

THE NEW FIFTH EDITION OF API 618 FOR RECIPROCATING COMPRESSORS— WHICH PULSATION AND VIBRATION CONTROL PHILOSOPHY SHOULD YOU USE?

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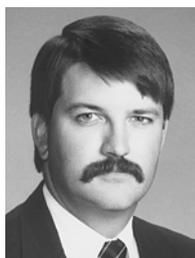
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ABSTRACT

The proposed Fifth Edition of API 618 (“Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services”) incorporates significant changes in the section concerning pulsation and vibration control. There are still to be

three “design approaches,” but the requirements to perform certain analyses that were presented as optional in the Fourth Edition will now be dependent on pressure pulsation and force levels determined from the acoustical simulation. The confusion concerning when piping forced mechanical response calculations should be performed, which originated in the Fourth Edition, has been eliminated; forced response calculations are not required to satisfy API 618 Fifth Edition when pulsation levels are controlled properly.

A separate article on pulsation and vibration control is being developed by the API 618 sub-task force on pulsation and vibration control as an appendix (annex) to API 618. This text will be a stand alone “RP” (Recommended Practices) document in the API system, which would then be referenced by API 618 as well as other API standards (e.g., API 674 for Positive Displacement Pumps) for which pulsation and vibration control are an issue. This document, to be issued in 2002, will discuss the different design philosophies inherent to the new edition of the standard.

The purpose of this tutorial is to provide the user with a working knowledge of good engineering practices for pulsation and vibration control of reciprocating machinery in relatively high mole weight gases (e.g., natural gas), as well as an indepth understanding of the proposed changes in API 618 and the differing design philosophies. Several case histories are used to illustrate why robust pulsation control is important for reciprocating compressor piping systems.

The authors are members of the API 618 sub-task force on pulsation and vibration control and each has over 20 years of experience in this field.

INTRODUCTION

In the 1950s and 60s, design techniques were developed using analog simulation tools for the control of pulsation in compressor piping systems. Acoustical designs utilizing reactive pulsation control (acoustic filtering), in combination with resistive elements (orifice plates) where necessary, became very successful in controlling pulsation levels transmitted to piping, piping shaking force, and bottle unbalanced force.

Over the last 20 years, digital techniques have progressed significantly as the speed and capacity of computers have developed, and today, digital techniques for acoustic simulation are in greater overall use worldwide than analog methods. However, in recent years there has also been a trend in some industry segments away from utilization of effective pulsation control techniques and toward more reliance on mechanical techniques to “control” vibration. There are several reasons for this disturbing trend.

First, the basic pulsation control technology has historically been proprietary to certain organizations. Many users of acoustical simulation software do not understand reactive pulsation control and/or their software does not permit them to be cost competitive

in designing reactive filter systems; resistive designs require significantly less engineering effort and technical expertise.

Another reason for this trend is the proliferation of finite element based structural dynamics software for piping. Virtually every pipe stress analysis package on the market today has some dynamic capabilities. Mechanical natural frequencies and forced vibration levels of complex piping systems, once modeled, can be calculated fairly easily; however, it is the lack of understanding of the limitations on the accuracy of these calculations that leads to serious problems and in some cases disastrous consequences. As will be shown herein, even if the structural dynamics calculations were extremely accurate, there is no justification for the risk involved by designing systems with inadequate pulsation control.

However, the new Fifth Edition of the API 618 (2001) standard will continue to include language concerning detailed mechanical response and natural frequency calculations, implying that these calculations are sufficiently accurate to be useful in the design stage. While such calculations can be performed to any degree of accuracy in theory, practical considerations put limits on the accuracy that is actually achievable. It is the goal of this tutorial to illustrate this point, and to present well-established design techniques that can reduce the dependence on expensive and problematic forced response analysis for the qualification of piping system designs.

SOURCES OF VIBRATION IN RECIPROCATING COMPRESSORS

Pulsation Excitation Mechanism

Reciprocating compressors generate flow modulations that in turn generate pressure pulsations. The flow modulations come about as a result of intermittent flow through the suction and discharge valves, as well as geometry effects due to the (finite) length of the connecting rod.

Figure 1 shows a schematic of a compressor cylinder. The suction flow (Q_S) enters the cylinder, and the discharge flow (Q_D) exits the cylinder. The velocity of the piston, shown in Figure 2, is approximately sinusoidal in shape. The deviation of the actual piston motion from the sinusoidal shape is due to the finite length of the connecting rod. As the ratio of the connecting rod length to the crank radius (L/R) is increased, the shape becomes more closely sinusoidal. The pressure pulsation generated by the compressor is proportional to the flow (Q_S or Q_D) modulation. Since the flow is based on the product of the piston velocity and the piston swept area, the shape of the discharge flow at the piston face is of the same shape as the piston velocity curve ($Q = \text{Area} \times \text{Velocity}$). Since the suction and discharge valves of each cylinder end (e.g., the head end) of a compressor are never open simultaneously, the suction and discharge piping systems are isolated acoustically. Therefore, we can look at the flow excitation of either the suction or discharge independently for the purpose of understanding the pulsation excitation mechanism.

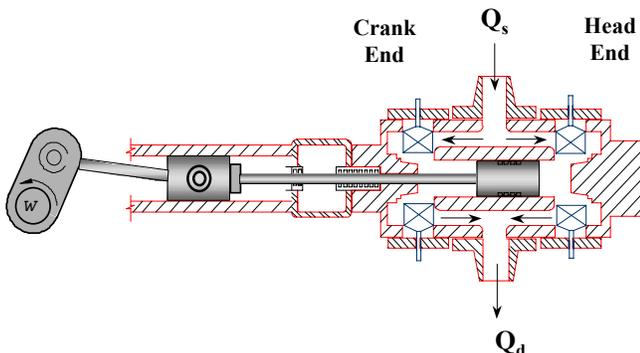


Figure 1. Reciprocating Compressor Slider Crank Mechanism.

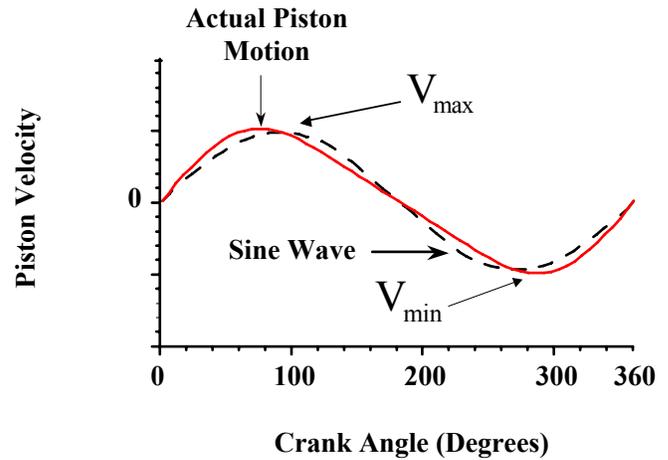


Figure 2. Piston Velocity for Slider Crank Mechanism.

Figures 3-6 show the effect of the valve action on flow through the discharge valves of a compressor. Figure 3 shows the discharge valve flow versus time for the head end of a cylinder. During compression, the suction and discharge valves are closed. When the pressure in the cylinder reaches the discharge back pressure, the discharge valve opens, and the flow versus time wave through the valve has the shape of a portion of the piston velocity curve shown in Figure 2. As the cylinder reaches top dead center (TDC), the discharge valves close, and the flow returns to zero.

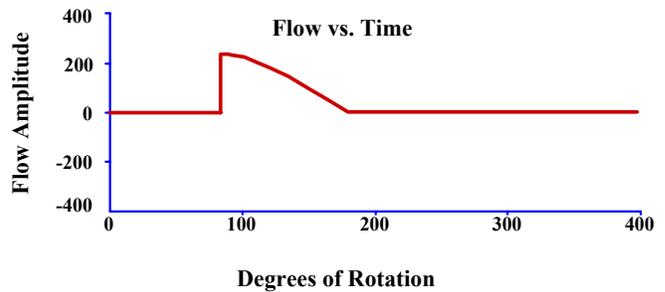


Figure 3. Single Acting Compressor Cylinder ($L/R = \infty$, Ideal Valves).

A frequency analysis of the flow wave of Figure 3 is shown in Figure 4. Due to the repetitive action of the compressor cylinder, excitation is generated only at discrete frequencies, which are multiples of the running speed. These frequencies are commonly referred to as harmonics. The highest amplitude occurs at $1 \times$ running speed, with the levels generally decreasing at higher harmonics.

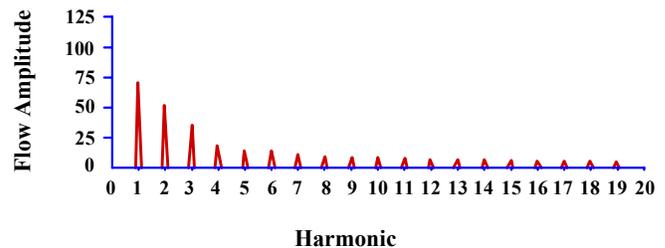


Figure 4. Flow Frequency Spectrum for Single Acting Cylinder.

