Evaluation of a Dynamic Variable Orifice for Reciprocating Compressor Pulsation Control

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ABSTRACT

Fixed orifice plates are effective in reducing reciprocating compressor pulsations over a narrow compressor operating range, however there is an associated pressure drop that adds to power consumption required by the compressor. Many applications require operation over a wide range of speed, pressure and flow conditions that may vary from low to very high flows. The system pulsation control design for such applications is almost always a compromise between pulsation control and pressure drop or power penalty.

Although one or more fixed orifice plates having specific diameters may be necessary and effective for pulsation control at one set, or even a range, of operating conditions, they may be ineffective and/or cause unacceptably high pressure drop and associated power consumption at other ranges of operating conditions. Therefore, it would often be advantageous to change one or more fixed orifice plate diameter(s) as operating conditions change. Yet, this is rarely considered as an alternative to extending range or improving efficiency, because it requires system depressurization, gas venting, manual labor and significant downtime.

This paper introduces a new technology, a dynamic variable orifice (DVO), for controlling pulsations and pulsation induced unbalanced forces in reciprocating compressor systems. The DVO can be manually or automatically adjusted while the compressor is pressurized and operating. This allows the orifice size to be adjusted for optimal pulsation control and efficiency as conditions change. Basic mechanical features, installation requirements, and operation of the DVO are described.

Results of two extensive acoustic analysis case studies of reciprocating compressors in typical compression applications are presented. The studies quantify the trade-offs and limitations in pulsation control, pressure drop and power consumption associated with a fixed set of orifices used over wide ranges of required operating conditions. They also quantify the benefits of modifying orifices to more optimal size(s) as operating conditions change.

Results of steady flow and dynamic compressor testing are presented that compare variable DVO equivalent orifice areas, or resistance to flow, with conventional flat plate orifice areas. These data provide validation of the DVO technology and application guidance for using the DVO for pulsation control in reciprocating compressor systems.

BACKGROUND

Common pulsation attenuation elements include pulsation bottles (expansion volumes, often containing internal baffles, multiple chambers and choke tubes), external choke tubes, additional pulsation bottles, and fixed orifice plates installed at specific locations in the both the suction and discharge side of each compressor cylinder. These pulsation attenuation devices can be used singly or in combination to dampen the pressure waves and reduce the resulting forces to acceptable levels. They typically accomplish pulsation attenuation by adding resistance to the system, which causes system pressure drop and energy losses both upstream and downstream of the compressor cylinders. The pressure losses typically increase as the frequency of the pulsation increases, and they add to the work that must be done by the compressor to move gas from the suction line to the discharge line [1].

Fixed orifice plates are one of the most common elements employed for pulsation control. They are relatively easy to install and low cost, often used at multiple locations throughout the system. Fixed orifices are thin metal sheets having a round hole of a specified diameter, located at the center of the pipe cross-section. Most typically, the Beta ratio, defined as the orifice hole diameter divided by the inside diameter of the pipe in which it is installed, is in the range of 0.5 to 0.7 [3]. However, smaller and larger Beta ratios are sometimes used. The orifice plate is retained between two adjacent pipe flanges that are held together with multiple threaded fasteners and sealed with flat gaskets. Once the flanges are installed, the orifice plates remain in place, and can only be removed or changed by stopping the compressor, completely venting all gas to atmospheric pressure, loosening all the flange threaded fasteners, removing the original orifice plates, installing new orifice plates with new gaskets, re-assembling and tightening the threaded fasteners, pressurizing the system with gas and restarting the compressor.

In the majority of applications, compressor operating conditions – speed, suction pressure and temperature, discharge pressure and temperature, displacement, effective clearance volume, and even the gas composition vary with time. Operating condition variations may be gradual over time, but are more often intermittent, changing frequently to higher or lower levels as dictated by the demands of the application. Some applications, e.g., natural gas transmission and gas storage, have extreme variations in operating conditions over time. In fact, the majority of reciprocating applications require operation over a wide speed range as well as multiple flow rates that range from low to very high flows.

Fixed orifice plates are effective in reducing pulsations over a narrow compressor operating range, however they cause an associated pressure drop that adds to the work and power consumption required by the compressor. The system pulsation control design is almost always a compromise between pulsation control and pressure drop or power penalty [2]. For example, a very restrictive (low Beta ratio) fixed orifice plate may be required to adequately dampen pulsations at certain operating conditions. However, at other operating conditions, the pulsations might be acceptable with a less restrictive (larger Beta ratio) fixed orifice plate or possibly with no orifice plate at all. In addition, a fixed orifice plate that controlled pulsations with a tolerable pressure drop and power penalty at some conditions, may cause excessive damping, pressure drop and power penalty at other conditions.

There are therefore multiple challenges when trying to achieve pulsation control with pulsation bottles and fixed orifice plates. A typical disclaimer by a pulsation control designer states that, "Orifice and choke tube diameters are selected to provide the optimum pulsation dampening and pressure drop over the entire operating range of the unit. Typically, the predicted pressure drop levels for the compressor will range from at or below API 618 allowable levels at normal and low flow conditions to above API 618 allowable levels at high flow conditions. Additionally, the pulsation dampening will be generally good at normal and high flow conditions, but may be marginal to poor at certain frequencies when operating at the minimum flow conditions."

Downtime, labor and lost production required for changing fixed orifice plates make this alternative undesirable in most cases. As a result, compressor systems tend to run with higher pressure and power losses or with higher pulsation driven vibration, and associated risk, than would be optimal if the orifice size were changed when dictated by operating conditions. The dynamic variable orifice, which is the subject of this paper, provides a practical alternative for changing orifice size during operation of a compressor.

CASE STUDIES

Two case studies of actual applications were conducted to provide examples of the penalties associated with having fixed orifice diameters as well as the benefits of having variable orifice diameters.

Case Study 1

The first study involved a single-stage, 4-throw, 5.5” stroke compressor with 8.75” diameter cylinders, driven by a 1200 rpm, 1500 HP gas engine in a gas gathering application. Over the life of the application, the suction pressure will vary with time as individual gas wells come on and off line. In addition, the suction pressure will trend to lower levels over longer periods of time as the gas wells mature and production volumes and pressure decline. In order to accommodate the wide range of operating conditions within the rated limits of the compressor and the engine driver, the operating speed, suction pressure, volumetric clearance and number of active compressor ends have to be varied, often by means of automatic controls. This type of application is very common, and the design of an optimal pulsation control system for such an application is often very challenging.

