Understanding the Pulsation & Vibration Control Concepts in the New API 618 Fifth Edition

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The Fifth Edition of API Standard 618 for reciprocating compressors will be published in the near future. This short course explains the changes from the previous editions and discusses the acoustical design philosophy required to achieve safe and reliable piping systems for reciprocating compressors.
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Abstract

The Fifth Edition of the API 618 specification is scheduled to be published in early 2005. Significant changes have been incorporated into the section concerning pulsation and vibration control. Specific guidelines are included to explain the necessity to perform certain analyses (that were optional in the Fourth Edition) based on pressure pulsation and shaking force levels determined from the acoustical simulation. This will hopefully reduce the ambiguity in determining whether or not forced mechanical response calculations should be performed.

The purpose of this short course is to provide the user with a working knowledge of good engineering practices for pulsation and vibration control for reciprocating machinery commonly used in the natural gas industry. An in-depth explanation of the changes in API 618 and the differing design philosophies will be presented. Several example cases illustrating design concepts will be used.

1. Introduction

In the 1950’s and 60’s, design techniques using analog simulation tools for the control of pulsation in compressor piping systems were developed. 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. 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 trend. First, the basics of pulsation control theory are not trivial and capabilities vary significantly among users of available software. Reactive pulsation control (acoustic filtering) requires more design effort. Generally, single surge volume designs in conjunction with resistive elements (orifice plates) 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. Most pipe stress analysis package on the market today claim some dynamic analysis capabilities. Mechanical natural frequencies and forced vibration levels of complex piping systems, once modeled, can be calculated fairly easily. However, the lack of understanding of the limitations on the accuracy of these calculations can lead 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.

The Fifth Edition of the API 618 specification will continue to include language concerning mechanical natural frequency and forced response calculations. 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 often problematic) mechanical natural frequency and forced response analyses for the qualification of piping system designs.
2. Basics of Excitation Mechanisms in Reciprocating Compressors

2.1. Pulsation Excitation Mechanisms

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

![Figure 1. Reciprocating Compressor Slider Crank Mechanism](image1)

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 = Area × 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, the flow excitation of either the suction or discharge can be considered independently for the purpose of understanding the pulsation excitation mechanism.

![Figure 2. Piston Velocity for Slider Crank Mechanism](image2)

![Figure 3. Single Acting Compressor Cylinder (L/R = ∞, Ideal Valves)](image3)

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 a single-acting
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 TDC, the discharge valves close, and the flow returns to zero.

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 integer multiples of the running speed. These frequencies are commonly referred to as “harmonics.” The highest amplitude occurs at $1 \times$ running speed, (for a single cylinder end) with the levels generally decreasing at higher harmonics.

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° apart in time. Therefore, the odd harmonics (in this idealized case) cancel, so that the cylinder flow excitation occurs at even harmonics of running speed ($2 \times$, $4 \times$, …).

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 integer 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.

2.2. 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× and 2× running speed, and are primarily a concern only in the immediate vicinity of the compressor.

3. Understand Limitations of Mechanical Modeling

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.

3.1. Inaccuracy of Mechanical Natural Frequency Calculations

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 contact the piping at some locations, which precludes accurate modeling assumptions.

Other items which influence the accuracy of these models are:

- Uncertainty of stiffness (six degrees of freedom) of clamps/hold downs and supporting structure
- Difficulty in accurately predicting coefficients of friction
- Nonlinear effects (e.g., gaps closing due to thermal growth)
- Uncertainty of “as-built” piping layout and dimensions, weights, etc.
- 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 and tubing
- Pipe supports (clamps, piers, etc.)
- 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

Other items which influence the accuracy of these models are:

- Uncertainty of “as-built” piping layout and dimensions, weights, etc.
- 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

Figure 7. Example of Vibration of Attached Branch Line Due to Base Excitation by Main Piping
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 structures. Typically, small branches attached to the main process piping are not even considered in mechanical natural frequency or forced 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 through pulsation control 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 p-p, while the actuator itself had vibration levels in excess of 50 mils p-p.

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 which occur in reciprocating compressor piping systems.

### 3.2. Effect of Inaccuracies of Mechanical Natural Frequencies

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: $\text{MNF} \propto \sqrt{k}$.