For a “perfect” double acting cylinder (symmetrical head end and crank end flows, $L/R = \infty$) the flow versus time contains two identical flow “slugs” 180 degrees apart in time. Therefore, the odd harmonics (in this idealized case) cancel, so that the nonzero

cylinder flow excitation occurs at even harmonics of running speed ($2\times, 4\times, \dots$). Actual cylinders have piston rods, differences in head end/crank end clearance volumes and finite length connecting rods, so that the two “flow slugs” generated each revolution are not identical (Figure 5). Therefore, even in double acting operation, the cylinder will, in general, produce flow excitation at all harmonics of running speed as shown in Figure 6. These flow harmonics act as excitations to the piping acoustics, and the acoustic resonances of the piping will amplify pulsation at particular frequencies.

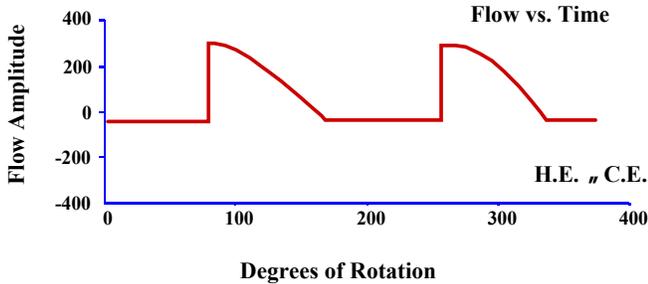


Figure 5. Unsymmetrical Double Acting Compressor Cylinder ($L/R = 5$, Ideal Valves).

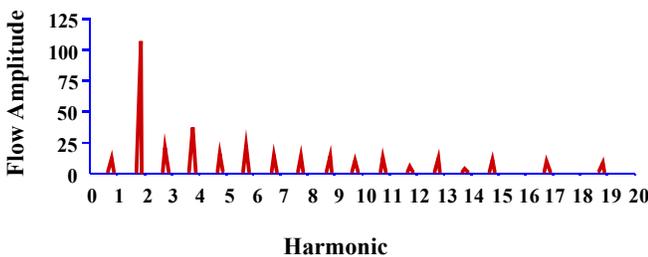


Figure 6. Flow Frequency Spectrum for Double Acting Cylinder.

Mechanical Excitation Mechanisms

In addition to acoustical excitation, another source of excitation in reciprocating compressor systems is mechanical excitation due to reciprocating inertial forces of the compressor itself, and cylinder “stretch” caused by internal pressure reaction forces acting on the cylinders and frame. These forces are typically strongest at $1\times$ and $2\times$ running speed, and are primarily a concern only in the immediate vicinity of the compressor.

A RISKY APPROACH—TUNING OF MECHANICAL NATURAL FREQUENCIES BETWEEN SIGNIFICANT EXCITATION HARMONICS

Real World Inaccuracies of Mechanical Natural Frequency Calculations

To avoid potential vibration problems in piping systems, the single most important concept is to avoid coincidence of mechanical natural frequencies with significant pulsation or mechanical excitation frequencies. However, field experience shows that the accuracy of predicted mechanical natural frequencies in piping systems is suspect even under the best of circumstances. Error margins of ± 20 percent are obtainable only in situations where accurate boundary conditions are known, and extensive, detailed modeling of both the piping system and the supporting structure is performed. Realistically, many mechanical natural frequencies cannot be calculated within a margin of 20 percent or even 50 percent. Inspection of real world chemical, gas transmission, and gas gathering stations reveals that in many cases, pipe supports have become loose or do not even touch the piping at some locations, which negates modeling efforts.

Other items that influence the accuracy of these models are:

- Uncertainty of stiffness (six degrees of freedom) of clamps/hold downs.
- Uncertainty of stiffness of clamp/hold down supporting structure.
- Difficulty in accurately predicting coefficients of friction.
- Nonlinear effects (e.g., gaps closing due to thermal growth).
- Uncertainty in attached weights (valves, actuators, etc.).
- Uncertainty of “as-built” piping layout and dimensions.
- Difficulty and complexity of modeling rack support structure.
- Uncertainties in soil stiffness effects on concrete piers.
- Settling of supports resulting in loss of piping contact.

A piping/structural support system is not a “polished machine part” for which finite element models are easily defined and analyzed. Furthermore, many vibration related problems are not associated with the main process piping itself, but with other attached components, examples of which are listed below:

- Valve actuators
- Tubing
- Conduit and cable trays in rack systems
- Inspection openings and instrument connections (thermocouples, pressure transducers)
- Flow measurement instrumentation
- Scrubber level control instrumentation
- Small branch connections (for instrumentation connections, vents, and drains)
- Instrument panels mounted on compressor decks

It is important to remember that even when the main process piping has low vibration, the main line can act as a base excitation to attached mechanically resonant branches. Often, small branches attached to the main process piping are not considered in mechanical natural frequency response modeling. Figure 7 shows conceptually how a branch, if resonant to the frequency of the vibration of the main line, can cause high vibration of the branch itself. Therefore, maintaining very low force levels in the piping is important. Figure 8 shows a valve actuator that vibrated so severely that the support bracket failed; the vibration of the main piping was less than 2 mils peak-to-peak, while the actuator itself had vibration in excess of 50 mils peak-to-peak.

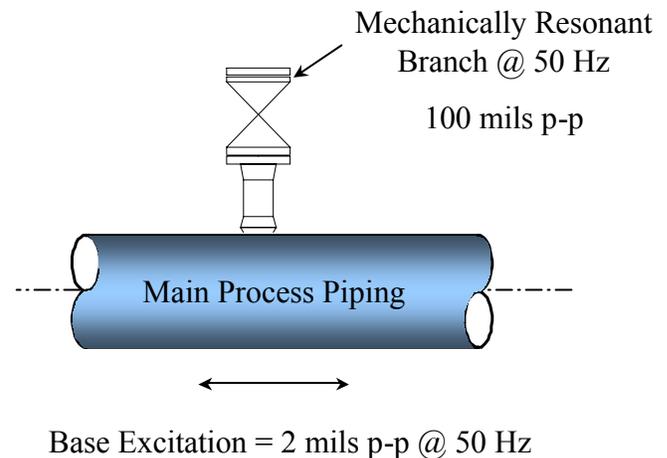


Figure 7. Example of Vibration of Attached Branch Line Due to Base Excitation by Main Process Line.



Figure 8. Valve Actuator Vibration—50 mils peak-to-peak at 24 Hz, 2 mils peak-to-peak on Main Line.

A detailed knowledge of the mechanical natural frequencies and response characteristics of the above components is generally not available in the design stage. Unfortunately, failures of small branch connections attached to main piping, as well as other items listed above, represent a large percentage of vibration related problems and actual failures that occur in reciprocating compressor piping systems.

The Effect of Inaccuracies in Mechanical Natural Frequency Calculations

Table 1 shows how errors in effective structural stiffness (due to pipe support system and structural stiffness of the piping itself) affect the accuracy of mechanical natural frequency (MNF) calculations based on the relation: $MNF \propto \sqrt{k}$.

Table 1. Effect of Uncertainty of Effective Structural Stiffness of Piping System on Actual Mechanical Natural Frequencies.

Range of Actual Effective Stiffness	Range of Actual M.N.F.
$\frac{1}{1.5} \times k_{calc} \rightarrow 1.5 \times k_{calc}$	$0.8f_{calc} \rightarrow 1.2f_{calc}$
$\frac{1}{2} \times k_{calc} \rightarrow 2 \times k_{calc}$	$0.7f_{calc} \rightarrow 1.4f_{calc}$
$\frac{1}{10} \times k_{calc} \rightarrow 10 \times k_{calc}$	$0.3f_{calc} \rightarrow 3f_{calc}$

As an illustration of the difficulty of predicting mechanical resonance frequencies in a piping system, consider a 900 rpm fixed speed compressor. The fundamental (1× running speed) frequency is 15 Hz. The frequencies of the first 10 harmonics are shown as bars on a graph in Figure 9.

Piping systems have numerous mechanical natural frequencies; in fact, they have an infinite number of natural frequencies. Figure 10 shows, conceptually, the locations of mechanical natural frequencies of a piping system superimposed on the pulsation spectrum. Generally, most designers agree that the minimum mechanical natural frequencies of the piping should be maintained at least 20 percent above the second harmonic of running speed when practical to avoid significant acoustical and mechanical excitation. The lowest piping mechanical natural frequency in Figure 10 is shown above 36 Hz (20 percent above 30 Hz), assuming this can be accomplished.

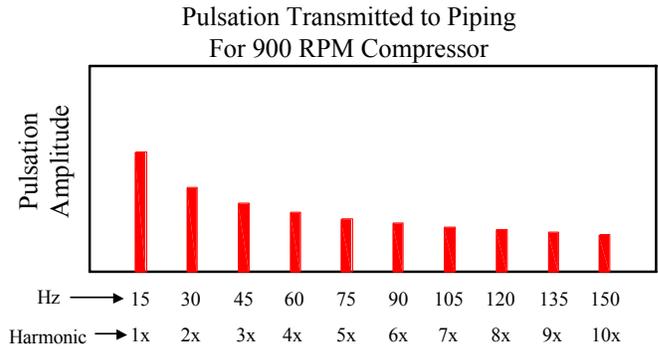


Figure 9. Pulsation Characteristics without Acoustic Filtering.

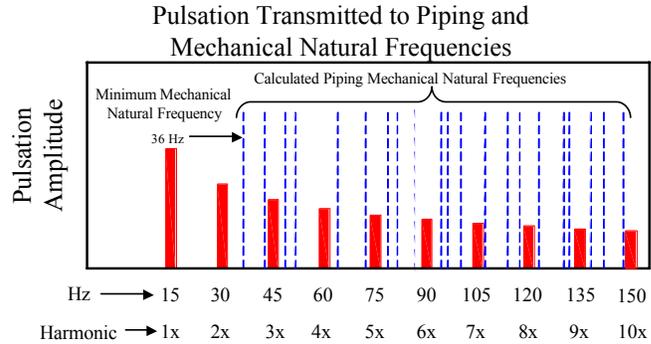


Figure 10. Pulsation Characteristics without Acoustic Filtering: Calculated Mechanical Natural Frequencies Superimposed.