The end user provided a total of 18 different operating conditions that defined the range over which the system was required to operate. A conventional flat plate orifice was placed at the inlet flange on the suction bottle, which was a two-chambered volume-choke-volume acoustic filter connected to two cylinders. Flat plate orifices were also placed at each cylinder suction flange. An identical configuration were used on the opposite side of the compressor for the other two cylinders.

Three-chambered volume-choke-volume bottles were used on the discharge, joining two parallel cylinders on each side of the compressor. Flat plate orifices were placed at each cylinder discharge flange. A flat plate orifice was also placed at the outlet flange on each discharge bottle.

The compressor and piping system was modeled and analyzed over the range of operating conditions to determine the pulsations throughout the system. The analysis results of 3 of the 18 specified operating conditions are presented in Table 1.

- Operating case 1 is a 1200 rpm operating point with all 4 cylinders in double-acting (DA) mode, but with volumetric clearance added to the cylinder head end to reduce the capacity to a rate of 86.5 million standard cubic feet per day (MMSCFD).
- Operating case 3 is a 1084 rpm operating point with 3 cylinders in single-acting (SACE) mode and 1 in DA mode with volumetric clearance added to the head to reduce capacity to a rate of 58.0 MMSCFD.
- Operating case 8 is a 1200 rpm operating point with all 4 cylinders acting in DA mode with no volumetric clearance added, for a maximum capacity of 149.9 MMSCFD.

As is customary, a common set of flat plate orifices was selected for all operating conditions. As shown in Table 1, a common set of orifices is far from optimal as conditions change. The set was selected to provide best overall performance at operating case 1, which is the highest power condition. At this condition, the suction (from the suction header to the compressor cylinder suction flange) and discharge (from the compressor cylinder discharge flange to the discharge header) pressure drops are 1.96% and 1.93% of line pressure, respectively. The suction and discharge pulsations are controlled to 1.9% and 1.3% of the line pressure, respectively, and the associated power consumed by the suction and discharge pressure drops is 2.60%.

A more optimal set of orifices for operating case 1 controls the suction and discharge pulsations to 2.2% and 1.4%, respectively, which would be acceptable for that case based on analysis of the resulting shaking forces. The larger diameter orifices in the optimal set resulted in suction and discharge pressure drops of 1.53% and 0.99%, respectively, with an associated power consumption of 1.69%. This translates to a savings of $7.35 in driver fuel cost per day, based on a fuel cost of $3.50/MMBTU. If the compressor were to operate at this operating condition all the time, with the assumption of the industry norm of 96% availability, use of the optimal orifice set would result in annual fuel savings of $2,575.44. Although the savings is significant, operating case 1 does not demonstrate the primary issue with the use of a common set of orifices.

Operating case 3 shows a different issue that occurs with the use of a common set of fixed pulsation control orifices. It is a low flow condition in which 3 of the 4 cylinders are operated in SACE mode. Power losses with the common set are 1.45%, however the pulsation control is not adequate. Suction and discharge pulsations with the common set are 11.8% and 5.8%, respectively. These are unacceptably high and result in a high risk of pulsation related vibration, meter measurement problems and other safety and reliability problems upstream of, within and downstream of the compressor system. A more optimal set of pulsation control orifices for this case result in suction and discharge pulsations of 7.2% and 5.6%, respectively. Although these pulsations are still higher than would be preferred, they are substantially better than the common orifice set and they represent the best practical alternative for this operating condition without more drastic redesign of the system. The resulting power consumption increases to 2.60%, however that is a reasonable premium for reducing the risk of pulsation related reliability problems. While this particular example is not perfect in all respects, it demonstrates a common issue that occurs with a common set of orifices.

Table 1: Case Study 1 – comparison of 1 common (1 set) and optimal (2 sets) of pulsation control orifices

	Optimal	Common	Optimal	Common	Optimal	Common
Operating Case	1		3		8	
Load Step	Cyl 1 - DA; added HE clearance Cyl 2 - DA; added HE clearance Cyl 3 - DA; added HE clearance Cyl 4 - DA; added HE clearance		Cyl 1 - SACE Cyl 2 - SACE Cyl 3 - SACE Cyl 4 - DA; added HE clearance		Cyl 1 - DA; nominal HE clearance Cyl 2 - DA; nominal HE clearance Cyl 3 - DA; nominal HE clearance Cyl 4 - DA; nominal HE clearance	
Suction Temperature (°F)	62		62		61	
Suction Pressure (psig)	705		735		850	
Discharge Pressure (psig)	981		981		1000	
Speed (rpm)	1200		1084		1200	
Power Required (HP)	1370		784		1245	
Flow Rate (MMSCFD)	86.5		58.0		149.9	
Suction Bottle Inlet Orifice Dia. (in.)	7.44	5.50	4.25	5.50	7.44	5.50
Cylinder Suction Flange Orifice Dia. (in.)	3.75	3.75	3.75	3.75	5.00	3.75
Cylinder Discharge Flange Orifice Dia. (in.)	3.75	3.50	3.75	3.50	5.50	3.50
Discharge Bottle Outlet Orifice Dia. (in.)	5.50	4.25	3.50	4.25	5.50	4.25
Suction Line Pressure Drop (psi)	10.8	13.8	11.6	6.1	11.0	33.6
Suction Line Pressure Drop (%)	1.53%	1.96%	1.58%	0.83%	1.29%	3.95%
Suction Line Power (HP)	13.7	17.4	9.8	5.2	19.2	58.5
Suction Line Pulsation (psi)	15.9	13.7	54.0	88.2	5.7	4.3
Suction Line Pulsation (% of average pressure)	2.2%	1.9%	7.2%	11.8%	0.7%	0.5%
Discharge Line Pressure Drop (psi)	9.7	18.9	15.6	8.9	12.1	52.1
Discharge Line Pressure Drop (%)	0.99%	1.93%	1.59%	0.91%	1.21%	5.21%
Discharge Line Power (HP)	9.4	18.2	10.6	6.2	18.4	79.2
Discharge Line Pulsation (psi)	13.9	13.0	55.7	57.6	1.7	1.7
Discharge Line Pulsation (% of average pressure)	1.4%	1.3%	5.6%	5.8%	0.2%	0.2%
Total Pressure Drop (psi)	20.5	32.7	27.2	15	23.1	85.7
Total Line Power (HP)	23.1	35.6	20.4	11.4	37.6	137.7
% System Power Cost	1.69%	2.60%	2.60%	1.45%	3.02%	11.06%
Daily Fuel Cost @ $3.50/MMBTU	$ 13.58	$ 20.93	$ 12.00	$ 6.70	$ 22.11	$ 80.97
Savings per day	$ 7.35		$ (5.29)		$ 58.86	
Savings per year at 96% utilization	$ 2,575.44		$ (1,854.32)		$ 20,624.12	