<table>
<thead>
<tr>
<th>Range of Actual Effective Stiffness</th>
<th>Range of Actual M.N.F.</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\frac{1}{1.5} \times k_{\text{calc}} \rightarrow 1.5 \times k_{\text{calc}}$</td>
<td>$0.8f_{\text{calc}} \rightarrow 1.2f_{\text{calc}}$</td>
</tr>
<tr>
<td>$\frac{1}{2} \times k_{\text{calc}} \rightarrow 2 \times k_{\text{calc}}$</td>
<td>$0.7f_{\text{calc}} \rightarrow 1.4f_{\text{calc}}$</td>
</tr>
<tr>
<td>$\frac{1}{10} \times k_{\text{calc}} \rightarrow 10 \times k_{\text{calc}}$</td>
<td>$0.3f_{\text{calc}} \rightarrow 3f_{\text{calc}}$</td>
</tr>
</tbody>
</table>

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 engineers agree that the minimum mechanical natural frequencies of the piping should be maintained at least 20 percent above the 2nd harmonic of running speed to avoid significant acoustical and mechanical excitation. Based on this design criterion, the lowest piping mechanical natural frequency is shown at 36 Hz (20 percent above 30 Hz) in Figure 10, assuming this can be accomplished.

![Figure 10. Pulsation Characteristics without Acoustic Filtering: Calculated Mechanical Natural Frequencies Superimposed](image)

Figure 11 shows the effect of uncertainty (due to calculation inaccuracy) of mechanical natural frequencies relative to harmonics of running speed for a 900 RPM compressor. The dashed vertical lines represent mechanical natural frequencies between 2× and 3×, 3× and 4×, and 4× and 5× running speed, etc. which happen to be “tuned” between harmonics. Assuming a ±20 percent mechanical natural frequency calculation accuracy (which is better than typically obtainable), the actual range of each natural frequency is also shown by the horizontal arrows. At all frequencies above 2× running speed, the possible range of actual mechanical natural frequencies between harmonics is too wide to avoid interference with the excitation harmonics. That is, the potential range of each mechanical natural frequency exceeds the frequency gap between harmonics. Therefore, in the design stage, it is not possible to tune any calculated mechanical frequencies above 2× running speed frequency away from pulsation excitation frequencies with any certainty.

![Figure 11. Pulsation Characteristics, No Acoustic Filtering: Range of MNF’s Superimposed](image)

Another important concept to remember is that even if the mechanical natural frequencies could be calculated within an accuracy of ±10 percent, significant amplification still occurs when the forcing frequency (i.e., the pulsation frequency) is within 10 percent (above or below) the mechanical natural frequency.

![Figure 12. Effect of Separation Margin on Amplification Factor](image)

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.

3.3. Inaccuracies of Forced Response Calculations

The new (Fifth Edition) API 618 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 forced mechanical response amplitude (vibration and stress) calculations is influenced by the same items which 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 – 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.

Forced 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 amplitudes (vibration and stress) at the design stage is generally impractical.

4. Pulsation and Shaking Force Control Using 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 due to the high speed of sound values, pulsation control can be accomplished through the use of ample surge volumes and resistive or pressure drop elements.

Acoustic filtering involves the use of two volumes joined by a relatively small diameter pipe, to create a volume-choke-volume filter. Figures 13 and 14 show various forms of the volume-choke-volume filter.

![Figure 13. Nonsymmetrical Volume-Choke-Volume Filter – Straight Choke Tubes](image)
These devices have 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:

$$f = \frac{c}{2\pi} \left( \frac{A_c}{L_c} \left( \frac{1}{V_1} + \frac{1}{V_2} \right) \right)$$

(1)

- $f$ = frequency (Hz)
- $c$ = speed of sound (ft/s)
- $A_c$ = Area of choke tube (ft$^2$)
- $L_c$ = $L_c + 0.6d_c$ (ft)
- $d_c$ = Choke diameter (ft)
- $V_1$ = Volume of primary bottle (ft$^3$)
- $V_2$ = Volume of secondary bottle (ft$^3$)

4.1. 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.

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× running speed pulsation levels can be controlled to any desired level.

4.2. Comparison of Pulsation Control Devices

Figures 17-21 compare pulsation in an infinite length discharge line (non-reflective boundary condition) of a compressor operating over a speed range of 700-1000 RPM. The assumption of a non-reflective boundary
eliminates acoustical resonances of the piping itself, and is a convenient method for comparison of the effectiveness of pulsation control devices.