Table 2 shows the effect of uncertainty on the location of mechanical natural frequencies relative to harmonics of running speed for a 900 rpm compressor. The frequencies between 1× and 2×, 2× and 3×, 3× and 4×, and 4× and 5× running speed shown in the table represent MNFs that are “tuned” between harmonics. Assuming a better than typical ±20 percent mechanical natural frequency calculation accuracy, the actual range of each natural frequency is also shown in the table. At all frequencies above 2× running speed, the actual possible range of mechanical natural frequencies between harmonics is too large; the actual range of the mechanical natural frequencies exceeds the frequency gap between harmonics. Therefore, in the design stage, it is impossible to tune any calculated mechanical frequencies above 2× running speed frequency away from pulsation excitation frequencies. Even the range between 1× and 2× running speed is within 10 percent of 2× running speed. Figure 11 shows this concept graphically.

Table 2. Range of Actual Mechanical Natural Frequencies Based on ±20 Percent Uncertainty (900 RPM Compressor).

	1×	2×	3×	4×	5×
Harmonic Frequency (Hz)	15	30	45	60	75
Calculated “Tuned” Mechanical Natural Frequency (Hz)		22.5	37.5	52.5	67.5
Actual Range (±20%) of Mechanical Natural Frequency (Hz)		18-27	30-45	42-63	54-81

Another important concept to remember is that even if the mechanical natural frequencies could be calculated within an accuracy of say, ±10 percent, significant amplification still occurs when the forcing frequency (i.e., the pulsation frequency) is 10 percent above or below the mechanical natural frequency. As shown in Figure 12, the effective amplification factor at frequencies 10 percent away from a particular mechanical natural frequency is 5:1 for all values of critical damping ratio less than 5

percent. Therefore, the excitation frequency need not be exactly coincident with the mechanical natural frequency to cause excessive vibration; a margin of 10 percent is not necessarily sufficient even when the exact natural frequency is known.

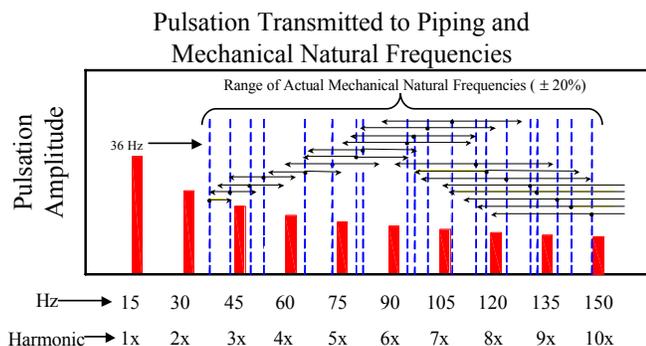


Figure 11. Pulsation Characteristics without Acoustic Filtering: Actual Range of Mechanical Natural Frequencies Superimposed.

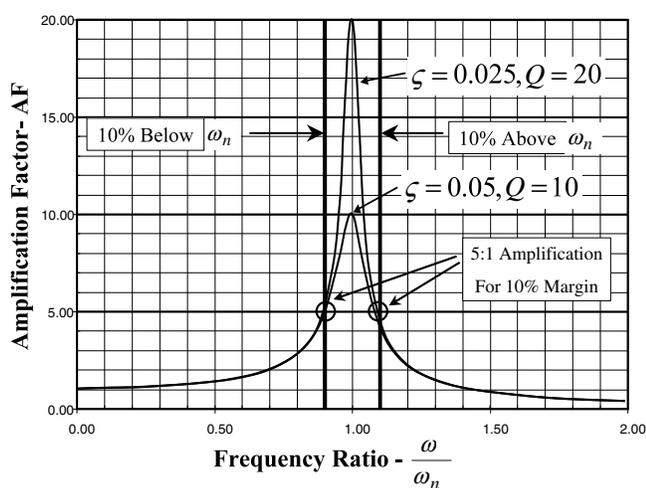


Figure 12. Effect of Separation Margin from Mechanical Natural Frequency on Amplification Factor.

Inaccuracies in Mechanical Forced Response Calculations

The new (Fifth Edition) API 618 (2001) standard allows that in the event the pulsation levels and force levels acting on runs of pipe exceed the amplitude guidelines, and separation margins are not met, forced response calculations may be performed to qualify the system. If the predicted vibration levels are below the vibration allowable guideline, then the pulsation and force levels are considered acceptable.

The accuracy of mechanical response amplitude (vibration and stress) calculations is influenced by the same items that affect the calculation of natural frequencies. Practically, because of the uncertainty of the actual mechanical natural frequencies, mechanical resonance must be assumed. If the condition of resonance is not assumed, the predicted response levels will almost always be low. The “Q” value (amplification factor), which may vary from 10 to 100, must also be assumed. This makes resonant amplitudes extremely difficult to predict. In fact, since the resonant amplitudes computed are defined by the assumed damping, the predicted results are arbitrary.

Mechanical response calculations are best left to situations in which field data are available to adjust the model to give the proper natural frequencies and damping; accurate simulation of forced piping mechanical response at the design stage is generally impractical.

A BETTER APPROACH TO VIBRATION CONTROL— PULSATION AND FORCE CONTROL THROUGH REACTIVE ACOUSTICAL FILTERING

Fortunately, many of the inherent difficulties of mechanical vibration and natural frequency prediction may be overcome through robust acoustical design. Whereas the mechanical natural frequencies of piping can be difficult to predict within ±20 percent or even ±50 percent, acoustical natural frequencies and therefore reactive filter frequencies can be calculated relatively accurately (within ±5 percent). Furthermore, the technique of acoustic filtering can be used effectively and confidently to control pulsation in relatively high mole weight, relatively low speed of sound systems (less than 2000 ft/s) in the design stage. (In low mole weight gas systems, where reactive filters are impractical, pulsation control can be accomplished through the use of resistive or pressure drop elements.)

Acoustic filtering involves the use of two volumes joined by a relatively small diameter pipe, which is known as a volume-choke-volume filter. Figures 13 and 14 show various forms of the volume-choke-volume filter. Such devices have the pulsation response characteristics shown in Figure 15. At frequencies above its characteristic resonance (f_H), transmitted pulsation levels drop off rapidly. Equation (1) is used to calculate the filter frequency, f_H , of an ideal filter with no piping attached:

$$f_H = \frac{c}{2\pi} \sqrt{\frac{A_c}{L'_c} \left(\frac{1}{V_1} + \frac{1}{V_2} \right)} \quad (1)$$

where:

- f = Frequency (Hz)
- c = Speed of sound (ft/s)
- A_c = Area of choke tube (ft²)
- L_c = Length of choke tube (ft)
- L'_c = $L_c + .6d$ (ft)
- d_c = Choke diameter (ft)
- V_1 = Volume of primary bottle (ft³)
- V_2 = Volume of secondary bottle (ft³)

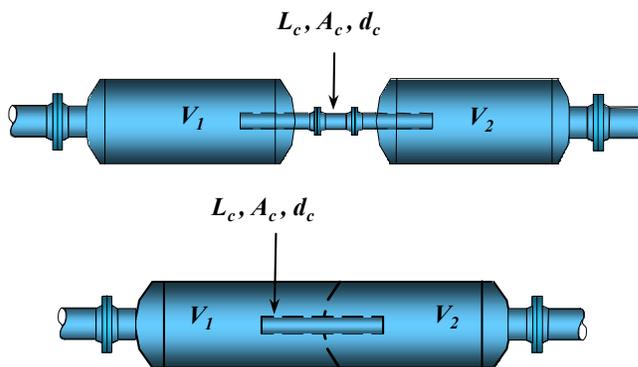


Figure 13. Nonsymmetrical Volume-Choke-Volume Filter—Straight Choke Tubes.

Design Using Acoustic Filtering in Conjunction with Good Mechanical Support Practices

Figure 16 shows how the pulsation control through use of such a filter controls vibration, eliminating the concern of uncertainty of piping mechanical natural frequency calculations.

- Pulsation and resulting force levels are controlled to insignificant levels above some cutoff frequency (usually below 1× running speed).
- Piping mechanical frequencies are placed well above this cutoff frequency.

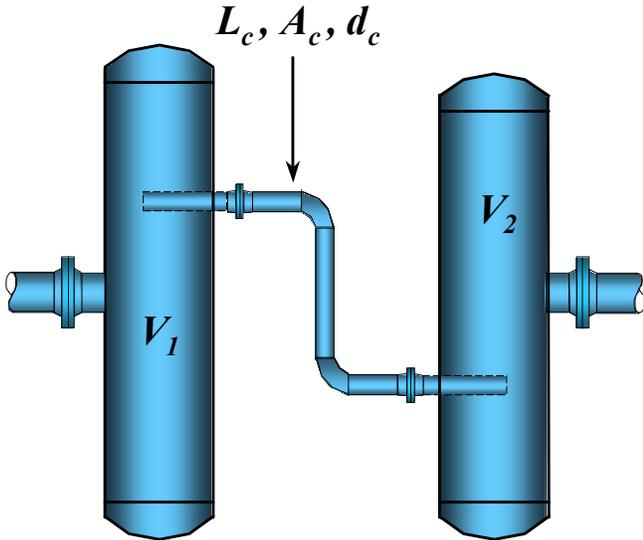


Figure 14. Nonsymmetrical Volume-Choke-Volume Filter—Elbows in Choke Tube.