Suction Line Pulsation (% of guideline limit)	192%	165%	636%	1005%	84%	64%
Discharge Line Pulsation (% of guideline limit)	127%	118%	509%	527%	46%	43%

Operating case 8 provides an example of another problem associated with using a common set of fixed pulsation control orifices in a compressor that must operate over a wide range of flow conditions. The common orifice set controls suction and discharge pulsations to 0.5% and 0.2%, respectively. This exceptional pulsation control comes with a significant power cost, however, for this low pressure ratio operating case, as the resulting power consumption is 11.06%. A more optimal set of pulsation control orifices for case 8 results in a power consumption of 3.02%. Suction and discharge pulsations remain very low, even with the optimal larger diameter orifice set. The power savings translates to $58.86 in driver fuel cost per day, based on a fuel cost of $3.50/MMBTU. If the compressor were to operate at this operating condition all the time, with the assumption of the industry norm of 96% availability, use of the optimal orifice set would result in annual fuel savings of $20,624.12.

In this case study, the options are limited to: (1) restricting the compressor operation to a limited operating range, i.e., a low flow of about 60 MMSCFD to a high flow of about 80 MMSCFD with the use of the common set of orifices, or (2) to frequently stop the compressor, vent the system to atmospheric pressure, physically unbolt ten sets of flanges to change the orifice plates to sets that are more optimal for the intended operation, reassemble the ten sets of flanges, re-pressurize the system, and restart the compressor. Option (1) could result in flow being limited by as much as 69.9 MMSCFD, or the difference between the desired 149.9 MMSCFD maximum capacity and the 80 MMSCFD limit imposed on the unit due to use of the fixed orifices. Based on a $3.50/MMBTU gas price, this lost production opportunity would be nearly $14,000 per day or more than $5.1 million annually. Option (2) is generally not a practical alternative because of its high cost, its labor intensity, the environmental impact from the more frequent venting of gas from the system to the atmosphere, and the fact that flow conditions are not always predictable or controllable, which could pose a risk to operational safety.

Case Study 2

The second study involved a two-stage, 4-throw, 4.5” stroke compressor with (2) 7.25” diameter cylinders and (2) 5.00” diameter cylinders, driven by a 1200 rpm, 1480 HP gas engine in a gas processing application. The end user provided a total of 13 different operating conditions that defined the wide range over which the system was required to operate.

Out of these cases, 2 were selected to represent the range of typical acoustic results.

- Operating case 1 is a 1200 rpm operating point with all 4 cylinders acting in double-acting mode, but with significant volumetric clearance added to the cylinder head end to reduce the capacity to a rate of 13 MMSCFD.
- Operating case 3 is a 1200 rpm operating point with both 1st-stage cylinders in single-acting (SACE) mode, and both 2nd-stage cylinders in double-acting (DA) mode to reduce the capacity to around 5.0 MMSCFD.

Table 2: Case Study 2 – comparison of 1 common (1 set) and optimal (2 sets) of pulsation control orifices

	Optimal	Common	Optimal	Common
Operating Case	1		3	
Load Step	Cyl 1 - DA; Min HE clearance Cyl 2 - DA; Min HE clearance Cyl 3 - DA; Min HE clearance Cyl 4 - DA; Min HE clearance		Cyl 1 - SACE Cyl 2 - DA; added HE clearance Cyl 3 - SACE Cyl 4 - DA; added HE clearance	
Suction Temperature (°F)	125		128	
Suction Pressure (psig)	400		400	
Discharge Pressure (psig)	2200		2200	
Speed (rpm)	1200		900	
Power Required (HP)	1439		445	
Flow Rate (MMSCFD)	14.82	13.09	4.936	5.005
1st Stage Suction Bottle Inlet Orifice Dia. (in.)	4.20	3.30	3.30	4.20
1st Stage Cylinder Suction Flange Orifice Dia. (in.)	4.20	2.40	2.40	4.20
1st Stage Cylinder Discharge Flange Orifice Dia. (in.)	3.30	2.40	2.40	3.30
1st Stage Discharge Bottle Outlet Orifice Dia. (in.)	4.20	3.30	3.30	4.20
2nd Stage Suction Bottle Inlet Orifice Dia. (in.)	2.80	1.60	1.60	2.80
2nd Stage Cylinder Suction Flange Orifice Dia. (in.)	2.20	1.60	1.60	2.20
2nd Stage Cylinder Discharge Flange Orifice Dia. (in.)	2.20	1.60	1.60	2.20
2nd Stage Discharge Bottle Outlet Orifice Dia. (in.)	2.80	1.60	1.60	2.80
Suction Line Pressure Drop (psi)	8.3	50.6	3.5	0.8
Suction Line Pressure Drop (%)	2.08%	12.66%	0.87%	0.20%
Suction Line Power (HP)	7.1	28.2	1.0	0.2
Suction Line Pulsation (psi)	3.4	3.8	11.8	23.9
Suction Line Pulsation (% of average pressure)	0.81%	0.93%	2.85%	5.77%
Interstage Pressure (psig)	835.0	805.7	934.7	934.5
Interstage Line Pressure Drop (psi)	24.3	62.4	5.9	2.4
Interstage Line Pressure Drop (%)	2.91%	7.74%	0.63%	0.25%
Interstage Line Power (HP)	9.8	24.1	0.5	0.3
Interstage Line Pulsation (psi)	2.9	0.8	16.3	24.7
Interstage Line Pulsation (% of average pressure)	0.34%	0.09%	1.72%	2.60%
Discharge Line Pressure Drop (psi)	5.6	92.5	9.0	1.1
Discharge Line Pressure Drop (%)	0.25%	4.20%	0.41%	0.05%
Discharge Line Power (HP)	1.7	14.2	0.3	0.0
Discharge Line Pulsation (psi)	9.8	5.6	3.3	17.0
Discharge Line Pulsation (% of average pressure)	0.44%	0.25%	0.15%	0.77%
Total Pressure Drop (psi)	38.2	205.5	18.4	4.3
Total Line Power (HP)	18.6	66.5	1.9	0.5
% System Power Cost	1.30%	4.62%	0.42%	0.12%
Daily Fuel Cost @ $3.50/MMBTU	$ 10.96	$ 39.11	$ 1.11	$ 0.31
Savings per day	$ 28.15		$ (0.75)	
Savings per year at 96% utilization	$ 9,862.91		$ (278.35)	
Suction Line Pulsation (% of guideline limit)	114%	55%	206%	440%
Interstage Line Pulsation (% of guideline limit)	63%	10%	189%	287%
Discharge Line Pulsation (% of guideline limit)	100%	29%	34%	91%

