MINIMIZING PRESSURE PULSATIONS IN A CENTRIFUGAL COMPRESSOR

Zheji Liu Dresser-Rand Company Olean, NY, USA Mark J. Kuzdzal Dresser-Rand Company Olean, NY, USA

ABSTRACT

As a centrifugal impeller rotates, it converts shaft mechanical power into fluid power - static and dynamic pressure. As a result of this aerodynamic process, the flow in the compressor has steady and unsteady components. The latter is undesirable as it can manifest itself as pressure pulsations that consequently cause machine vibration, noise and alternating stress in the impeller. Typically a centrifugal compressor has strong pressure pulsation amplitude at the blade passing frequency and /or its harmonics. The installation of diffuser vanes to enhance the compressor peak efficiency further increases the magnitude of these pressure pulsations. Dominant pulsations typically occur at the impeller exit and diffuser entrance region, and are a major excitation source to a centrifugal compressor and its associated piping. This internal aeroacoustic energy source propagates pressure waves through various paths in the compressor system and eventually couples to the structure and impeller and causes structural vibration and sound radiation.

This paper focuses on a technique to control pressure pulsations in a centrifugal compressor. Two examples along with test data will be discussed.

INTRODUCTION

The flow delivered by a compressor has an unsteady component that can manifest into various forms of pressure pulsations. Regardless of the compressor type, whether it is a turbo compressor or a positive displacement compressor, it always produces pressure pulsations to some degree as an unavoidable by-product. Severe pressure pulsations can have detrimental effect to the compressor system.

In a positive displacement type compressor, gas is intermittently flowing into and out of a cylinder. Since the discharge ports open only for a small fraction of the duration of each cycle, the flow through the discharge ports is discontinuous. This discontinuous process repeats itself as a cyclic operation. As a result, it causes the flow to pulsate in the intake and discharge lines. Pressure pulsations in the suction and discharge plenums affect the valve response and the mass flow rates through the suction and discharge passages, which, in turn, influence the compressor performance [1-3]. In addition, pressure pulsations can also cause valve fluttering, piping vibration, and noise.

Compared to positive displacement compressors, centrifugal compressors utilize a relatively smooth and continuous compression mechanism. As a result, centrifugal compressors produce much less excitation in the low frequency range. Instead, a centrifugal compressor produces its dominant excitation at a relatively high frequency. Typically, the dominant excitation is associated with the blade passing frequency of the impellers. This relatively high frequency acoustic excitation (often described as pressure pulsations) is mainly a noise and vibration issue rather than a aerodynamic performance issue. Acoustic energy travels in the compressor flow paths and eventually couples to the compressor structures. Consequently, internal pressure pulsations manifest into external vibration and noise radiation. Pressure pulsations are occasionally responsible for causing structural damage to compressor piping system, particularly at pipe fittings, nipples, and instrumentation joints.

Pressure pulsation control is often necessary for positive displacement compressors because of the performance drawback resulted from pressure pulsations. Discharge mufflers are commonly used inside refrigeration compressors. Large suction and discharge "bottles" are essential parts of industrial reciprocating compressors. On the other hand, most centrifugal compressors have been designed without any internal pressure pulsation control device. This paper presents an effective product for reducing pressure pulsations in centrifugal compressors.

PRESSURE PULSATIONS IN TURBO COMPRESSORS

Pressure pulsations generated by a centrifugal compressor have a strong discrete component at the blade passing frequency. This dominant tonal frequency component is generated by the rotor alone and the rotor/stator interaction.

In a compressor stage without inlet guide vanes and diffuser vanes, discrete pressure pulsations are generated by the direct pressure field of the impeller. As the impeller rotates, the steady but circumferentially varying pressure field attached to its blades also spins with it. This rotating pressure field is sensed as a periodic time variation of pressure by a stationary observer. This type of pressure pulsations is generated by the spinning impeller alone. Any distortion of the flow field around the impeller creates additional discrete pressure pulsations.

In a centrifugal compressor with diffuser vanes, the rotor/stator interaction between impeller blades and diffuser vanes is the primary cause of pressure pulsations. Rotor/stator interaction can be further classified into two types of interaction: wake interaction and potential field interaction. As a wake from an impeller blade impinges on a downstream diffuser vane, the velocity deficit in the wake causes the diffuser vane to experience a momentary change in angle of attack due to the variation of the in-flow velocity in the wake. Consequently, a fluctuating load is imposed on the diffuser vane and a pulsating pressure field is generated.

