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## **GUIDELINES IN PULSATION STUDIES FOR RECIPROCATING COMPRESSORS**

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### **ABSTRACT**

While new gas compression in pipeline service tends to be dominated by centrifugal machines, reciprocating compressors still have a significant place in the industry.

Specific dynamic design is required to ensure reliable and efficient operation of all reciprocating compressor installations. This requirement is particularly significant in pipeline installations, because the compressor is intended to be in service for many years, and because high efficiency is important for economic reasons.

It is widely recognized that the design of these types of installations should include a "pulsation study". A pulsation study involves analysis of the proposed installation to predict pulsation, vibration, and stress levels. Further, a pulsation vibration control scheme is developed as part of the overall design. The objective is to ensure that predicted pulsation and vibration levels meet guidelines while limiting associated pressure drops and horsepower losses to acceptable levels.

Various guidelines have been used in these studies, but the most commonly used standards are in API 618. While this standard was not originally intended for pipeline service, in reality it represents the best design standard available for high specification reciprocating compressor installations in any application.

Recently, work has been done to upgrade the API 618 design standard. One of the changes in the proposed new 5<sup>th</sup> edition is the addition of unbalanced force guidelines to the existing pressure pulsation guidelines. Much discussion occurred regarding the need for and the advisability of making the addition.

Real examples show designs in which a reduction of pressure pulsation is accompanied by an increase in unbalanced forces, illustrating the need for an unbalanced force guideline. It is also shown that problems can occur due to unbalanced

forces in parts of the piping system not currently addressed by the pulsation guidelines in API 618.

The paper compares the current 4<sup>th</sup> Edition versus the draft 5<sup>th</sup> Edition. Comments are made on the applicability of the various guidelines.

While API 618 is the best available design document, the addition of force guidelines will help API 618 do a better job for industry.

### **INTRODUCTION**

While new gas compression in pipeline service tends to be dominated by centrifugal machines, reciprocating compressors still have a significant place in the industry. Depending on the specific application there are advantages and disadvantages to using each type of compressor. Some of these are briefly discussed in this paper.

For installations where reciprocating compressors are used, detailed dynamic design is usually required to ensure a successful installation. Of the various design requirements discussed in this paper, pulsation control is one area where it may be difficult to find applicable guidelines or standards. The pulsation control section of API (American Petroleum Institute) 618 is most commonly used for these types of studies. While API 618 standard represents the best design standard available for high specification reciprocating compressor installations in any application, it falls short in some areas. Some of the recent work done to upgrade API 618, and other design guidelines not covered in the standard are discussed here.

### **NOMENCLATURE**

Acoustical/acoustics – in this paper refers to the pressure fluctuations in the compressor piping system, not noise.

## **CENTRIFUGAL VS RECIPROCATING**

Some of the advantages of centrifugal compressors over reciprocating are: lower installation cost per brake horsepower, lower maintenance costs, and smaller footprint. Some of the advantages of reciprocating compressors over centrifugal are: ability to cover a wider range of operating parameters, and higher efficiency (especially off of the design point).

## **DESIGN CONSIDERATIONS**

Table 1 lists some of the main design considerations for both centrifugal and reciprocating compressors. As indicated in the table it is our experience that the design considerations for a reciprocating compressor are no more demanding than for a centrifugal compressor. The area of pulsation control and flow disturbances is the area where there is a significant difference in design requirements between centrifugal and reciprocating compressors.

Various guidelines have been used to design pulsation control for reciprocating compressors, but the most commonly used standards are in API 618. While API 618 was not originally intended for pipeline service, in reality it represents the best design standard available for high specification reciprocating compressor installations in any application.

### **API 618**

API 618 is a set of guiding design principles published by the American Petroleum Institute, nominally intended for Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services (formerly Reciprocating Compressors for General Refinery Service). The most recent edition of API 618, the 4<sup>th</sup> Edition published in June 1995, provides a good starting point for designing pulsation vibration control for reciprocating compressor packages. Although this Edition represents the most comprehensive standard available for guidance on pulsation control, the upcoming 5<sup>th</sup> Edition will be an improvement. Below is a summary of the Pulsation Control section of the 4<sup>th</sup> Edition of API 618. Where applicable, clarification is provided and additional guidelines proposed for the 5<sup>th</sup> Edition are discussed.

API 618 suggests three levels of analysis, Design Approaches 1, 2 and 3. Guidelines are given for specifying the appropriate design approach. As well, several specific design criteria are presented in the 4<sup>th</sup> Edition of API 618: initial commercial pulsation bottle sizing, cylinder side pulsation guideline, allowable pressure drop, line side pulsation guidelines and a cyclic stress limit.

### **Design Approach 1 (DA1)**

Design Approach 1 recommends pulsation control design through the use of proprietary and/or empirical techniques. Although the standard states that acoustical modeling is not

required, it states that guideline pulsation levels at the line side of the bottle are to be met. Without modeling an appropriate amount of the piping system the pulsation levels at the line side of the bottle cannot be accurately determined. There are no plans to address this discrepancy in the 5<sup>th</sup> Edition of 618.

### **Design Approach 2 (DA2)**

Design Approach 2 recommends pulsation control design through the use of proven acoustical simulation techniques. API does not define what constitutes a *proven* acoustical analysis technique. The first acoustical analyses were performed in the 1950's using the Passive Analog. Later, Active Analog systems were used. Starting in the early 1970's, Digital Computers were used with "frequency domain" models. Improved computing power now permits use of Time Domain modeling on Digital computers.