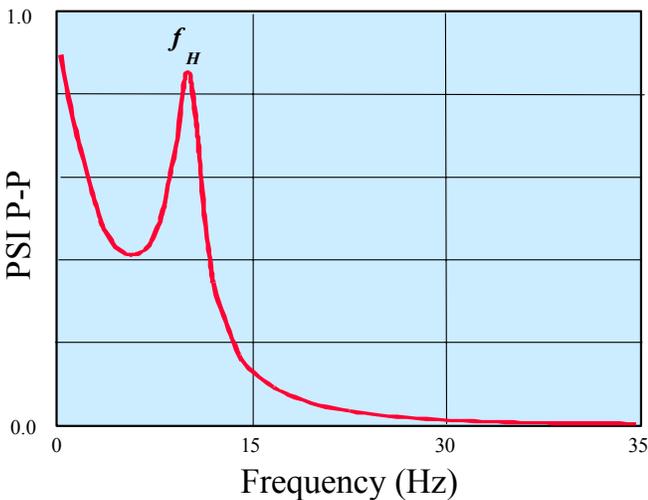


Figure 15. Typical Frequency Response to Flow Excitation of Volume-Choke-Volume Filter System (No Passbands).

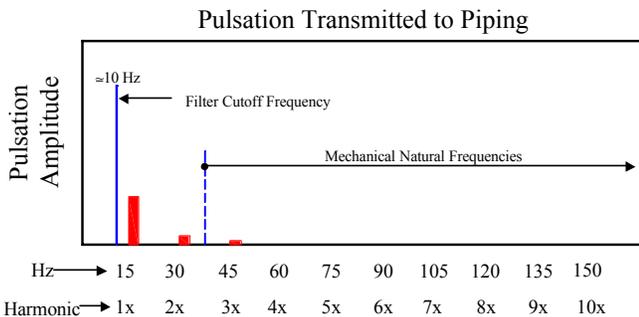


Figure 16. Pulsation Characteristics with Acoustic Filtering.

With this design concept, the mechanical natural frequencies of the piping are well above the lower significant harmonics of pulsation, removing the concern over the exact location of the various mechanical natural frequencies. Within practical limits, even $1\times$ running speed pulsation levels can be controlled to any desired level.

Comparison of Pulsation Control Devices in Discharge Piping Systems

Figures 17, 18, 19, 20, and 21 compare various pulsation suppression techniques in an infinite length discharge line (nonreflective boundary condition) of a compressor operating over a speed range of 700 to 1000 rpm. The assumption of a nonreflective boundary eliminates acoustical resonances of the piping itself, and is a convenient method for comparison of the effectiveness of pulsation control devices.

The pulsation control treatments are:

- None (no surge volume, Figure 17).
- A simple surge volume with a volume equal to 50 percent of the volume calculated using the API 618 Design Approach 1 sizing formula ($1/2 \times$ API surge volume, Figure 18).
- A simple surge volume with a volume equal to 100 percent of the volume calculated using the API 618 sizing formula ($1 \times$ API surge volume, Figure 19).
- Volume-choke device (Figure 20).
- Volume-choke-volume filter with $f_H < 1\times$ running speed (Figure 21).

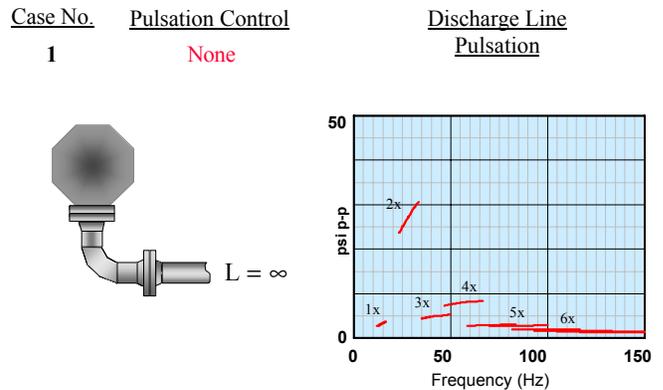


Figure 17. Comparison of Pulsation Control Devices—Case 1.

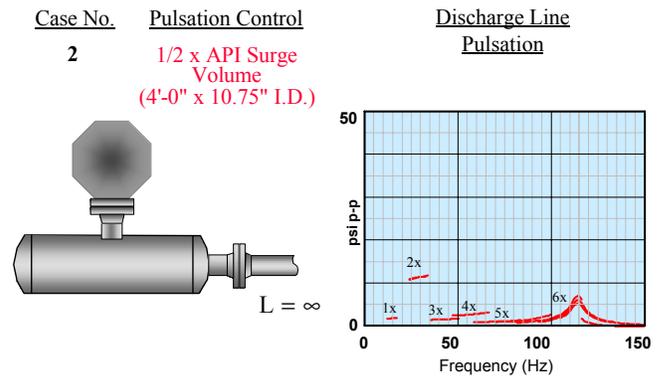


Figure 18. Comparison of Pulsation Control Devices—Case 2.

Comparison of the pulsation amplitudes for the five cases shows the significant reduction in pulsation levels obtained by the use of acoustic filtering. (Note that no resonances occur in the piping because of the assumed infinite length line boundary conditions; therefore, these cases can only be used for relative comparison of pulsation amplitudes.)

Mechanical Analogies and Interpretation of Surge Volumes and Filters

Surge Volumes

At frequencies below the length resonances of the bottles themselves, volume bottles act predominantly as acoustic

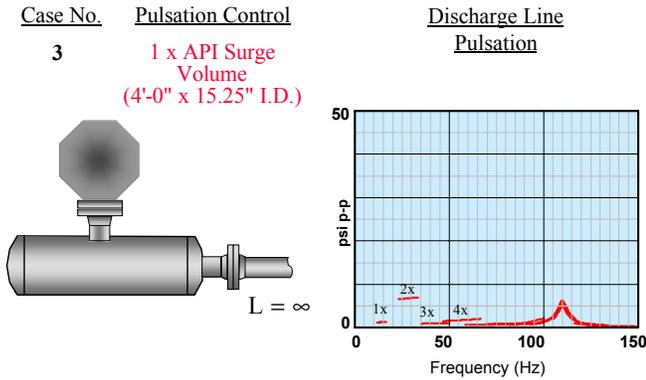


Figure 19. Comparison of Pulsation Control Devices—Case 3.

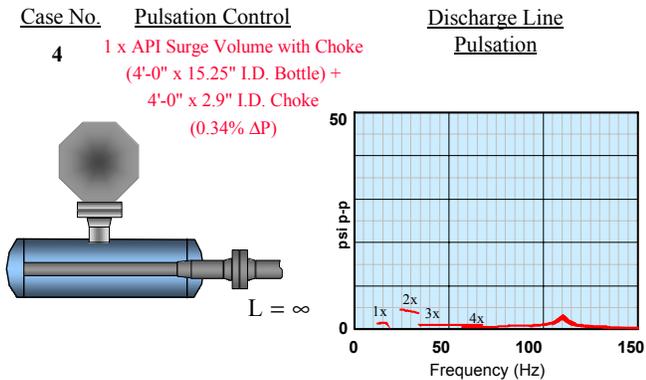


Figure 20. Comparison of Pulsation Control Devices—Case 4.

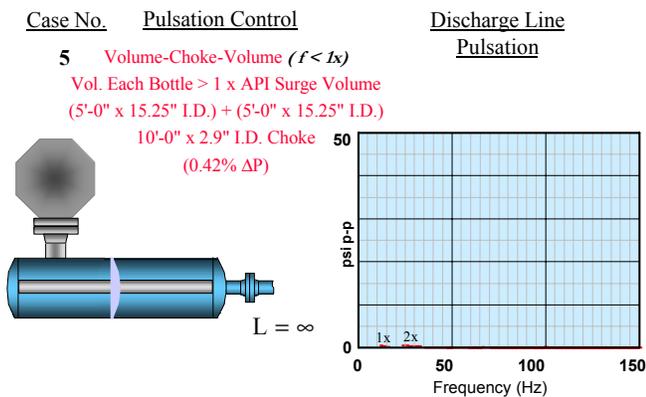


Figure 21. Comparison of Pulsation Control Devices—Case 5.

compliance. Acoustical compliance is analogous to mechanical flexibility as shown in Figure 22. The pipe beyond the surge volume contains the gas that has mass and elastic properties. This fluid is set into a vibratory state by the motion imposed upon it by the piston. If a highly flexible (low stiffness) element is placed between the piston face and the pipe fluid, the piston motion is essentially isolated from the fluid in the piping, and less vibration (and therefore less pressure variation) of the fluid occurs. This is similar to the concept of vibration isolation commonly used in machinery.