UNBALANCED FORCES OVER API ALLOWABLE

As is customary, a common set of flat plate orifices was selected for all operating conditions as tabulated in Table 2. The data in Table 2 shows that, with a common set of orifices, it is not possible to operate at both of these specified conditions. Selecting the set to provide best overall performance at operating case 1, which is the highest power condition, results in unacceptable pulsation levels for operating case 3. On the other hand, selecting a common set that controls pulsations at case 3 results in very high pressure drops that result in a fuel cost penalty of $28.15 per day, or $9,862.91 per year. More importantly, it results in 1.73 MMSCFD less flow from the compressor, a reduction of 11.7%. To a producer, this has a potential economic value of $6055 per day, or more than $2.1 million annually, in lost production.

Both of the foregoing case studies show the potential for operating improvements if there were a practical method of changing orifice sizes as operating conditions change. The rest of this paper describes a new technology that provides such a solution.

DESCRIPTION AND OPERATION OF THE DVO

The dynamic variable orifice (DVO) provides a practical means of changing the effective orifice sizes to optimal values in response to changing compressor operating conditions [4]. It can be adjusted while the compressor is operating and pressurized, and allows the user to increase or decrease the effective Beta ratio.

The DVO consists of a rotatable upper ported plate aligned with a fixed lower ported plate, both having a central round port. As shown in Fig. 1, the central port, which corresponds to the minimum DVO Beta ratio, is similar to a fixed orifice plate. As the upper ported plate is rotated relative to the lower fixed ported plate, additional ports are opened allowing flow to pass in parallel with the central port, increasing the DVO's Beta ratio. The Beta ratio is at a maximum when the ports are fully open. The DVO Beta ratios can be selectively designed from a minimum of typically 0.4 to 0.5, or lower, up to a maximum of 0.7 to 0.8, or higher, if necessary.

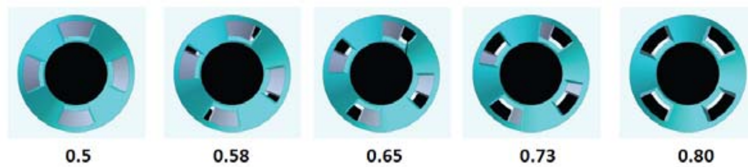

Fig. 1: The effective Beta ratio of the DVO is increased from a minimum of 0.5 [left] to a maximum of 0.8 [right] as the upper plate is rotated relative to the fixed lower plate.

The shapes of the upper and lower ported plates can be flat as shown in Fig. 2A and 2B, conical as shown in Fig. 3A and 3B, or a combination thereof. The upper ported plate is typically rotated (Fig. 3B) in one direction about the lower fixed ported plate to reduce the effective orifice size, and in an opposite direction to increase the effective orifice size.

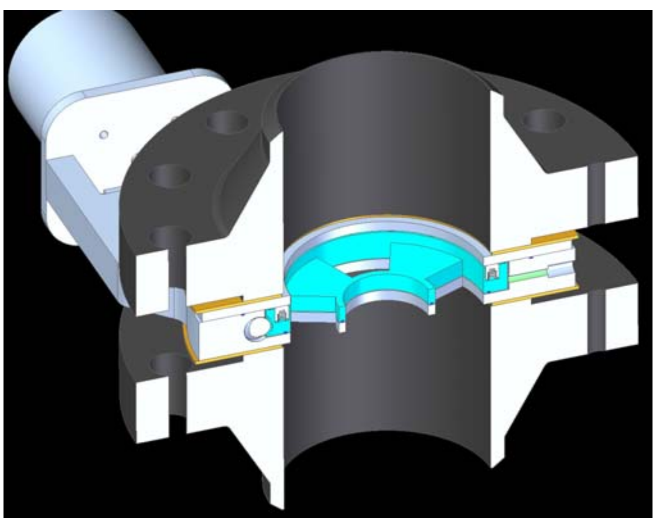

Fig. 2A: Flat DVO showing fixed lower ported plate held between pipe flanges and rotatable upper ported plate.

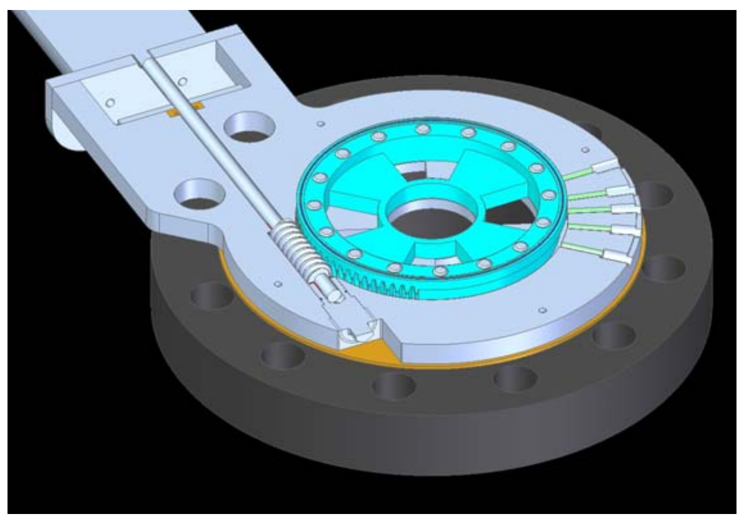

Fig. 2B: A worm gear rotates the upper ported plate to align its ports relative to ports in the fixed lower ported plate to increase or decrease the DVO Beta ratio.