For this level of analysis, a mechanical piping system analysis, where piping supports and clamps are recommended, is also specified. Guidelines for cylinder side pulsation, pressure drop and line side pulsation levels apply to this level of analysis.

As part of the acoustical portion of this analysis acoustical unbalanced forces are to be determined and controlled. However, no guidelines are given for unbalanced forces. The lack of definition of what unbalanced force levels are acceptable is one of the most significant shortcomings in the 4<sup>th</sup> Edition. Unbalanced force guidelines are proposed for piping and along the axis of pulsation bottles in the 5<sup>th</sup> Edition. Another significant area that is not addressed in the 4<sup>th</sup> Edition is a guideline of any sort (pulsation or unbalanced force) for the part of the system between the compressor cylinder flanges and the line side pulsation bottles. The pulsation bottle unbalanced force guideline for forces acting along the axis of the bottle, proposed for the 5<sup>th</sup> Edition, partially addresses this area. However, the unbalanced forces acting between the compressor cylinder and pulsation bottle will likely only briefly be mentioned in a note. Case Study 1 emphasizes the need to consider unbalanced forces acting between the compressor cylinder and pulsation bottle.

### **Design Approach 3 (DA3)**

Design Approach 3 starts with the same analysis defined in DA2 and extends the mechanical analysis to detailed finite element modeling and forced response analysis. The focus of a DA3 analysis is to use both acoustical and mechanical methods to arrive at the most efficient and cost effective design. The way the 4<sup>th</sup> Edition is written dictates that extensive analysis is required to optimize the design. It has been well proven that a less rigorous analysis, centered around controlling pulsation and unbalanced force levels and avoiding mechanical resonance, can also result in an optimized design. DA3 proposed for the 5<sup>th</sup> Edition follows a hierarchy approach, whereby if certain

**Table 1: Design Requirements**

<b>Process</b>	<b>Centrifugal Compressors</b>	<b>Reciprocating Compressors</b>
<b>Foundation</b>	Foundation must be solid and strong enough to prevent distortion causing misalignment.	High speed separable units more inherently balanced, thus foundation design less of an issue than for low speed integrals. Design no more demanding than for centrifugal.
<b>Thermal</b>	Compressor connections more sensitive to thermal forces exerted by the piping.	Nature of compressor and piping layout makes thermal concerns more easily dealt with. Thermal design is at worst case no more demanding than for centrifugal.
<b>Efficiency/Control Systems</b>	More complex control system is required to ensure that surge and stone wall are avoided. May be more efficient at specific design point, however, turn down is achieved by either speed or recycle, both of which reduce efficiency.	Not a concern in pipeline service. Ability to cover a wide range of operating parameters without significant efficiency loss but utilizing clearance pockets, stepless type unloaders and valve unloaders. Speed and recycle can also be used.
<b>Rotor Dynamics and Driver Selection</b>	Lateral analysis is required. Torsional analysis is required to ensure torsional natural frequencies are not coincident with one or two time run speed. If a motor drive is selected a gear box is required, making the lateral and torsional analyses more complex. Centrifugal compressor can be direct coupled to a turbine.	Lateral analysis is not required. Torsional analysis, including torsional natural frequencies calculations and a forced response analysis, is required. If a turbine drive is selected, a gear box is required, making the torsional analysis more complex. Reciprocating compressors can be direct coupled to a motor.
<b>Flow Disturbances and Pulsation Control</b>	Flow induced pulsation, pressure fluctuations generated at vane passing frequency, and flow disturbances at the inlet and outlet of the compressor can cause severe vibration problems or loss of efficiency. Flow induced pulsation are potentially generated when gas flows past dead leg tees. A relatively simple analysis can be done to avoid flow induced problems. Pressure pulsation generated at vane passing frequency can excite shell modes of silencers, piping and screens. If any of these modes are excited noise can be a problem, or appendages off of piping can fail. Dealing with these types of problems can be very difficult as the pulsation tends to be three dimensional. Flow disturbances, on the suction side in particular, can adversely affect compressor performance. All of these types of problems tend to be more significant on the newer high efficiency centrifugal compressors. Tighter clearances and generally a less robust design are the main causes.	Pressure pulsation are generated in reciprocating compressors, not just at run speed but at multiples of run speed up to as high as 250 Hz. Pulsation generated at low compression ratio operating conditions, which are typical of pipeline installations, tend to be particularly severe. High pulsation at the compressor valves can adversely affect compressor performance and valve life. Pulsation away from the compressor couples with piping and vessel geometry to produce unbalanced, or shaking, forces. Unbalanced forces must be controlled and the piping and vessels properly supported to avoid high vibration and stresses, which can lead to failures. Through the use of acoustical modeling pulsation control devices can be designed to ensure pulsation and unbalanced forces are controlled and mechanical modeling can be used to ensure adequate supporting. Although theoretically flow-induced pulsation can be generated in a reciprocating installation, these types of problems are rarely seen.

acoustical and mechanical guidelines cannot be met, then more detailed analysis is required.