Filters

Volume-choke-volume filters have, in addition to two compliance components (two volumes), a choke tube that acts as an acoustical inductance to resist changes in velocity of the fluid contained in the choke tube. As for the single surge volume, these lumped compliance and inductance properties are valid at

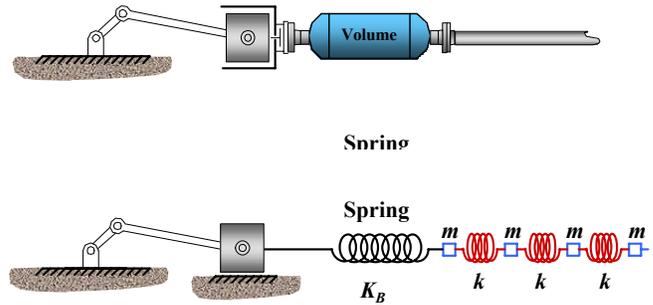


Figure 22. Mechanical Analogy of Surge Volume.

frequencies below the open-open resonant frequencies of the choke tube length, and the closed-closed resonant frequencies of the bottle lengths. The mechanical analogy of such a filter is a high flexibility (volume) in series with a large mass (choke) and another high flexibility (volume) as shown in Figure 23. At frequencies above the resonant frequencies of the mass spring system, the piston motion is isolated due to the momentum characteristics of the choke tube fluid. The acoustic filter has characteristics analogous to those of L-C filters used in electrical systems.

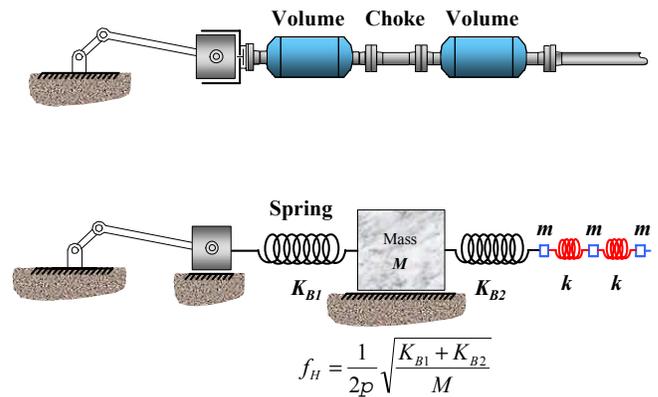


Figure 23. Mechanical Analogy of Volume-Choke-Volume Filter.

Design Procedure for “Heavy” Gases Using Reactive Pulsation Control (Acoustic Velocity < 2000 ft/s)

The use of reactive filtering in conjunction with control of mechanical natural frequencies results in a safe margin between significant pulsation induced forces and mechanical natural frequencies. The procedure for designing reactive filters is:

- Determine volume-choke-volume filter design to filter all harmonics of running speed. Generally, the filter frequency is set at 50 to 80 percent of 1× running speed for heavy gases, or between 1× and 2× running speed for lighter gases.
- Perform pulsation simulation to determine pulsation levels and acceptability of filter design. Determine maximum frequency (f_p) of significant pulsation and force in piping.
- Determine minimum allowable mechanical natural frequency (f_m) based on (f_p). Set $f_m \geq 1.5 \times f_p$.
- Locate vibration restraints near all concentrated masses (e.g., valves).
- Use pipe support span tables (Table 3) to determine additional support locations based on f_m .
- Determine minimum stiffness (k) of each support: $k \geq 2 \times$ lateral span stiffness = $\frac{2 \times 48 EI}{l^3}$ (l = support span).

Table 3. Pipe Support Span Spacing Table.

Natural Frequency	Maximum Span Spacing (ft)								
	Nominal Pipe Size / Outside Diameter (in)								
	2	4	6	8	10	12	14	16	18
10 Hz	2.375	4.500	6.625	8.625	10.75	12.75	14.00	16.00	18.00
20 Hz	13.3	18.3	22.2	25.3	28.3	30.8	32.3	34.5	36.6
30 Hz	9.4	12.9	15.7	17.9	20.0	21.8	22.8	24.4	25.9
40 Hz	7.7	10.6	12.8	14.6	16.3	17.8	18.6	19.9	21.1
50 Hz	6.7	9.2	11.1	12.7	14.1	15.4	16.1	17.3	18.3
60 Hz	5.9	8.2	9.9	11.3	12.7	13.8	14.4	15.4	16.4

Use of this acoustic filtering concept in conjunction with control of minimum piping mechanical natural frequencies ensures that resonance is avoided.

CHANGES FOR API 618 FIFTH EDITION

Residual Nonresonant Force Evaluation

A significant change currently being made to API 618 (2001) is with regard to pulsation induced unbalanced forces acting on piping runs. Although this is not a new concept, past versions of API 618 specified limits only for pulsation levels. The new standard will address allowable force levels for nonresonant conditions.

Figure 24 shows how acoustically induced forces are calculated for a portion of the piping system (assuming centrifugal effects of dynamic gas/fluid flow at elbow are small). The straight run of pipe between elbows is considered to be a rigid body, and the force acting along the run is the sum of the force acting at the elbows at each end as defined in Equations (2) and (3):

$$\Sigma F = \vec{F}_A + \vec{F}_B \tag{2}$$

$$\Sigma F = \vec{P}_A \frac{\pi id^2}{4} - \vec{P}_B \frac{\pi id^2}{4} \tag{3}$$

where \vec{P}_A, \vec{P}_B are vectors representing amplitude and phase of pulsation at points A and B at a particular frequency. These force calculations are easily made based on the known piping geometry and calculated pulsation levels at the elbows.

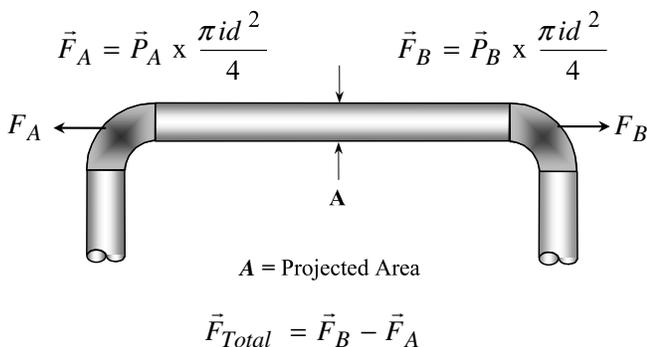


Figure 24. Dynamic Force on Piping Run.

While calculating the forces acting on the piping is fairly straightforward, determining an acceptable force level is quite difficult. It is important to realize that the shaking force guideline for the new API 618 (2001) standard is based on specific nonresonant configurations. In general, much higher force levels may be tolerated at frequencies below the lowest mechanical natural frequencies; however, force levels should be controlled to very low levels at frequencies near or above the lowest piping mechanical natural frequency.

A force evaluation criteria proposed and used by the authors' is given in Table 4 for the allowable forces acting on each piping

restraint in a straight run of pipe between elbows (or other force coupling points). The force per restraint is calculated as the force in the piping segment divided by the number of restraints that resist axial vibration of the piping run. Note that the important concept of these criteria is that forces at higher frequencies should be controlled to much lower levels, since the accuracy of mechanical natural frequency calculations is such that the only reasonable engineering assumption that can be made is that resonance can potentially occur at the higher frequencies.

Table 4. Allowable Force Guideline.

Support/Restraint System	Allowable Force (lbs p-p) Per Piping Restraint in Straight Run	
	$f < 0.8 f_{mf}$ (Non-resonant)	$f \geq 0.8 f_{mf}$ (Potentially Resonant)
Ground Supported	100 × NPS	10 × NPS
Rack Supported	50 × NPS	5 × NPS

where:

- f = frequency of force (Hz)
- f_{mf} = lowest mechanical natural frequency of piping (Hz)
- NPS = nominal pipe size (in)

Care should be taken to apply force criteria with caution. Very low force levels in main piping may cause very low vibration levels in the main line; however, if branch piping, appurtenances, instrumentation lines, etc., are resonant at the same frequency, very high vibration of the attached elements can occur.

Definition of Mechanical Response Calculation "Triggers"

API 618, Third Edition (1986), described in general terms the concepts of resonance avoidance and the use of filtering techniques. The Fourth Edition (1995) added descriptions of various procedures (Appendix M) in an attempt to clarify specific procedures to meet the requirements of Design Approaches 2 and 3. Procedure M.7 of the current Fourth Edition is as follows:

A piping system dynamic stress analysis calculates the mechanical system responses and associated mode shapes. The significant predicted pulsation forces are imposed on the piping to the extent necessary in order to calculate the expected vibration and stress amplitudes at the critical points in the system. These stresses are compared to the levels identified in 3.9.2.2.1.

With this short paragraph, Design Approach 3 analyses were forced with providing a level of calculation that is subject to a great deal of interpretation. What are the critical points in the piping system? What pulsation forces are significant?

Based on the authors' involvement with the API 618 sub-task force, it appears that in Europe this paragraph has been taken literally. Finite element models of piping systems are routinely made, and piping forced response calculated. This apparently fits in well with their normal design procedure since:

- The bottles are designed prior to knowing the piping details.
- Acoustic filtering is usually not used.
- This mechanical analysis is used to design and justify more complex supports and anchoring systems to control vibration.

Table 5 shows the procedures specified by Appendix M of the Fourth Edition for each of the three design approaches. Table 6 describes each of these procedures.

The flow chart shown in Figure 25 describes the work process used to satisfy the three design approaches for the new Fifth Edition of API 618 (2001), and various analysis procedures

Table 5. Summary of Analysis Procedures of Fourth Edition for Design Approaches 1, 2, and 3.

API 618, Existing Fourth Edition Appendix M – Pulsation Design Studies	
Design Approach 1	Includes M.1
Design Approach 2	Includes M.2 through M.4
Design Approach 3	Includes M.2 through M.8
Optional	M.9 through M.11

Table 6. Description of Analysis Procedures for Fourth Edition for Design Approaches 1, 2, and 3.

