THE EFFECT OF COMPRESSOR CYLINDER DESIGN ON NOZZLE PULSATIONS:

FIELD TEST RESULTS OF THE DRESSER-RAND DDV PIPELINE CYLINDER

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Introduction

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 Dynamic Discharge Valve (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.

This paper will 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 will be discussed. Field measurements taken in February 2008 regarding cylinder nozzle pulsations and cylinder performance will be presented. The DDV pulsation and performance characteristics are unlike most common cylinder designs.

Figure 1. The DDV Pipeline Cylinder

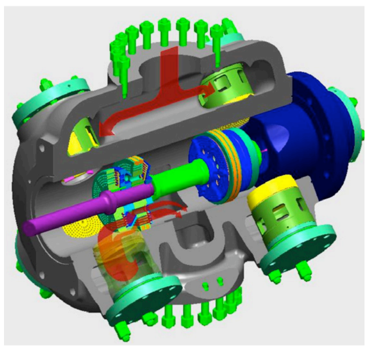

Figure 2. Cutaway of the DDV Pipeline Cylinder

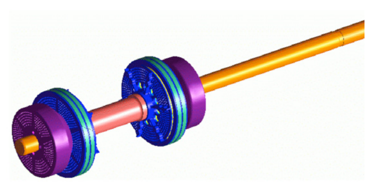

Figure 3. Piston, Piston Rod, and Dynamic Discharge Valves

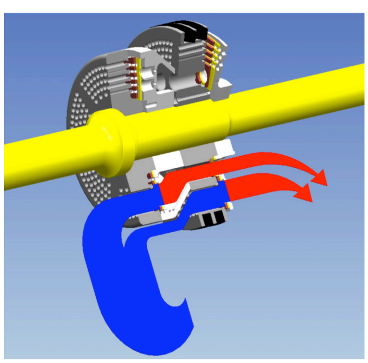

Figure 4. Cutaway of the Piston/Dynamic Double Deck Discharge Valves

As illustrated with Figure 1, the DDV cylinder looks very similar to a common cylinder design. 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 of the cylinder. Discharge gas passageways still exit at the lower cylinder flange, but 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 with 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 ten years ago as the VIP (Valve In Piston) cylinder. The DDV differs from the VIP in that is has multiple radial inlet valves in the barrel whereas the VIP has a single axial inlet valve at each end.

Test Results

Typically, it is desired to make the internal gas passages and cylinder nozzles 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 passage ways 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. It is difficult for a pure quarter wave response to be sustained 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.

Design characteristics of the discharge portion of the cylinder are more unique than the suction design characteristics. The cylinder discharge passageways utilize conventional lengths but the valve location utilizes 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

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Figure 5. Field Site for Pulsation and Performance Testing of DDV Cylinder

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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 was obtained for the 4-cylinder unit (750-1000 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 para 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 is depicted in Figures 6 and 7. The suction cylinder valve maximum pulsation amplitude was approximately 23 psi at 52 Hz during the double acting operation of the cylinder. The discharge cylinder nozzle maximum pulsation amplitude was approximately 12 psi 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, depending on the load case. Discharge cylinder nozzle maximum pulsation amplitudes ranged from 5 to 12 psi.

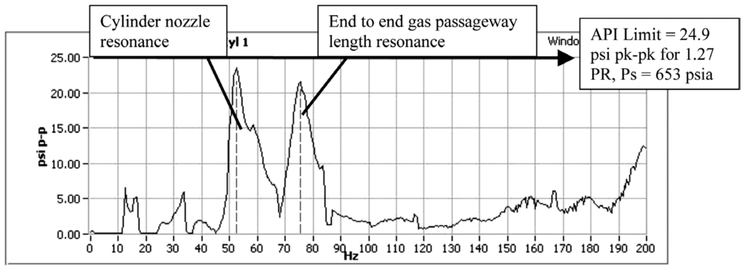

Figure 6. Suction Pulsation Field Data Taken at the Crank End Valve Cap of a Double Acting Cylinder

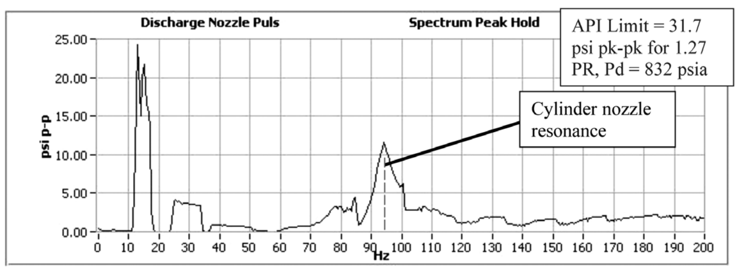

Figure 7. Discharge Pulsation Field Data Taken at the Cylinder Nozzle of a Single Acting Cylinder

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 percent 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 was collected on cylinders 1 and 4 for different load steps. The majority of the 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 due to the presence of a large amplitude channel resonance in measurement connection to the compressor cylinder. A channel correction was used to analyze this 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 percent. The scatter in efficiency due to this measurement channel resonance is such that PV card efficiencies do not 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.