### **Guidelines for Specifying Design Approach**

The current edition of API 618 provides a chart of rated horsepower versus discharge pressure to define the *appropriate* level of analysis. This chart implies that any compressor over 500 HP should warrant a Design Approach 3 analysis. In our experience the chart is very conservative and narrow in its focus. The modifications for the 5<sup>th</sup> Edition take a less conservative approach and consider horsepower per cylinder rather than total compressor horsepower.

The authors have taken what is outlined in API 618 and 35 years of combined company experience to develop a risk-rating chart, shown in Figure 1. This rating system helps determine the level of design analysis by rating the risk associated with a wide range of factors that are specific to the compressor station. The chart is a useful tool in identifying how much design analysis should be conducted. It takes into account known risk factors, such as horsepower per cylinder and the range of operating conditions. It also weights more subjective factors such as personal experience and how critical the unit is to the process (e.g. How much money do you lose if the unit is not running?).

To fill out the chart, enter a number between 0 and 10 for each risk factor. If any particular factor is viewed as more or less important, enter a value in the "Your rating factor" to weight the factor appropriately. Multiply the entries in the "score" and "importance factor" columns and place the result in the "modified score" column. Add up the Score and Modified Score columns to get the final total and percentage. Use the percent score to determine what level of analysis should be considered.

### **Initial Commercial Pulsation Bottle Sizing**

The intent of the recommended sizing section in API 618 was to allow the purchaser to solicit competitive bids (i.e. suppliers would be quoting based on the same size pulsation bottles). Final pulsation bottle sizes will be as defined by the pulsation study. Occasionally this criterion is misconstrued as an absolute minimum size requirement. Additional wording is proposed for the 5<sup>th</sup> Edition to clarify this point.

Techniques of modeling part of the system to size bottles in the design stage are likely to be part of the 5<sup>th</sup> Edition. It is preferable to avoid this short-cut in the design stage. Design of the bottles with the complete piping system included in the models will minimize the risk of problems.

### **Cylinder Side Pulsation Guideline**

A maximum allowable unfiltered (overall) pressure pulsation level, to be applied at the compressor cylinder flange, is defined. The purpose of this guideline is to protect the

compressor valves and performance from excessive pulsation. This guideline does not, in general, protect the system from excessive unbalanced forces at discrete frequencies. The 5<sup>th</sup> Edition does not directly remedy this deficiency.

### **Allowable Pressure Drop**

A guideline for maximum allowable pressure drop, based on steady flow through the pulsation suppression devices at the manufacturer's rated capacity is given. The wording in the 4<sup>th</sup> Edition allowed for the interpretation of the pressure drop guideline only needing to apply to the manufacturer's guaranteed design point, typically only one point, rather than over the full range of operating conditions. As well the dynamic component of pressure drop is not addressed. For pipeline installations where it is important to minimize pressure drop over the full range of operating conditions, these two points can be significant. Tentatively the 5<sup>th</sup> Edition of API 618 will clarify that the pulsation guideline is to be applied to all conditions. As a minimum, the 5<sup>th</sup> Edition will likely acknowledge the concept of dynamic flow component resulting in higher pressure drop than what is calculated based on just the mean flow. At this time, considering the dynamic component of pressure drop will not likely be a requirement. Depending on the system configuration, the dynamic component of pressure drop can be significant. For pipeline applications in particular, it is in the end user's best interest to specify that the total (static plus dynamic) pressure drop through pulsation suppression devices be considered.

Pressure drop added by the pulsation control devices equates to power loss. The evaluation of losses should not stop at a pressure drop number; the pressure drop should be converted to power. For conditions where the unit is operating at full load, any pressure drop (power loss) will result in decreased capacity. For conditions where the unit is operating below full load any pressure drop (power loss) will result in more power to move the same amount of gas.

### **Line Side Pulsation Guidelines**

Different line side pulsation guidelines are presented for a DA1 versus DA2 and DA3. For DA1, empirical bottle design, the pulsation guideline is basically a percent of line pressure to be met at any frequency. For DA2 and DA3, detailed acoustical analysis, the pulsation guideline is a function of line pressure, line size and frequency. The 4<sup>th</sup> Edition does state that pulsation levels should not be used as the sole criterion for design pulsation control. However, as mentioned previously, the reference to evaluating unbalanced forces is lacking. Case Study 2 discusses a compressor installation where only considering the reduction in pulsation would have lead to insufficient pulsation control from an unbalanced force and vibration point of view. The proposed changes to API 618 will guide people to consider unbalanced forces as well as pulsation.

**Figure 1 - Risk Rating Chart for Reciprocating Compressors**

**Project Description:**

**Entered by:**

**Date:**

	1	2	3	4	5	6	7	8	9	10	Score	Weight Factor	Modified Score	
Service/Gas	Sweet	H <sub>2</sub>				Sour			Heavy, SpG>1			1.0		
Load Steps	1 Step, DA	Few Load Steps			DA & SA		> 50% turndown		Infinite Variation			1.0		
Speed	Fixed	Medium Range			Wide Range Attended		Wide range, remote control. RPM > 1200.				1.0			
HP/cylinder	<150		<300		<500					>1000		1.0		
# Stages # Cylinders *	2 stg 4 cyl	2 stg 6 cyl	# of single nozzle bottles 1 multiple			1 stage 3 cylinders or greater			1stg 2 cyl			1.0		
Compression Ratio / Stage	1.7		1.7 - 3.5		1.4 to 1.7		> 3.5 or < 1.3				1.0			
# units online	1			2		3		4		5+		1.0		
Suction Pressure	Fixed							> 50% change. Very low pressures				1.0		
Discharge Pressure **	Fixed						>2000 psig	> 50 % change >3000 psig	>5000 psig**			1.0		
% of Rated Rod Load				80%			90%					1.0		
Location, Ease of Field Fixes	Convenient. Close to fab shop								Remote Platform			1.0		
Efficiency	Not Critical									Critical		1.0		
API Pulsation Limit	Not Critical									Critical		1.0		
Vibration Limits	Not Critical									Critical		1.0		
Orifice Meter Accuracy	Not Critical									Critical		1.0		
Unit Criticality	Spared	Standby						Plant fails if unit goes down.				1.0		
Your Experience	Proven Design								New, complex design.			1.0		
Project Profile	Low									High		2.0		
<b>Notes:</b>												<b>Total</b>	<b>190</b>	