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Pulsation Control</th>
<th>Discharge Line Pulsation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>None</td>
<td></td>
</tr>
</tbody>
</table>

Figure 17. Comparison of Pulsation Control Devices – Case 1

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Pulsation Control</th>
<th>Discharge Line Pulsation</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>1/2 x API Surge Volume (4'-0&quot; x 10.75&quot; I.D.)</td>
<td></td>
</tr>
</tbody>
</table>

Figure 18. Comparison of Pulsation Control Devices – Case 2

The pulsation control treatments are:

- None (no surge volume, Figure 17)
- A simple surge volume with a volume equal to 50% of the volume calculated using the API 618 Design Approach 1 sizing formula (1/2 × API Surge Volume, Figure 18)
- A simple surge volume with a volume equal to 100% of the volume calculated using the API 618 sizing formula (1 × API Surge Volume, Figure 19)
- Volume-choke device (Figure 20) (note relatively small improvement over Case 3)
- Volume-choke-volume filter with $f_H < 1 \times$ running speed (Figure 21)

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 condition; therefore, these cases can only be used for relative comparison of pulsation amplitudes.)

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Pulsation Control</th>
<th>Discharge Line Pulsation</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>1 x API Surge Volume (4'-0&quot; x 15.25&quot; I.D.)</td>
<td></td>
</tr>
</tbody>
</table>

Figure 19. Comparison of Pulsation Control Devices – Case 3

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Pulsation Control</th>
<th>Discharge Line Pulsation</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>1 x API Surge Volume with Choke (4'-0&quot; x 15.25&quot; I.D. Bottle) + 4'-0&quot; x 2.9&quot; I.D. Choke (0.34% $\Delta P$)</td>
<td></td>
</tr>
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</table>

Figure 20. Comparison of Pulsation Control Devices – Case 4

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Pulsation Control</th>
<th>Discharge Line Pulsation</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>Volume-Choke-Volume ($f_H &lt; 1 \times$) Vol. Each Bottle &gt; 1 x API Surge Volume (5'-0&quot; x 15.25&quot; I.D.) + (5'-0&quot; x 15.25&quot; I.D.) + 10'-0&quot; x 2.9&quot; I.D. Choke (0.42% $\Delta P$)</td>
<td></td>
</tr>
</tbody>
</table>

Figure 21. Comparison of Pulsation Control Devices – Case 5
4.3. Mechanical Analogies and Interpretation of Surge Volumes and Filters

4.3.1. Surge Volumes

At frequencies below the length resonances of the bottles themselves (passbands), volume bottles act predominantly as acoustic compliance. Acoustical compliance is analogous to mechanical flexibility as shown in Figure 22. The pipe beyond the surge volume contains the gas which 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 becomes more 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.

Figure 22. Mechanical Analogy of Surge Volume

4.3.2. Filters

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

4.3.3. 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 choke diameter and approximate length based on allowable pressure drop.
- Determine volume-choke-volume filter design to filter all harmonics of running speed. Generally, the filter frequency is set at 50-80 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 2) to determine additional support locations based on $f_m$.
• Determine minimum stiffness (k) of each support: $k \geq 2 \times \frac{\text{lateral span stiffness}}{\ell^3}$ ($\ell =$ support span).

5. Changes for API 618 Fifth Edition

5.1. Residual Non-Resonant Force Evaluation

A significant change currently being made to API 618 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 non-resonant 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):

$$\Sigma F = \overline{F}_A + \overline{F}_B$$

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

where: $\overline{P}_A$, $\overline{P}_B$ are vectors representing amplitude and phase of pulsation at points A & B at a particular frequency. These force calculations are easily made (when using digital techniques) based on the known piping geometry and calculated pulsation levels at the elbows.

While calculating the forces acting on the piping is fairly straightforward, determining an acceptable force level can be quite difficult. It is important to realize that the shaking force guideline for the new API 618 standard is based on specific non-resonant configurations.

<table>
<thead>
<tr>
<th>Table 2. Pipe Support Span Spacing Table</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Span Spacing (ft)</td>
</tr>
<tr>
<td>Natural Frequency</td>
</tr>
<tr>
<td>-------------------</td>
</tr>
<tr>
<td>10 Hz</td>
</tr>
<tr>
<td>20 Hz</td>
</tr>
<tr>
<td>30 Hz</td>
</tr>
<tr>
<td>40 Hz</td>
</tr>
<tr>
<td>50 Hz</td>
</tr>
</tbody>
</table>

Use of this acoustic filtering concept in conjunction with control of minimum piping mechanical natural frequencies provides a high level of confidence that resonance will be avoided.