Since the PV card efficiencies can not be used in this case, an alternate procedure for evaluating cylinder efficiency has been utilized. 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

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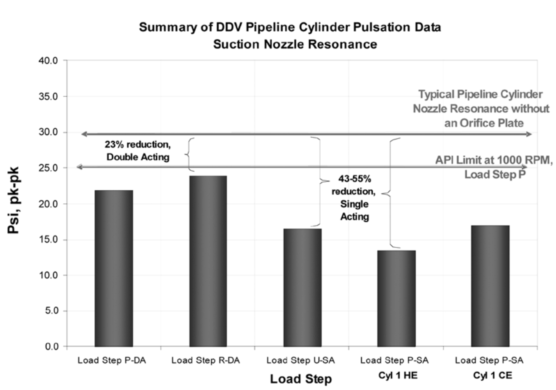

Figure 8. Suction Pulsation Field Data Summary – DDV Pulsations as Compared with that of Typical Cylinder

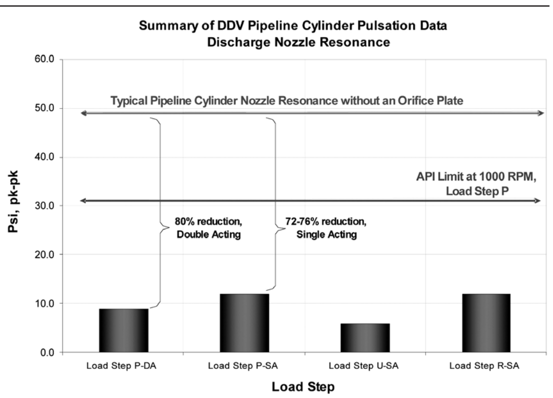

Figure 9. Discharge Pulsation Field Data Summary – DDV Pulsations as Compared with that of Typical Cylinder

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with speed and load of the compressor. Capacity data and 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 1-2 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.

This 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 9a and 10. 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 (DR 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 (DR 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.

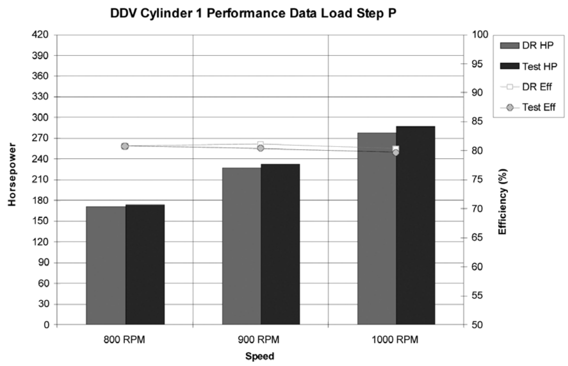

Figure 9a. Cylinder 1 Horsepower (bars) and Efficiency (line) Results from DR Performance Prediction and Field Test PV Cards

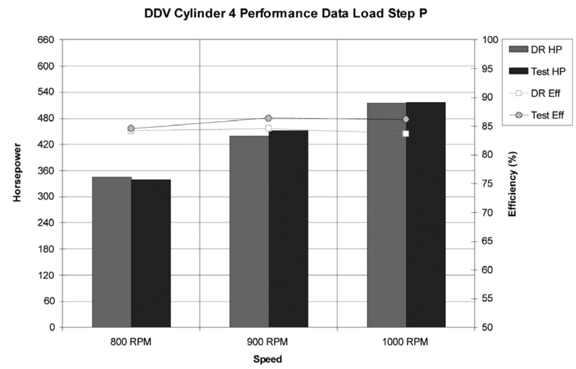

Figure 10. Cylinder 4 Horsepower (bars) and Efficiency (line) Results from DR Performance Prediction and Field Test PV Cards

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

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 Point

The DDV is a unique cylinder design which 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 non-defined 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 and does 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. If typical flange orifice plates had been used (with a pressure drop of 0.5%), the BHP/MMSCFD would increase by approx 3-4% 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.