\* Applies to separable compressors only.

\*\* For system pressures greater than 5000 psig a Level 5 analysis is recommended.

**%**

**Analysis Level Rating Scheme**

<b>% Score</b>	<b>Level</b>	<b>Description</b>
<b>0 - 15</b>	<b>0</b>	Sound Basic Design and Drawing Review
<b>15 - 25</b>	<b>1</b>	Empirical Bottle Design and/or Mechanical Review (API 618 DA 1)
<b>25 - 40</b>	<b>2</b>	Empirical Design and Limited Modeling (API 618 DA2)
<b>40 - 70</b>	<b>3</b>	Comprehensive Mechanical and Acoustical Study (API 618, Studies M.2 through M.5)
<b>70 - 85</b>	<b>4</b>	Level 3 plus More Detailed Mechanical Analysis (API 618 DA3)
<b>85 - 100</b>	<b>5</b>	API 618 Design Approach 3, 4th Ed. plus More Detailed Mechanical Analysis

**Consider a start up check at any level.**

**Cyclic Stress Guideline**

The cyclic stress guideline that will be used in both editions is that the pulsation or mechanically-induced vibration shall not cause cyclic stresses in excess of the endurance limits of the materials used.

One of the concerns with analyses centered around cyclic stresses is that it is difficult to confirm the analysis in the field. Although stresses can be measured in the field it is a very time consuming process. It is much more straight forward to measure vibration levels. Design stage vibration guidelines are proposed for the 5<sup>th</sup> Edition of API 618. Vibration guidelines to be used for field checks will still be lacking in the 5<sup>th</sup> Edition.

**Recommended Practices**

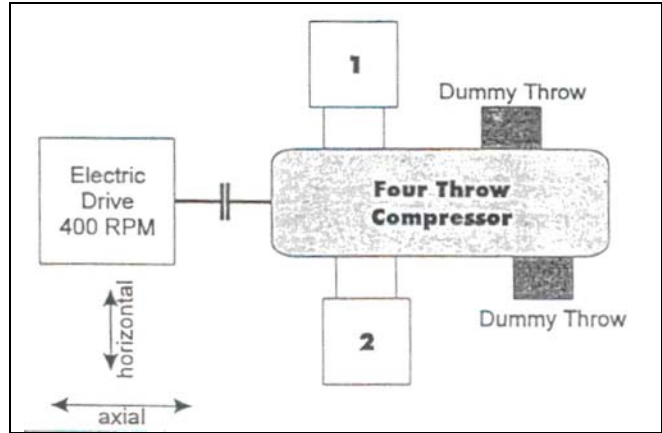
API is planning on publishing a Recommended Practices document which is intended to describe, discuss and clarify the Pulsation and Vibration Control section of API 618, 5<sup>th</sup> Edition. Together API 618, 5<sup>th</sup> Edition, and the Recommended Practice document will provide a good set of guidelines and some background on the fundamentals of pulsation in reciprocating compressors systems.

**CASE STUDY 1 – VERTICAL FORCES BETWEEN THE BOTTLE AND THE CYLINDER**

**The Problem**

Two four throw compressors had been experiencing failures of pistons and rods over some years. On a visit to the site in February, 1994, the following problems were found:

- cylinder supports were loose or broken
- discharge bottle wedge supports were broken
- joints had fretted, shown by red dust (iron oxide) present at most of the joints in the cylinder supports
- anchor bolts and grout under the crosshead guide had broken
- high vibrations were observed on the concrete pedestal under the outboard cylinder support.



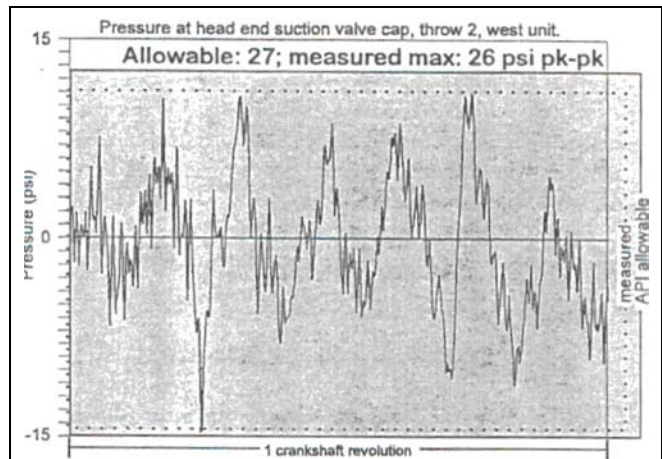
**Figure 2** Rod and piston failures over many years made this unit uneconomic to run.