![Figure 24. Dynamic Force on Piping Run](image-url)
In general, much higher force levels may be tolerated at frequencies below the lowest mechanical natural frequency; however, force levels should be controlled to very low levels at frequencies near or above the lowest piping mechanical natural frequency.

The force evaluation criteria proposed is based on the allowable vibration chart shown in Figure 25 and the effective support stiffness. It is the responsibility of the designer to determine the appropriate stiffness to apply. Note that the important concept of this 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.

Figure 25. Allowable Vibration Chart

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.

5.2 Design Process Flow Charts

API 618, Third Edition (1986), described in general terms the concepts of resonance avoidance and the use of filtering techniques. In Appendix M of the Fourth Edition (1995), descriptions of various procedures were added in an attempt to clarify the specific procedures necessary to meet the requirements of the Design Approaches 2 and 3. Procedure M.7 of the Fourth Edition read 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.”

This paragraph is somewhat ambiguous and subject to a great deal of interpretation. For example, what are the critical points in the piping system? What pulsation forces are significant and to what extent is it necessary to calculate vibration and stress amplitudes? What is meant by “response” – natural frequency or forced response amplitude?

Table 3 shows the procedures specified by Appendix M of the Fourth Edition for each of the three design approaches. Table 4 describes each of these procedures.

Table 3. Fourth Edition Analysis Procedures

<table>
<thead>
<tr>
<th>API 618, Existing Fourth Edition Appendix M</th>
<th>Pulsation Design Studies</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design Approach 1</td>
<td>Includes M.1</td>
</tr>
<tr>
<td>Design Approach 2</td>
<td>Includes M.2 through M.4</td>
</tr>
<tr>
<td>Design Approach 3</td>
<td>Includes M.2 through M.8</td>
</tr>
<tr>
<td>Optional</td>
<td>Are M.9 through M.11</td>
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Table 4. Description of Fourth Edition Analysis Procedures

<table>
<thead>
<tr>
<th>Design Approach 1 (Existing Fourth Edition)</th>
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<tbody>
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<td>M.1</td>
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<table>
<thead>
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<th>Design Approach 3 (Existing Fourth Edition) (Includes Design Approach 2 Plus M.5-M.8)</th>
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Because of the ambiguity in the Fourth Edition concerning which procedures are mandatory (especially for Design Approach 3) and which are done only when certain conditions are not met, three distinct design philosophies have evolved. They are:

- acoustic control
- shaking force control
- vibration control.

Acoustic control employs pulsation suppression device (PSD) designs that generally result in the lowest pulsation and shaking forces in the piping system of the three philosophies. It is based on providing pulsation levels at or below the historical API 618 (i.e. Third Edition and Fourth Edition) requirements such that there is never enough excitation force reaching the piping system to require forced response analysis.

Shaking force control utilizes the shaking forces in the piping system (while not specifically attempting to comply with the pulsation limits) to determine the amount of acoustic control that will be used in the PSD design. This technically complies with the wording of the Fourth Edition because of the note that states “Compliance with the pulsation limits should not be the sole design criteria”.

Designers using either of these first two philosophies will however, whenever possible, employ sufficient acoustic control such that the resulting pulsation reaching the piping system does not require a forced mechanical response analysis.

The vibration control philosophy, however, begins with the premise that a forced mechanical response analysis of the piping system is an integral part of the design process. This philosophy seems to be preferred by European designers who interpreted the Fourth Edition to require that M.7 analysis is mandatory for Design Approach 3. Finite element models of piping systems are routinely made, and piping forced response is calculated. This apparently fits well with their normal design and construction procedures since:

1. The PSDs are designed prior to knowing the piping details. Therefore, the pulsation control is not optimized to the piping system.
2. The mechanical analysis is used to design and justify more complex supports and anchoring systems to control vibration.
3. Significantly more precision is utilized in constructing and maintaining the piping system to match the mechanical model.
Each of these design philosophies can be utilized to create a system with acceptable pulsation and vibration levels. With this in mind, the Fifth Edition was written to address all three. To provide further technical explanation of the application of these philosophies, a new document has also been created by API. It is RP-688 “Recommended Practice for Pulsation and Vibration Control of Positive Displacement Machinery”.

However, even though all three philosophies can be successfully utilized, the default philosophy for API 618 machines is “acoustic control” for the reasons explained in this paper. If the user wants to have a system designed using either of the other philosophies, they will have to explicitly select it on the data sheet.