Design Approach 1 (Existing Fourth Edition)	
M.1	Analytical Study (No Acoustical Simulation)
Design Approach 2 (Existing Fourth Edition)	
M.2	Acoustical Analysis (Acoustical Simulation Study)
M.3	Performance Analysis (Pulsation and Pressure Drop Effects)
M.4	Mechanical Piping System Analysis (Piping Natural Frequencies)
Design Approach 3 (Existing Fourth Edition) (Includes Design Approach 2 Plus M.5-M.8)	
M.5	Mechanical Compressor Manifold System Analysis (Manifold Natural Frequencies)
M.6	Compressor Manifold System Vibration and Dynamic Stress Analysis (Forced Vibration Analysis)
M.7	Piping System Dynamics Stress Analysis (Forced Vibration Analysis)
M.8	Calculation of Dynamic and Static Stresses for Pulsation Suppressor Internals

corresponding to the Fourth Edition are noted. Design Approach 1 (DA-1) is not changed from the previous editions and involves no simulation of the system. The pulsation suppression devices (PSD) are sized based on the formula included in the specification or by vendor proprietary methods.

If a simulation is to be done, there is a new optional “pre-study” that can be performed. This comes from a common practice in Europe, also referred to as a “damper check.” This simulation is conducted prior to the finalization of the piping layout and sizes the PSDs using nonreflective piping. The intent is to allow procurement of the PSD vessels earlier in the project timeline, which is desirable from a commercial standpoint. However, from a technical standpoint, this is not the best approach since the attached piping affects the optimum PSD design.

Once the piping layout is finalized, a complete acoustic simulation is carried out. If Design Approach 2 (DA-2) is specified, the pulsation and pressure drop criteria must be met and a table of maximum clamp spacing is developed based on avoidance of resonance with any significant pulsation energy. No evaluation of shaking forces or compressor manifold system response is required.

Design Approach 3 (DA-3) includes evaluation of the mechanical response characteristics of the compressor manifold system and piping, just as for the previous editions. The key changes include the introduction of shaking force criteria and specific steps that can be taken to satisfy DA-3. As shown in the flow chart, if the pulsation and pressure drop allowables are met, and there are no mechanical natural frequencies of the manifold system or piping that are coincident with significant pulsation energy, and the nonresonant shaking forces are acceptable, then the system is acceptable and the analysis is complete. This is the technical approach that has served the industry well for decades. If, however, the above-mentioned criteria are not met, specific additional steps can be taken to justify (with calculations) that the system may be acceptable even though it violates the DA-3 criteria.

The DA-3 criteria again are low pulsation, low shaking forces, and nonresonant mechanical systems. If these criteria are violated, the additional steps include calculation of vibration levels for comparison to a newly included allowable vibration curve and ultimately calculation of cyclic stresses if the vibration criteria are not met.

A close examination of these steps reveals that one should not like to be in the situation of having to resort to these analyses.

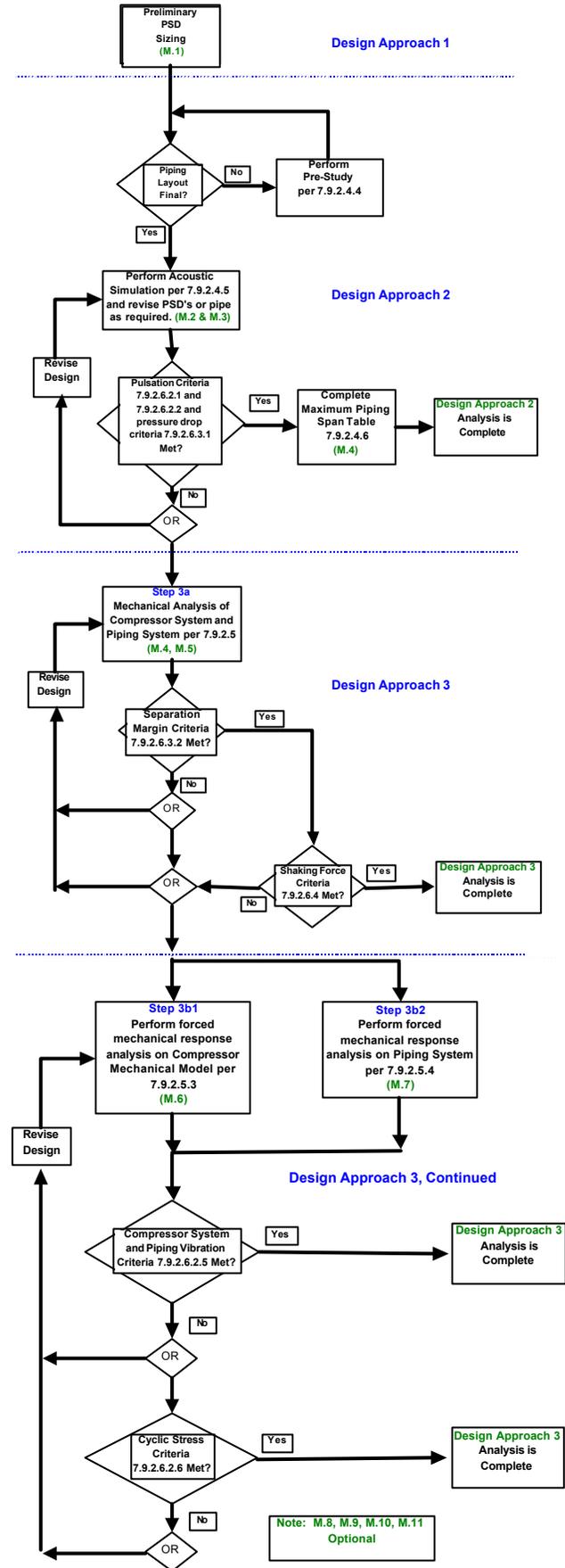


Figure 25. Flow Chart for API 618 Fifth Edition.

First, to be in the position of having to calculate vibration levels requires that the shaking forces are either too high or there is a mechanical natural frequency of the manifold system or piping that violates the separation margin. If the shaking forces are too high, the vibration levels should also calculate to be too high, since the shaking force criteria were derived from the allowable vibration curve. If the separation margin is violated, then for practical purposes the chances of the system being resonant are high, which is obviously an undesirable situation.

Finally, if the point is reached where the cyclic stresses must be computed, this means that the forces are high, the vibration is high, and the system is likely resonant. It follows then that the cyclic stresses, if computed properly, will likely be significant. Even if the calculations show the stresses to be acceptable, given the limitations on the accuracy of these calculations, a system requiring steps 3b1 or 3b2 to satisfy DA-3 is necessarily a high risk system. In addition, the stress calculations typically apply only to the main process piping, not the numerous branch connections, instrumentation, etc., that are responsible for the majority of failures in industrial piping.

The flow chart (Figure 25) illustrates that Design Approach 3 can be satisfied by any of the design steps, 3a, 3b1, or 3b2. This is an improvement over the Fourth Edition in that Step 3a clearly satisfies the design criteria as originally intended in the Third Edition. *Progression to Steps 3b1 and 3b2 are only required if the designer has failed to control pulsation levels properly and/or the system does not meet the separation margin guidelines (i.e., the potential for resonance exists).*

This is not to say that structural analysis tools are of no practical use. They can be used with good success in the design stage to avoid resonance if the limitations are understood, and they are extremely useful in designing corrections when used in conjunction with field measurements.

Table 7 shows analysis procedures that will be optional in the new Fifth Edition.

Table 7. Optional Procedures (New Fifth Edition).

M.8	Dynamic & Static Stress Calculations for Pulsation Suppression Internals
M.9	Compressor Valve Dynamic Response Study
M.10	Pulsation Suppression Device Low Cycle Fatigue Analysis
M.11	Piping System Thermal Flexibility

CASE HISTORIES ILLUSTRATING LIMITS OF MECHANICAL MODELING

Failure of 10 Inch Suction Pipe

This first example illustrates the false sense of security created by a design that (with good intention) met the API 618 Fourth Edition (1995) requirements. An acoustical simulation was performed; however, pulsation filters were not designed and a poor mechanical layout was used. The basic design philosophy of avoiding resonance was not followed. Instead, a structural analysis of the system was conducted, including forced response calculations to justify the design even though high pulsation and potential resonances existed within the speed range. The calculations predicted low vibration and stress levels. The authors were consulted to assist the user company after a failure occurred in the main suction piping.

The layout, showing long unsupported spans is illustrated in Figure 26. The failure occurred in the 10 inch suction piping at the location indicated. Field tests showed that the piping mechanical natural frequency (Figure 27) was excited by pulsation energy (Figure 28) within the operating speed range. Measured vibration levels were well above the allowables. The measured vibration mode shape showed that the failure location was as would be expected (Figure 29).



Figure 26. Failure of 10 Inch Suction Piping.

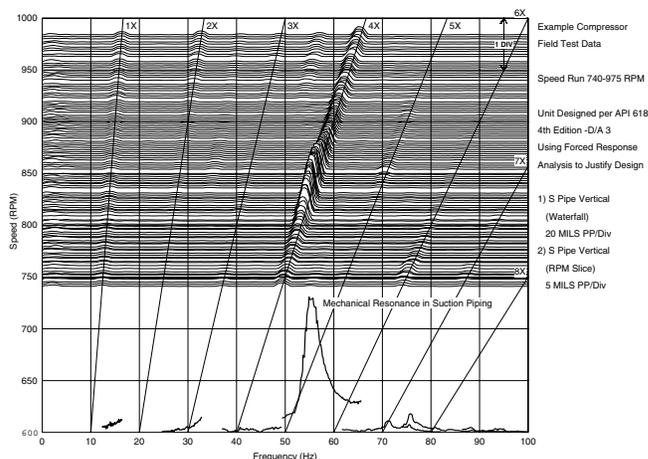


Figure 27. Mechanical Resonance of Suction Piping.