**Measurements**

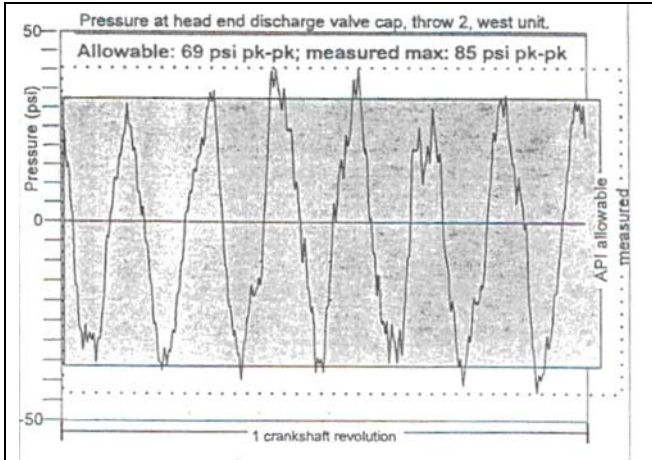
Excessive rod loads were ruled out as a cause of the breakage, because rod loads during the tests were within or at the manufacturer’s guidelines.

Pulsations were measured in the time domain at the suction and discharge valve caps (Fig. 3 & 4).

Suction side pulsations were within the API 618 compressor side guideline (CPL), while discharge side pulsations were about 120% of the guideline.



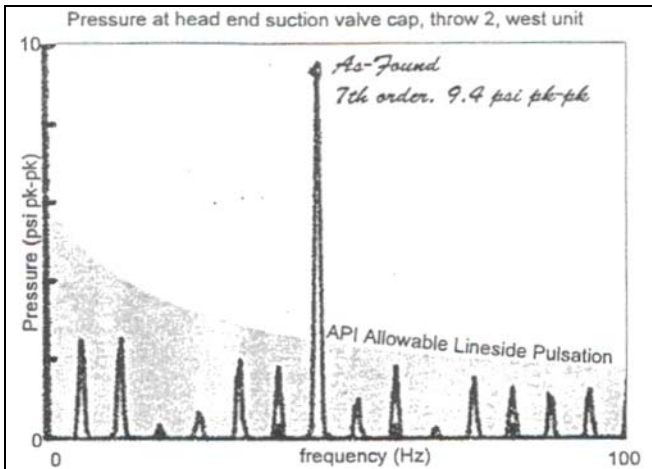
**Figure 3** Suction side overall pulsation was within guideline.



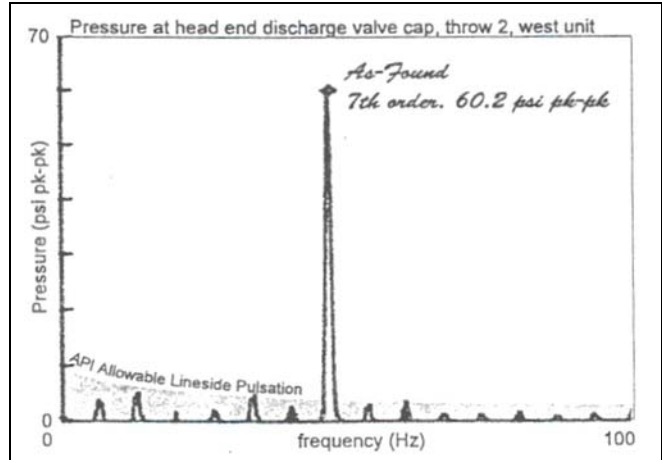
**Figure 4** Discharge side overall pulsation was about 120% of guideline.

A spectrum of the as-found suction valve cap pulsation is shown in Fig. 5, and of the discharge valve cap pulsation in Fig.6.

The API 618 line side pulsation guideline (DPL) is included as a reference in each plot, since there is currently no recognized pulsation limit based on discrete frequencies for the compressor side piping. In both cases, the DPL has been greatly exceeded at seven times crankshaft speed (7X).