The flow chart shown in Figure 26 describes the work process used to satisfy the three design approaches in the new Fifth Edition of API 618 with various analysis procedures corresponding to the Fourth Edition noted. Design Approach 1 (DA1) is not changed from the previous editions and involves no simulation of the system. The PSDs 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 non-reflective piping to provide pulsation levels that are no greater than 80% of the API allowable. The intent is to allow procurement of the PSDs earlier in the project time line, 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.

![Figure 26. Work Process Flow Chart for API 618 Fifth Edition](image-url)
Once the piping layout is finalized, a complete acoustic simulation is carried out. If Design Approach 2 (DA2) is specified, the pulsation, pressure drop and the new PSD shaking force criteria must be met. A table of maximum clamp spacing is developed based on avoiding resonance with any significant pulsation energy. No evaluation of piping 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 as for the previous editions. The key changes include the introduction of shaking force criteria and specific steps which 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 non-resonant 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 does not comply with all of the DA-3 criteria.

The DA-3 criteria again are low pulsation, low shaking forces and non-resonant mechanical systems. If these criteria are not met, the additional steps include calculations 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 it is preferable not to be in the situation of having to resort to these analyses. First, to be in the position of having to calculate vibration levels requires that either:

- the shaking forces are too high, or
- there is a mechanical natural frequency of the manifold system or piping that violates the separation margin, or
- the PSD design has not been optimized for the piping system.

If the shaking forces are too high, one would also expect the calculated vibration levels to be high, since the shaking force criteria was derived from the allowable vibration curve. If the separation margin is not met, then for practical purposes the chances of the system being resonant is high, which is obviously an undesirable situation. If the PSDs were not optimized for the actual piping system, then it is usually necessary to provide additional volumes, orifices, etc. in the piping system to obtain additional acoustic control and engineered restraints to provide mechanical control.

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 and 3b2 to satisfy DA-3 is necessarily a higher 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 26) 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 is only required if the designer has failed to control pulsation levels to the specified levels 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 5 shows analysis procedures, which are optional in the Fourth Edition and will continue to be optional in the new Fifth Edition. In the Fifth Edition, they will be found in the main text of the appropriate paragraphs as “when specified…” items, instead of listed separately in Appendix M. They will not be mandatory for Design Approach 3.

Table 5. Optional Procedures (New Fifth Edition)

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6. Case Histories

6.1. Failure of 10” 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 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 27. The failure occurred in the 10” suction piping at the location indicated.

Field tests showed that the piping mechanical natural frequency (Figure 28) was excited by pulsation induced forces (Figure 29) 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 30).
6.2. 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 31 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 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 32. This was within the range of excitation by the 3rd harmonic for this variable speed machine (250-300 RPM). Excessive vibration levels occurred (Figure 33), and a suction bottle nozzle failure resulted.
6.3. Gas Transmission Station Piping System

Figure 34 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.

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 35. The ultimate solution was to install volume-choke-volume filters for each unit.
6.4. Typical Packaged High Speed Designs

Figures 36-39 compare layout requirements for simple surge volumes and reactive filter designs. These examples are for a single stage, 4-throw compressor typical of many packaged high-speed units. The reactive acoustical design (Figures 37 and 39) utilizes internal choke tubes and extended length discharge bottles. The scrubbers are utilized as secondary suction volumes. These designs generally do not require significant 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 40-41). This is important since the choke tube connecting the scrubber and suction bottle is subject to significant mechanical and acoustical excitation.

Figures 40 and 41 show special dual choke tube designs which are required to eliminate certain acoustical modes on compressors with wide speed ranges (e.g., 2:1 turn down).

Figures 42 and 43 show special dual choke tube reactive filter designs to eliminate these modes.
7. Conclusions

1. 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 (e.g. natural gas) 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.

2. 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 by justification system acceptability through mechanical response calculations (of questionable validity). This is not to say that all systems designed this way are unsafe or unreliable. Such systems can operate successfully, 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.

3. 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 control philosophy (Step 3a), or through the use of the higher risk philosophies based on mechanical forced response calculations of steps 3b1 or 3b2.

4. Step 3a (pulsation control) has been adopted by API as the default design philosophy for satisfying Design Approach 3 requirements. Caution is advised whenever Steps 3b1 or 3b2 are employed.

5. The authors’ experience, as presented in this paper, shows that robust pulsation control (though the use of reactive acoustical filters in relatively high mole weight gases or ample surge volumes with resistive elements in lighter mole weight gases) is required to achieve safe and reliable piping systems of reciprocating compressors.
References