The second type of interaction is caused by the local periodic interference between the potential fields of impeller blades and down stream diffuser vanes. That is, the pressure field of an impeller blade disturbs a diffuser vane and the pressure field of a diffuser vane disturbs an impeller blade. These disturbances cause pressure pulsations with high amplitude at the blade passing frequency and higher harmonics.

Increasing the distance between impeller trailing edges and diffuser vane leading edges can soften the rotor/stator interaction and thereby lower the pressure pulsations. But this can adversely affect the aerodynamic performance of the compressor.

Skewing or leaning diffuser vane leading edges with respect to impeller trailing edge or vice versa is a good engineering practice to weaken the impeller/diffuser interaction and to reduce pressure pulsations. This makes an impeller trailing edge pass by a diffuser vane not instantly but with an extended time period. Consequently, the more gradual passing by of impeller blades relative to the diffuser vanes reduces pressure pulsations generated by rotor/stator interaction.

CONTROL OF PRESSURE PULSATIONS BY DUCT RESONATOR ARRAY

Pressure pulsations are primarily caused by rotor/stator interaction, or impeller/diffuser vanes interaction in a centrifugal compressor. The diffuser region, particularly clearance space between impeller trailing edges and diffuser vane leading edge, becomes an ideal location for pressure pulsation control. The duct resonator array is such a pressure pulsation control device.

In the form of a rugged one-piece configuration, the duct resonator array is manufactured out of a solid plate with acoustic cavities machined from one side and perforated holes drilled on the other side. By setting the duct resonator array inside an annular space machined into an existing diffuser wall, the acoustics cavities are capped by the underlying diffuser wall at one side but are vented to the diffuser channel via the perforated holes at the other side. As the duct resonator array is flush mounted inside the diffuser wall, it becomes integral to the diffuser wall and does not alter the diffuser channel. Because of its one-piece unitary design that requires no bonding or welding, the duct resonator array has high mechanical strength that fits well to serve the severe high pressure and high velocity environment of a centrifugal compressor. This product was first introduced to centrifugal compressors in 2000. The integrity of the design has been demonstrated over the past 10 years in over 200 compressors.

A duct resonator array installed inside compressor flow paths minimizes the compressor internal acoustic energy by reducing pressure pulsation. Because the duct resonator array controls pressure pulsations close to the source, it provides an internal energy relief for the whole compressor system. This not only provides the compressor internal components such as the diffuser vanes with a smoother operating environment, but also propagates less acoustic energy to upstream and downstream pipes. Less in-pipe pressure pulsations means less piping vibration and less noise radiation to the ambient. Multiple field tests confirms that the duct resonator array can achieve noise reduction of 10 to 20 dBA [5-7] and piping vibration level reduction of 90 percent. In addition, a properly designed duct resonator array does not create equipment accessibility issues, pressure loss or reduction in compressor efficiency.

As an engineered product, a duct resonator array is designed to attenuate a specific acoustics source in a frequency range where high amplitude of pressure pulsations, vibration, and noise occurs. Because most centrifugal compressors tend to produce high amplitude of pressure pulsations, vibration, and noise at the blade passing frequency (BPF), most of the duct resonator arrays are tuned or designed to attenuate the dominant energy at this frequency.

Duct resonator arrays can be applied to turbo compressors in two different forms. The first type is a flat plate duct resonator array mounted inside the compressor diffuser. This allows the duct resonator array to be very close to the source and thereby to prevent the pressure pulsations from leaving the compressor diffuser. The closer a control device is installed to the source, the more effective it becomes. For most compressors, this is the preferred approach.

A second type is a pipe duct resonator array. It takes the form of a pipe spool piece that is mounted adjacent to the compressor inlet or discharge nozzle.

TEST COMPRESSOR AND INSTRUMENTATION

A contract compressor, as shown in Figure 1, was utilized as the test vehicle to measure pressure pulsations. It is a single-stage unit designed for the gas transmission service. A 26-inch diameter impeller with 17 blades is mounted on the non-drive end of the shaft to form an overhang compressor. Low Solidity Diffuser (LSD) vanes are installed inside the diffuser to boost the peak efficiency of the compressor. When running at the design speed of 8917 rpm, the compressor has a power input of 9987 horsepower.



Figure 1 - Pipeline Direct Inlet Compressor – Head Removed

The client purchased the compressor with a duct resonator array for noise reduction. The duct resonator array was installed on the hub side of the diffuser while the LSD vane plate was mounted on the opposite side of the diffuser, or the shroud side, as shown in Figure 2. The contract compressor automatically provides one configuration (with array) for the test. By replacing the duct resonator array with a blank filler plate, a second configuration or the baseline build was created for comparative testing.