**Figure 5** As-found spectrum on the suction side shows high pulsation at 7X shaft speed.

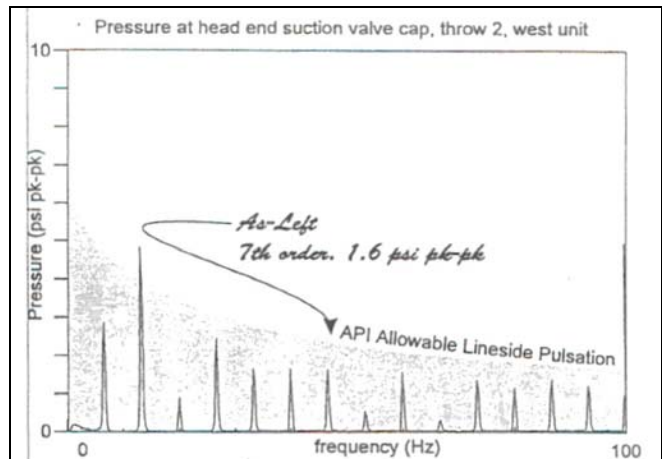


**Figure 6** As-found spectrum on the discharge side shows pulsation at 7X shaft speed is about 15 X DPL.

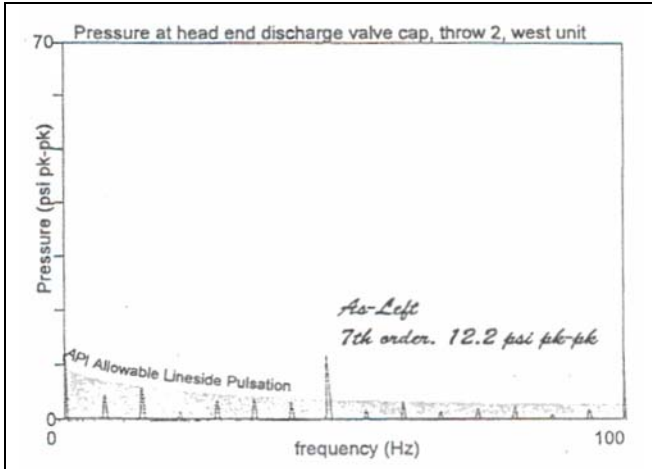
### Modifications to the System

As-left readings (Fig. 7 & 8) were taken after orifice plates were installed in both the suction and discharge nozzles. As shown in Fig. 7, the as-left pulsation at the seventh order on the discharge side still exceeded the line side guideline by about 3X, even though it was only about 20% of the as-found reading.

Three years later, the compressor is still working well.



**Figure 7** The orifice plate significantly reduced pulsation levels.



**Figure 8** The orifice reduced pulsation at 7X to about 20% of as-found; still three times DPL.

**Vertical Forces Between Bottle and Cylinder**

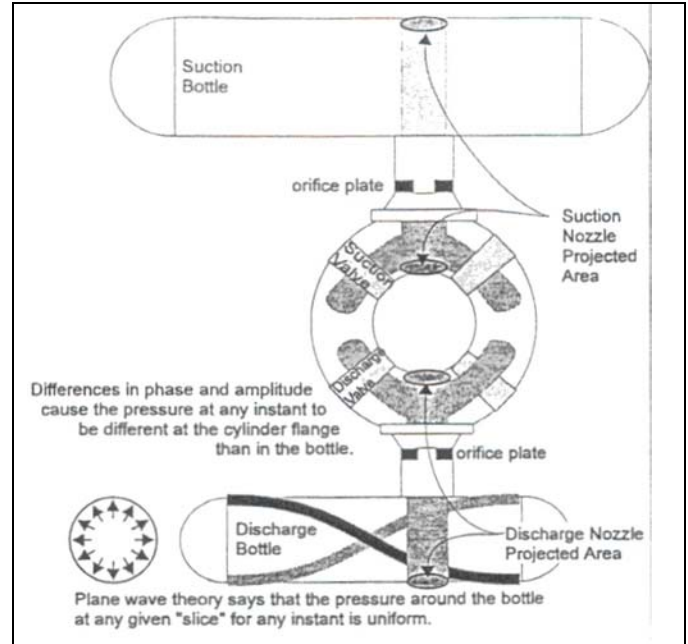
Figure 9 shows a cross-section of the suction and discharge bottles and cylinder on one side of the machine. The standing wave pattern of the pressure pulsation carries from the valves through the cylinder gas passages and the cylinder nozzle into the bottle, on each side of the cylinder. There is no direct connection between the two standing waves, because the suction and discharge valves are never open at the same time.

There is a difference between the amplitude and phase of the pulsation at different points on the standing wave.

Varying pressure acting on an area produces a varying force. In the vertical direction, the areas of concern are the projected areas of the inside diameters of the suction and discharge nozzles (Fig. 9) on the bottle and the cylinder passages.

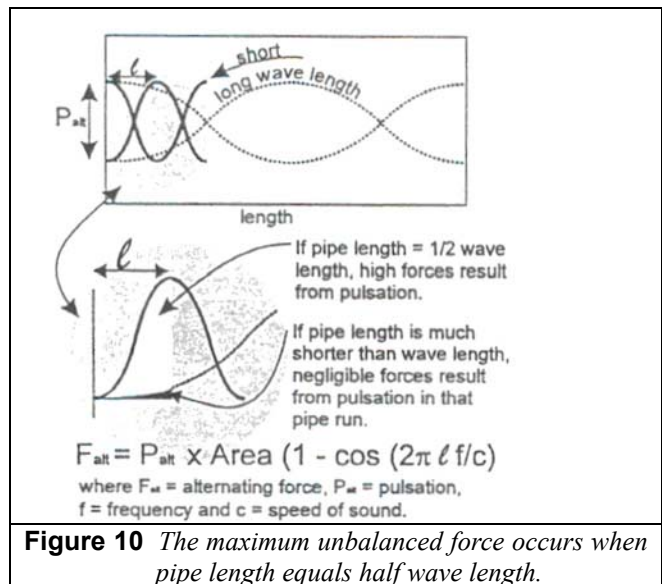
A computer model was used to predict pulsations, and the model was verified with field measurements (Figs. 6 – 8). The worst vertical forces in the discharge nozzles were calculated to be about 7000 pounds peak to peak (pk – pk) and in the suction nozzles about 1100 pounds pk – pk, both at the seventh order. It is clear that these forces are sufficient to cause the problems found.

Note that over the short length between the cylinder and the bottle, there is very little change in pressure amplitude at low frequencies (Fig. 10). The unbalanced force generated in short pipe runs by pressure pulsations at low orders of run speed (long wave lengths) will be much lower than at higher orders, all other things being equal.