The lower plate has an integral flange that fits between pipe flanges, supports and contains the entire assembly, and provides mounts for an actuator and provisions for control position sensors, if needed. The worm gear can be driven manually using an external handle or wrench or automatically using a small electric or pneumatic actuator.

Selected DVO orifice positions can be locked with a spring loaded detent arrangement (not shown) or sensed and actively controlled via feedback from one or multiple electronic sensors (when a more sophisticated control approach is necessary).

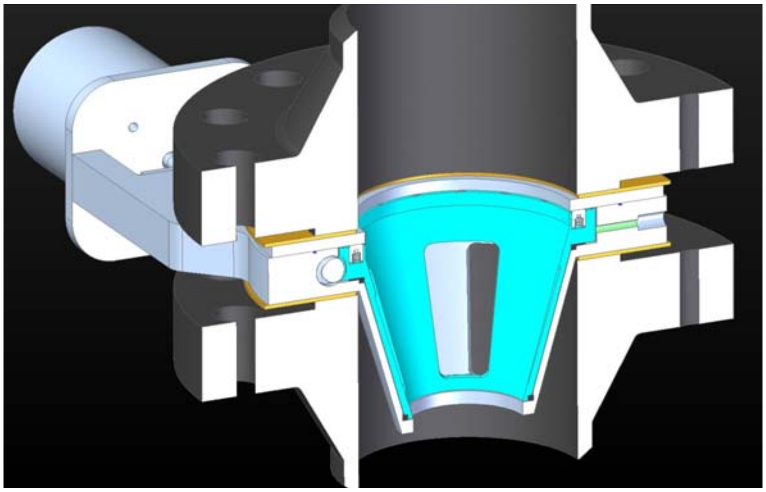

Fig. 3A: Conical DVO showing fixed lower ported plate held between pipe flanges and rotatable upper ported plate.

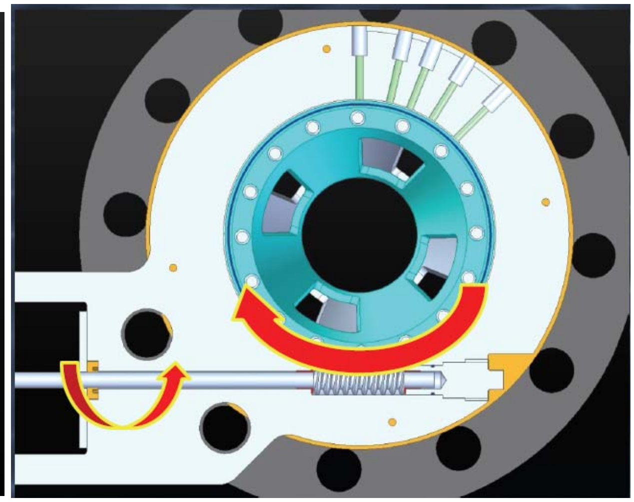

Fig. 3B: A worm gear rotates the upper ported plate to align its ports relative to ports in the fixed lower ported plate to increase or decrease the DVO Beta ratio.

The basic DVO mechanical structure, positioning mechanism and seals are designed for 1500 psig.. DVOs can be designed for essentially any size pipe size and 600# or 900# ANSI flange ratings to provide a range of Beta ratios as may be required within this pressure limit. Higher pressure applications may be possible with special designs.

The positioning mechanism and flange spacing are the same for the flat and conical DVO configurations. The flat DVO can be slipped between properly spaced mating flanges, however its maximum Beta ratio is less than that of the conical DVO. The conical DVO can provide a wider range of Beta ratios, however, it requires removal of a pipe spool piece for installation and removal of the DVO assembly. The shape of the conical DVO typically results in less pressure drop for a given Beta ratio or flow area.

PRODUCTION PROTOTYPE TEST – STEADY FLOW

Due to its unique shape, initial tests focused on measuring the steady flow behavior of the conical DVO. A 4” (pipe ID) plastic conical DVO assembly was manufactured using a 3-D rapid-prototype printer. Flow tests were conducted on a standard flow bench with atmospheric air at 74 to 80°F. The DVO was positioned at Beta ratios ranging from a minimum of 0.5 and a maximum of 0.94. Comparative flow tests were conducted on conventional round hole, flat plate orifices with the same Beta ratios. A few comparative tests were also conducted with a flat DVO model. Fig. 4 is a plot of representative pressure drop data for the conical DVO and the flat DVO compared with conventional flat plate orifices with round holes.

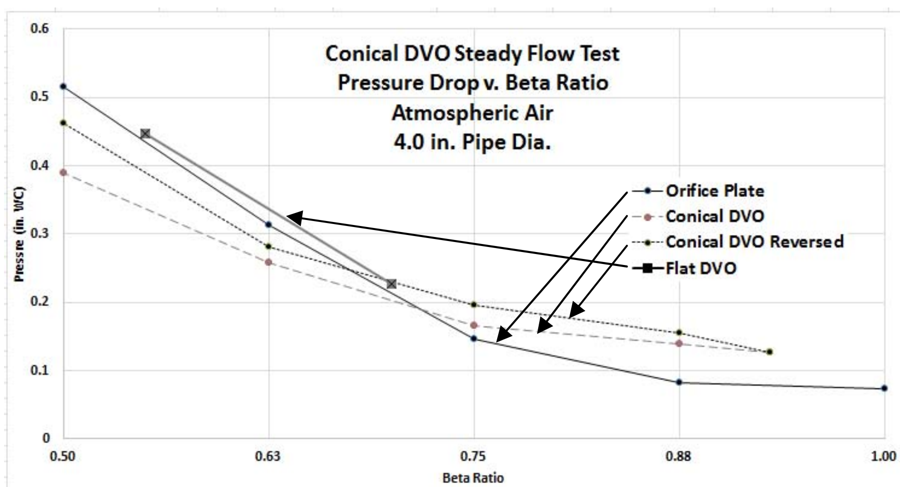

Fig. 4: 4” Conical DVO, flat DVO and standard orifice pressure drop

The conical DVO was tested with flow entering the large end of the cone and then reversed so that flow entered the small end of the cone. As expected, pressure drop is lower with flow entering the large end of the cone. Reversing the flow increases the pressure drop at all Beta ratios, although the difference is insig-nificant as the Beta ratio approaches about 0.9. More interesting is the observation that the DVO has less pressure drop than a normal flat plate orifice at low Beta ratios.