Six high sensitivity dynamic pressure probes were used to measure pressure pulsations. Three probes were mounted at the diffuser entrance and are circumferentially spaced in one blade pitch. The other three probes are spaced radially outward at a diameter above the LSD trailing edge. All probes are located on the opposite side of the array, so the replacement of the array with the blank filler plate does not interfere with the instrumentation.

The radial position of the first three probes, marked as 130K, 130L, and 130M (Figure 2), are upstream of the duct resonator array. The other three probes marked as 150K, 150L, and 150M are further downstream and face the duct resonator array or the blank filler plate on the opposite side of the diffuser channel.

The compressor was first run with the duct resonator array. Next, the blank filler plate was installed in place of the duct resonator array and the same test procedure was repeated.



Figure 2 - Compressor Layout



Figure 3 - Dynamic Pressure Transducer Locations

TEST CONDITION

The duct resonator is an engineered product that is specially tuned to provide the maximum acoustical performance at the compressor field condition. However, the production test performed in the OEM factory often applies the volume reduction method to "scale" the field condition to a production test condition. The compressor under the production test typically uses a different type of gas at lower pressure and density, consumes lower power, and runs at lower speed. This volume reduction method works well for aerodynamic performance purpose but not necessarily for acoustics. If the product test condition is not acoustically "scaled" properly, the full impact of the duct resonator array may not be seen in the production test.

With the above in consideration, the production test condition was carefully modified and selected so that the full acoustics benefit of the duct resonator array could be demonstrated in such a test. The following compares the major parameters of the production test condition to that of the field condition:

Parameters	Production	Field
Mole Weight	28.01	18.2
Speed (rpm)	7995	8917
Inlet Temperature (F)	100	86
Inlet Pressure (psia)	92	694
Horsepower (hp)	1272	9987



The compressor was run at five different test points on the performance map, as shown in Figure 4, one design point, three points toward the overload side of the curve and one point toward the surge side of the curve. These same test points were first run for the compressor with the duct resonator array and repeated for the compressor with the blank filler plate.

PRESSURE MEASUREMENT DATA

The pressure pulsation signature of the compressor without duct resonator array can be seen in Figures 5-6, which plot the pressure data measured by the 3 upstream probes and the 3 downstream probes, respectively.

The pressure pulsation spectra measured at all six locations have the highest amplitude at the blade passing frequency -2265 Hz. Tonal peaks at the 2nd and the 3rd harmonics are also present but are much lower in amplitude than the BPF. The pressure pulsation signature is highly discrete and tonal. This dominant tonal noise component is recognized as the first target for reduction.

Pressure pulsations measured at the same radius of the diffuser are not the same in magnitude. Although the three probes at each radius are spaced within one blade pitch, the pressure pulsation amplitude varies in the circumferential direction. This spatial variation of pressure pulsations is related to the pressure waveform distributed circumferentially. It makes the measured data sensitive to the location of the probe.

Figure 5 - Diffuser Upstream Pulsations – Point 3

Figure 6 - Diffuser Downstream Pulsations – Point 3

Pressure Pulsation Reduction at Overload Point (Point 3)

Although five different test points were run to compare the pressure pulsations of the compressor with and without the duct resonator array, pressure data measured from three points were selected for discussion. They are points 3, 4 and 5 in the compressor performance map of Figure 4.

Figures 7-12 give the pressure pulsations measured at point 3. In each plot, the red dotted curve represents the pressure pulsation spectrum without duct resonator array while the solid blue curve is the pressure pulsation spectrum with duct resonator array. The peak amplitude at BPF is tabulated in Table 1 for comparison.

Probe	e Location	Amplituc	le at BPF
	(Overload Point)	w/o Array	w/ Array
	130K	0.247	0.514
	130L	1.638	2.233
Upstream	130M	0.790	0.625
	150K	1.285	0.026
	150L	0.622	0.007
Downstream	150M	0.735	0.012

Table 1 – BPF Pressure	Pulsation An	plitude at Pt	. 3
------------------------	--------------	---------------	-----

Comparing the pressure pulsations measured by the 3 upstream probes, as shown in Figures 7-9, indicates that the BPF amplitude of the pressure pulsations increased at two upstream locations (130K and 130L) and decreased at one upstream location (130M) with duct resonator array. As the pressure pulsations also vary circumferentially, it is not conclusive whether this variation is due to the duct resonator array or measurement repeatability in the circumferential direction.