**Figure 9** Vertical forces due to pulsation can develop between the cylinder and the bottle.

Consequently, the line side pulsation guideline (DPL) should not be used between the bottle and the cylinder. The DPL would restrict low order pulsations excessively. There is no need for such restrictions. Moreover, any attempt to design a system to meet the DPL in the cylinder and nozzle would be impractical. An unbalanced force guideline makes more sense for this portion of the system, combined with the current compressor side guideline.

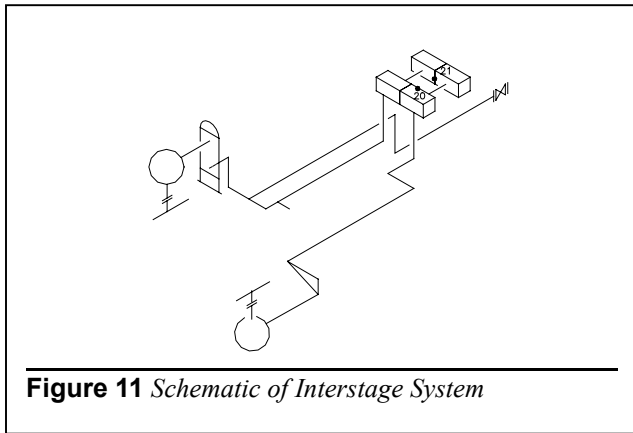


**Figure 10** The maximum unbalanced force occurs when pipe length equals half wave length.

## CASE STUDY 2 – COOLER FORCES

### The Problem

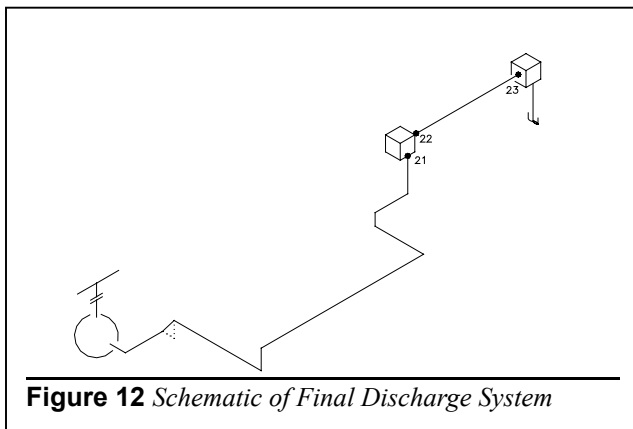
A large horizontal cooler in a two stage, high pressure compressor installation was experiencing marginal to high vibration. In addition, the aftercooler section was under sized, causing excessive pressure drop and horsepower loss. Schematics of the interstage and final discharge systems are shown in Figures 11 and 12.



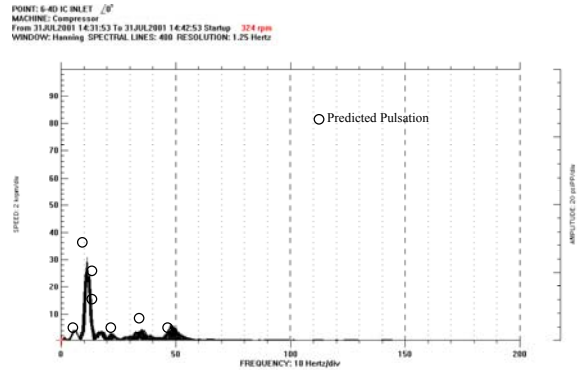
**Figure 11** Schematic of Interstage System

### New Aftercooler Section

The aftercooler section was a single pass cooler, 48 ft. long. A new, larger cooler section, of the same length, was proposed to address the high pressure drop issue. Since the vibration levels on the cooler were already a concern, the effects of changing out the cooler section were evaluated prior to making the physical change. A digital, time domain acoustical model of the existing second stage discharge system was used to predict pulsation-induced unbalanced forces across the existing aftercooler. Field measured pulsation was compared to predicted levels to confirm the accuracy of the model (Fig. 13).