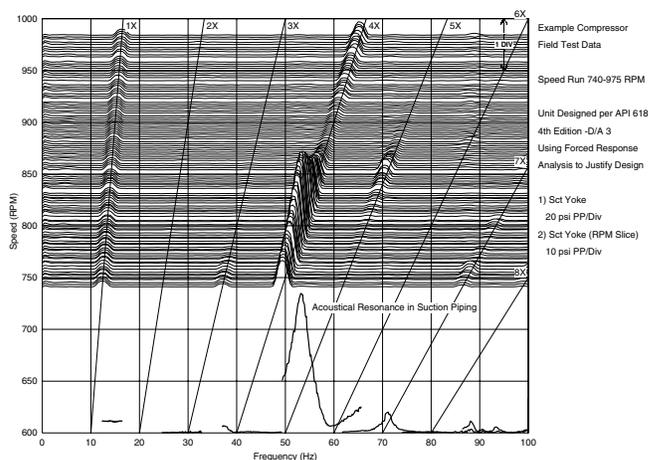


Figure 28. Acoustical Resonance in Suction Piping.

Failure of Suction Bottle Nozzles

Even if pulsation levels are controlled to acceptable levels, excessive vibration can occur if mechanical natural frequencies exist near significant mechanical excitation frequencies. Figure 30 shows a suction bottle system in which a volume-choke-volume filter was constructed in a single vessel. From a pulsation control standpoint, this design was effective. However, it resulted in a large

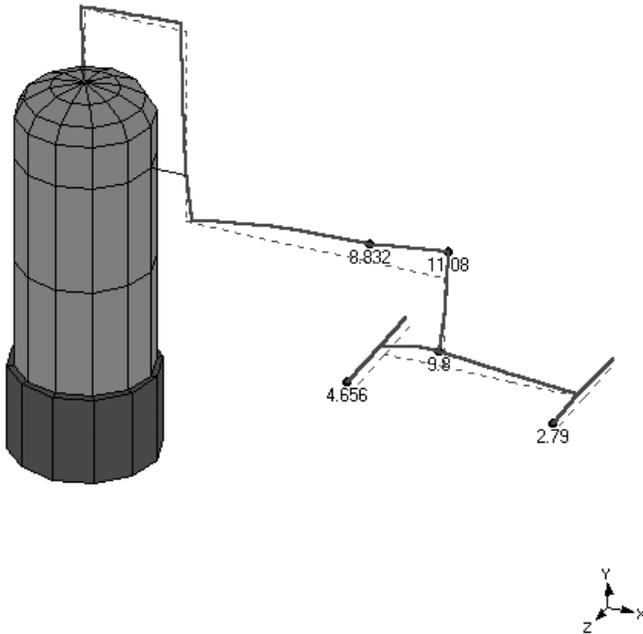


Figure 29. Measured Vibration Mode Shape.

diameter, relatively heavy suction bottle, which is difficult to dynamically support. It was practically impossible to raise the calculated mechanical natural frequency of the suction bottle cantilever mode above the range of expected excitation frequencies, so the designer specified *longer* suction nozzles to place this mode in between the third and fourth harmonics. Once again, there was too much confidence placed in the ability to calculate the mechanical response with such precision. The predicted frequency was 17.4 Hz, while the measured frequency was 15 Hz, Figure 31. This was within the range of excitation by the third harmonic for this variable speed machine (250 to 300 rpm). Excessive vibration levels occurred (Figure 32), and a suction bottle nozzle failure resulted.

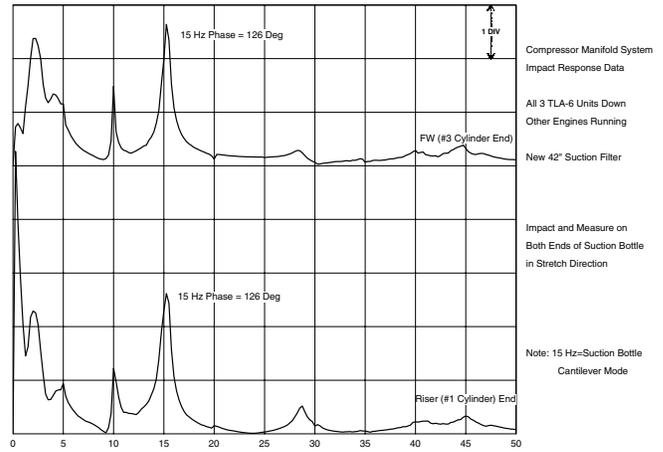


Figure 31. Impact Response Measurements of Suction Bottle.

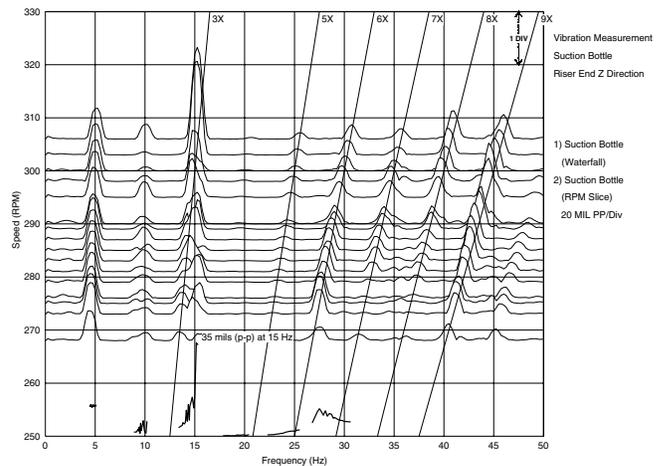


Figure 32. Measured Vibration on 42 Inch Suction Bottle.

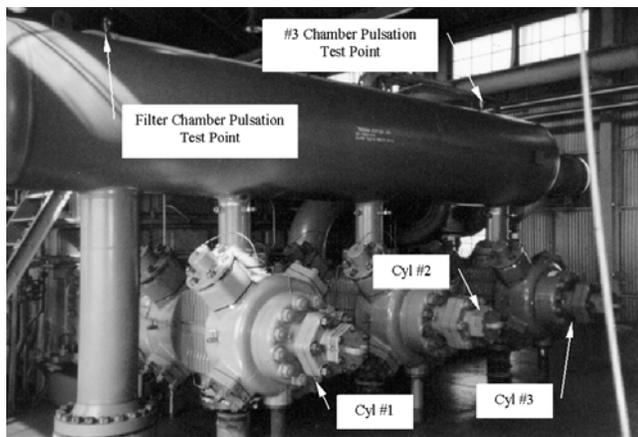


Figure 30. 42 Inch OD Filter in Single Bottle.

Gas Transmission Station Piping System

Figure 33 shows a sketch of a six-compressor discharge system for a gas transmission and storage facility. Six units, with four different compressor types, were operating in a common dual header system for storage or withdrawal. Compressors were added to the system over a 25 year period as demand increased. Unit 1 was the only compressor initially. It utilized volume-choke-volume filters and operated without problems. Unit 2 was added a year or

so later and the secondary volume was eliminated to save on costs. Vibration levels remained acceptable. A few years later, the third unit was added and the logic was that since Unit 2 was acceptable without an acoustical filter, the addition of Unit 3 without filtering should also be acceptable. At this point, minor problems began to occur with pipe clamps and instrument tubing. A fourth higher speed unit was installed that did not interact significantly with the other three units. Unit 5 was installed after a few more years, again without a filter. Chronic problems developed with severe vibration occurring at certain operating conditions. The sixth unit was added including a volume-choke-volume filter, but the vibration problems persisted. Pipe clamps in the discharge header repeatedly failed and pneumatic actuators experienced excessive vibration and failures.

Field testing showed that several system acoustic resonances were excited by the unfiltered units. This pulsation energy caused excessive vibration of the headers and also coincided with mechanical natural frequencies of several valve actuators (Figure 34). The ultimate solution was to install volume-choke-volume filters for each unit.

TYPICAL PACKAGED HIGH SPEED DESIGN

Figures 35, 36, 37, and 38 compare layout requirements for simple surge volumes and reactive filter designs. These examples are for a single stage, four-throw compressor typical of many packaged high-speed units. The reactive acoustical design (Figures 36 and 38) utilizes internal choke tubes and extended length discharge bottles. The scrubbers are utilized as secondary suction volumes. These designs generally do not require significant

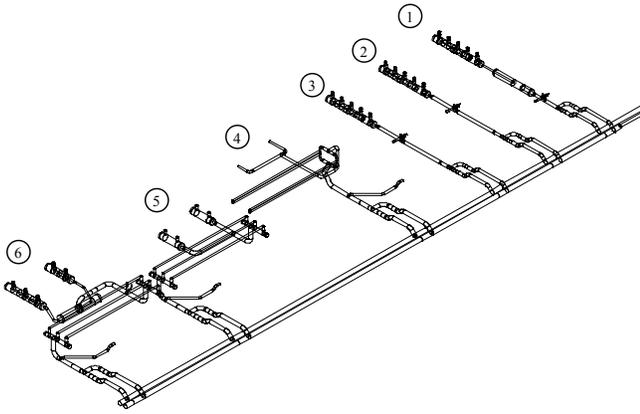


Figure 33. Gas Transmission Discharge Piping System Model—Six Compressors in Parallel.