The flat DVO that was tested had a limited range of flow areas, but its pressure drop tracked closely with the conventional flat plate orifice over the range that was tested. Both DVOs and a conventional orifice plate have about the same pressure drop at 0.7 Beta ratio, but the conical DVO has more pressure drop than a conventional orifice plate at higher Beta ratios.

PRODUCTION PROTOTYPE TEST – DYNAMIC FLOW

In order to evaluate the pulsation attenuation of the DVO, tests were run on a closed loop reciprocating test compressor. The compressor is a 4.5” stroke Knight KOA set up as a single-cylinder, single-stage unit with no suction pulsation bottle as shown in Fig. 5. The cylinder has a 10.25 in. bore with one large suction valve and one large discharge valve per end. The unit is powered by an electric motor with VFD that can vary speed from 600 to 1200 rpm.

Fig. 5: Closed loop test compressor showing orifice location

As shown in the schematic in Fig. 6, the loop is set up with various coolers, bypasses and throttle valves that can be used to control operating conditions over a wide range. Flow can be measured with an ASME orifice meter and the unit is equipped with a computerized data acquisition system.

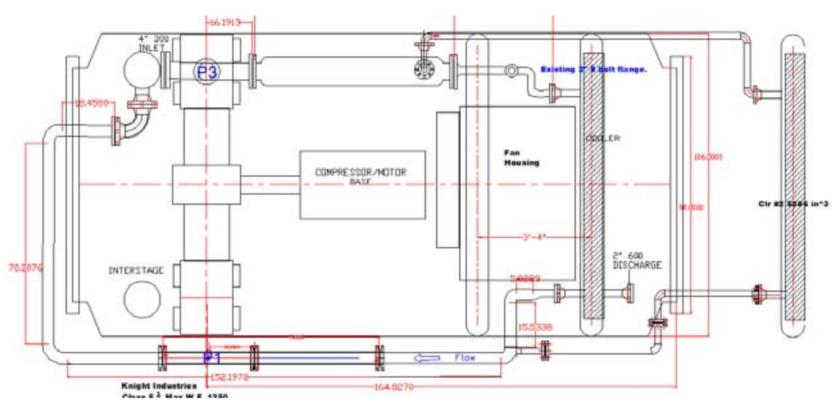

Fig. 6: Schematic of closed loop test compressor showing test cylinder at upper left.

Comparative tests were conducted on a flat version of the DVO at several settings and compared with conventional flat plate orifices having the same Beta ratios. The DVO and the orifice plates were installed sequentially in the 6.07” ID line at the cylinder suction flange. The DVO port geometries are shown in Fig. 7A and 7B for Beta ratios of 0.70 and 0.55, respectively. At the smallest Beta ratio of 0.4, all of the flow goes through the 2.426” diameter center port. Pulsation was measured at the line side flange immediately upstream of the orifice with the compressor operating on 100% nitrogen gas. Tests were run with the cylinder set up as single-acting crank end (SACE) and then double-acting (DA), and the speed was swept through a range from 600 to 1200 rpm.

Operating the compressor in single-acting mode at suction and discharge pressures of 60 psig and 90 psig, respectively, with no orifice in the line, the pulsation was observed to peak at a speed of 1046 rpm. Tests were then repeated six times at the same operating conditions, first with flat orifice plates and then with the flat DVO, each with Beta ratios of 0.70, 0.55 and 0.40. Fig. 8A shows the peak-to-peak (P/P) pulsation frequency response with no orifice installed. Interestingly, even though the cylinder is single-acting, the primary pulsation is at 2x running speed. This is a unique characteristic of the test cylinder, which has a single, large suction valve for a relatively large bore diameter. Most cylinders of this size would have two smaller suction (and discharge) valves. It appears that with the relatively high molecular weight gas, the internal pressure cannot fully unload across the piston face, resulting in significant compression, and high parasitic losses, occurring in the deactivated end. With no orifice, the peak response at this operating condition is 2.96 psi P/P at 34.8 Hz (2x rpm) as shown in fig. 8A. This response is 4.0% of absolute line pressure, and therefore a significant pulsation. There is also a small resonant response at 30.3 Hz and a small pulse at 17.3 Hz, which is 1x running speed.