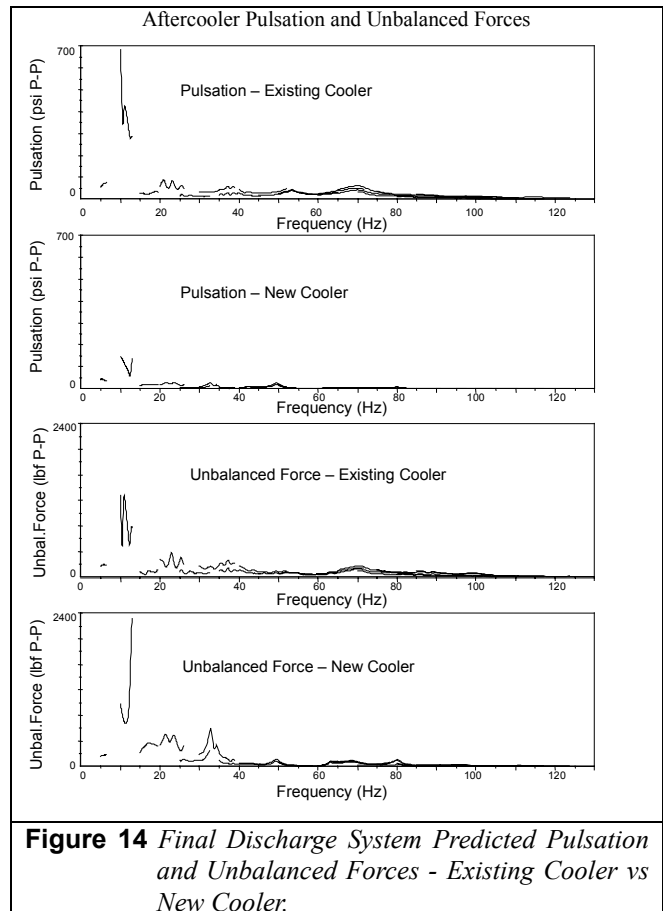


**Figure 12** Schematic of Final Discharge System



**Figure 13** Field Measured Pulsation vs Predicted at Aftercooler Outlet.

Once the new cooler section was added to the acoustical model the pulsation levels in and around the cooler were reduced (Fig. 14). However, contrary to intuition, even though the pulsation levels were lower with the new cooler the unbalanced force levels across the cooler showed a significant increase (Fig.14). The increase in unbalanced force levels was due to the increased cross sectional area of the new cooler section.



**Figure 14** Final Discharge System Predicted Pulsation and Unbalanced Forces - Existing Cooler vs New Cooler.

Given the predicted increase expected in the aftercooler unbalanced forces the cause of the cooler vibration was further investigated. It was determined that the intercooler unbalanced forces were making a significant contribution to the cooler vibration. The intercooler section consisted of two passes, also 48 ft. long, physically located next to the aftercooler section. Beta was asked to investigate what system changes would be necessary to lower the unbalanced forces in both the interstage and final discharge coolers as well as throughout other areas. The acoustical analysis revealed that secondary volumes were required to significantly reduce the pulsation and unbalanced forces. Less aggressive modifications were successful in lowering pulsation and unbalanced force levels in select areas.

Coolers are being used more often in pipeline installations to increase the mass of gas being pumped into the main pipeline. Coolers are just one area where it is very important to evaluate unbalanced forces, not just pulsations, when determining if pulsation control is sufficient.

### CONCLUSION

Reciprocating compressors have a definite place in the pipeline industry. For installations where reciprocating machinery are the best fit for the compression required, the design of the installation should be no more difficult than for a similar centrifugal installation.

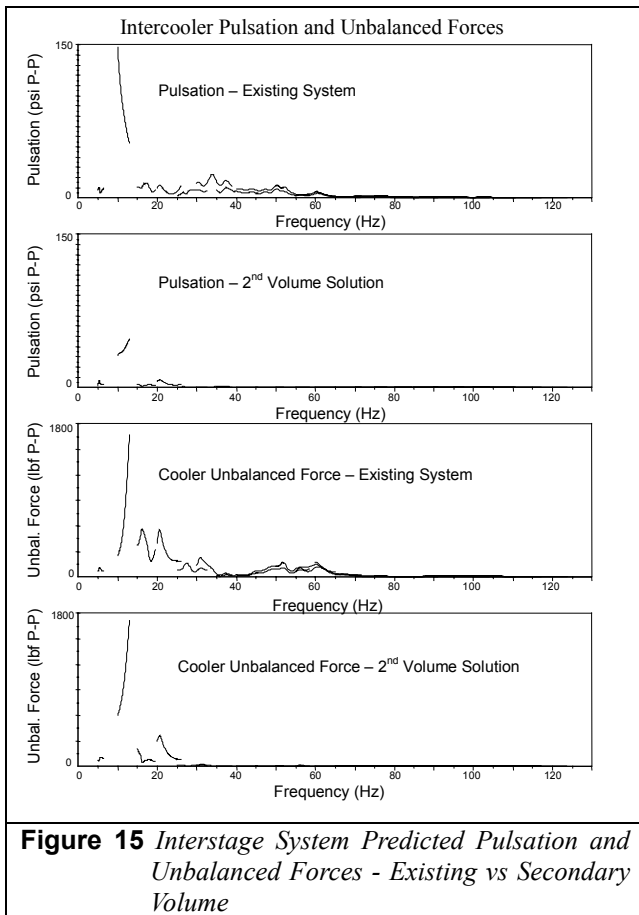
The area of pulsation and vibration control is one area where there is a significant difference in design requirements between centrifugal and reciprocating compressors. For reciprocating compressor installations API 618, 4<sup>th</sup> Edition, provides a good starting point for outlining design guidelines. Many of the areas that can be viewed as deficient in the 4<sup>th</sup> Edition of API 618 (e.g. lack of unbalanced force guidelines and vagueness in pressure drop guideline) are addressed in the proposed 5<sup>th</sup> Edition. In addition the Recommended Practice document API is planning on publishing around the same time as the 5<sup>th</sup> Edition of API 618, will provide even better guidance to any industry using reciprocating compressors.

### ACKNOWLEDGMENTS

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The secondary volume solution for the interstage significantly lowered pulsation levels throughout the system (Fig. 15), and unbalanced force most everywhere. Again contrary to intuition, the intercooler unbalanced forces remained basically unchanged (Fig. 15). In this case the relative difference in pulsation levels between the cooler header volumes remained about the same, even though the absolute levels were much lower with the secondary volume installed. Modifications to the recycle line were required to significantly lower the intercooler unbalanced forces.