However, it is very clear that the duct resonator made a dramatic impact on reducing the pressure pulsations at the

downstream locations. As compared in Figures 10-12 and the last three rows of Table 1, the BPF pressure pulsation peaks measured by all downstream probes were almost completely eliminated. The duct resonator array has a significant impact on the pressure pulsations where it is installed and downstream.

Pressure Pulsation Reduction at Design Point (Point <u>4</u>)

Pressure pulsations measured at the design point (point 4) are compared in Figures 13-18. Table 2 compares the peak pressure pulsation amplitude at BPF. Similar to the data measured at the overload point, the BPF amplitude of the pressure pulsations increased at two upstream locations (130K and 130L) but decreased at one upstream location (130M) with duct resonator array.

At the downstream locations of the diffuser where the duct resonator array is installed, the pressure pulsations at BPF were completed removed, as seen in Figures 16-18 and the last three rows of Table 2.

Probe Location		Amplitude at BPF	
	(Design Point)	w/o Array	w/ Array
	130K	0.264	0.435
	130L	1.603	2.208
Upstream	130M	0.946	0.258
	150K	1.200	0.030
	150L	0.495	0.016
Downstream	150M	data lost	0.011

Table 2 – BPF Pressure Pulsation Amplitude at Pt. 4

Pressure Pulsation Reduction at Surge Point (Point 5)

Pressure pulsations measured at the surge point (point 5) are compared in Figures 19-24. Table 3 compares the peak pressure pulsation amplitude at BPF. Similar to the data measured at the overload point and the design point, the BPF amplitude of the pressure pulsations increased at two upstream locations (130K and 130L) but decreased at one upstream location (130M) with duct resonator array.

At the downstream locations of the diffuser where duct resonator is installed, the pressure pulsations at BPF were completed removed, as seen in Figures 22-24 and the last three rows of Table 3.

Table 3 – BPF Pressure Pulsation Amplitude at Pt. 5

Prob	e Location	Amplitud	le at BPF
	(Surge Point)	w/o Array	w/ Array
	130K	0.367	0.530
	130L	1.813	2.421
Upstream	130M	0.765	0.427
	150K	1.090	0.030
	150L	0.898	0.012
Downstream	150M	0.905	0.011

Figure 7 - Probe at Location 130K; Overload Pt. 3

Figure 8 - Probe at Location 130L; Overload Pt. 3

Figure 9 - Probe at Location 130M; Overload Pt. 3

Figure 10 - Probe at Location 150K; Overload Pt. 3

Figure 11 - Probe at Location 150L; Overload Pt. 3

Figure 12 - Probe at Location 150M; Overload Pt. 3

Figure 13 - Location 130K; Design Pt. 4

Figure 14 - Location 130L; Design Pt. 4

Figure 15 - Location 130M; Design Pt. 4

Figure 16 - Location 150K; Design Pt. 4

Figure 17 - Location 150L; Design Pt. 4

Figure 18 - Location 150L; Design Pt. 4

Figure 23 - Location 150L; Surge Pt. 5

Figure 24 - Location 150M; Surge Pt. 5

NOISE MEASUREMENT DATA

The effect of duct resonator array on compressor noise reduction has been well documented by numerous internal reports and technical papers [5-7]. Although the purpose of this study was primarily to quantify the pressure pulsation reduction benefit of the duct resonator array, noise data was also measured for reference. For brevity, sound pressure levels measured at just one overload point, with and without the duct resonator array, are included in the paper, as shown in Figure 25. The sound pressure data was recorded at 3 feet from the compressor discharge pipe with a B&K 2260 sound investigator.

With the duct resonator array, the sound pressure amplitude at the blade passing frequency was reduced by 26.5 dB, and the overall sound pressure level was reduced by 15 dB. Noise reduction of such magnitude is typical of the duct resonator array. In addition to reducing pressure pulsations, duct resonator arrays also reduce structure noise and vibration.

Figure 25 - Sound Pressure Levels Measured at 3 feet from Discharge Pipe; Overload

FIELD APPLICATION

A field application of using a duct resonator array to reduce pressure pulsations, piping vibration, and noise is presented next as a supplement to the in-house experimental study. It shows that a duct resonator array can also be installed external of a compressor as an individual pipe spool. As previously discussed, it also demonstrates that the duct resonator array technology works equally well in the field.