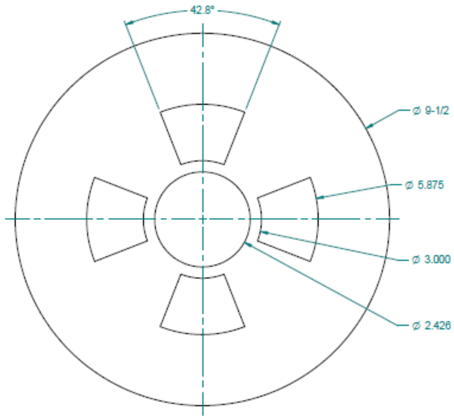

Fig. 7A: DVO port geometry at β = 0.70

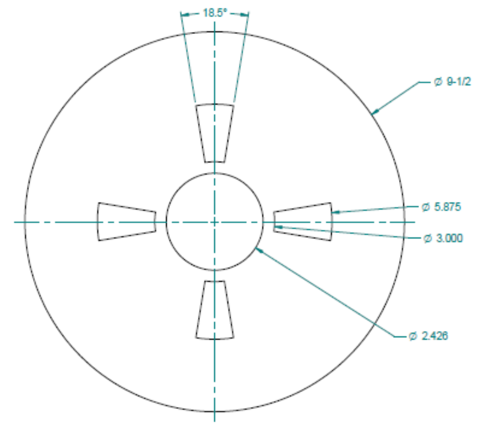

Fig. 7B: DVO port geometry at β = 0.55

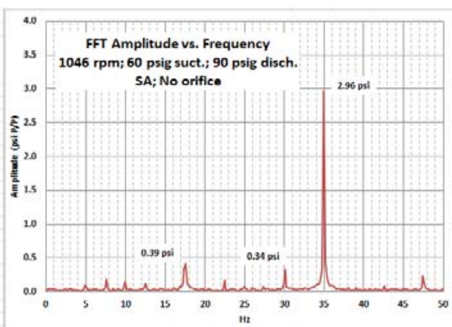

Fig. 8A: SA pulsation – no orifice

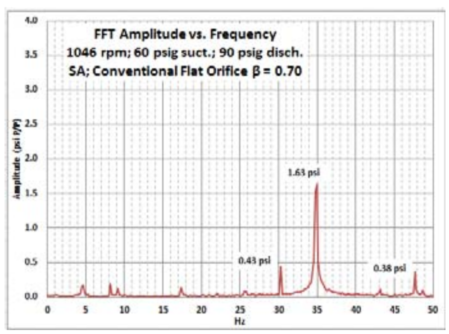

Fig. 8B: SA pulsation – flat orifice β=0.70

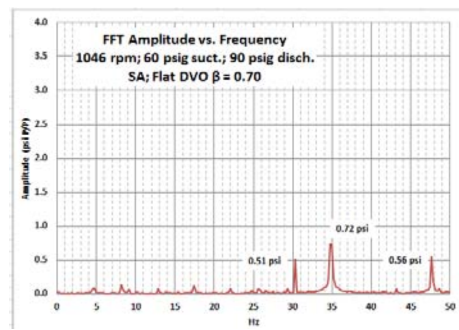

Fig. 8C: SA pulsation – flat DVO β=0.70

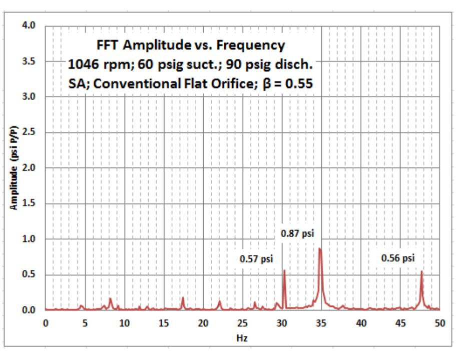

Fig. 8D: SA pulsation – flat orifice β=0.55

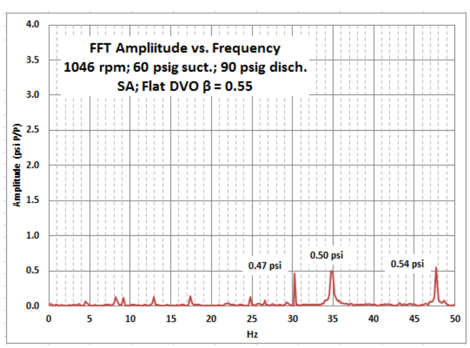

Fig. 8E: SA pulsation – flat DVO β=0.55

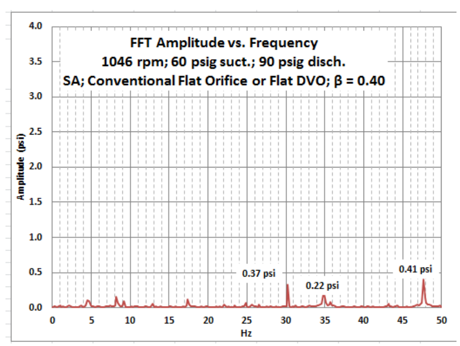

Fig. 8F: SA pulsation – β=0.40

Adding a conventional flat plate orifice with Beta ratio of 0.70 at the suction flange reduces the 2x response to 1.63 psi P/P and the 30.3 Hz resonance to 0.40 psi P/P as shown in Fig. 8B. Fig. 8C shows that the flat DVO with the same 0.70 Beta ratio further reduces the 2X response to 0.72 psi P/P and the 30.3 Hz resonance increases to 0.51 psi P/P. Interestingly, a secondary resonance at about 47.7 Hz also kicks up to 0.38 psi P/P with the flat orifice and to 0.56 psi P/P with the DVO with 0.70 Beta ratio.

Reducing the conventional flat plate orifice Beta ratio to 0.55 reduces the 2X response to 0.87 psi P/P with no significant effect on the 30.3 Hz resonance as shown in Fig. 8D. Fig. 8E shows that the flat DVO with the same 0.55 Beta ratio is much more effective, with the 2x response reduced to 0.50 psi P/P. The 30.3 Hz resonance drops slightly to 0.47 psi P/P and the 47.7 Hz resonance increases slightly to 0.54 psi P/P.

The last single-acting test was with a Beta ratio of 0.4. The geometry was the same for both the flat plate orifice and the flat DVO, i.e., a round 2.426” dia. round hole. Fig. 8F shows that the responses at all frequencies are 0.41 psi P/P or less. Pulsation is controlled to 0.5% of line pressure, but an orifice that small is usually exceptionally restrictive, which would result in large pressure drop and horsepower penalties.