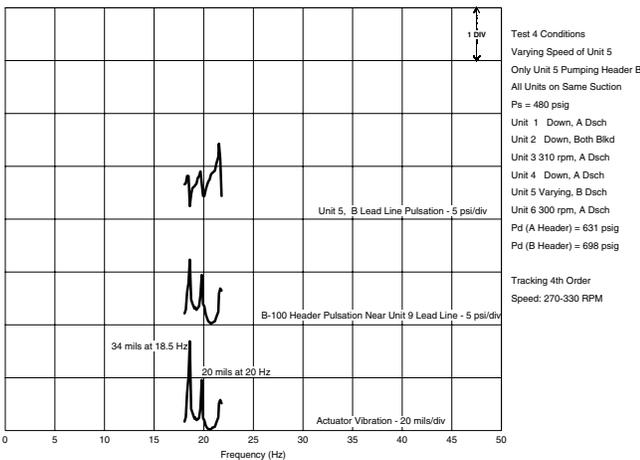


Figure 34. Measured Pulsation and Vibration in Discharge System.

additional skid area or fabrication and material costs, while offering superior pulsation and bottle force control as compared to simple surge volumes and orifice plates. Note that two scrubbers are used to avoid the type of poor layout that results when a single scrubber is used (Figures 39 and 40). This is important since the choke tube connecting the scrubber and suction bottle is subject to significant mechanical and acoustical excitation.

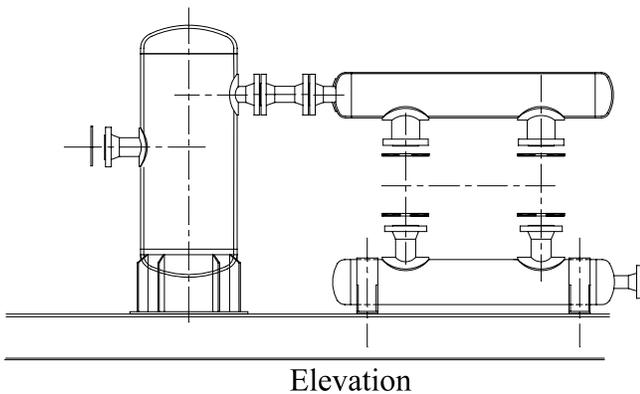
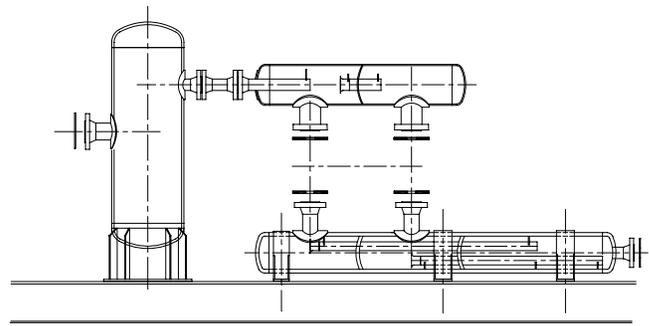


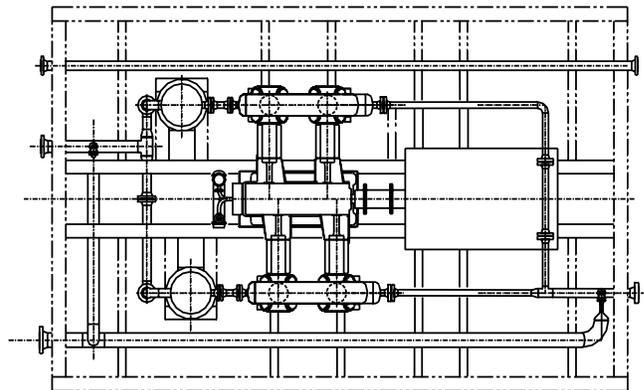
Figure 35. Skid Mounted Compressor Layout without Reactive Acoustical Filters.

Figures 41 and 42 show special dual choke tube designs that are required to eliminate certain acoustical modes on compressors with wide speed ranges (e.g., 2:1 turn down).



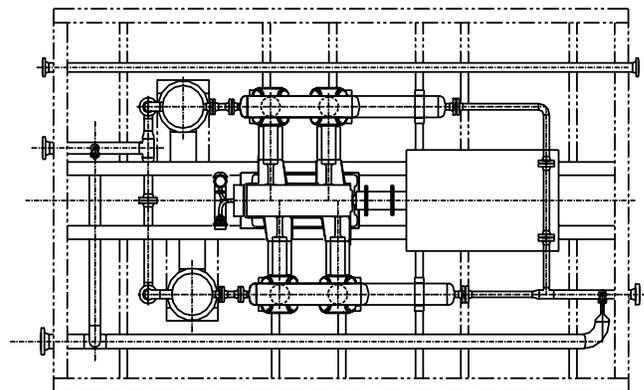
Elevation

Figure 36. Skid Mounted Compressor Layout with Reactive Acoustical Filter Designs.



Plan

Figure 37. Plan View of Layout without Reactive Acoustical Filters.



Plan

Figure 38. Plan View of Layout with Reactive Acoustical Filters.

CONCLUSIONS

- The intent of API 618 at the Third Edition was that Design Approach 3 meant effective pulsation control. This usually required that reactive filtering be used in relatively high mole weight systems. From the user's perspective, a Design Approach 3 system was a safe and reliable system. This standard served the industry well for many years.
- The Fourth Edition attempted to define the steps required to qualify a piping system, in the event that the allowable pulsation levels were exceeded. This created confusion and led to systems being designed with less emphasis on pulsation control, and



Figure 39. Poor Suction Layout for Skid Unit Resulting from Use of Single Scrubber.



Figure 40. Alternate View of Poor Suction Piping Layout Resulting from Use of Single Scrubber.

justified with mechanical response calculations of questionable validity. This is not to say that all systems designed this way are unsafe or unreliable. Such systems can work acceptably well, but it is more often the result of good fortune rather than accurate response calculations. Since the Fourth Edition was published, numerous users have been in the situation of having purchased a “Design Approach 3 System,” yet ending up with unacceptable and potentially catastrophic results.

- The Fifth Edition will clarify the confusion that resulted from the addition of the language concerning mechanical forced response calculations in the Fourth Edition. The user will now be able to determine if the system meets Design Approach 3 by the use of the technically sound pulsation and shaking force control philosophy (Step 3a), or through the use of the higher risk philosophies based on mechanical forced response calculations of steps 3b1 or 3b2.
- The authors’ experience, as presented in this paper, shows that robust pulsation control through the use of reactive acoustical filters is required to achieve safe and reliable piping systems of reciprocating compressors.

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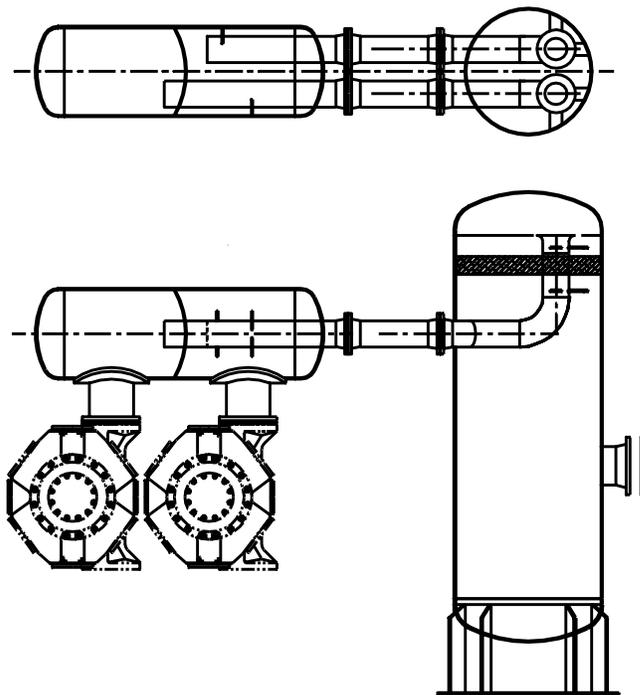


Figure 41. Example of Dual Choke Tube Reactive Filter Design Using Scrubber Volume—Suction.

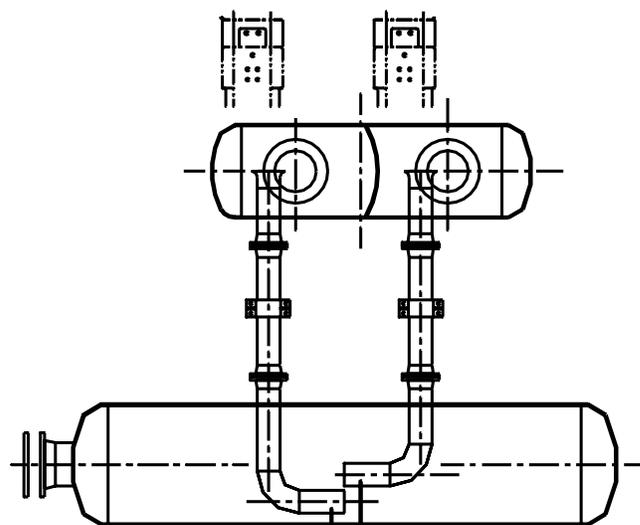


Figure 42. Example of Dual Choke Tube Reactive Filter Design with Secondary Bottle—Discharge.

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