High level vibration and noise were experienced from an 8-inch sidestream discharge pipe of a 42 MW compressor installed in a chemical plant [8]. The high level vibration near 480 Hz caused a crack to develop in that process pipe. To correct this structural integrity issue, a duct resonator array in the form of a pipe spool, as depicted in Figure 26, was designed and installed on the compressor sidestream discharge nozzle. The choice of this external pipe array over the internal diffuser duct resonator array was based on the short

turnaround time. The former requires no machine disassembly and re-assembly.

Figure 26 - Sketch of Pipe Array

Field measurements performed before and after the installation of the pipe duct resonator array demonstrated the effectiveness of this solution. As shown in Figure 27, pressure pulsations measured upstream (P1) and downstream (P2) of the pipe array are compared between the measurements with and without the pipe array. The pressure pulsations downstream of the array were completely eliminated. The compressor system has been successfully running in the field since the retrofit.

Figure 27 - Comparison of Pulsation Data Upstream and Downstream of a Pipe Array

CONCLUSIONS

A 16.8 MW overhung centrifugal compressor with a 26inch diameter impeller was instrumented on the OEM's test stand with six dynamic pressure transducers mounted at various locations in the shroud side of the diffuser. The opposite side of the diffuser (hub side) was then configured with two different designs: a traditional solid diffuser wall to simulate a baseline compressor and a diffuser wall with a duct resonator array. The compressor was run under identical conditions with both configurations. Flow rates were varied from overload to surge on a performance map while dynamic pressure at six locations was measured. For the build with a traditional solid diffuser wall, strong pressure pulsations at the blade passing frequency were measured at all six locations. For the build with the duct resonator array installed, the blade passing frequency pressure pulsation was eliminated downstream of the duct resonator array. This confirms that the duct resonator array installed in the diffuser of a centrifugal compressor is an effective device to reduce pressure pulsation. As a result, compressor vibration and noise emission were also reduced. Ambient sound pressure readings for both configurations align well with the measured dynamic pressure data and a 15 dB reduction was measured.

Duct resonator arrays can also be installed in the compressor process pipes to minimize pressure pulsations. A successful field application of a pipe duct resonator array to a 42 MW centrifugal compressor that previously experienced elevated vibration, noise and pressure pulsations from an 8-inch sidestream pipe were presented. Significant pressure pulsation reduction was realized after the introduction of a duct resonator array in the process pipe.

ACKNOWLEDGMENTS

The authors would like to thank Dr. Jose Gilarranz for help in instrumentation, Mr. Rick Antle for help in data acquisition, and Dresser-Rand for permission to publish this paper.

REFERENCES

- Soedel, W., "Gas Pulsations in Compressor and Engine Manifolds," Short Course Text, 1978, Ray W. Herrick Laboratories, Purdue University.
- Liu, Z. and Soedel, W., "Discharge Gas Pulsations in a Variable Speed Compressor," Proceedings of the 1994 International Compressor Engineering Conference, July, 1994, West Lafayette, Indiana
- Liu, Z., and Soedel, W., "Using Gas Dynamic Model to Predict the Supercharging Phenomenon in a Variable Speed Compressor," Proceedings of the 1994 International Compressor Engineering Conference, July, 1994, West Lafayette, Indiana
- 4. Arndt, N. K. E., "Experimental Investigation of Rotor-Stator Interaction in Diffuser Pumps," Ph.D. Dissertation, California Institute of Technology, 1988.
- Liu, Z., Kuzdzal, M. J., "Noise Control of an 11,000 horsepower Single Stage Pipeline Centrifugal Compressor," Proceedings of ASME Turbo Expo 2007, May 8-11, 2007, Montreal, Canada.
- Liu, Z., Jahnke, W., Marczak, M., and Kiteck, P., "Reducing Compressor Station Ambient Noise Level By Controlling Compressor Internal Noise Source,"

Proceedings of the International Pipeline Conference 2002, September 29 - October 3, 2002 Calgary, Alberta, Canada.

- Liu, Z., Kuzdzal, M. J., and Flood, Einar Jr., "Noise Reduction Of A Multistage Export / Reinjection Centrifugal Compressor Through the Use of Duct Resonator Arrays," GMRC 2005, Cincinnati, Ohio.
- Liu, Z., Kuzdzal, M.J., Atkins, K., and Rials, R., "Diagnosis and Solution of High Noise and Vibration Issues After a Propylene Refrigeration Compressor Rerate Proceedings of ASME Turbo Expo 2010: Power for Land, Sea and Air, June 14-18, 2010, Glasgow, UK.