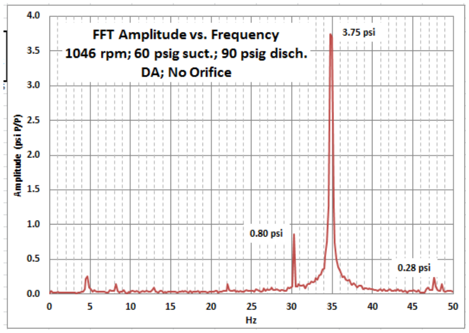

Fig. 9A: DA pulsation – no orifice

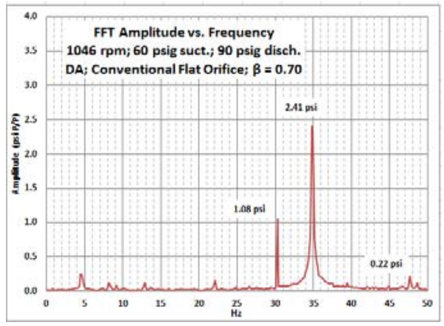

Fig. 9B: DA pulsation – flat orifice β=0.70

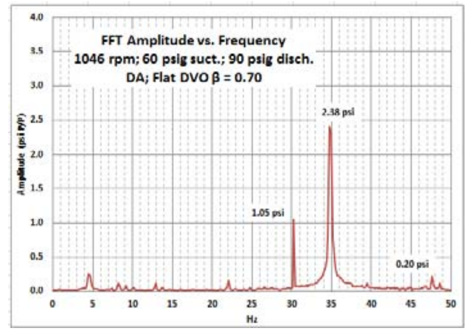

Fig. 9C: DA pulsation – flat DVO β=0.70

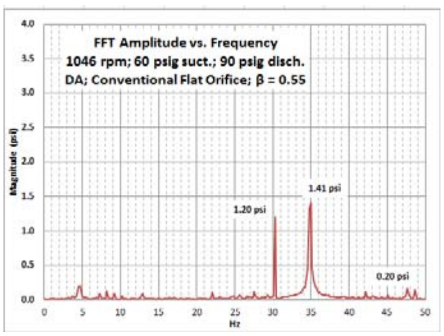

Fig. 9D: DA pulsation – flat orifice β=0.55

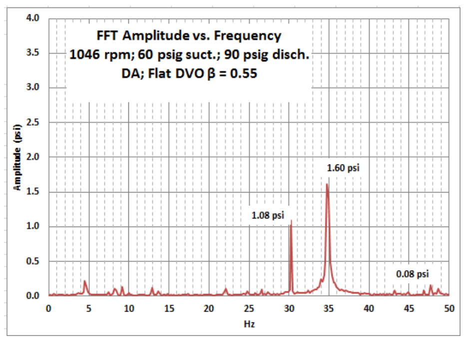

Fig. 9E: DA pulsation – flat DVO β=0.55

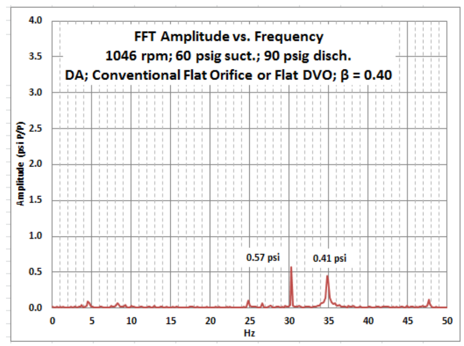

Fig. 9F: DA pulsation – β=0.40

Another battery of tests was conducted with the test compressor operating in double-acting mode. As shown in Fig. 9A through Fig. 9F, the 2x pulsation was much higher in double-acting mode. With no orifice, the measured pulsation was 3.75 psig at 34.8 Hz as shown in Fig. 9A. This is 5% of line pressure. A conventional flat plate orifice with Beta ratio of 0.7 reduces the 2x pulsation to 2.41 psi, but the 30.3 Hz resonance increaase from 0.80 to 1.08 psi as shown in Fig. 9B. The flat plate DVO with Beta ratio of 0.7 shows very similar pulsation control results in Fig. 9C.

Fig. 9D shows the pulsation with a conventional flat plate orifice with a 0.55 Beta ratio. The 2x pulsation is reduced to 1.41 psi, but the 30.3 Hz resonance jumps up to 1.20 psi. Fig. 9E shows the results with the flat DVO with Beta ratio of 0.55. Unlike the single-acting results, in the double-acting tests, the DVO was slightly less effective than the conventional orifice, although both are effective for pulsation control. The 2x pulsation was higher at 1.60 psi, but 30.3 Hz pulsation was 1.08 psi. Fig. 9F shows the results for a Beta ratio of 0.4, with maximum pulsation of 0.57. Pulsation is controlled to 0.76% of line pressure, but an orifice that small is usually exceptionally restrictive.

CONCLUSIONS

A number of conclusions can be drawn from the investigations presented in this paper.

1. Changing the Beta ratio of pulsation dampening orifices as a compressor’s operating conditions change can optimize the trade-offs between pulsation control and pressure drop and power consumption penalty caused

by the orifices. As a result, operating ranges can often be significantly extended, pulsation control can be improved, fuel costs can be reduced and capacity can be increased.

2. The dynamic variable orifice (DVO) provides a practical means of changing the Beta ratio of a pulsation control orifice while a compressor system is pressurized and operating.

3. The multi-port flat DVO has been demonstrated to be very similar to, and sometimes even more effective in reducing pulsation, than a conventional flat plate orifice having the same effective Beta ratio.

4. Steady flow pressure tests show that the conical DVO causes less pressure drop than a conventional flat plate orifice at Beta ratios below about 0.7. At Beta ratios of 0.5, the conical DVO had 26% less pressure drop than a conventional flat plate orifice. This difference declined to the point of being about the same as the conventional orifice at 0.7 Beta ratio. At higher Beta ratios above 0.7, the conical DVO had slightly more pressure drop than a conventional flat plate orifice, for example 3% at 0.75 Beta ratio.

5. Steady flow pressure tests at Beta ratios of 0.55 to 0.70 show that the flat DVO has 3 to 5% more pressure drop than a conventional flat plate orifice. This is typically not a significant difference because the pressure drop penalty is diminished with larger Beta ratio orifices.

6. Operating compressor, dynamic flow, tests show that the pulsation control of the flat DVO is overall very similar to conventional flat plate orifices. At some frequencies and operating conditions, the flat DVO was more effective, and in a few cases slightly less effective. The differences were likely within the accuracy of the measurements.

RECOMMENDATIONS

Upon completion of the current mechanical and performance testing that is partially described in this paper, the DVO will be ready for field introduction in systems operating up to 1500 psig. A schedule of Beta ratio settings can be established by the pulsation analysis consultant during the system design process. The DVO can then be manually or automatically adjusted to change the DVO's effective Beta ratio as operating conditions change. If conditions change slowly and/or infrequently, then manual adjustments may be practical. If conditions change quickly and/or frequently, automatic control via PLC algorithms may be required.

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