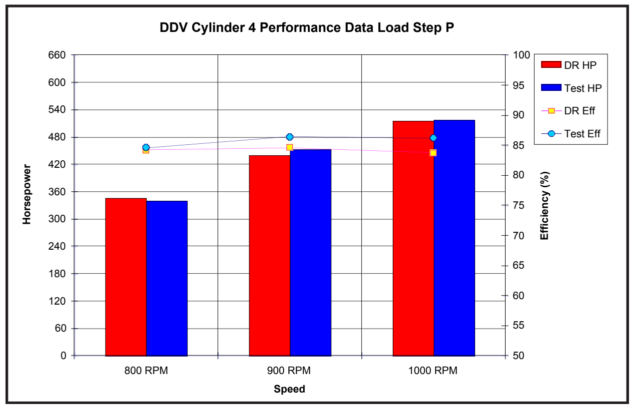
■ Table 1: Test data for Cylinder 1 with field-test efficiency.

Cylinder 4 - Load Step P					
Speed (RPM)	Ps (psig)	Pd (psig)	Field Test Power (hp)	Test Efficiency (%)	Loading Condition (SA = Single Act, DA = Double Act)
800	659.9	802.1	340.3	84.6	DA
850	654.4	812.6	390.4	87.9	DA
900	650.1	825.2	451.9	86.4	DA
950	648.3	829.9	477.4	86.2	DA
1000	646.1	836.5	516.6	86.2	DA

■ Table 2: Test data for Cylinder 4 with field-test efficiency.



■ Figure 10: Cylinder 1 horsepower (bars) and efficiency (line) results from D-R performance prediction and field-test PV cards.



■ Figure 11: Cylinder 4 horsepower (bars) and efficiency (line) results from D-R performance prediction and field-test PV cards.

provide an accurate indication of cylinder performance. This measurement channel resonance has nothing to do with the pulsation level in the cylinder or the design of the cylinder but only with the test transducer connection to the cylinder.

The PV card efficiency is the ratio of the ideal card area (a part of the PV card area) to the total area of the card. This ratio is sensitive to the distortion of the card from the measurement channel (not nozzle) pulsations. The PV card efficiency is not consistent with speed or pressure ratio.

Because the PV card efficiencies cannot be used in this case, an alternate procedure for evaluating cylinder efficiency has been used. The heat rate of the cylinder is determined by comparing the total indicated power (from the PV card) to the work done by the cylinder, that is, the mass flow times the operating pressure ratio. The total indicated cylinder power is affected somewhat by the measurement channel resonance but is less sensitive to the channel pulsation than the PV card efficiency.

The indicated power (total area) of the PV card shows a consistent result with speed and load of the compressor. Capacity data, as well as the total indicated power, is consistent such that heat rate (total indicated power divided by power delivered) shows a consistent trend with respect to compressor speed. The efficiencies for Cylinder 1 and 4 shown in Tables 1 and 2 are the inverse of the heat rate calculated as discussed and represent the mass flow times pressure ratio work delivered, divided by the total indicated power (PV card area) of the cylinder.

These tabularized data from the field testing (as provided in Tables 1 and 2) is compared to the thermodynamic enthalpy rise data calculated by Dresser-Rand in Figures 10 and 11. The calculated PV card-derived horsepower and efficiencies were taken from suction nozzle to discharge nozzle, which includes valve losses. In this case it appears that the valve loss horsepower is low and the difference between the cylinder power (D-R hp) and the nozzle-based indicated power (PV hp) is small to within the measurement uncertainty. The efficiencies calculated from the Dresser-Rand performance program using total cylinder power (D-R Eff) and the PV card-derived efficiencies using measured field data (PV Eff) are in close agreement. Within an experimental uncertainty of a few percent, the horsepower and efficiencies from the PV card-derived method and the cylinder performance predictions show very good agreement.

The collected field data and the calculated enthalpy rise data based on nozzle pressure and temperature are similar, and the predicted values correspond fairly well to the field data. The small difference between the predicted results and the field results provides confidence in designing an accurate DDV cylinder system using the available software.

Summary Points

The DDV is a unique cylinder design that features a fully circumferential inlet valve chest and discharge valve as the piston. This geometry allows for much greater valve area than conventional cylinder designs. The greater valve area and generous inlet valve passageway area minimizes valve losses, which is a major component of compression inefficiency on low compression ratio applications. These low valve losses are reflected in the measured compression efficiency of 78 to 81% for single-acting operation and 84 to 88% for double-acting operation. These are improved efficiencies (over conventional high-speed compressors) running at a compression ratio of 1.2 to 1.25 and 1000 rpm.

By featuring a large inlet passage volume in conjunction with the nondefined end condition, this cylinder design (in most cases) achieves the API cylinder flange pulsation limits without the use of an inlet orifice plate, which is typically required on a conventional cylinder. Similarly, on the discharge side the very short passage and dynamic valve result in low pulsation levels that do not require a flange orifice plate, which again is typically required on a conventional cylinder.

Cylinder flange orifice plates add a significant amount of loss on low-ratio applications. For the Cylinder 4 load step P test at 1000 rpm, the measured horsepower is 516.6 hp (385.2 kW). If typical flange orifice plates had been used (with a pressure drop of 0.5%), the bhp/MMscfd would increase by approx 3 to 4% at the same compression ratio. The resulting efficiencies for the DDV pipeline cylinder represent the improvement gained through not requiring the flange orifice plate on either the suction or discharge nozzle. ■